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Life Cycle Costs –

Reciprocating Compressors in the Focus of Function, Economics and Reliability

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Inhalt / Contents

LIFE CYCLE COSTS -
Reciprocating Versus Rotating Technology in Natural Gas Compression

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability May 17th / 18th, 2001, The Hague

Abstract:

Modern natural gas compression is a very diversified business both globally and from application to application. This article provides a short and comprehensive comparison between two leading gas compression technologies. Due to fundamental differences in the technologies, a direct comparison is difficult and some generalisations must be made. It will be shown the principal benefits and disadvantages of both technologies and the impact of these characteristics to the operating costs of typical applications.

Today’s market is divided between centrifugal compressors driven by gas turbines and reciprocating compressors driven by electric motors or gas fuelled, spark ignited piston engines. The selection is based, in some degree, on either traditional application for the service type or a clear technological benefit over the other.

Naturally there are many aspects with impact on the decision making process. One of the most important is reliability of the package (driver and compressor). Local norms, codes and regulations are a jungle, as well as diverse end user preferences, all of which have to be clarified on a case to case basis: and these have an impact on the investment price as well. Generally it can be said that with smaller size units the most important aspects are investment price and quick delivery. With larger units, life cycle costs take on a greater degree of significance in the process.

1 Problemanalyse

Life Cycle Costs: Was ist das? Jeder benutzt diesen Terminus Technicus, viele benutzen diese Vokabel heutzutage, weil es modern ist über Geld, über Kosten zu reden, aber auch weil in nahezu allen Firmen und Institutionen klar geworden ist, dass mit Verbesserungen, Leistungssteigerungen, Kostensenkungen, usw. allein langfristig keine wirklichen Erfolge erzielbar sind. Man denkt heute in den modernen Industrieländern in Vorbereitung von größeren Investitionsentscheidungen nicht nur an die Investitionskosten an sich, sondern an die Summe der Kosten, die entstehen, während die geplante Investition ihre geplanten oder auch ungeplanten Dienste verrichtet.

Besonders wichtig sind für unsere Branche vor allem

- Energiekosten
- Schmierstoffkosten
- Wartungskosten
- Ersatzteilkosten
- Kosten, die durch evtl. relative kurze Nutzungsdauer und damit zusammenhängenden Wiederbeschaffungskosten ,entstehen,

aber auch

- Finanzierungskosten, etc.

Ein Wust von stark von einander abhängigen Details, die nicht in einer "Weltformel" algorith-mierbar sind. Es gibt aber, und das ist ein Segen und eine Chance für uns (auch für unsere Wettbewerber in unserer Branche "Reciprocating Compressors"), keine "eierlegende Wollmilchsau".

Die Aufgabe des Verkäufers für Kolbenkompressoren-Anlagen wird zunehmend geprägt durch die Vermittlung von Kenntnissen aufbauend auf betriebswirtschaftlichen und sonstigen kaufmännischen Zusammenhänge seines Kunden, die reine Auflistung von technischen Vorteilen seines Produktes und seines Preises gehört schon seit längerem der Vergangenheit an. Analog hierzu gilt das für den Einkäufer von Kompressoren bzw. den Entscheider "kaufen oder nicht", seine Analysen werden immer komplexer und immer mehr auf die Lebenszeit seiner Anlage bezogen.

1.1 Randbedingungen einer Studie "Kolben im Vergleich zum Turbo"

Um Betrachtungen zu den Life Cycle Costs einer Kompressorenanlage - und dann auch noch im Vergleich dieser zu anderen, die nach anderen Technologien arbeiten - machen zu können, sind zunächst viele Voraussetzungen definieren, die man unverändert konstant bei der Bearbeitung dieser Studie belässt:

- Eine ganz wichtige Voraussetzung ist, dass im Rahmen dieser Studie nur das Medium 'Erdgas' betrachtet wird. Es ist völlig klar, dass bei Kompressorenanlagen, die andere Gase zu verdichten haben, andere Gesetzmäßigkeiten gelten.
- Die erarbeiteten Ergebnisse (vielleicht sogar Gesetzmäßigkeiten oder Trends) gelten nur im betrachten Antriebsleistungsbereich von ca. 1 MW bis ca. 10 MW.
- Die Saugdrücke der untersuchten Beispiele liegen im Bereich 20 - 100 bar und die Enddrücke nicht über 400 bar. Bei anderen Drücken und Druckdifferenzen werden auch tendenziell andere Ergebnisse erwartet, vor allem bezüglich der Abgrenzung "Kolbenverdichter" zum "Turboverdichter"
- Sehr kleine und sehr große Kompressoren unterliegen sicherlich völlig anderen Regeln und Trends (extreme Groß-Serienproduktionen bzw. Einzelfertigungen, etc.) als die im Rahmen dieser Arbeit erfassten, die sich auf zum Beispiel Volumenströme von ca. 10.000 bis ca. 200.000 m3 (Vn)/h beschränkt.

2 Kompressorentechnologie im Vergleich

Diese Studie liefert einen Vergleich zweier führender Verdichtungstechnologien, wobei die heutige Erdgasverdichtung sowohl regional als auch entsprechend der Anwendung verschiedenartig ist. Wegen grundlegender Unterschiede zwischen diesen Verfahren ist ein direkter Vergleich schwierig, so dass einige Vereinfachungen gemacht werden müssen. Diese Studie zeigt die prinzipiellen Vor- und Nachteile beider Technologien sowie deren Auswirkungen auf die Betriebskosten. Der heutige Markt ist aufgeteilt in turbinengetriebene Turboverdichter, elektromotorisch getriebene Turboverdichter und sowohl e-motorengetriebene Kolbenverdichter als auch gasmotorischgetriebene Kolbenverdichter.
Die Wahl für die eine oder andere Verdichtervariante fällt entweder aus traditionellen Gründen oder aufgrund klarer technischer Vorteile eines Verdichtertyps für den speziellen Anwendungsfall. Bei geringen Antriebsleistungen, bis ca. 2000 kW, kommen fast ausschließlich Kolbenverdichter zum Einsatz. Bei höheren Leistungen steigt der Anteil der Turboverdichter.

2.1 Kompressionsverfahren

Der Kolbenkompressor ist eine Hubkolbenmaschine, die ein gegebenes Gasvolumen pro Hubbewegung verdichtet, unabhängig von Gasdichte, -druck und -temperatur. Das verdichtete Gas wird gegen den Systemdruck auf der Auslassseite ausgestoßen.
Kolbenkompressoren können einen großen Volumenstrombereich abdecken. Dazu bedient man sich einer Drehzahlsteuerung ebenso wie Methoden der Volumensteuerung (Ventilabhebung, Schadraum-veränderung, etc.) Dieser große Einsatzbereich wird bei einem hohen Wirkungsgrad erreicht.
Ein Turboverdichter ist eine dynamische Maschine, bei der die Verdichtung durch Umwandlung kinetischer Energie zu Druck erfolgt. Um dies zu erreichen, muss die Verdichtergeometrie (d.h. Turbinenschaufeln, Diffuser, etc.) speziell für den Betriebspunkt ausgelegt werden. Diese exakte Auslegung schränkt den Einsatzbereich jedoch ein. Um eine Turboanlage in einem akzeptablen Einsatzbereich betreiben zu können (normalerweise darf der Volumenstrom nur 80 bis 105% des Auslegungswertes betragen), muss ein Drosselklappen- und Bypass-System installiert werden, welches den Wirkungsgrad stark verringert.
Turboverdichter haben einen weitaus kleineren Bereich hohen Wirkungsgrades als Kolbenkompressoren (Bilder 1 und 2).
Der Kolbenkompressor mit seiner nahezu isen-tropen Verdichtung hat in den meisten Fällen klare Wirkungsgradvorteile vor dem Turboverdichter, der nach einem weniger effizienten polytropen Verfahren arbeitet. In der Verdichtungsstufe eines Kolbenkompressors wird nur wenig Wärme aufgrund von Gasturbulenzen erzeugt.

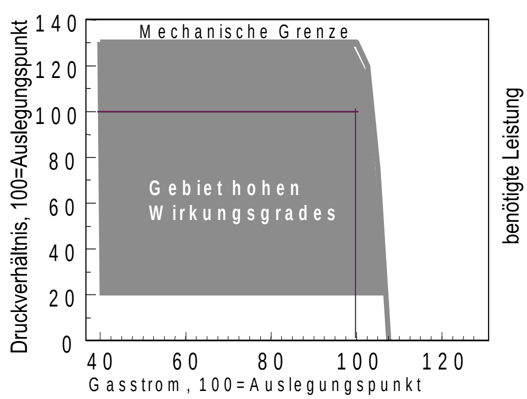

Abb. 1: Kennlinie des Kolbenkompressors

Um das gleiche Verdichtungsverhältnis mit einem Turbokompressor zu erreichen, benötigt man mehrere Verdichtungsstufen (üblicherweise 7 bis 9 mal mehr), und aufgrund von Leckagen zwischen den Stufen wird mehr Wärme durch Turbulenzen erzeugt. Zusammen mit den Verlusten, die durch die polytrope Verdichtung entstehen, hat der Turboverdichter einen geringeren Wirkungsgrad als der Kolbenverdichter. Der Unterschied im Gesamtwirkungsgrad der Anlage wird um so größer, je höher das Verdichtungsverhältnis und der Enddruck sind.

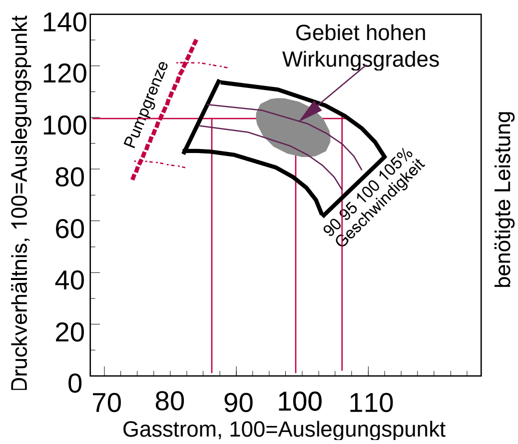

Abb. 2: Kennlinie des Turbokompressors

In Anwendungen mit geringem Verdichtungs-verhältnis und großem Volumenstrom, wie sie z.B. bei Pipelineanwendungen auftreten, können Turboverdichter jedoch einen höheren Wirkungsgrad erreichen. Bei diesen Anwen-dungen spielen die Ventilverluste der Kolben-kompressoren eine große Rolle.

2.2 Leistungsdiagramme

Bild 3 zeigt ein typisches Leistungsdiagramm einer 3000 kW Kompressorstation mit einer Druckerhöhung von 16,5 bar auf 45 bar. Dieses Diagramm zeigt den Enddruck beider Kompressortypen in Abhängigkeit vom Verhältnis Volumenstromkapazität /Volumenstrom. Der Auslegungspunkt liegt bei 58000 Nm3 /h mit 45 bar.

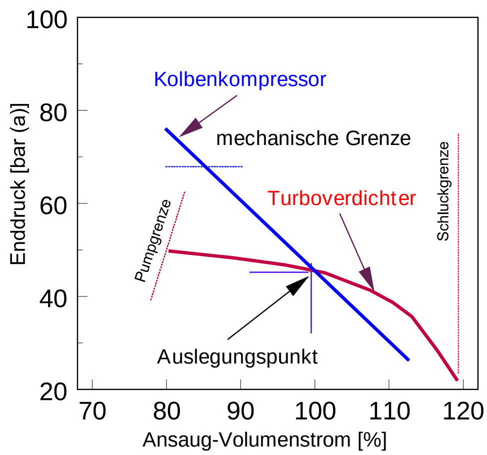

Abb. 3: Vergleich der Kompressorenkennlinien (Turbo und Kolben)

Sehr selten ist der Auslegungspunkt der einzige Betriebspunkt und fast immer ist eine Regelung erforderlich, entweder um die Kapazität konstant zu halten, oder um sie zu ändern. Typische Schwankungen sind Änderung in der Gastemperatur oder des Saugdrucks, die beide einen Einfluss auf die Dichte des Gases und damit auch auf den Volumenstrom und die notwendige Leistung haben.

Abb. 4: Typische gasmotorisch getriebene Kolbenkompressorenanlage (Buchholz/D)

Der Kolbenkompressor hat eine sehr steile Kennlinie, was bedeutet, dass geringe Änderungen im Gasstrom große Änderungen im Enddruck bewirken. In der Praxis bedeutet das, dass jede Veränderung des Saugdrucks, der Gastemperatur oder des Volumenstroms eine merkliche Änderung im Enddruck bewirkt, die einfach zu überwachen ist und als Eingangsvariable für die Regelung des Prozesses fungiert. Um den Druck und die erforderliche Leistung auf dem gewünschten Niveau zu halten, muss nur die Geschwindigkeit des Antriebs entsprechend geregelt werden. Auch der Kompressor selbst kann mit automatischen Regelsystemen bestückt werden, die z.B. den Schadraum oder die Füllung des Kompressors verändern und damit auch den Enddruck sowie den Volumenstrom. Durch diese Regelmöglichkeiten, die keine Energieverluste im System verursachen, bieten Kolbenkompressoren hohe Wirkungsgrade bei verschiedenen Betriebzuständen.

Der Turboverdichter hat eine relativ flache Leistungskurve, was bedeutet, dass sich der Enddruck nur wenig verändert, selbst bei großen Änderungen des Saugdrucks, der Gastemperatur oder des Volumenstroms. Trotzdem haben diese Veränderungen einen direkten Einfluss auf die benötigte Leistung und verschieben den Betriebspunkt erheblich. Aus diesem Grund benötigen Turboverdichter ein empfindliches und teures Regelsystem, um den gewünschten Betriebspunkt einzuhalten.

Turboverdichter sind fest ausgelegte Maschinen für einen Betriebspunkt, die keine Regelung des Verdichtungsverhältnisses oder des Volumenstroms wie beim Kolbenkompressor erlauben. Aufgrund dieser mangelnden Flexibilität müssen energiezehrende Saugdruckventile und Bypass-Systeme eingesetzt werden, um den Betriebspunkt zu erreichen. Diese Regelsysteme verringern einen bereits niedrigen Wirkungsgrad und erhöhen die Investitions- und Betriebskosten.

Es ist zwar grundsätzlich möglich, die Geschwindigkeit des Turboverdichters zu verändern, aber das hat wiederum einen negativen Einfluss auf den energetischen Wirkungsgrad der Antriebsmaschine. Um einen neuen Auslegungspunkt zu realisieren, können Veränderungen an der Maschine durchgeführt werden, doch sind diese Möglichkeiten sehr begrenzt, teuer und zeitaufwendig.

Abb. 5: Typische gasturbinen-getriebene Turboverdichtereinheit

3 Verschiedene Technologien bei der Anwendung "Untergrund-Erdgas-speicherung"

Aus der Vielzahl der Anwendungen für Kolbenverdichter im Bereich der chemischen, petro-chemischen und anderen Industrien soll hier eine spezielle Nische, die Verdichtung von Erdgasen zur Speicherung in Untergrundspeichern, betrachtet werden.

3.1 Erdgasspeicherung

Bei dieser Art der Speicherung wird das Erdgas in großen Mengen in natürliche, gasdichte geologische Formationen oder in speziell hergestellte Hohlräume eingepresst und bei Bedarf wieder in das Versorgungsnetz rückentspannt. Hierzu werden ausgeförderte Gas- oder Öllagerstätten, gleichartig angeordnete geologische Formationen die jedoch wassergefüllt sind (Aquifere) und speziell für die Erdgasspeicherung errichtete Salzkavernen genutzt.
Die Zwischenspeicherung von Erdgasen erfolgt zur Erhöhung der Versorgungssicherheit und zur Vergleichmäßigung des Gasbezuges bei schwankendem Verbrauch. Bei der Betriebs-führung eines Speichers sind sowohl die saisonalen Schwankungen (Sommer/Winter) als auch Verbrauchsspitzen entscheidende Kenngrößen. Wirtschaftliches Idealziel ist ein konstanter Gasbezug über das gesamte Jahr und eine 100%ige Ausnutzung des Speichervolumens am Ende des Betrachtungszeitraums. Die Erreichbarkeit dieses Zieles wird durch die Genauigkeit der Verbrauchsprognose und die vorzuhaltende Versorgungsreserve begrenzt. Um jedoch keinen zusätzlichen betrieblichen Einschränkungen zu unterliegen, müssen die Ein- und Ausspeichereinrichtungen über ein Höchstmaß an Flexibilität verfügen. Nur so kann der Speicher nah am jeweils aktuellen Bedarf
gefahren werden.

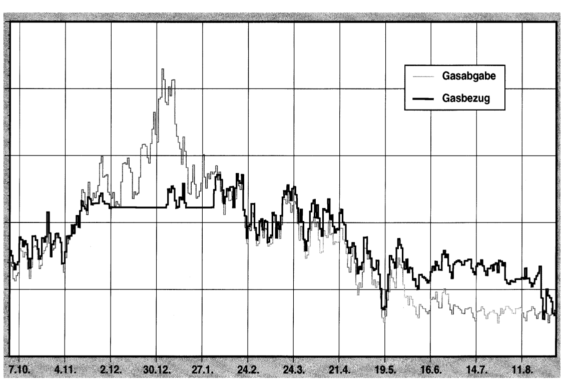

Abb.6: Vergleich Gasbezug/Gasabsatz

Neben diesen bisherigen Aufgaben für Erdgasspeicher, die immer auf ein Versorgungsgebiet und die individuelle Situation des Versorgungsunternehmens bezogen sind, ergeben sich aus der Liberalisierung des Gasmarktes neue Einsatzmöglichkeiten. Die verbrauchsunabhängige Nutzung freier Durchleitungskapazitäten in fremden Netzen und der Handel mit Spotmengen ist nur mit der Nutzung von Speichern sinnvoll, um aus der Veränderung der Abgabepreise eine zusätzliche Wertschöpfung zu erzielen.
Die Speichervolumina betragen 50 x 10^6 m³(V_n) bis 3000 x 10^6 m³(V_n) je nach Gasumsatz des Betreibers und geologischen Möglichkeiten. Die maximalen Speicherdrücke liegen zwischen 100 bar und 320 bar und werden durch die Teufenlage und geologischen Gegebenheiten bestimmt. Auch die minimalen Speicherdrücke sind nicht frei wählbar, sondern müssen die Bedingungen der Lagerstätte oder des Salzstockes berücksichtigen und liegen zwischen 40 bar und 120 bar.

Betrachtet man den Verlauf des Gasverbrauchs in Relation zum Gasbezug, so ergeben sich Zeiten des Gasüberschusses (= Einlagerung) und des Zusatzbedarfs (= Auslagerung). Aus der Größe der Abweichung lassen sich die erforderlichen Injektions- und Auslagerraten bestimmen. Die Injektionskapazitäten liegen zwischen 25.000 m³(V_n)/h und 400.000 m³(V_n)/h, die Auslagerungsraten zwischen 100.000 m³(V_n)/h und 1.200.000 m³(V_n)/h.

Abb. 7: Turboverdichter im Einsatz "Untergrund-Erdgasspeicherung"

Die Injektion von Spotmengen erfordert, neben dem verfügbaren Speicherraum, tendenziell auch eine Erhöhung der Einlagerkapazitäten. Während die Kompressoren bisher für die Füllung des Speichers während der gezeigten Schwachlastzeiten dimensioniert wurden, erfordern Spotmengen die schnellstmögliche Einlagerung unter Berücksichtigung der verfügbaren Durchleitungskapazitäten.

3.2 Betriebsanforderungen

Aus den vorgenannten Rahmenbedingungen lassen sich die Betriebsanforderungen für die Auslegung der Ein- und Ausspeicheranlagen
festlegen. Die Einspeicheranlagen bestehen im wesentlichen aus Filter, Mengenmessung, Gasverdichter, Gaskühler und Ölabscheider.

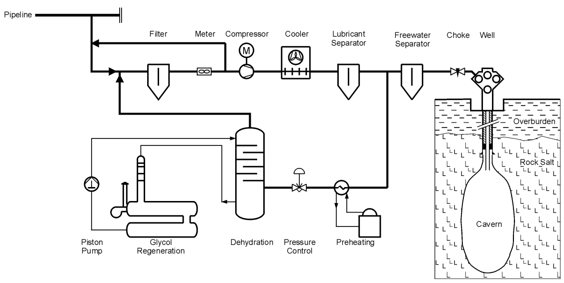

Abb.8: Prozessfließbild Erdgasspeicherung

Die oben erwähnten Einspeichermengen werden im Zusammenhang mit den durch die Pipelineanbindung gegebenen Saugdrücken und den notwendigen Redundanzen betrachtet. Geht man von mindestens 2 x 50 % als Sicherung der Verfügbarkeit aus, ergeben sich die typischen Auslegungsdaten für einen Verdichter in einem Erdgasspeicher:

- Saugdrücke: 30...85 bar (Pipelinedruck)
- Enddrücke: 50...320 bar (Speicherdruck)
- Fördermenge: 12.000...200.000 m³(V_n)/h

Um auf Basis dieser Daten zu einer optimalen Maschinenauswahl zu kommen, sind die betrieblichen Randbedingungen zu betrachten.

- Verfahrensbedingt müssen die Verdichter in der Lage sein, von den anstehenden Pipelinedrücken die jeweils zur Verfügung stehende Überschussmenge auf den Speicherdruck zu verdichten, der dem aktuellen Speicherfüllstand entspricht.
- Der Verdichterbetrieb erfolgt intermittierend. Neben längeren Einlagerperioden im Sommer erfolgen kurzzeitige Einlagerungen in verbrauchsarmen Zeiten wie z.B. an Wochenenden oder Unterschieden im Tages- und Nachtverbrauch.
- Die jährliche Gesamtlaufzeit liegt bei ca. 2000 h. Die Einsatzbereitschaft muss für ca. 6000 h gewährleistet sein.
- Die Maschine muss für das Fördermedium Erdgas in unterschiedlichen Zusammensetzungen und mit den zulässigen Begleitstoffen geeignet sein.
- Die "Life-Cycle-Costs" für Verdichteranlagen für die Injektion des Gases stellen einen wesentlichen Posten der Gesamtkosten eines Erdgasspeichers dar. Geringe Beschaffungskosten und ein wirtschaftlicher Betrieb der Kompressoren sind daher ein wesentliches Auswahlkriterium.

3.3 Einsatzgebiete verschiedener Verdichter im Untergrund-erdgasspeicher

Die Einsatzmöglichkeiten der bisher beschriebenen Verdichter lassen sich nur im Bezug zu den Anforderungen bestimmen. Bei der Verdichtung von Erdgas zur unterirdischen Speicherung z.B. treten Druckverhältnisse Φ von 1 bis maximal 18 auf, wobei effektive Ansaugvolumenströme $\dot{V}_{eff}$ von 250 bis 10.000 m³/h und Enddrücke p_2 von 40 bis 350 bar realisiert werden können.

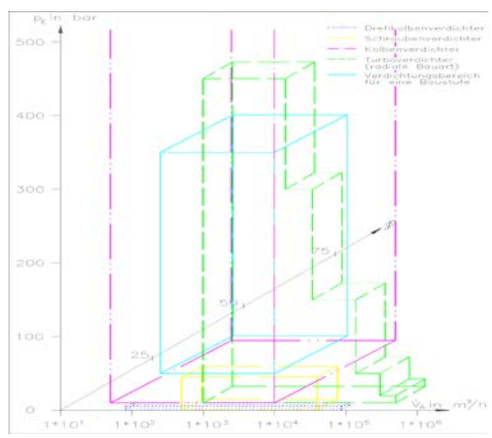

Abb. 9: Einsatzgebiete verschiedener Kompressorentypen

- *Hubkolbenverdichter*
- *Drehkolbenverdichter*
- *Schraubenverdichter*
- *Turboverdichter (radialer Bauart)*

Im Bild 9 sind die Einsatzgebiete verschiedener Verdichter und Verdichtungsanforderungen einer Baustufe eines Untergrund-Erdgasspeichers dargestellt

Es ist auch zu erkennen, dass sich für den Einsatz auf Untergrund-Erdgasspeichern nur Hubkolben-verdichter (Langsam- und Schnell-Läufer) und Turboverdichter radialer Bauart eignen.

Es zeigt sich, dass die Kolbenverdichter den gesamten Bereich und Turboverdichter nur den Bereich bis zu einem Druckverhältnis von maximal 5 der für die Erdgasspeicherung erforderlichen Verdichtung abdecken. Es stellt sich die Frage, welcher Verdichtertyp eingesetzt werden soll. Die Auswahl erfolgt primär nach wirtschaftlichen Kriterien:

- Investitionskosten
- Spezifischer Leistungsbedarf
- Betriebskosten
- Aufwendungen für den Bau und die Neben-anlagen.

Doch nicht nur die monetär bewertbaren Aufwendungen spielen eine Rolle, auch die Verfügbarkeit der Maschinen und der Service des Herstellers gewinnen an Bedeutung.
Gerade in den ersten Baustufen der Speicheranlage, in der noch keine Redundanz vorliegt, ist eine hohe Betriebssicherheit ausschlaggebend für den Speicherbetrieb.

Aus diesen größtenteils wirtschaftlichen Kriterien lassen sich technische Parameter ableiten, die im Bild 10 für beide Verdichter-Bauarten betrachtet werden.

Die Gegenüberstellung der beiden Verdichter-Bauarten zeigt, dass der Hubkolbenverdichter wesentliche betriebliche Vorteile aufweist. Diese liegen vor allem in der besseren Regelbarkeit auf Grund der Fördercharakteristik (Bild 3) und damit in der optimalen Abdeckung der einzelnen

Kriterium	**Hubkolbenver-dichter** mehrstufig, Boxerbauart	**Radial Turboverdichter** mehrstufig, einwellig
Enddruck und Druckverhältnis	werden von beiden Verdichtertypen erreicht, jedoch beim Turboverdichter im Bereich ho- her Druckverhältnisse und Enddrücke mit hö- heren Investitionskosten	
Wirkungsgrad	von einer Mindestlast ab annähernd konstant, η ist ca. 0,9	leistungsabhängig, dadurch erhöhte Ver-luste bei Teil- und Überlast η bis ca. 0,8
Betriebsbereich und Regelbarkeit	großer Betriebs-bereich, damit Abdeckung aller Betriebspunkte möglich, Regelung im gesamten Lei-stungsbereich	eingeschränkter Be-triebsbereich im Lei-stungsbereich von 70 bis 105 % regelbar
Änderung der Gas-zusammensetzung	unempfindlich gegen jegliche Veränderungen dieser Art	führt zu einer Verän-derung des Betriebs-bereiches, abhängig u.a. von Gasdichte
Auswirkungen des Arbeitsprinzips	oszillierender Prozess; Schwingungen, Pulsationen - aufwendige Funda-mente; zusätzliche Massenkräfte größerer Platz-bedarf; Schmierung des Kolbens Öl im Arbeitsmittel	rotierender Prozess, keine freien Massen-kräfte, kleinere Rei-bungsverluste; höhere Betriebsdrehzahlen, ölfreies Arbeitsmittel
Wartungs- und Reparaturaufwand	Ersatzteilhaltung ca. 20 % vom Auftragswert	Ersatzteilhaltung ca. 10 % vom Auftragswert
Leckageverluste	geringe Leckagen	Verluste nehmen mit höheren Gasdrücken und geringer werden- der Gasdichte zu
Verfügbarkeit	geringer, aber stark vom Antrieb abhängig	höher, aber stark vom Antrieb abhängig

Abb. 10: Technische Kriterien zur Verdichter-auswahl

Betriebspunkte. Der niedrige Leistungsbedarf bei hohen Wirkungsgraden und die hohe Zuverlässigkeit sind weitere Vorteile.

Doch es werden auch neue Grundkonstruktionen für die Turboverdichter entwickelt und es ist abzuwarten, wie sich die Einsatzmöglichkeiten der Getriebeturboverdichter verändern.

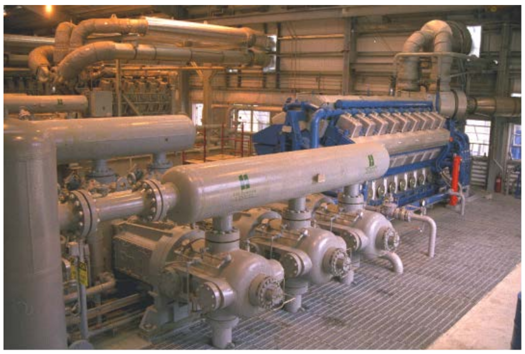

Abb.11: Kolbenverdichteranlage auf einem Untergrundgasspeicher (Moss Bluff / Texas / USA)

Des Weiteren unterteilen sich die Hubkolbenverdichter in Langsam- und Schnell-Läufer. Der Unterschied zwischen beiden Maschinentypen ist die Drehzahl und damit die Beanspruchung der bewegten Teile. Die geringere Belastung, vor allem der Ventile, wird als Vorteil für den Langsam-Läufer gewertet, da der Verschleiß nicht so hoch ist, wie bei doppelter oder dreifacher Drehzahl.

Die Verschleißfestigkeit der bewegten Teile eines Hubkolbenverdichters hat sich durch die Weiterentwicklung der Werkstoffe jedoch erhöht. Deshalb ist es fraglich, ob es bei Schnell-Läufern wirklich zu einem größeren Wartungsaufwand und niedrigeren Standzeiten kommt. Eine Aussage lässt sich treffen, wenn die Reparatur- und Wartungskosten und die Verfügbarkeit an konkreten Objekten miteinander verglichen werden.

Um das Risiko für den Betreiber der Anlage zu minimieren, der keine Informationen über den Verschleiß und den Wartungsaufwand der Maschine besitzt, können langfristige Wartungsverträge mit dem Verdichterhersteller abgeschlossen werden. Eine Bewertung der unterschiedlichen Maschinentypen ist anhand der angebotenen Wartungsverträge möglich.

Die höhere Drehzahl der Schnell-Läufer besitzt bei der Wahl des Antriebes einen wesentlichen Vorteil. Da die Antriebe meist mit höheren Drehzahlen laufen (Gasmotor), lassen sich die Verdichter ohne technischen Aufwand direkt an die Antriebsmaschine koppeln.

Der Vergleich zeigt, dass es nicht den Standardverdichtertyp für die Speicherung von Erdgas gibt, sondern die Auswahl von projektspezifischen Parametern, wie Betriebspunkten, Regelungsbereich, Volumenstrom usw. und der Betriebsart abhängt.

Des Weiteren sind alle anderen Kombinationen von Verdichtern und Antrieben theoretisch möglich.

Eine Bewertung im Zusammenhang mit dem Antrieb zeigen die Bilder 12 und 13.

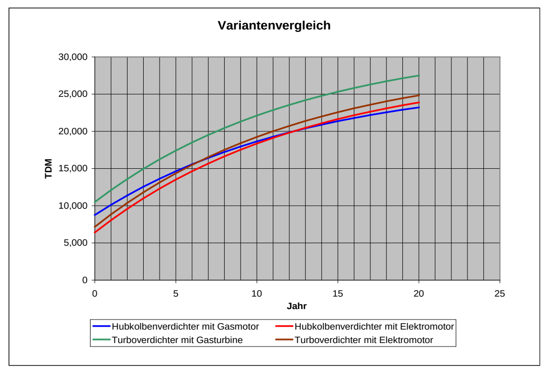

Abb. 12: Variantenvergleich (Turbo und Kolben)

Kriterium	E-Motor	Gasturbine	Gasmotor
Energiekosten	Hoch	niedrig	niedrig
Energieart	elektrische Energie	Erdgas	Erdgas
Drehzahl [min^{-1}],	< 300 bis 6.000	5.000 bis 20.000	750 bis 1.800
Wirkungsgrad	0,94 bis 0,98	0,30 bis 0,37	bis 0,42
Drehzahlregelung	Frequenzumrichter	vorhanden	vorhanden
Verfügbarkeit	sehr hoch	hoch	niedriger
Investitionskosten	niedrig	hoch	niedrig bis mittel
Wartungsaufwand	sehr gering	höher	sehr hoch
Genehmigungsverf.	einfach	nach BImSchG	nach BImSchG
Platzbedarf	niedrig	hoch	hoch
Energieversorgung	abhängig von EVU	unabhängig von EVU	

Abb. 13: Variantenvergleich (Turbine, Gasmotor und E-Motor als Antrieb)

3.4 Parameter für Maschinenaus-wahl

Die Auswahl der Verdichter wird durch eine Vielzahl von Parametern definiert. Der spätere Betreiber legt die Ausgangsparameter fest. Sie bilden die Grundlage für die gesamte Auswahl und geben in Form des Designpunktes die Verdichtungsarbeit und somit die Leistung des Verdichters vor. Von der Herstellerseite aus werden die technischen und wirtschaftlichen Parameter für die jeweilige Maschine vorgegeben und die rechtlichen Grundlagen werden in Verbindung mit den Umweltbedingungen durch die externen Parameter definiert.

Abb.14: Kompressorenanlage auf einem Speicher in West-Texas

Die Aufgabe des mit der Planung beauftragten Ingenieurs besteht darin, die verschiedenen Parameter zu ermitteln und zu bewerten, die einzelnen Anbieter von Verdichtern und möglichen Antriebseinheiten anzufragen, die daraus folgenden Angebote zu vergleichen und die technisch sinnvollste und kostengünstigste Verdichter-Antriebs-Kombination auszuwählen. Die Schwierigkeiten einer solchen Arbeit liegen zum einen in der Ermittlung und Bewertung der einzelnen Parameter.

Zum anderen ist ein Vergleich der Angebote trotz definierter Anfrageparameter ohne zusätzliche Anpassungsarbeiten in der Regel nicht möglich, denn es gibt immer technische Lösungen, die zwar einfacher und kostengünstiger sind, die aber von den Ausgangsparametern des Speicherbetreibers abweichen. In diesen Fällen muss geprüft werden, ob das Angebot bzw. die daraus folgenden Leistungsparameter für die erforderliche Speicheraufgabe ausreichen.

3.4.1 Ausgangsdaten einer Erdgas-speicheranlage

Der Speicherbetreiber bestimmt aus der Speichergeometrie und der geplanten Fahrweise des Verdichters die einzelnen Betriebspunkte. Der maximale Betriebspunkt ist der Designpunkt, nach dem der Verdichter dimensioniert wird. Er beinhaltet folgende Parameter:

- den Normvolumenstrom ($\dot{V}_{Nmax}$) in m³/h,
- den Saugdruck (p_1)
- den Enddruck (p_2) in bar.

Des Weiteren werden vom Betreiber die Parameter

- Saugtemperatur (t_1),
- max. Ausgangstemperatur (t_2) nach dem Kühler,
- Taupunkt
- die Gaszusammensetzung vorgegeben.

Nach diesen Parametern können die Verdichterhersteller die Verdichterleistung berechnen und ihre Maschinen für diese spezielle Verdichtungsaufgabe auslegen. Die anderen Betriebspunkte müssen durch eine adäquate Regelung fahrbar sein. Die Regelung des Verdichters muss vom Verdichterhersteller für jedes Projekt mit Hilfe von Leistungsdiagrammen neu berechnet und angepasst werden.

Aus der Gaszusammensetzung, dem jeweiligen Druck und der Gastemperatur lassen sich die Realgasfaktoren Z_1 - für den Ansaugzustand - und Z_2 - für den Endzustand - berechnen. Dafür liegt eine Vielzahl von Programmen vor, die auf verschiedenen Lösungsansätzen (z.B. Benedict-Webb-Rubin) beruhen. Mit Hilfe dieser Stoffwerteprogramme lässt sich außerdem noch der Isentropenexponent (κ) bestimmen, der für die Berechnung der Verdichterleistung benötigt wird.

Der Bauherr gibt weiterhin meistens die Zahl der zu installierenden Verdichtereinheiten vor. In der Regel wird pro Ausbaustufe ein Verdichter eingesetzt, da die Forderung nach einer redundanten Auslegung zumeist hinter die Forderung der Kostenminimierung zurücktritt.

Es besteht die Möglichkeit, dass zwei oder mehrere kleinere Verdichter einem großen vorgezogen werden, da die Redundanz eine höhere Priorität besitzt oder in der geforderten Leistungsklasse keine Verdichter verfügbar sind. Weitere Gründe wären eine nicht ausreichende Energieanbindung oder die

Antriebe entsprechen in dieser Größenordnung nicht den gesetzlichen Anforderungen bzw. können nicht zur Verfügung gestellt werden. Die Fahrweise - ferngesteuerter oder Vor-Ort-Betrieb - ist ebenfalls ausschlaggebend für die Auswahl bzw. die Auslegung des Verdichters und wird vorgegeben. Außerdem ist im Endausbau die Errichtung eines Grundlastverdichters zur Abdeckung von addierten Leistungen angebracht.

3.4.2 Technische und wirtschaftliche Parameter des Verdichters

Auf Basis der im Designpunkt festgelegten Daten berechnet der Verdichterhersteller die maximale Verdichtungsleistung und dimensioniert auf Grundlage seiner Baureihen die Arbeitsmaschine. Daraus ergeben sich weitere, für die Auswahl wichtige technische Parameter, wie

- den Liefergrad λ[1],
- den volumetrischen Wirkungsgrad η_s[2]
- den mechanischen Wirkungsgrad η_m
- die Drehzahl n
- die Druckverluste Δp_1 und Δp_2
- die Stufenzahl z_{St}
- die Kurbelzahl z_K[1]
- die Schallemissionen L_E und
- die Abmessungen des Verdichters (H, B, L)

[1] nur für Hubkolbenverdichter
[2] nur für Turboverdichter

Der Liefergrad, der volumetrische Wirkungsgrad bzw. bei Turboverdichtern der isentrope Wirkungsgrad und die Druckverluste werden zur Berechnung der im Designpunkt verlangten maximalen Verdichterleistung benötigt. Mittels des mechanischen Wirkungsgrades lässt sich die Kupplungsleistung berechnen, die wiederum in Verbindung mit der Drehzahl die Auswahl des Antriebes bestimmt.

Die Stufenzahl und bei Hubkolbenverdichtern die Kurbelzahl bestimmen die Bauart und Größe des Verdichters, woraus sich die Maße ergeben, die die Größe und damit die Kosten der Verdichterhalle bestimmen. Die Schallemissionen werden zur Einschätzung eventuell nötiger Schutzmaßnahmen benötigt.

Neben den technischen Parametern liefern die Hersteller zusätzlich wirtschaftliche Daten, die die Grundlage für den Vergleich der verschiedenen Varianten bilden. Zu diesen wirtschaftlichen Parametern gehören:

- die Investitionskosten für den Verdichter und dessen Nebenanlagen
- die Aufwendungen für Montage und Inbetriebnahme
- die Ersatzteilkosten
- die Wartungskosten
- der Verbrauch von Schmiermitteln und
- die Leckgasmengen.

Auf die wirtschaftliche Betrachtung haben die Investitionskosten zwar einen großen Einfluss, stellen aber nur einen Teil der Aufwendungen dar. Beim Einsatz von Verdichtern kommen noch die Wartungskosten und die Kosten für den Verbrauch von Betriebsmitteln hinzu. Um die Wartungskosten schon im Vorfeld abschätzen zu können, werden Wartungsverträge mit den Herstellern abgeschlossen. Somit wird zwar das Risiko auf den Hersteller übertragen, dieser erhält aber auch Informationen über seine Maschine und kann aus den Ergebnissen Rückschlüsse für die Konstruktion ziehen.

Einen beträchtlichen Kostenblock bilden die Energiekosten.

Wichtig für den wirtschaftlichen Vergleich ist die Abgrenzung des Liefer- und Leistungsumfanges, damit den einzelnen Varianten vergleichbare Basis zu Grunde liegt. Doch gerade in diesem Punkt liegen bei der Verdichterauswahl besondere Probleme, den schon geringe Abweichungen vom Designpunkt können, wenn auf bereits eingesetzte Anlagen zurückgegriffen werden kann, auf Grund von Lernkurveneffekten zu einer erheblichen Reduzierung der Investitionskosten führen. Es stellt sich somit für jedes Projekt neu die Frage, inwieweit alternative Varianten betrachtet werden.

Ein weiteres Problem besteht darin, dass nicht alle Parameter bewertet werden können. So ist die Verfügbarkeit der Maschine ein grundlegender Parameter für die Auswahl, lässt sich aber nicht direkt in eine Wirtschaftlichkeitsbetrachtung einbeziehen. Solche Parameter müssen über die Methode der gewichteten Faktoren in den wirtschaftlichen Vergleich integriert werden.

3.4.3 Externe Parameter für die Gesamt-Anlage

Neben den vom Betreiber der Verdichter bzw. vom Hersteller des Verdichters bestimmten Parametern fließen auf Grund des gesetzlichen Rahmens und anderer Umweltbedingungen weitere Parameter in die Auswahlbetrachtung ein.

Zu den gesetzlichen Rahmenbedingungen gehören unter anderem die Anzeigepflicht der gesamten Anlage, die Auflagen bzw. Genehmigungsverfahren auf der Grundlage des Bundes-Immissionsschutzgesetzes (BimSchG). Letzteres gibt in seinen Verordnungen (TA Lärm) die Grenzwerte für Lärmemissionen vor.

3.4.4 Wesentliche Parameter der Verdichterantriebe

Die Auswahl des Antriebes hängt, wie die des Verdichters, von den verschiedensten Parametern ab. Ausgangspunkt bilden wiederum die Anforderungen des Betreibers.

3.4.4.1 Ausgangsparameter für den Antrieb aus Sicht des Investors

Die entscheidenden Parameter neben der indirekten Leistungsvorgabe sind die Energiekosten. Die Kosten für Erdgas ergeben sich in den meisten Fällen aus internen Verrechnungssätzen, da der Speicherbetreiber selbst oft Gasversorger ist. Beim Bezug von Elektroenergie muss der Speicherbetreiber mit dem jeweiligen Stromversorger verhandeln.

Die Entscheidung erfolgt häufig zu Gunsten der Elektroenergie, da die Stromlieferanten gewillt sind, langfristige Lieferverträge abzuschliessen.
Die damit verbundenen Preise fordern dies heraus.

3.4.4.2 Technische und wirtschaftliche Parameter des Antriebsherstellers

Aus der Berechnung der Verdichterleistung-- ergibt sich die Kupplungsleistung. Auf der Basis dieser Leistung wird unter Beachtung von Anfahrmomenten der Verdichterantrieb dimen-sioniert.

Von Seiten des Antriebsherstellers ergeben sich auf Grundlage der Leistungsberechnung folgende technische Parameter:

- der Wirkungsgrad η
- die Drehzahl n
- die Maße (L, H, B) und
- die Schallemissionen L_E.

Mit der Wahl des Verdichters ist auch die Drehzahl festgelegt. Für die Antriebsauswahl lässt sich daraus ableiten, ob eine direkte Kupplung des Antriebes mit dem Verdichter erfolgen kann oder ob ein Getriebe zwischen-geschaltet werden muss, das wiederum mit zusätzlichen Aufwendungen verbunden ist.

Die Größe der Antriebe bildet zusammen mit den Maßen der Verdichter die Grundlage für die Dimensionierung der Verdichterhalle. Die Gasmotoren und -turbinen sind zumeist größer als die elektrischen Antriebe. Das führt zu höheren Aufwendungen beim Bau der Verdichterhalle. Ein ähnliches Verhältnis ergibt sich bei der Betrachtung der Schallemissionen. Bei den Verdichtern und E-Motoren sind sie bedeutend niedriger als bei gasgetriebenen Antrieben, d.h. ausschlaggebend für die Schallschutzmaßnahmen sind beim Einsatz von Gasmotoren oder Gasturbinen diese selbst, während bei der Verwendung von elektrischen Antrieben der Verdichter die Hauptemissionsquelle ist.

Für gasgetriebene Antriebe ergeben sich außerdem folgende zusätzliche Parameter:

- der erforderliche Gasdruck
- der Abgasmassenstrom
- die Abgastemperatur und
- die Emissionswerte für CO, NO_X und NMHC[1].

[1] Nicht-Methan-Kohlenwasserstoffe

Die Abgastemperatur der Gasmotoren bzw. -turbinen sind die Basis für die Dimensionierung der gesamten Abgasanlage und die Grundlage für mögliche Genehmigungsverfahren nach BimSchG.

Die Emissionswerte dürfen die vorgegebenen Grenzwerte nach der „Technischen Anleitung zur Reinhaltung der Luft“ (TA Luft) nicht überschreiten. Für gasgetriebene Maschinen ist außerdem die Anbindung an das vorhandene Gasnetz von Bedeutung (erforderlicher Gasdruck).

Bei elektrischen Antrieben entfällt zwar die gesamte Abgasanlage, die gasseitige Anbindung und die Genehmigungspflicht nach BimSchG, dafür benötigt der Motor einen Anschluss an das Stromnetz des jeweiligen Elektroenergieversorgers. Von der Herstellerseite aus ergeben sich folgende zusätzliche Parameter:

- der Leistungsfaktor
- der Nennstrom und
- der Anlaufstrom.

3.4.4.3 Externe Parameter, die den Antrieb bestimmen

Die gesetzlichen Anforderungen gerade an die Gasmotoren und -turbinen haben einen grundlegenden Einfluss auf die Auswahlentscheidung, denn für Gasturbinen generell und für erdgasbetriebene Gasmotoren ab einer Feuerungswärmeleistung von 1 MW ist ein Genehmigungsverfahren nach BimSchG erforderlich. Es kann außerdem durchaus möglich sein, dass die hohe Umweltbelastung am Standort des Speichers keine weiteren Kraftanlagen mit innerer Verbrennung mehr zulassen.
Ausschlaggebend für den Einsatz von E-Motoren ist das vorhandene Netz, das folgende Parameter vorgibt:

- die Anschlussleistung
- die Nennspannung und
- die Kurzschlussleistung.

Weiterhin ist zu prüfen, ob das Netz den Anforderungen des E-Motors entspricht, d.h. verkraftet das E-Netz die erforderlichen Anlaufströme und

Spannungseinbrüche. Sind die Anforderungen des Motors nicht zu realisieren,
muss durch zusätzliche Maßnahmen, wie Frequenzumrichter oder ähnlichen Anlagen der Anlaufstrom reduziert werden.

4 Maschinenvariantenvergleich

Auf der Grundlage von "Technischen Anfragespezifikationen zur Abgabe eines Richtpreises" werden in der Regel typische technische Verdichtervarianten von verschiedenen Herstellern angefragt, die sich wie folgt gliedern:

A. 1-stufiger, 4-kurbeliger Boxerverdichter mit Elektromotorenantrieb
B. 1-stufiger, 4-kurbeliger Boxerverdichter mit Gasmotorenantrieb
C. 1-gehäusiger Topfverdichter (2-stufig) mit Gasturbinenantrieb

Bei diesen Angeboten (pro Variante 3 bis 5 Hersteller) handelt es sich um technische Angebote mit einer Reihe von offenen Fragen, die noch nicht beantwortet werden konnen. Dies sind:

- Betriebspunkten mit entsprechenden Leistungsangaben
- Wartungs- und Servicekosten
- Wirkungsgraden

Demzufolge konnen Betriebskostenvergleiche nur als angenommene, aus unserer Erfahrung und langjähriger Kenntnis kommende, Daten zugrunde gelegt werden kennen.

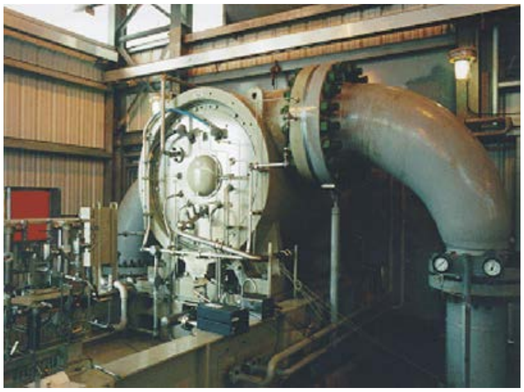

Abb.15: Typische Gasturbine als Antrieb für einen Turboverdichter

Der Investkosten-Vergleich beinhaltet nicht die Errichtung eines evtl. notwendigen Gebäudes, bei unserem Vergleich wird von einem vorhandenen genügend großen Gebäude ausgegangen.

Abb. 16: Typische e-motorische angetriebene Kolbenkompressorenanlage (Kraak/D)

Die vorliegende Betrachtung wird nur für die Arbeitsmaschine mit Antrieb durchgeführt. Die Betriebsmittel wie Wasser, Öl, Steuerluft und Stickstoff wurden nicht betrachtet, da sie ebenso wie die zusätzlich benötigte Elektroenergie für den Normalbetrieb als neutral angesehen werden können. Sie werden somit in den Betriebskosten-vergleich nicht mit einbezogen.

Technische Unterschiede

A. Drehzahl: 372 rpm
B. Drehzahl: 1.000 rpm
C. Drehzahl; 15.500 rpm

A./B./C : ohne Getriebe

A. 35 - 55 bar auf 60 - 75 bar bei 30.000 Nm3/h
35 - 55 bar auf 75 - 100 bar bei 40.000 Nm3/h

B. 35 bar auf 100 bar bei 33.000 Nm3/h
55 bar auf 100 bar bei 53.000 Nm3/h

C. 55 bar auf 90 bar bei 22.000 Nm3/h
35 bar auf 55 bar bei 14.000 Nm3/h
55 bar auf 90 bar bei 46.000 Nm3/h

4.1 Investkosten

Kosten für Kompressor + Antriebsmaschine:

A. Richtpreis: 1.956.500,00 EURO
B. Richtpreis: 1.500.000,00 EURO
C. Richtpreis: 3.350.000,00 EURO

Kosten für E-Installation:

A. 600.000,00 EURO
B. 600.000,00 EURO
C. 610.000,00 EURO

Fundamente:

A. 18.000,00 EURO
B. 10.000,00 EURO
C. 12.000,00 EURO

Brenngasversorgung:

A. entfällt
B. 45.000,00 EURO
C. 60.000,00 EURO

Kühlwassersystem:

A. 30.000,00 EURO
B. 12.000,00 EURO
C. entfällt

Gas/Luftkühler:

A. enthalten in Maschinenkosten
B. enthalten in Maschinenkosten
C. 65.000,00 EURO

Steuerluftsystem:

A. 47.000,00 EURO
B. entfällt
C. entfällt

Stickstofferzeugungsanlage:

A. entfällt
B. entfällt
C. 75.000,00 EURO

Schallschutzmassnahmen:

A. 75.000,00 EURO
B. 75.000,00 EURO
C. 75.000,00 EURO

Fundamente, etc.:

A. 9.500,00 EURO
B. 9.500,00 EURO
C. 9.500,00 EURO

Montage:

A. 51.000,00 EURO
B. 34.000,00 EURO
C. 50.000,00 EURO

Stationspiping, icl. Entspannungssystem:

A. 250.000,00 EURO
B. 379.000,00 EURO
C. 334.000,00 EURO

Baustelleneinrichtung:

A. 50.000,00 EURO
B. 50.000,00 EURO
C. 50.000,00 EURO

Transport, etc.:

A. ist in den Angeboten enthalten
B. 50.000,00 EURO
C. 100.000,00 EURO

Gaswarn-/Brandwarnanlage:

A. 110.000,00 EURO
B. 110.000,00 EURO
C. 110.000,00 EURO

Feuerlöschsystem:

A. entfällt
B. entfällt
C. 80.000,00 EURO

Kosten für Abnahmen, etc.:

A. 50.000,00 EURO
B. 50.000,00 EURO
C. 50.000,00 EURO

Engineeringkosten:

A. 280.000,00 EURO
B. 278.000,00 EURO
C. 391.000,00 EURO

GESAMTKOSTEN FÜR DIE KOMPRESSORENANLAGE, incl. NEBENANLAGEN

A. 4.065.000,00 EURO
B. 3.712.000,00 EURO
C. 5.850.000,00 EURO

4.2 Betriebskostenvergleich

Durchschnittliche Antriebsleistung:

A. 1.500 kW
B. 1.500 kW
C. 1.600 kW

Wirkungsgrad:

A. 0,9
B. 0,45
C. 0,3

Betriebskosten pro Betriebsstunde:

A. 208,00 EUR/ h
(Basis: 0,12 EUR/kWh)
B. 33,50 EUR/h
(Basis: 0,01 EUR/kWh)
C. 53,50 EUR/h
(Basis: 0.01 EUR/kWh)

Betriebskosten pro Jahr:

A. 938.000,00 EUR/a
B. 150.000,00 EUR/a
B. 241.000,00 EUR/a

Service- und Wartungskosten:

A. 210.000,00 EUR/a
B. 260.000,00 EUR/a
C. 167.000,00 EUR/a

GESAMTKOSTEN FÜR DEN BETRIEB DER ANLAGE (ENERGIE, SERVICE, WARTUNG):

A. 1.148.000,00 EUR/a
B. 410.000,00 EUR/a
C. 408.000,00 EUR/a

Diese Ergebnisse zeigen in erstaunlicher Deutlichkeit den Trend für und gegen die betrachteten Kompressorentechnologien und die unterschiedlichen Antriebstechnologien, es muss natürlich auch bedacht werden, dass die Richtigkeit dieser Untersuchung nur unter den untersuchten Randbedingungen richtig ist:

- Untergrundspeicher (Aquifer)
- Durchschnittliche Einspeisemenge: 30.000 Nm3/h
- Einspeisedauer: 187,5 Tage = 4.500 Betriebsstunden
- Aktivgas: 135.000.000 Nm3
- Saugdruck: 35 bar (a)
- Enddruck: 100 bar (a)

4.3 Life-Cycle-Kosten

Basierend auf den Investitionskosten und den Kosten für Energie, Wartung und Service, einem üblichen Zinssatz, Inflationsrate, etc. (Discount rate = 10 %, was z.Zt. sehr konservativ ist), einer üblichen und realistischen Preissteigerung von 3 % pro Jahr und einer angenommenen Lebensdauer für die Kompressorenanlage von z.B. 10 Jahren lassen sich die Life-Cycle-Costs wie folgt ermitteln. Das Ergebnis zeigt sehr deutlich, dass der von uns näher untersuchte Fall, eindeutige Vorteile für die gas-motorisch getriebene Kompressoren-anlage gegenüber dem Turboverdichter und noch deutlicher gegenüber dem e-motorisch getrie-benen Kolbenverdichter ausweist.

Year	Oper. Costs Reciprocating Engine driven 3 % esc. (EURO's)	Oper. Costs Reciprocating E-Motor driven 3 % esc. (EURO's)	Difference (EURO's)	Discount 10 %	NPV savings (EURO's)
1	3.712.000,00	4.065.000,00	353.000,00	1,000	353.000,00
1	410.000,00	1.148.000,00	738.000,00	1,000	738.000,00
2	422.300,00	1.182.440,00	760.140,00	0,909	690.967,00
3	434.969,00	1.217.913,00	782.944,00	0,826	646.712,00
4	448.018,00	1.254.451,00	806.433,00	0,752	606.438,00
5	461.458,00	1.292.084,00	830.626,00	0,683	567.317,00
6	475.302,00	1.330.847,00	855.545,00	0,621	531.293,00
7	489.561,00	1.370.772,00	881.211,00	0,564	497.003,00
8	504.248,00	1.411.895,00	907.647,00	0,513	465.622,00
9	519.375,00	1.454.252,00	934.877,00	0,467	436.588,00
10	534.957,00	1.497.880,00	962.923,00	0,424	408.279,00

NPV of the savings over a 10 year period (net present value) **5.941.219,00 EURO's**

The life cycle costs for an engine driven reciprocating compressor are less than for an e-motor driven reciprocating compressor (version B is cheaper than version A)

Abb. 17: Life cycle cost savings in NPV während einer Zeit von 10 Jahren (Gasmotor/Kolbenverdichter E-Motor/Kolbenverdichter)

Year	Oper. Costs Centrifugal 3 % esc. (EURO's)	Oper. Costs Reciprocating E-Motor driven 3 % esc. (EURO's)	Difference (EURO's)	Discount 10 %	NPV savings (EURO's)
1	5.850.000,00	4.065.000,00	1.785.000,00 (plus)	1,000	1.785.000,00 (plus)
1	408.000,00	1.148.000,00	740.000,00 (minus)	1,000	740.000,00 (minus)
2	420.240,00	1.182.440,00	762.200,00 (minus)	0,909	692.840,00 (minus)
3	432.847.00	1.217.913,00	785.066,00 (minus)	0,826	648.465,00 (minus)
4	445.833,00	1.254.451,00	808.618,00 (minus)	0,752	608.081,00 (minus)
5	459.208,00	1.292.084,00	832.876,00 (minus)	0,683	568.854,00 (minus)
6	472.984,00	1.330.847,00	857.863,00 (minus)	0,621	532.733,00 (minus)
7	487.173,00	1.370.772,00	883.599,00 (minus)	0,564	498.349,00 (minus)
8	501.788,00	1.411.895,00	910.107,00 (minus)	0,513	466.885,00 (minus)
9	516.842,00	1.454.252,00	937.410,00 (minus)	0,467	437.770,00 (minus)
10	532.347,00	1.497.880,00	965.533,00 (minus)	0,424	409.386,00 (minus)

NPV of the savings over a 10 year period (net present value) **3.818.363,00 EURO's**

The life cycle costs for a turbine driven centrifugal compressor are less than for an e-motor driven reciprocating compressor (version C is cheaper than version A)

Abb. 18: Life cycle cost savings in NPV während einer Zeit von 10 Jahren (Turbine/Turboverdichter E-Motor/Kolbenverdichter)

Year	Oper. Costs Centrifugal 3 % esc. (EURO's)	Oper. Costs Reciprocating Engine driven 3 % esc. (EURO's)	Difference (EURO's)	Discount 10 %	NPV savings (EURO's)
1	5.850.000,00	3.712.000,00	2.138.000,00 (plus)	1,000	2.138.000,00 (plus)
1	408.000,00	410.000,00	2.000,00 (minus)	1,000	2.000,00 (minus)
2	420.240,00	422.300,00	2.060,00 (minus)	0,909	1.873,00 (minus)
3	432.847.00	434.969,00	2.122,00 (minus)	0,826	1.752,00 (minus)
4	445.833,00	448.018,00	2.185,00 (minus)	0,752	1.643,00 (minus)
5	459.208,00	461.458,00	2.250,00 (minus)	0,683	1.537,00 (minus)
6	472.984,00	475.302,00	2.318,00 (minus)	0,621	1.439,00 (minus)
7	487.173,00	489.561,00	2.388,00 (minus)	0,564	1.347,00 (minus)
8	501.788,00	504.248,00	2.460,00 (minus)	0,513	1.262,00 (minus)
9	516.842,00	519.375,00	2.533,00 (minus)	0,467	1.183,00 (minus)
10	532.347,00	534.957,00	2.610,00 (minus)	0,424	1.107,00 (minus)

NPV of the savings over a 10 year period (net present value) **2.122.857,00 EURO's**

The life cycle costs for an engine driven reciprocating compressor are less than for a turbine driven centrifugal compressor (version B is cheaper than version C)

Abb. 19: Life cycle cost savings in NPV während einer Zeit von 10 Jahren (Turbine/Turboverdichter Gasmotor/Kolbenverdichter)

5 Zusammenfassung

Das Untersuchungsbeispiel zeigt einige Fakten auf, die dem Kolbenkompressor einen starken Wettbewerbsvorteil gegenüber gleich großen Turboverdichtern verschaffen.

Eine wichtige Erkenntnis, die aus dieser Arbeit gezogen werden kann, ist die, dass die Ökonomie des Einsatzes von Turboverdichtern drastisch zurückgeht, sobald sie nicht mehr unter Norm-Bedingungen eingesetzt werden oder der Betriebspunkt häufig vom Auslegungspunkt abweicht. Eine weitere wichtige Erkenntnis ist, dass die gesamten Investitionskosten einer Turboverdichteranlage beachtlich höher liegen als die Einzelkosten für die Turbine und den Verdichter zusammen. Dies liegt vor allem an den teuren Regelsystemen und den hohen Installationskosten. Außerdem lässt sich sagen, dass die Konkurrenzfähigkeit von Kolbenkompressorenanlagen steigt, je höher das Verdichtungsverhältnis ist. Wenn es sich jedoch um niedrige Verdichtungsverhältnisse handelt, wie z.B. bei Pipelineanwendungen, kann der Kolbenkompressor nur dann wirtschaftlich eingesetzt werden, wenn sich die Betriebszustände häufig ändern.

Die Abbildungen 20, 21, 22 zeigen sehr deutlich die Abhängigkeit der Investitionskosten-Vorteile des Kolbenverdichters gegenüber dem Turbo-verdichter, die bei relativ kleinen Leistungen eindeutiger sind als bei grösseren.

Die Unterschiede/Vorteile werden sicherlich noch deutlicher bzgl. der Life-Cycle-Costs, wie wir an unserem Fallbeisspiel unter 4.3 belegen können. Hier verschiebt sich auch häufig der optimale Einsatz zugunsten des Kolben-verdichters in Richtung grösserer Leistungen.

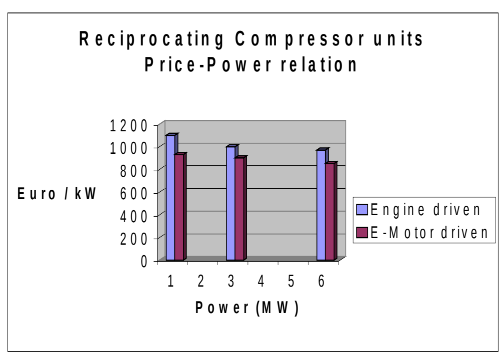

Abb.20:: Investitionskosten von Kolbenverdichteranlagen bei verschiedenen Leistungsklassen

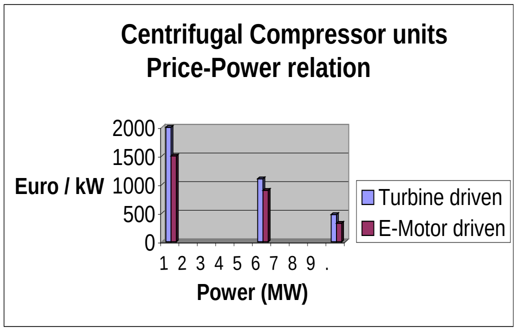

Abb.21:: Investitionskosten von Turboverdichteranlagen bei verschiedenen Leistungsklassen

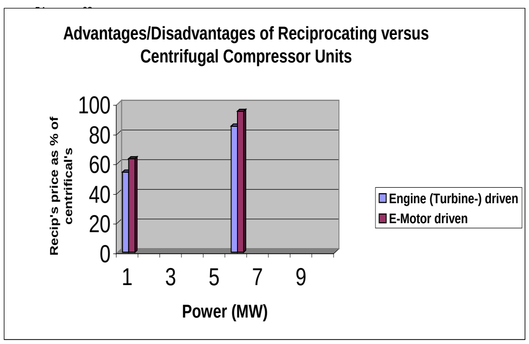

Abb.22: Vor- und Nachteile von Kolbenverdichter- und Turboverdichteranlagen bei verschiedenen Leistungsklassen unter dem Gesichtspunkt Investitionspreis

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Current Life Cycle Costs also / or Especially for High Speed Piston Compressors

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability May 17th / 18th, 2001, The Hague

Abstract:

Acting as a market leader for the manufacture of reciprocating compressors in Germany and as a global player worldwide the NEA-Group has also increased activities in high compression technology starting in the middle of 2000. The compressors in high pressure applications which are employed up to max. 3500 bar, with drive lines that exceed quite often 10.000 kW, are usually the vital centre of the manufacturing units driven by them. Therefore highest reliability and long lifetime are an absolute must for such machines. Propositions, how technological progress in life cycle costs can be achieved also with such machines are made and discussed. Problems concerning the use of modern stocks of assembly units, diagnostic means, improved lubrication systems and improvements in the cylinder area which increase the lifetime are considered in detail.

1 Einleitung

Bei den Höchstdruckkompressoren, in Fachkreisen oft auch einfach Hyper genannt, sind wie kaum bei einer anderen Maschinenkategorie höchste Verfügbarkeit gefragt, denn diese Maschinen bestimmen dominierend den Prozess der Kunststoffherstellung (Polyethylen). Wenn diese Maschinen abgeschaltet werden müssen – aus welchem Grund auch immer – steht die gesamte Prozesskette und die Verluste gehen schnell in sechsstellige DM-Beträge. Deshalb muss hier die Zuverlässigkeit die höchste Priorität bilden. Solche Gesichtspunkte sollten auch bereits bei Investitionsvorhaben berücksichtigt werden. Qualität kostet Geld und ein etwas höherer Preis kommt sehr schnell mit der Betriebszeit wieder in positive Regionen. Derartige Kompressoren beinhalten außerdem, trotz ihrer wahrlich nicht geringen Anschaffungskosten, nur etwa 5 % der Kosten, die eine solche Maschine über ihre gesamte Lebensdauer verursacht. Neben den reinen Anschaffungskosten, einschließlich derer zur Aufstellung, sind die Faktoren

- Energiekosten,
- Zuverlässigkeit und
- Wartungskosten (einschließlich Wartungsaufwand)

mit zu berücksichtigen.

Bei der Auswahl des Höchstdruckkompressors ist die Palette der Anbieter begrenzt. Aber wie bereits erwähnt, zahlt sich die eingekaufte Qualität beim Erzeugnis aus. Diesem Standpunkt verschließt sich vermutlich keiner der Anwender. Besonders die eigentlichen Betreiber wollen dies bei der Investitionsentscheidung berücksichtigt wissen. (In diesem Zusammenhang sei erwähnt, dass die Investitionsvorbereitung oft von Engineeringfirmen getätigt wird und die eigentlichen Betreiber noch nicht so stark dabei beteiligt sind, d.h. bei den Verhandlungen über Auswahl des jeweiligen Höchstdruckkompressors oft nicht mit am Tisch sitzen.) Leider werden ihre Belange in den Projekten mit unterschiedlichen Kostenverantwortungen allzu oft unzureichend vertreten und es wird erst mal nur an die kurzfristigen Kosten – den Preis der Anlage – gedacht.

Auf lange Sicht taugen aber vorrangig die Lebenszykluskosten als Entscheidungskriterium. Selbstverständlich verdienen auch die Energiekosten eine Betrachtung. Aber für die in jedem Falle vorgegebenen Grundparameter Saugdruck, Enddruck, Förderstrom und Eigenschaften des Fördermediums sind die Energiekosten wegen der unbeugbaren physikalischen Grundlagen keine Frage von größeren Abweichungen des Wirkungsgrades. Tatsächliche Differenzen des Wirkungsgrades bewegen sich im Rahmen von < 1 %, also oft nur in zehntel Prozenten. Diese daraus resultierenden Kostenunterschiede sind unter anderem nicht mit Kosten der Inspektionsintervalle, Stillstandszeiten während der Inspektion und Lebensdauer-Zyklen der wichtigsten Verschleißbaugruppen vergleichbar. Vor allem unvorhergesehene und damit ungeplante Ausfälle dieser Prozessmaschinen sollten deshalb mit Sicherheit ausgeschlossen werden. Eine Alternative dazu sind zielgerichtete Diagnosemittel, die im weiteren Verlauf des Vortrages noch angesprochen werden.

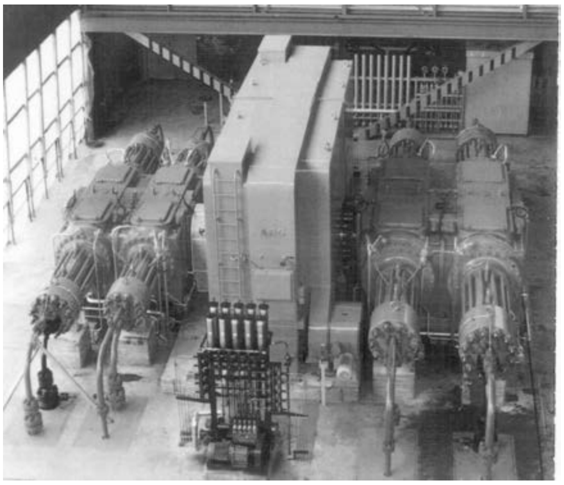

Bild 1 Gesamtansicht einer Anlage[1]

2 Technische Beschreibung der Maschine

Die Konstruktion der Hochdruck-Kolbenkompressoren in Polyäthylenanlagen unterscheidet sich wesentlich von der üblicher Kolbenkompressoren. Bedingt durch den anstehenden Enddruck von bis zu 3500 bar sind sowohl die Bauteile des Triebwerkes als auch die Ausführung des Verdichterteils den enormen Belastungen entsprechend angepasst. Zum besseren Verständnis sollen hier einige wesentliche Merkmale dieser Art Kompressoren näher erklärt werden:

So verlässt man hier die beim Boxerkompressor gängige Kurbelkröpfung, um eine Reduzierung der Beanspruchung von Pleuel und Lagern zu erreichen. Für jeweils zwei gegenüberliegende Zylinder existiert nur eine Kurbel mit einem Kreuzkopf und einer Pleuelstange. Dabei bewegen sich alle Bauteile in ein und derselben Achse. Bei dieser Anordnung wirkt nicht die Summe, sondern nur die Differenz der anstehenden Gaskräfte der gegenüberliegenden Plunger auf das Triebwerk. Ein Moment entsteht hier nicht. Ferner gewährleistet diese Konstruktion eine Umkehrung der Lastrichtung am Kreuzkopfbolzen, d.h. es wird ein Lastwechsel oder noch ausdrucksvoller ein Anlagewechsel erzielt.

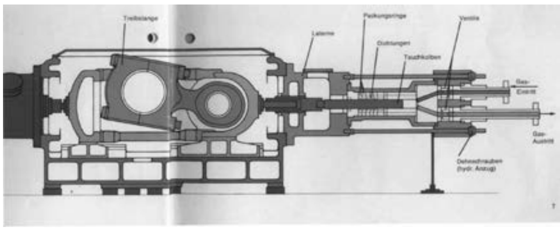

Bild 2 Schema-Zeichnung Kurbeltrieb[1]

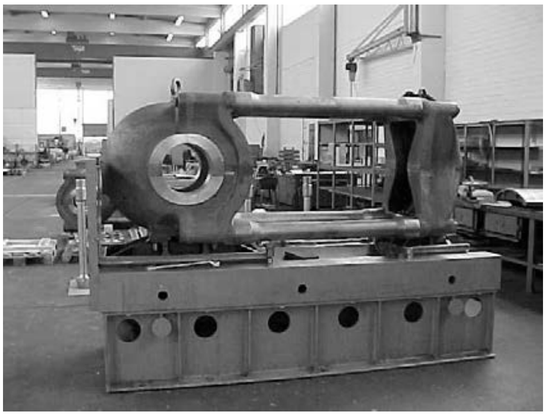

Bild 3 Kreuzkopf-Traverse

Wie Sie an dem hier gezeigten Bild erkennen, besteht der Kreuzkopf aus zwei Teilen, die mittels sogenannter Umführungsstangen miteinander verbunden sind. Der eigentliche Hauptkreuzkopf, der über den Kreuzkopfbolzen die Verbindung zum extrem kurzen Pleuel herstellt, überträgt über seinen großflächigen Gleitschuh die Normalkraft auf die Führung. Die Drehrichtung der Maschine ist immer so gewählt, dass die Normalkraft nach unten gerichtet ist. Der kürzere Gleitschuh der Traverse – das ist der dem Hauptkreuzkopf gegenüberliegende Teil des gesamten Gebildes – dient nur zur besseren Führung. Das Triebwerksgehäuse der GHH-Verdichter weist durch doppelt vorhandene Längswände und zusätzliche Verrippung ein hohes Trägheitsmoment und somit eine entsprechende Biegesteifigkeit auf. Der Hochdruckzylinder bzw. die Hochdruckzylinder – meistens sind die Maschinen wegen der großen Druckdifferenz zweistufig ausgeführt – und die dazugehörige Plungerdichtung zählen wohl zu den am stärksten belasteten Bauteilen. Die aus dem extrem hohen, pulsierenden Gasdruck resultierenden Belastungen der einzelnen Komponenten erfordern sowohl eine spezielle, bei üblichen Kolbenkompressoren nicht angewandte Art der Konstruktion als auch eine besonders sorgfältige Herstellung.

Grundsätzlich sind zwei Arten der Gasabdichtung bekannt:

- Ausführung des Kolbens mit Kolbenringen
- Plungerausführung mit Hochdruckpackung

Die weitaus größte Anzahl der von Neuman & Esser betreuten GHH-Höchsdruckkompressoren sind mit Plunger und Hochdruckpackung ausgerüstet. Hier verweisen wir auf die bei der ELENAC durchgeführten Umbau- und Optimierungsmaßnahmen, über die anlässlich des 4. Workshops Kolbenverdichter bei der Firma Kötter berichtet wurde[2]. Durch den Umbau auf Plungerausführung konnten die Standzeiten der Zylindereinheit erheblich verbessert werden.

Die Hochdruckpackung oder auch Plungerabdichtung ist das Kernstück der Zylindergruppe. Bei den hier herrschenden hohen Drücken können übliche Konstruktionen wie einteilige Kammerringe - das gleiche gilt für die Zylinder selbst – nicht mehr zur Anwendung kommen. Zur Vermeidung von Durchmessersprüngen und daraus resultierenden Spannungsspitzen werden die Gehäuse nicht als einteilige Kammern, sondern in Scheibenbauweise ausgeführt. Gehäuseringe und Zylinderbüchse sind je nach Druckbereich bis zu zweifach geschrumpft. Durch die Schrumpftechnik wird in den Bauteilen eine Druckvorspannung erzeugt und somit eine günstigere Spannungsverteilung über den gesamten Querschnitt erreicht. Auch die Zylinderköpfe sind entsprechend dieser Schrumpftechnik gebaut. Die hier eingebauten Arbeitsventile sind federbelastete Kegel. Saug- und Druckventile sind gleich, ein verkehrter Einbau wird jedoch durch eine Sicherung ausgeschlossen.

An die Oberflächenbearbeitung des Plungers werden natürlich höchste Anforderungen gestellt. Er besteht entweder aus Vergütungsstahl mit Hartmetallpanzerung oder aus massivem, gesinterten Hartmetall. Für einen sauberen Geradlauf sorgt die Plunger-Feinführung, die vorhandene Lauftoleranzen der Kreuzkopfführung kompensiert.

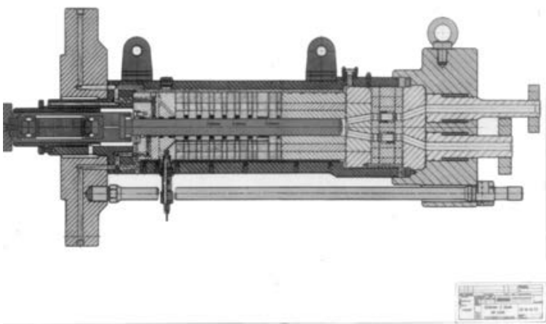

Bild 4 Zylinder / Packung[1]

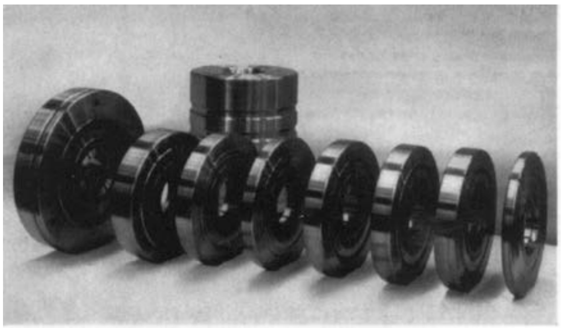

Bild 5 Packung[1]

3 Erweiterung des NEA-Produktbereiches

Wie sicher viele der hier interessiert zuhörenden Fachleute bereits in unseren NEA-NEWS[3] gelesen haben, hat die NEA-Group ab 15. Juni 2000 die Exklusivrechte mit Erwerb der gesamten technischen Dokumentation von MAN Turbomaschinen GHH Borsig für die GHH-Höchstdruckkolbenkompressoren erworben. Damit sind außer den vor diesem Termin ohnehin schon ausgeführten Wartungsarbeiten auch alle wesentlichen Ersatzteillieferungen als Inhouse-Fertigung möglich geworden. Die NEA-Group will diese Wartungs- und Inspektionsarbeiten kurzfristig weltweit mit einer gezielten Weiterentwicklung und Modernisierung der nun teilweise seit etwa 20 Jahren in Betrieb befindlichen Höchstdruckkompressoren verbinden. Die NEA-Group gesellt sich damit zu den anderen zwei Großen der Höchstdruckkolbenkompressoren-Hersteller Sulzer Burckhardt und Nuovo Pignone und bekundet mit dem gewachsenen Kompetenzbereich ihr umfassendes Kolbenkompressoren-Know-how.

Mit den über 50 bestens ausgebildeten und überaus erfahrenen NEAC-Servicetechnikern kann das NEA-Unternehmen exzellent organisierte Höchstdruck-Maschinenanlagen-Inspektionen anbieten, die in kürzester Stillstandszeit abgewickelt werden und somit geringste Kosten verursachen. Hinsichtlich der Weiterentwicklung und Modernisierung sollte nicht unerwähnt bleiben, dass in den letzten Jahren bei NEA eine eigene F+E-Gruppe mit FEM-Rechentechnik installiert wurde, die über Rahmenvereinbarungen eng mit den territorial quasi in der "Nachbarschaft" agierenden, bekannten und geschätzten Instituten der RWTH Aachen sowie der FEV Motorentechnik zusammenarbeitet. Mit Hilfe von Finite-Elemente-Analysen (FEA) können neue Strukturen hinsichtlich thermischer und mechanischer Belastungen Deformationen, Massenbelegung und Material untersucht und optimiert werden.

Ebenso widmet sich der Zentralbereich Technik der NEA-Group der Störungsfrüherkennung und Kostenminimierung im Lebenszyklus der Höchstdruckkompressoren, denn die Optimierung der auftretenden Lebenszykluskosten bei diesen Maschinen hat gezeigt, dass auch die Betreiber in der Frage der Verfügbarkeit und Verhinderung bzw. Früherkennung von System- oder Bauteilfehlern einen der dominierenden Faktoren für die Minimierung der Gesamtkosten sehen.

4 Analyse der Lebensdauer der wichtigsten Bauelemente

Im Laufe von vielen Produktionsjahren konnte das für die Wartung und Betriebssicherheit in den einzelnen Betreiberfirmen zuständige technische Personal einen reichen Erfahrungsschatz mit den Eigenheiten der Höchstdruckkompressoren sammeln. So wurden - wie bereits oben erwähnt - umfangreiche Schadensuntersuchungen und Statistiken mit daraus resultierenden Optimierungsmaßnahmen durchgeführt.

Im Durchschnitt ergeben sich für die wichtigsten Bauteile folgende Standzeiten:

- Kurbelwellen- und Kurbelzapfenlager:	7 bis 8 Jahre
- Kreuzkopfbolzenlager:	3 bis 4 Jahre
- Zylinder:	3 bis 5 Jahre
- Packungen:	3 bis 5 Jahre
- Plunger:	3 bis 5 Jahre
- Ölwechsel nach:	1 bis 4 Jahren

5 Beispiel einer Schadensbehebung im Jahre 2000

Fast zeitgleich mit der Übernahme der GHH-Lizenz bot sich der Firma Neuman & Esser die Gelegenheit, ihr Können und ihre Leistungsfähigkeit unter Beweis zu stellen. In der Polyäthylenanlage eines Betreibers von GHH-Höchstdruckkompressoren hatte sich ein gravierender Triebwerksschaden ereignet. Mehrere Brüche am geschlossenen Auge einer Pleuelstange eines dreikurbeligen, zweistufigen Kompressors hatten dazu geführt, dass das ungeführte Pleuel bzw. die Treibstange das Triebwerk zerstören konnte.

Hier einige technische Daten der Maschine:

Leistung:	5.600 kW
Enddruck:	3.157 bar
Anzahl der Stufen:	2
Liefermenge:	23.800 Nm^3/h

Bild 6 Beschädigtes Kurbelgehäuse

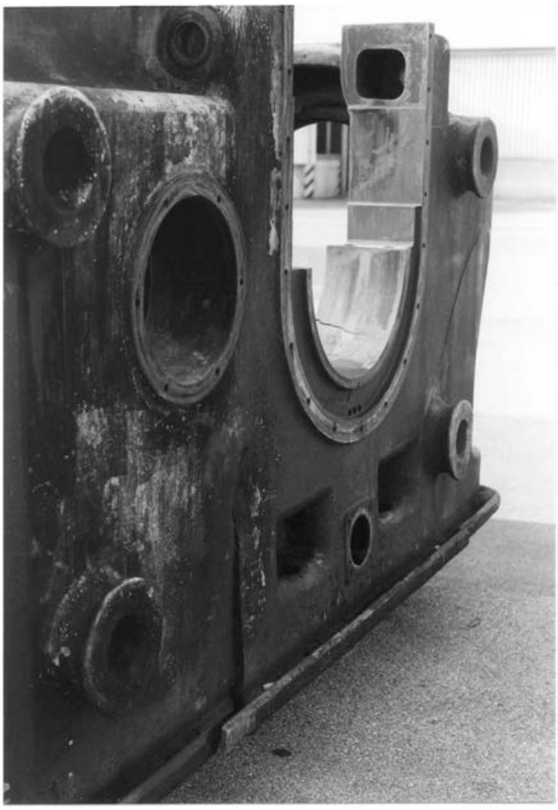

Bild 7 Gerissener Lagerstuhl

Durch die unkontrollierten Bewegungen des Pleuels waren regelrechte Löcher in das Triebwerksgehäuse geschlagen, so dass es komplett ersetzt werden musste. Die Kreuzköpfe konnten durch teilweises Auftragsschweißen wieder verwendet werden. Bei der Schadensaufnahme wurden durch eingehende Untersuchungen mehrere kleine Risse in der Oberfläche des kleinen Treibstangenkopfes festgestellt. Durch die hohe dynamische Belastung ist eine Fortpflanzung eines oder auch mehrerer Risse denkbar, was dann schließlich zum Bruch bzw. Aufreißen des Pleuelauges führte. Nach einem mittels FEM durchgeführten Festigkeitsnachweis des Pleuelauges wurde dieses soweit optimiert, dass die Bruchsicherheit um 6 % erhöht werden konnte.

Dies konnte durch folgende zwei Maßnahmen erreicht werden:

Die Wandstärke des Pleuelauges wurde unter Ausnutzung der vorhandenen Bearbeitungszugaben am Rohteil um 10 mm verstärkt. Zusätzlich wurde der gefährdete Bereich einem Shot Peening ausgesetzt. Hierbei handelt es sich um ein Kaltbearbeitungsverfahren, bei dem die Werkstückoberfläche mit einem kugelförmigen Strahlmittel behandelt wird. Durch die Verdichtung der Außenhaut erzeugt man eine hohe Druckvorspannung, die die im Betrieb auftretende Zugspannung reduziert. Natürlich hätte man die Optimierung der Pleuelstangen auch durch die Wahl eines höherfesten Werkstoffes vornehmen können. In diesem Falle waren jedoch aus Termingründen drei neue Pleuel bereits in dem herkömmlichen Material bestellt und vorgeschmiedet, ehe die Schadensuntersuchung abgeschlossen war.

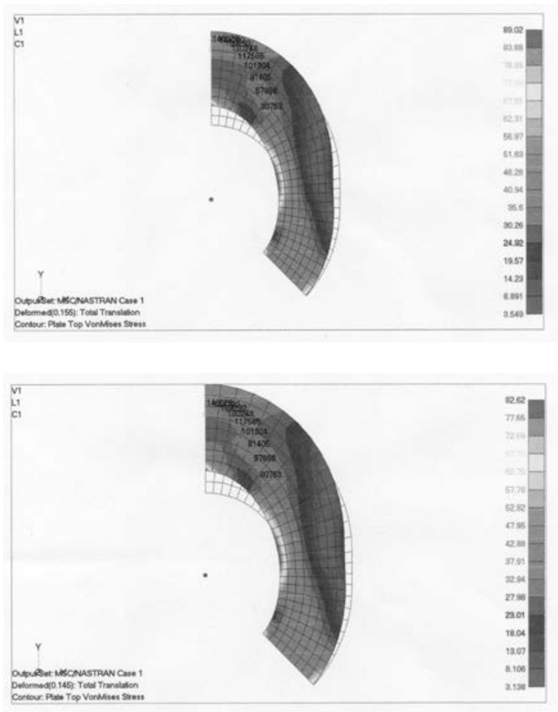

Bild 8 FEM-Pleuel

Außer einem neuen Kurbelgehäuse und den verstärkten Pleuelstangen wurde das Triebwerk erstmalig mit Fertiglagern ausgerüstet.

Zunächst noch ein paar Worte zur Bearbeitung des Kurbelgehäuses:

Nach dem Abguss des 34 Tonnen schweren Gehäuses wurde es in unserer Übach-Palenberger Fertigungsstätte zunächst bis auf eine Schnittzugabe von 3 mm vorgearbeitet, anschließend spannungsarm geglüht und im letzten Schritt auf Fertigmaß bearbeitet.
Da die Bearbeitung eines Werkstückes solchen Ausmaßes in unserem Werk nicht an der Tagesordnung ist, wurden über den üblichen Rahmen hinaus mit verschiedenen Abteilungen, unter Einbeziehung aller möglichen Gesichtspunkte, die einzelnen Arbeitsschritte festgelegt. Für das Ausspindeln jeder einzelnen Lagerstelle mit Hilfe eines Winkelkopfes wurde das Gehäuse hochkant auf dem Palettenfeld der Bearbeitungsmaschine in X- und Z-Richtung ausgerichtet und aufgespannt. Bei der späteren Bearbeitung der Zylinderanlageflächen sollte das Gehäuse vom Drehtisch des Bearbeitungszentrums aufgenommen werden, um eine beidseitige Bearbeitung in einer Aufspannung zu ermöglichen. Beim Ausrichten und dem hierbei durchgeführten Vermessen der Lagergasse stellte man aufgrund des Gehäuseüberhanges über den Drehtisch und aufgrund seines Eigengewichtes Deformationen fest, die zu Maßabweichungen von bis zu 0,7 mm führten. Dies hatte dann eine geänderte Vorgehensweise der weiteren Bearbeitung zur Folge. Die Endbearbeitung erfolgte in einer der späteren Fundamentaufstellung entsprechenden Aufspannung.

Das gleiche Problem stellt sich für den Transport des Triebwerksgehäuses zur Baustelle. Auch hier ist eine exakte Ausrichtung auf dem Transportfahrzeug unbedingt erforderlich, um o.g. Verformungen zu vermeiden.

Der hier geschilderte Vorgang soll nur zeigen, dass zur erfolgreichen Durchführung solcher Arbeiten außergewöhnliche Kenntnisse in der spanenden Fertigung, eine äußerst flexibel reagierende Arbeitsplanung bzw. –vorbereitung sowie ein für diese Zwecke geeigneter Maschinenpark nötig ist.

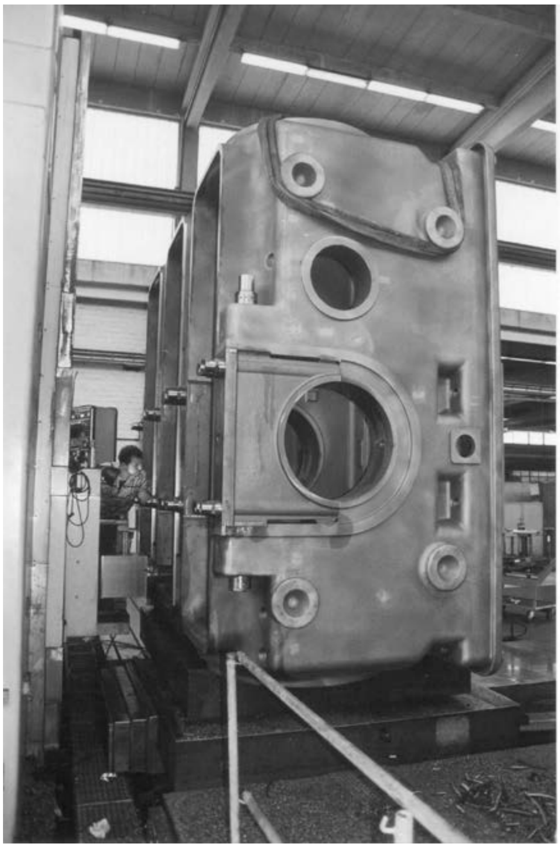

Bild 9 Bearbeitung Kurbelgehäuse

Bild 10 Transport Kurbelgehäuse

Nicht nur die fachlichen Anforderungen bei diesem Reparatur-Projekt waren hoch angesetzt. Die äußerst harten Forderungen des Kunden für die Wiederaufnahme der Polyäthylenproduktion 6 Monate nach dem Schadensfall waren eine zusätzliche Herausforderung. Die Anlage konnte 2 Wochen früher als geplant wieder in Betrieb genommen werden, was mit einem entsprechenden Bonus des Kunden honoriert wurde.

Nach Abschluss der mechanischen Bearbeitung des Kurbelgehäuses erfolgte sowohl für die Hauptlager als auch für die Kurbelzapfenlager ein Lagereinbauversuch, um den Wechsel zum Fertiglager vollziehen zu können, was bei Kompressoren kleinerer Leistungen bereits vor etlichen Jahren geschah. Die GHH-Höchsdruckkompressoren wurden bei ihrer Auslieferung mit mehrteiligen Kurbelwellenlagern ausgerüstet, die ein aufwendiges Einschaben und Einstellen erforderten. Dies ist besonders arbeits- und somit kostenintensiv, wenn solche Arbeiten vor Ort durchgeführt werden müssen. Bild 11 zeigt den Einbau einer Fertiglagerschale in der Neuman & Esser-Montagehalle. Diese Lager werden einbaufertig geliefert und bedürfen lediglich kleiner Korrekturen beim Einbau.

Bild 11 Einbau eines Fertiglagers

6 Kostenreduzierung durch Modernisierung und Detailverbesserung

Die bei den Hypern erzielte Verfügbarkeit zeigt, dass im Wesentlichen die hohen Anforderungen der Betreiber nach Zuverlässigkeit und hoher Lebensdauer erfüllt werden. Dies findet auch unter dem Abschnitt "Analyse der Lebensdauer der wichtigsten Bauelemente" seine Bestätigung. Es wird aber auch sichtbar, dass zum Teil noch recht große Laufzeitdifferenzen bestehen. Außerdem treten trotzdem – wenn auch selten – noch immer überraschend Schäden mit zeitaufwendigen Reparaturen auf, wie z.B. in dem unter Abschnitt 5 geschilderten Schadensfall mit Ursache eines Pleuelaugenrisses mit nachfolgender Zerstörung des Kurbelgehäuses.

Wie ist also eine weitere Senkung der Life Cycle Costs noch zu erreichen?

Bei Neuman & Esser vertritt man die Meinung, dass in erster Linie alle bereits vorgefallenen und aufgetretenen Schäden akribisch analysiert und ausgewertet werden müssen, entsprechende Veränderungen vorzunehmen sind und / oder diese mit gezielten Diagnosemitteln für die Zukunft gesichert als Wiederholungsschäden auszuschließen.

Des Weiteren sollten die regelmäßigen Stillstands- und Wartungsarbeiten in ihrem zeitlichen Ablauf verkürzt und die Intervalle nach Möglichkeit verlängert werden.

Auch kostenintensive Hilfsstoffe, wie z.B. Schmieröl, sollten einer höchstmöglichen Nutzungszeit durch gezielte Untersuchungsmethoden zugeführt und damit Betriebskosten minimiert werden. Bezüglich der Schadensvorbeugung über Maschinendiagnose stehen neben umfassenden Diagnosepaketen für Höchstdruckkompressoren (Prognost-NT der Firma Kötter Prognost Systeme) auch Softwaremodule oder bauteilbezogene Diagnosemittel zur Anwendung bereit.

Letztere bieten sich z.B. für die Temperaturüberwachung bei Lagern an. Dazu muss bemerkt werden, dass die Überwachung der Lagertemperaturen bei den feststehenden Kurbelwellenhauptlagern kein Problem darstellt und schon lange Zeit Stand der Technik ist. Komplizierter liegen die Verhältnisse bei den sich bewegenden Lagern – den Kurbelzapfen- und Kreuzkopfbolzenlagern. Bekannt sind Überwachungsmodule, bei denen Temperaturfühler Signale bilden und über Kabel weiterleiten. Die Signalweiterleitung bzw. –übertragung ist das eigentliche Problem. Neuman & Esser setzt hierfür ein drahtloses Übertragungssystem der Firma CMR aus Frankreich ein. Die Änderung des gemessenen Temperaturwertes wird durch Magnetfeldänderung abgebildet und bei Vorbeidrehen des Senders am Pleuelauge gegenüber dem Empfänger am Kurbelgehäuse pro Kurbelumdrehung übertragen (siehe Bild 12).

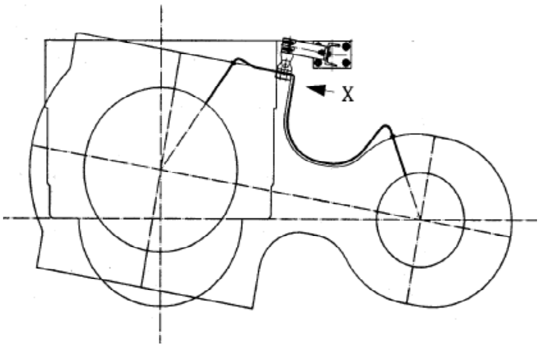

Bild 12 Messung der Lagertemperatur am Kreuzkopfbolzenlager

Dieses System wurde von Neuman & Esser erstmals in Ex-geschützter Ausführung in einem Kompressor verwendet.

Als ein weiteres hilfreiches Diagnosemittel für die Kompressordiagnose wurde die Überwachung mittels Schwingungsaufnehmer erkannt. Damit können sich lösende Verbindungsschrauben, fest werdende oder ausschlagende Lager o. ä. sehr früh und noch vor Eintreten großer Schäden diagnostiziert und durch "Notaus"-Abschaltung verhindert werden. Auch dafür gibt es abgerüstete Diagnosemodule, die in bestehende Überwachungs-einrichtungen integriert werden können. Natürlich braucht es dazu Kenntnisse über die bestens geeigneten Stellen zur Anbringung der Schwingungsaufnehmer.

Die Diagnose der Bauteile im Zylinderbereich setzt häufig eine mitlaufende Indizierung des Druckver-laufes voraus, welche aber wegen der hohen Drücke mit sonst üblichen Schraubanschlüssen nicht vorge-nommen werden kann. Dafür gibt es aber nun auch Möglichkeiten durch Dehnungsmessungen, z. B. an Zylinderbefestigungsschrauben, die die Indizierung in guter Näherung ermöglichen.

Bei allen den vorgeschilderten Diagnosemitteln muss natürlich darauf verwiesen werden, dass damit die eigentlichen Ursachen der Schäden nicht verhindert, sondern nach Maßgabe nur früh genug erkannt werden. Die vollkommenere Methode liegt verständlicherweise in der Ausschaltung von Ursachen. Hierfür stehen aber heute neue Hilfsmittel in Form vervollkommneter Software-systeme für FEM-Berechnungen, Material-prüfungen u. ä. zur Verfügung.

Ein weiterer Schritt in Richtung Verkürzung der Stillstandszeiten bzw. Verlängerung der Lebens-dauer wird von Neuman & Esser durch Substitution von einzuschabenden Gleitlagern mittels Fertig-lagern sowie des Umbaus von Kolben mit Kolben-ringen in Plungerkolben betrieben. Auch die abgestimmte Vergrößerung der Auflagefläche der Fixatoren unter dem Gehäuserahmen zielt in vorgenannte Richtung der Kostenminimierung.

[1] Fotos wurden verschiedenen GHH-Prospekten entnommen

[2] Kötter-Tagungsband 4. Workshop Kolbenver-dichter Oktober 2000: R. Grabowski / K. Wolf “Laufzeitverhalten der Zylinder von Ethylen-Höchstdruckverdichtern“

[3] NEA-NEWS, Ausgabe 1, Oktober 2000

DOTT. ING. MARIO COZZANI s.r.l.
VALVOLE PER COMPRESSORI
Sistema Qualità certificato ISO 9001 dal 1996

New Rotary Valve Actuated by Electronic Control / New Profile for Thermoplastic Shutters of Compressors Valves

by:

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Italy

Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability May 17th / 18th, 2001, The Hague

Abstract:

At present, valves for reciprocating compressors function with automatic alternating movement; also, in order to carry out a control of the capacity, pneumatic or electro-hydraulic devices are used. Dott.Ing. Mario Cozzani Srl presents a new type of patented valve; the valve in question works with a rotational movement of the shutters and is operated by an electromechanical device controlled by an electronic driver. This solution permits complete control of the valve's working cycle, and, consequently, the regulation of the compressor capacity.
With the introduction of thermoplastic material for the manufacture of valve shutters, flat or shaped profiled shutters are being developed: while the first ensure a good sealing capability, the latter ensure a high fluidodynamic efficiency. Fluidodynamic research carried out on shutter profiles by Dott.Ing. Mario Cozzani Srl, have permitted the creation of a new patented profile. The new profile is substantially different from the existing ones and offers high fluidodynamic efficiency combined with good sealing capacity.

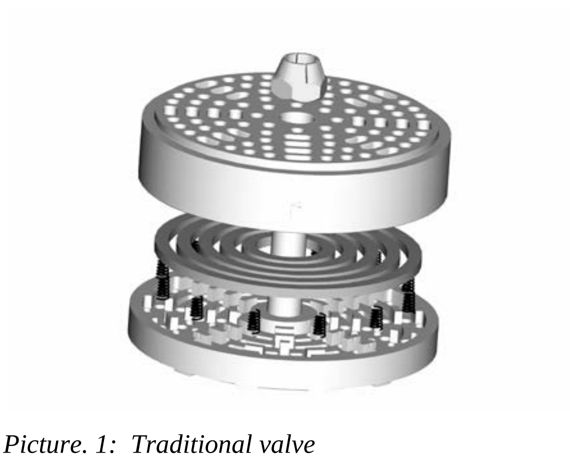
1 Introduction

Since 1946 Cozzani company has been designing and producing reciprocating compressor valves.
Motominima, the original name, in 1975 changed to Dott.Ing. Mario Cozzani s.r.l., after the company founder. Today the company is still owned and run by the Cozzani family.
The factory is located in Arcola, near La Spezia, Italy.
In 1996 the Quality System of Cozzani S.r.l. was certified in accordance with the ISO 9001 Norm.
Dott.Ing. Mario Cozzani s.r.l. technical department is fully equipped with sophisticated calculation programs and benefits from its experience of over 50 years. In recent years Cozzani's research program has had, as its goal, the improvement of the efficiency and working life of its valves. These studies have led to the birth of two new products: a rotary valve with electronic drive and a thermoplastic shutter with a concave shape. This paper contains a brief description of the studies made by Cozzani to reach the final design of these new products.

2 Chapter 2 – Rotary valve

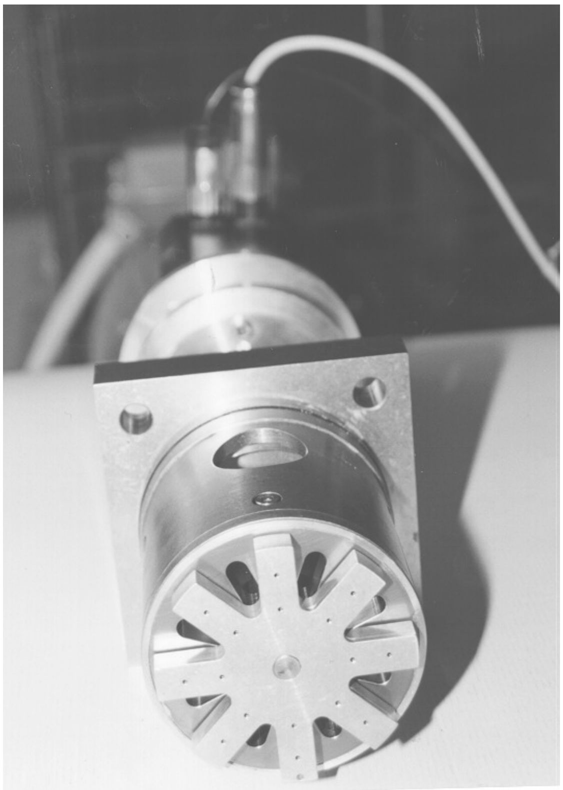
2.1 Valve design

Reciprocating compressor valves work by an alternative movement of the shutter, a typical valve with flat thermoplastic shutters is below (Picture 1).



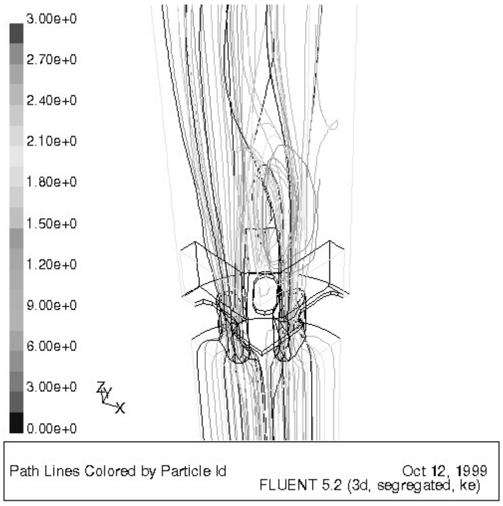
Picture. 1: Traditional valve

The working cycle of a traditional valve is automatic; this means that the shutter movement is due to the difference of pressure between the inner and outer part of the cylinder and by the force made by some elastic components which are part of the valve itself. The aforementioned movement is usually made at high frequencies and at every working cycle, the shutter is subjected to a double "shock" (one bumping into the seat and one into the guard), the life of the valve depends on how long the shutter resists wear due to its working conditions. To eliminate these "shocks", which are responsible for the valve wear, Cozzani considered it opportune to radically change the valve working movement, to a rotary movement. As for the rotary movement there still is, of course, some kind of wear, but of another nature; now the shutters rotate together with the guard and slide on the seat, so wear is due to the friction between the shutter and seat materials. The new kind of valve involves a simplified geometry: less geometrical discontinuity in the valve channels and therefore good efficiency for the gas flow (see Picture 2).



Picture. 2: Rotary valve

To maintain a good fluidodynamic efficiency of the rotary valves different geometric configurations have been tested in the Cozzani laboratory compressor. Many theoretical studies by CFD software were also carried out to choose the geometry which gives lower flow losses; this is an important aspect in valve design because it can imply energy saving for the compressor. One of the fluidodynamic studies is represented in Picture 3, it is evident how the fluid passes directly through the valve channels. The only discontinuity is due to the passages through the seat as can be seen in the CFD simulation represented in Picture 3, in fact there in no lift.

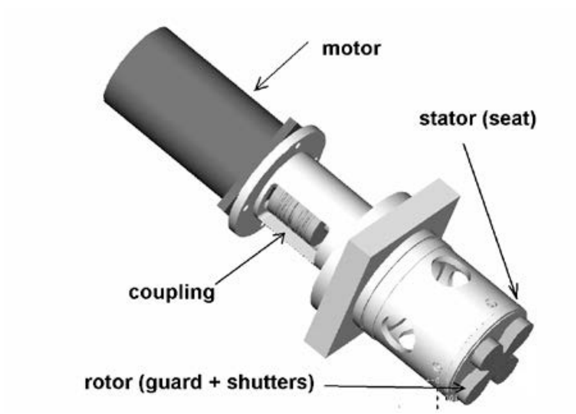


Picture. 3: CFD study of gas flow in rotary valve

The rotating component must have low inertia to avoid the need for large motors to activate the valve.

2.2 Architecture of rotary valve

In the new valve the shutters movement is no longer automatic but it is due to an electric motor driven by an electronic driver, as explained in the following chapters. Shutters and channels are designed so as not to leak during the working life of the valve, elastic components are studied to recover any gap due to friction wear, all components are designed to work in a static way to avoid the typical dynamic problems of springs. Special couplings are used to connect the valve shaft to the motor (see Picture 4). The presence of elastic components is not critical since they work in a static way only to recover the gap due to wear between shutters and seat.

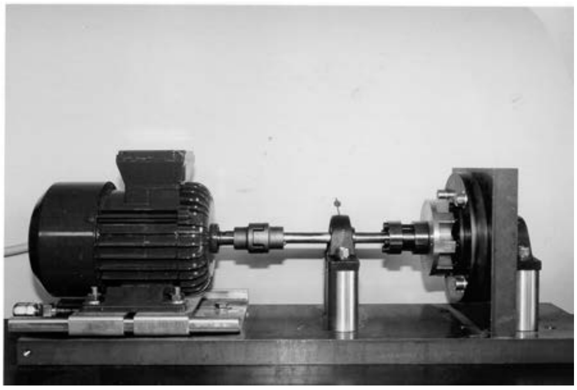


Picture. 4: Rotary valve assembly.

As can be seen in the previous picture the valve is a compact body including its cover.

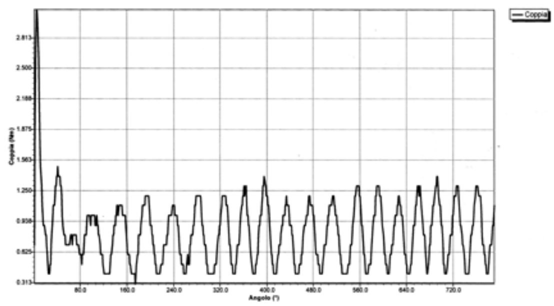
2.3 Testing materials

Preliminary research regarding the materials to be used in the rotary valve was made by Cozzani in its test facilities. Different low friction thermoplastic materials were tested by a machine simulating the movement of the new valve. Some of these materials behaved very well with regards to wear and friction. In particular one has been tested for 10000 hours of simulated compressor life without any particular problems, working continuously at 1400 rpm. In Picture 5 the laboratory-testing machine used for the material evaluation is shown.



Picture. 5: Wear testing machine

Picture 6 shows a diagram of the tested torque value to enable the valve rotation at different speeds.

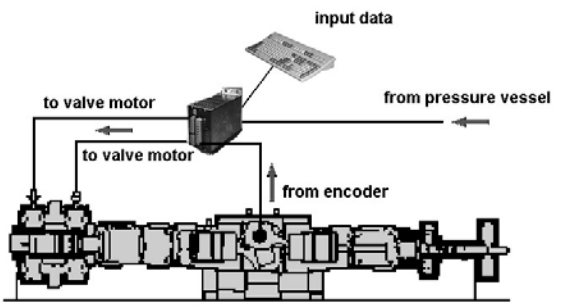


Picture. 6: Torque testing diagram

This latest test was carried out to verify the dimensions of the motors needed for valve rotation and to evaluate the friction coefficient between seat and shutter materials.

2.4 Electronic control

As mentioned beforehand, to control the valve working cycle the motor must receive the signal from an electronic driver. But how is the velocity profile of a valve motor defined?
Cozzani electronic driver receives the signal from an encoder solidly mounted with the compressor crankshaft. In this way the motor starts its velocity profile at the desired crank angle. Of course the desired crank angle is defined by the compressor working cycle and then it must be input in the driver as cycle control parameter using software.
The electronic driver also contains all the parameters to completely define the velocity profile of the motors, these parameters are defined by Cozzani depending on the compressor working conditions, then they are transmitted to the driver using software. In the picture below a simple scheme of the Cozzani electronic control is represented. The sensor signal coming from pressure vessel can be used for capacity control as described in the following chapter. Other pressure sensors mounted inside the cylinder or flow sensor placed in the piping can be used for compressor monitoring.

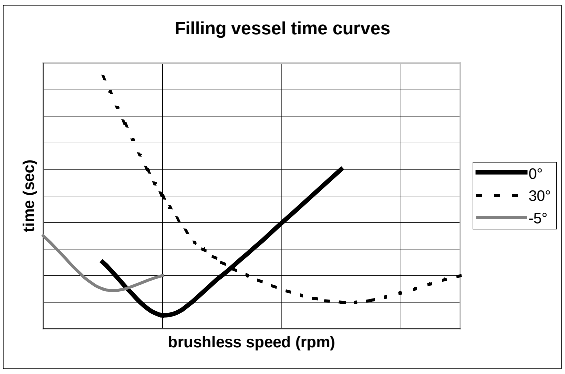


Picture. 7: Electronic control scheme

Of course, depending on the compressor working conditions, explosion proof, waterproof, dust proof, cryogenic or other special motors can be used.

2.5 Capacity control

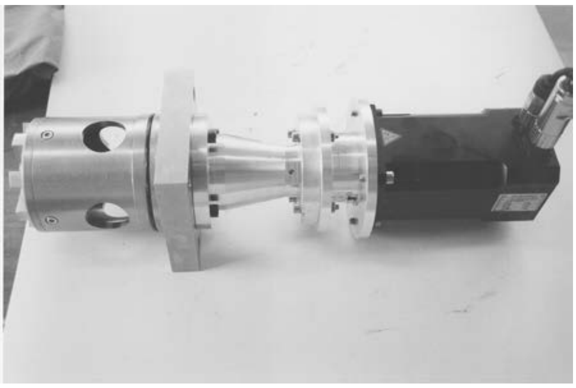
Since the electronic control permits the total control of the valve working cycle, consequently, it permits the control of the compressor working cycle. By changing the velocity profile of the valve motor the valve closure can be delayed and, in this way, the compressor capacity can be varied both without steps and in a step-by-step way. The valve does not need any more pneumatic or hydraulic components, the plunger is no longer necessary, the valve itself can determine the capacity value. How is the valve able to know when the capacity must be reduced? An electric signal coming, for instance, from a pressure or capacity sensor mounted in a pressure vessel or along the piping, can activate the program in the electronic driver so that the valve delays its closure to reach the desired level of capacity. In this way Cozzani electronic system is suitable not only for optimising the valve working cycle but also as capacity controller. In Picture 8 we can see some experimental curves relating to the time taken for filling a vessel with air at a pressure of 4 bar. Each curve is correlated to a different crack-angle value at which the working cycle of the valve starts. As can be seen, by varying the speed of the motor different values of capacity can be achieved.



Picture. 8: Filling vessel time curves

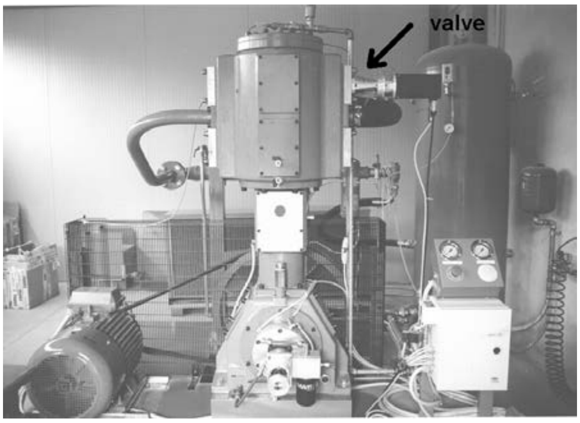
2.6 Compressor tests

The first tests of the rotary valve on a compressor were made in Cozzani laboratory test room. The test compressor is a one-stage single HE effect air compressor working at a speed of 765 rpm. The valves tested were all suction valves (see Picture 9) because greater losses were found in this type of valve and also in order to test the capacity variation.



Picture. 9: Rotary valve used for tests

The compressor (see Picture 10) is equipped with a pressure sensor in order to evaluate the pressure inside the cylinder. The sensor is connected to a digital oscilloscope which enables one to see the diagram pressure-time and thereby evaluate losses during the working cycle. A temperature sensor is also mounted in the cylinder and an encoder is solidly mounted to the crankshaft. The compressor suction pressure is atmospheric and the delivery pressure is 4 bar abs. In the following picture a photograph of this compressor equipped with a suction rotary valve is shown. In the background there is a pressure vessel equipped with a pressure sensor to test the capacity control.



Picture. 10: Cozzani compressor laboratory test

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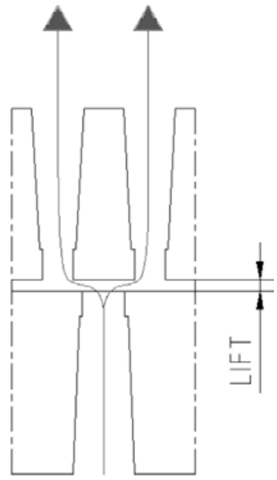
3 Chapter 3 – New shutter profile

As it is generally known, plane shutters do not have a good fluidodynamic efficiency while contoured shutters can have lower reliability at high temperatures. For these reasons Cozzani tried to find a new shape for the shutter section in order to have efficiency combined with good leakage features.

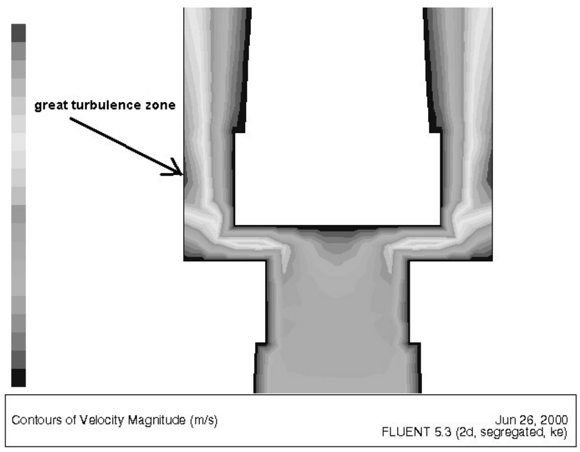
3.1 Flat shutters

In the case of a valve with flat shutters (rings or plates) the fluidodynamic efficiency is relatively low due to the different geometrical discontinuity that the flow finds during its passage through the valve (see Picture.11).



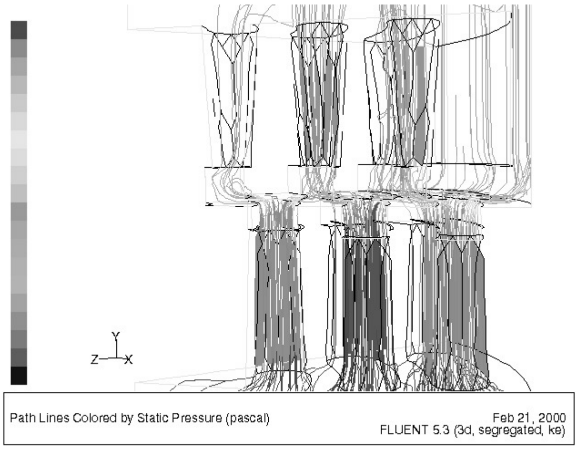
Picture. 11: Flow passage in flat shutter valve

As one can see, the flow meets different obstacles during its passage and the lift parameter has great influence in the section area and therefore in the pressure loss calculation.



Picture. 12: Fluid velocity in flat shutter valve

The above picture represents one of the CFD studies Cozzani makes on the fluidodynamic efficiency of valves. This picture shows the fluid velocity distribution in a delivery valve. The higher value of speed is not in correspondence with the minimum passage section but in a zone located behind this. This means a greater turbulence zone and therefore a loss in pressure. Tridimensional simulations were also made to calculate the value of the loss and Picture 13 is an example of these studies.

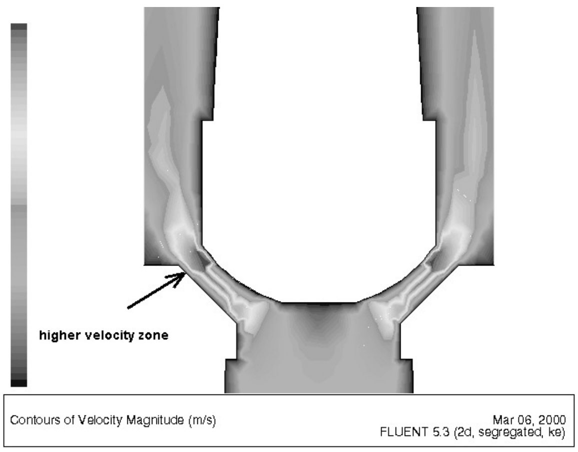


Picture. 13: Tridimensional CFD study

One of the advantages of using plane shutters is that they have few problems with regard to leakage. In fact a good design of this type of shutter together with that of the seat of the valves can ensure good working performance (valve sealing with increasing temperature, hence increasing diameters, is ensured by the sliding of two plane surfaces, one belonging to the shutter the other to the seat).

3.2 Contoured shutters

In this case great aerodynamic efficiency is found and the higher fluid velocity corresponds to the minimum passage section as can be seen in the following figure (Picture 14).



Picture14:Fluid velocity in contoured shutter valve.

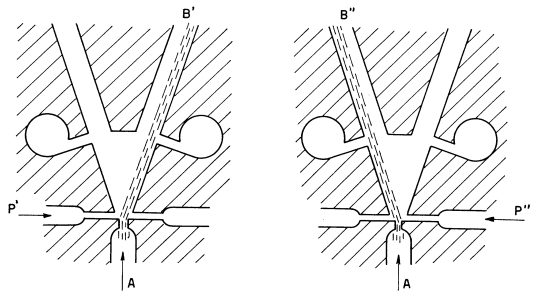
This type of shutter, on the other hand, presents some working difficulties during valve maintenance due to the particular shape of the seats (conical). Moreover, depending on the material used and on the working temperature of the compressors, there can be some problems of leakage due to large-scale deformation of thermoplastic materials compared to steel ones.

3.3 Fluid turbulence inside valves

What can be said about shutter fluidodynamic drag? One of the current tendencies is that of increasing compressor speed, this implies a high fluid velocity through the valves. So if the Reynolds number is calculated the value found determines a turbulent flow inside the valve channels. So even with an aerodynamic configuration of shutters a turbulent and not laminar aerodynamic flow must be considered. As the flow is substantially turbulent Coanda effect[i] can be taken into account.

3.4 Coanda effect

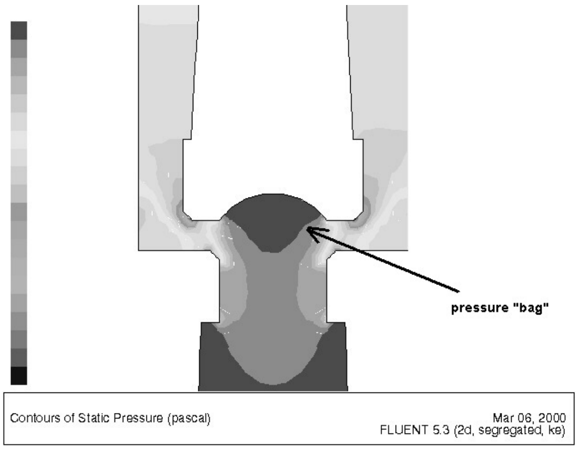
In 1930 Enrico Coanda discovered a fluidodynamic phenomenon regarding the possibility of driving a turbulent flow by pilot pressures (see Picture.15). An inlet turbulent flow (A) can be driven by an inward flow pressure (P‘, P‘‘) towards B‘, B“ . This phenomenon is largely used for pneumatic logical elements, helicopters and aeroplane nozzles.



Picture. 15: Coanda effect scheme

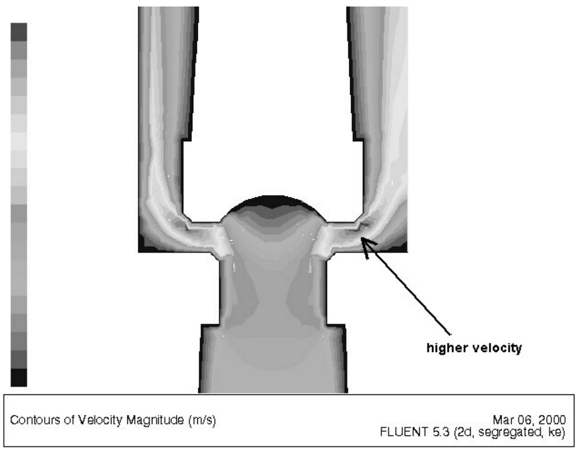
3.5 New concave profile shutters

How can the Coanda effect be used in automatic compressor valves? By creating a pilot pressure in the central part of the shutter section so that the fluid is driven towards the guard channels. In fact cutting the shutter section, so as to give it a concave shape, a pressure “bag” is created during the valve working cycle. This pressure is created by a small part of fluid which takes on a rotation motion and generates a convex shape which drives the remaining fluid towards the guard channels and therefore lower pressure losses are found (see Picture 16).



Picture. 16: Fluid pressure in concave shutter

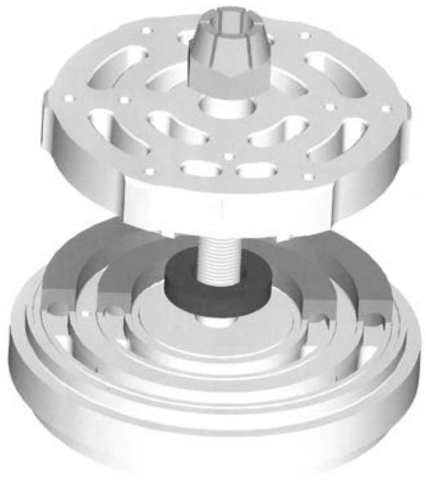
What influence does the new geometrical configuration of these shutters have as regards the fluid velocity field? As can be seen in Picture 17 now the higher value of velocity corresponds with the minimum section of passage. This is a very important aspect because it means lower turbulence and less kinetic energy spent during the passage of fluid. All this leads to lower pressure loss and better valve efficiency. Regarding leakage problems: this type of valve shutter closes the fluid passage due to the contact of two plane surfaces (one belonging to the seat the other to the shutter) this is the same method used by plane shutters. So, there is a new generation of shutters which have the same reliability, regarding leakage, as the plane shutters, but also have better fluidodynamic efficiency.



Picture. 17: Fluid velocity with concave shutter

3.6 Compressor tests

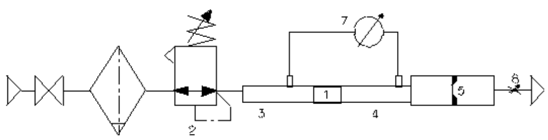
This new shutter geometrical configuration was, at first, tested in the Cozzani laboratory. The compressor used is an air compressor whose features have been shown in Chapter 2.6. An assembly of a valve with these new shutters named CC type is shown in Picture 18.



Picture. 18: CC valve type assembly

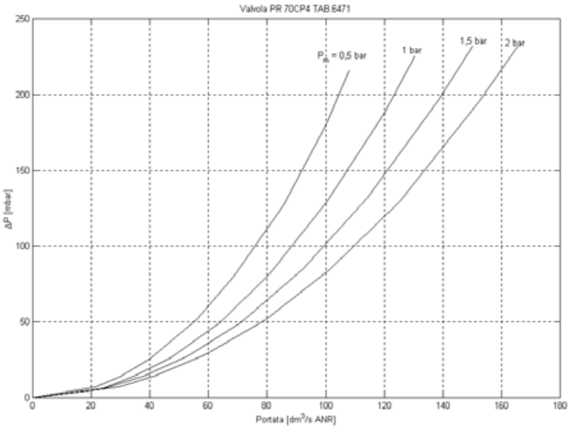
3.7 Laboratory tests

Cozzani tested its CC valves at one of the most important Italian University fluidodynamic laboratories. These tests have been done according to ISO 6953/1, a plan of the test bench is shown in Picture 19.

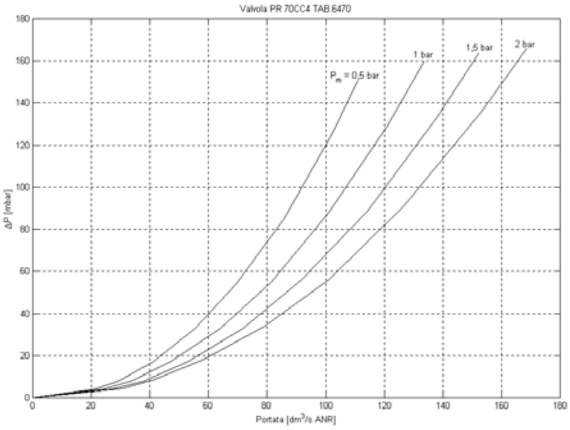


Picture. 19: Test bench plan

The results of these studies show the good fluidodynamic efficiency of the CC valves compared to the flat shutter valves (see Picture 20 showing a flat valve capacity and pic. 21 showing a CC valve capacity).



Picture. 20: Flat shutter capacity vs. pressure loss



Picture. 21: CC shutter capacity vs. pressure loss

The CFD analysis made by Cozzani shows the improved theoretical efficiency of the CC valve compared to a traditional valve with flat shutters. An experimental improvement in capacity of 20% is estimated for the CC valves compared to flat shutter valves

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[i] Coanda effect from the name of the Rumanian scientist Enrico Coanda (1886-1972), who discovered this fluidodynamic phenomenon in 1930.

BASF

Failure on an Ethylene Compressor by Fatigue Fracture of the Piston Rod

by :

Dipl. Ing. (FH) Heinrich Ochs
Leiter Kolbengruppe

Betriebswerkstätte Sondermaschinen

BASF AG

Ludwigshafen, Deutschland

Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability May 17th-18th, 2001, The Hague

Abstract:

In a company of BASF AG, Ludwigshafen there was a damage at an ethylene reciprocating compressor. Suction valve and fixing screws were torn out of the cylinder, 6th stage and thrown through the compressor hall. The process could be run down safely. There was nobody injured. Reasons for this event and aspects of how to prevent damage are shown.

1 Einführung

Zusatzstoffe für Kunststoffe, Kraftstoffe und Kosmetika sind einige der etwa 8000 Verkaufsprodukte, die von der BASF AG am Verbundstandort Ludwigshafen, Deutschland hergestellt werden.

Ethylen ist Ausgangsstoff für die Herstellung einer speziellen Klasse der genannten Produkte. Es wird zunächst mit einem 6-stufigen 2-kurbeligen Vorverdichter (Fa. GHH, Typ F2y 30/40 G, Bj. 1962) auf ca. 300 bar verdichtet. Mit einem 2-stufigen, 2-kurbeligen Nachverdichter (Fa. GHH, Typ B29/25, Bj. 1960) wird das Ethylen auf einen Prozessdruck von ca. 2000 bar hochverdichtet. Die Maschinen wurden im Lauf der Jahrzehnte an die Werkstoff-Entwicklungen auf dem Gebiet der Packungs-, Kolbenring- und Beschichtungs-materialien angepasst.

2 Ereignis

Plötzlicher Druckabfall nach der 5. und 6. Stufe des Kolbenverdichters, sowie das Ansprechen der Gaswarngeräte löste die Abschaltung der Anlage aus.
Der Prozess konnte sicher abgefahren werden. Personen wurden nicht verletzt.

Vor Ort zeigte sich folgendes Bild:

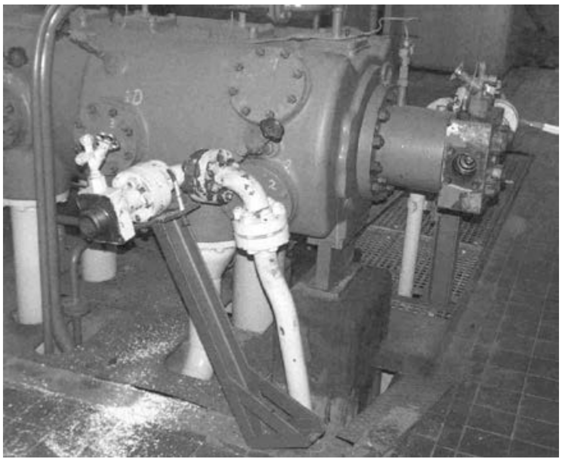

Bild 1: Blick auf Stufenzylinder 2./3. Stufe und 6. Stufe mit abgerissener Rohrleitung

Das Ventildruckstück war mit der Saugleitung aus der Saugseite des Zylinders der 6. Stufe herausgerissen. (Bild 1). Die Saugleitung hatte sich dabei um 180° gedreht. Die Bodenverankerungen der Rohrleitungsstütze war abgerissen.

Die Bruchstücke der Dehnschraubenbolzen ragten aus dem Zylinderkopf heraus (Bild 2). Das Saugventil und zwei der insgesamt vier gerissenen Dehnschraubenbolzen mit Mutter wurden ca. 20 Meter weit vom Kolbenverdichter entfernt gefunden. Der Hallenboden war durch ausgetretenes Zylinderschmieröl stark verschmutzt.

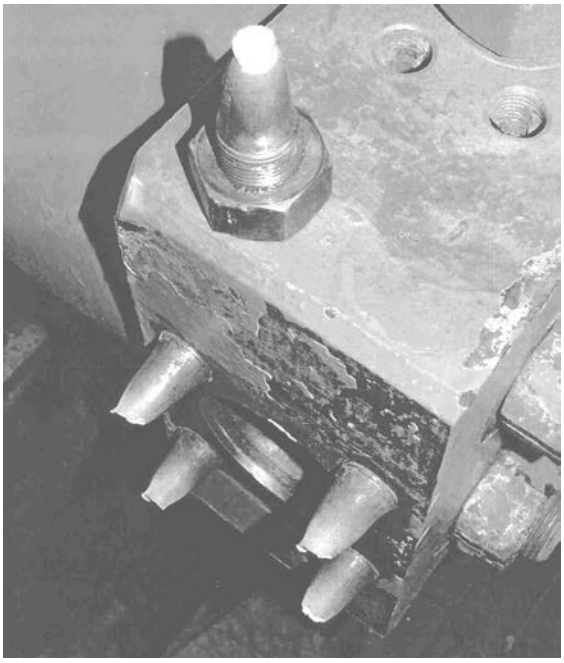

Bild 2: Ansicht von Zylinderoberseite auf Saugseite 6. Stufe

3 Befundaufnahme

Zylinder und Stufenkolben 6. Stufe

Nach der Demontage des Zylinderkopfes und des Zylinders der 6. Stufe wurde der Abriss der Kolbenmutter des Stufenkolbens sichtbar. (Bild 3).

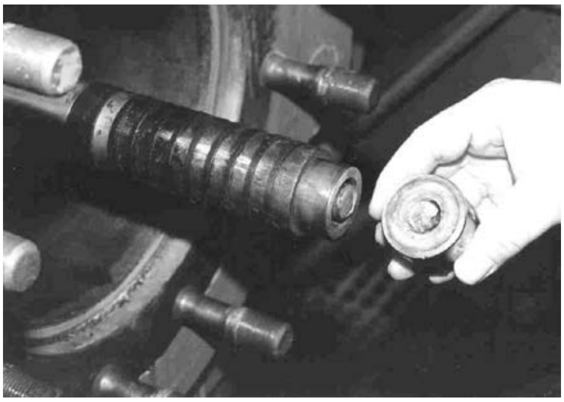

Bild 3: Abgerissene Kolbenmutter des eingebauten Stufenkolbens 6. Stufe

Im Zylinderkopf der 6. Stufe lag die abgerissene Kolbenmutter (Bild 4) mit Teilen des Kolbenführungsringmaterials verklemmt vor dem Auslasskanal der Druckseite.

Bild 4: Abgerissene Kolbenmutter des Stufenkolbens 6. Stufe

Die verklemmte Kolbenmutter musste mit Kraftaufwand aus dem Zylinderkopf entfernt werden.

In Bild 5 ist die Ansicht auf den druckseitigen Auslasskanal des Zylinderkopfes der 6. Stufe zu erkennen. Im Zylinderkopfraum sind Reste des Führungsringmaterials sichtbar.

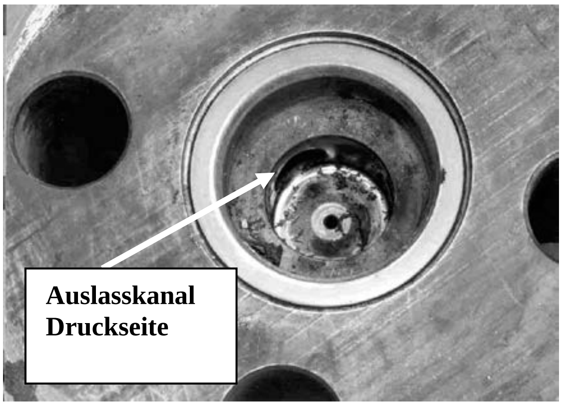

Bild 5 : Ansicht auf druckseitigen Auslasskanal Zylinderkopf 6. Stufe

Eine Detailaufnahme der Kolbenmutter im Zylinderkopf zeigt Bild 6. Das Ringmaterial ist bereits entfernt.

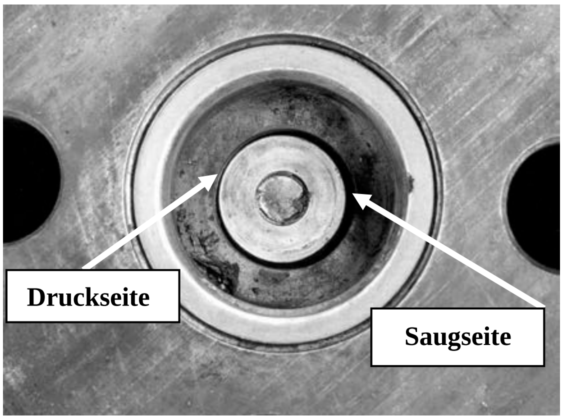

Bild 6 : Kolbenmutter im Zylinderkopf 6. Stufe (Detail). Druckseite links im Bild, Saugseite rechts im Bild

Auf der linken Bildseite ist die Kolbenmutter vor dem Auslasskanal der Druckseite zu sehen. Die Öffnung vor der Saugseite ist in der rechten Bildseite als Sichel erkennbar.

Eine Übersichtsaufnahme des Zylinderkopfes 6. Stufe mit abgerissenen Dehnschraubenbolzen der Saugseite (rechte Bildseite), abgerissene Kolbenmutter und intakten Dehnschraubenbolzen (linke Bildseite) zeigt Bild 7.

Bild 7 : Kolbenmutter im Zylinderkopf 6. Stufe

Rohrleitungssystem

Im Rohrleitungssystem und den angeschlossenen Apparaten nach der 5. Stufe Druckseite wurden keine Querschnittsverengungen oder Fremdkörper gefunden. Die Sicherheitsventile zeigten auf dem Prüfstand keine Abweichungen des eingestellten Ansprechdruckes.

4 Untersuchungsergebnisse

Der Stufenkolben 6. Stufe (Bild 8) ist mit Kolben- und Führungsringbüchse (Pos. 4) ausgeführt. Diese Büchse wird mit einer Mutter M 20x2 (Pos.7), einem Sicherungsblech (Pos.8) und einer Kontermutter M 20x2 (Pos. 9) auf der Kolbenstange (Pos.2) befestigt.

Mit einer Sechskantmutter W 99x1/6“ (Pos. 10) wird der Stufenkolben 6. Stufe in den Stufenkolben der 2. und 3. Stufe eingebaut. Das radiale Spiel wird durch den O-Ring (Pos.1), das axiale Spiel durch das Anziehen der Sechskantmutter (Pos. 10) bestimmt.

Die Zentrierung erfolgt über einen Einspannbund.

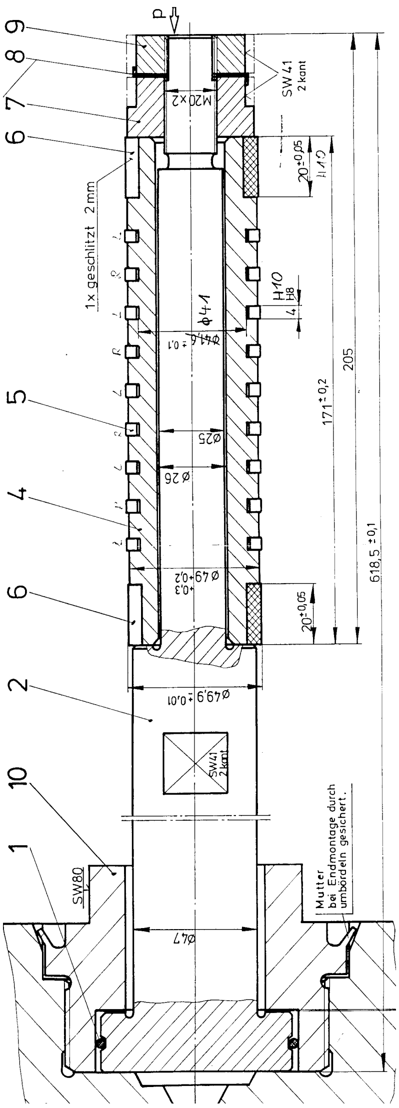

Bild 8 : Zeichnung Stufenkolben 6. Stufe

Der demontierte Stufenkolben wies zwei Bruchstellen auf.

Eine Bruchstelle befand sich am Einspannbund des Kolbens (Bild 9). Der Bruch ist ein Schwingungsbruch mit Bruchausgang an der umlaufenden Entlastungsnut. In der Bruchfläche sind Rastlinien im Rissverlauf zu erkennen.

Bild 9: Bruchstelle am Einspannbund Stufenkolben 6. Stufe

Die andere Bruchstelle befand sich im Befestigungsgewinde M 20x2 der Büchse (Pos.4) mit Mutter (Pos 7).

Bild 10 zeigt den Stufenkolben 6. Stufe mit den Einzelteilen Kolbenstange mit Gewindestummel (Pos. 2), Büchse (Pos. 4), Muttern (Pos. 7-9) und Führungsringe (Pos. 6).

Bild 10: Bruchstelle im Befestigungsgewinde M 20 x 2 des Stufenkolbens 6. Stufe

Bild 11 zeigt die Bruchstelle im ersten tragenden Gewindegang der Kolbenmutter (Pos. 7) des Stufenkolbens (Pos. 2).

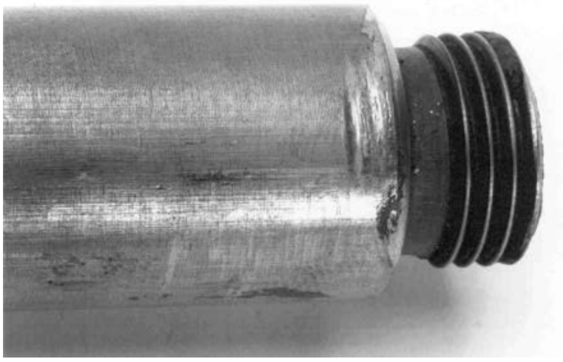

Bild 11: Bruchstelle im Gewinde von Pos. 2

In Bild 12 ist ein Ausschnitt der Bruchfläche des Gewindes mit der Gewindeflanke (unterer Bildteil) dargestellt.

Auf der Gewindeflanke fallen dunkle Flecken auf. Es handelt sich hierbei um Korrosionsmulden, die mit Führungsringmaterial bzw. vercracktes Schmiermittel gefüllt sind. Bruchausgänge sind über weite Teile des Umfanges verteilt.

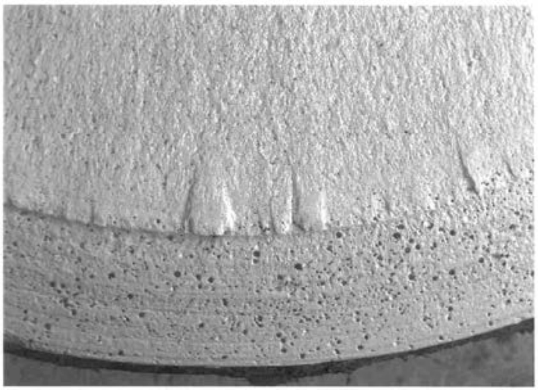

Bild 12: Detailansicht Bruchfläche mit Gewindeflanke von Stufenkolben (Pos 2)

An Stellen , an denen die ursprüngliche Struktur erhalten blieb, wurde eine REM- Untersuchung durchgeführt.
In der REM- Aufnahme (Bild 13) sind neben den Korrosionsmulden parallele Nebenrisse ersichtlich, die als Schwingungsbruch zu bewerten sind.

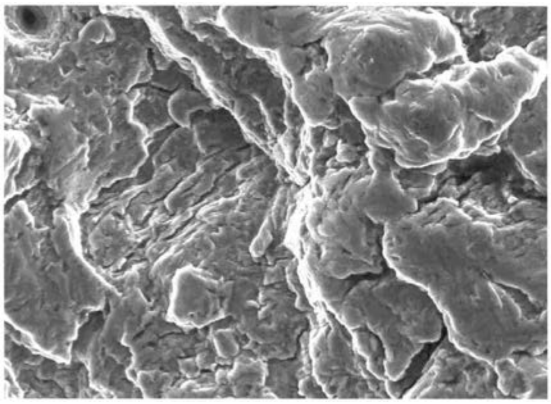

Bild 13 : REM-Aufnahme (2000:1) Bruchfläche mit parallelen Nebenrissen

Operation and Maintenance of Reciprocating Compressors in the New Millennium

by:

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BP

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th / 18th, 2001, The Hague

Abstract:

Reciprocating Compressors of all sizes are key to the operation of many of the processes found in the Refining and Chemicals Industry. Prior to the 1980's, many of the installations were spared machines. This enabled time based maintenance to be carried out by the large maintenance workforces then employed. Since the 1980's, the number of maintenance personnel has been reduced significantly and many installations now continuously run both machines to satisfy the demands of de-bottlenecked plants. This paper reviews the changes to maintenance and the operation of Reciprocating Compressors that have occurred in the last 20 years and why these changes effect not only the end user, but everyone involved with Reciprocating Compressors. This paper also looks at the future of the Reciprocating Compressor in the Refining and Chemicals industry and how the OEM, valve manufacturer, ring manufacturer and others have a role to play in the reliable low total life cost operation of the Reciprocating Compressors.

1 Introduction

Reciprocating Compressors are an integral part of the equipment requirements for many processes. They provide a relatively low throughput with a potentially high differential pressure - although some compressors are operated with a low differential pressure. They operate with a wide spectrum of process gases and pressures.

For the purpose of this paper, I will be concentrating on the Refinery / Petrochemical duties. The bulk of the compressors in these areas are horizontal mutli-cylinder compressors - although there are some vertical compressors on specialist duties.

2 A review of the past

In order to ascertain the present position with Recip compressor maintenance and operation, it is important to understand what has happened in the past and how this has affected our perception and understanding of this class of machine now.

2.1 A lesson from history

“Recip compressors are a pain in the neck”, “Recips are nothing but a maintenance workout!”, “Recips are out of date and should be consigned to a museum where they belong, along with the other Dinosaurs!”.

These are some of the sentiments of Engineers that I have spoken to in the past regarding Reciprocating Compressors. Whatever your feelings about Reciprocating Compressors, you have to accept that for a significant number of Process requirements we can’t live without them. One of the hurdles to installing Reciprocating Compressors in new installations is the perception of un-reliability. This is often the result of a lack of understanding coupled with a legacy of the 1970’s, when a lot of poor installations were completed. The fundamental concept of the reciprocating compressor is sound and the Total Life Cost of a reciprocating compressor compared to a screw or centrifugal compressor, is lower. The areas of difference include spares holding, power costs, turnaround time and initial installation costs.

If you were to go back in time, say 25-30 years, you would have probably found Assets / Plants that had their own dedicated Maintenance Teams. These teams often included a Senior Mechanical Engineer and a Trainee Engineer(s) who was gaining experience to take over a senior engineering role. These teams also had Foremen who had worked their way up through the ranks and had worked on a particular area all their working lives. There were also time served craftsmen who had carried out the overhauls on the machines every year for the last x years. These people knew their equipment well and often could recite the maintenance history of the machine of verbatim. This system had one key advantage over the system that operates today - the retention of corporate experience / knowledge. It also provided a level of knowledge redundancy not seen today.

Experience that was learnt the hard way was retained and handed on through the apprentice system - don’t get me wrong, there were also downsides as well, but we will discuss those in a minute. Another plus point for this period, is the good record keeping that often went with equipment of this era. Often you will find that for any machine over 20 years old, there is a good set of maintenance records up to about 15 - 20 years ago. Then the Dark Ages began! From my persoanl experience, it is nearly impossible in some instances to find out what was done or why to a particular machine during this period, because of the quality of the record keeping. It might be worth adding at this point that it was not only the Recips that suffered but so did most other types of main machinery. So what happened? A number of changes took place and there was a big push to cut overheads. This meant trimming down the manning levels, reducing the trainee schemes and loading up the people left with the workload of the personnel who had already left. There was also a gradual increase in the bureaucracy that the Engineer had to contend with. This did not only come from within the Company, there was also a gradual increase in bureaucracy that originated from European legislation during this time. This period also saw the introduction of the computerised maintenance management systems. I am not suggesting that these are all bad, but during the transition, significant amounts of valuable information was lost. Everyone saw the computer systems as a cure for all ills, but the old adage of “ rubbish in, rubbish out” applied to these systems as much as any others.

The upshot of this was that we lost a lot of the experienced personnel and corporate knowledge. The reduced numbers meant that often things were skimped on, work routines were put to one side, preventative maintenance was cut back to fit in with plant shutdown time frames and budgets, and most of the remaining personnel were overloaded with work. At this point, we slipped from planned maintenance into the trough of reactive maintenance - “fire fighting” ruled the day. This represented the “Dark Ages” for machinery.

As I mentioned before, this early period was not the complete utopia that it might seem, there was a lot of industrial tension, demarcation was prevalent and promotion was literally through “dead man’s shoes”. But what about the recips during this period?

2.2 Reciprocating Compressors before the Dark Ages.

Appearances can be deceptive. The general perception today is that Recip Compressors were more reliable during the “Pre-dark ages era” than they are now. Lets examine what was happening during this period. We had large maintenance resources at our disposal, often with dedicated compressor maintenance teams. The installations were mostly spared and because of the lack of any suitable Condition Monitoring technology, the maintenance of the machines was solely time based - this was derived from manufacturers recommendations and experience - i.e. empirical.

This type of maintenance regime resulted in parts being replaced prematurely and inherent unreliability was often masked by the ability to changeover to the spared machine and carry out maintenance on the run. The un-reliability of the machines during this time frame was not an issue, as there was no apparent impact on manufacturing output because of the sparing policy. The impact was on the maintenance budgets of the time but as these weren’t too closely inspected, the relatively high maintenance costs of the recips was accepted.

2.3 The Dark Ages

During this period experienced personnel gradually became a thing of the past, manning levels were taken to a minimum and the majority of main machinery suffered as a result. Also during this period, de-bottlenecking began to make an appearance. This involved identifying process bottlenecks to extra output and overcoming them. Much of the equipment of the time was simply replaced with bigger and more modern versions or simply uprated as in the case of the centrifugal compressors. Remember though that most Reciprocating Compressor installations were spared, in fact Reciprocating Compressors were often the only major items of machinery that were. So from a process point of view, it was a simple procedure to increase compression throughput - just run both compressors together. This introduces a number of interesting scenarios. We now have two compressors running in parallel - that were not originally designed to do so. One of these compressors will most likely be running on partial through-put. If you cast your mind back to the original concept of sparing recips, you will realize that we removed the safety margin of being able to change over to the spare when things go wrong.

The unreliability inherent in some of the early designs and installations now came back to haunt the end user with a vengeance. On top of all this we were moving further and further away from having experienced engineers at the helm, with the time to sort out the problems.

During this time, it began to dawn on the end-user that the Original Equipment Manufacturer (OEM) was taking him for a ride as far as spares were concerned, so he began to turn to the 3rd party reverse engineering companies for some of his spares. Also during time the wear parts manufacturers started to become more proactive by dealing directly with the end-user. This all resulted in the OEM becoming more and more divorced from the end-user. The OEM then became nothing more than a supply and forget shop for new recip compressors.

This was a detrimental situation for both sides; on the one hand, the OEM was losing out on the feedback from the end-user, which is vital for product development; and the end-user was loosing out on the injection of experience and technical knowledge from the OEM. To further compound the situation, the reverse engineered parts were not always successful and reliability suffered even more.

During this time a lot of work was done by the valve manufacturers and wearing parts suppliers. This work was aimed at curing the symptoms rather than the root causes of the problems. The reason for this is because they did not have any control over the root causes. They were trying to sell solutions to problems which were more fundamental than the parts involved. It is difficult to solve a valve problem when the root cause is liquid condensing in the cylinders.

2.4 Coming out of the Dark Ages

I don’t think that we have fully recovered from the “Dark Ages” as such but there has been a lot of progress in recent years. The “Cinderella” of the process industry is starting to come out of the Kitchen! There have been a number of improvements over the last few years in all aspects of the Recip Compressor. Therefore, lets examine some of these before we look at what our aspirations are for the future.

2.4.1 Maintenance

On the maintenance front, the OEM's are being involved more and more in compressor overhaul work. There are no longer sufficient personnel within the end-user organizations to cover major overhauls on Reciprocating Compressors and all the other work that has to take place during a plant shutdown.

In the past, maintenance on Reciprocating Compressors was nearly always carried out on the run. Now with no spare capacity available, without reducing plant output, Reciprocating Compressor maintenance has to be done during shutdowns; this imposes a set time schedule on the work and often causes resourcing issues. Therefore, the OEM's are now being involved more and more in overhaul work, and in a lot of cases the Service Dept does better than the Manufacturing Dept within an OEM's organization.

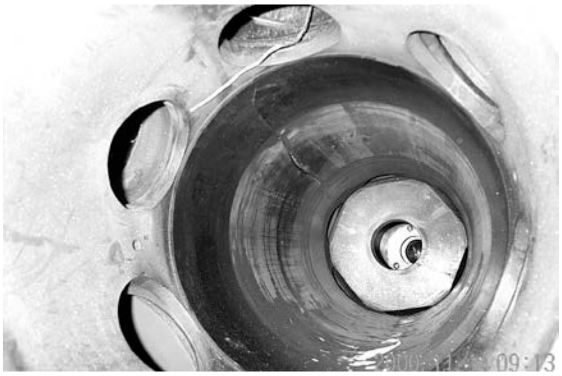
The problem with this situation is that the Reciprocating Compressor overhauls have to be timed to coincide with the plant outages. This often involves 1 to 2.5 years continuous operation between maintenance intervention - here we come back to the question of perceived and actual unreliability of Recips!

Another issue that has arisen is the fact that some 3rd party engineering companies will often modify parts to suit their own design and to ensure that further new parts are only bought from them. This applied to wear components, rods, packings, wipers and housings This has often resulted in confusion with spares, especially as more and more parts are now being purchased from the OEM, who are unaware of the changes that have occurred.

Original data is also an issue, with a lot of valuable information missing or simply out of date, it is often difficult to determine what tolerences and sizes should be used. With the loss of experienced personnel it is often difficult to fill in the gaps - especially as a number of the OEMs have since gone out of business or been taken over. With the de-bottlenecking of plants, there is often a resultant change in operating conditions - i.e. gas composition, pressures, temperature or throughput. This has occasionally happened without a full understanding of the implications to the compressors of the changes. This has in a few isolated cases, had a detrimental effect on the reliability of the compressor.

The ability to identify problem areas has also been hampered by the lack of quality overhaul records. This has also impacted on the evaluation of modifications and new materials.

Another down side to all this, is the fact that - like the rest of us - the recip installations are not getting any younger. There are a lot of installations that are 30 + years old. With these older machines we are seeing subsidence of the foundations, cracking of the foundations, mis-alignment problems etc. This is resulting in some severe remedial actions having to be taken, which cost money and takes time and resources to accomplish.



Picture 1: Cracked liner believed to have been caused by thermal expansion.

Yet it still has to be said that there are a lot of installations that have operated successfully during the "Dark Ages". When these successful installations are looked at carefully, It turns out that in theory some should have failed years ago! This must say a lot for the manufacture and over-design of a lot of the early recips. We have on the odd occasion come across some horrendous misalignments and web deflections, which should, according to the book, have caused a serious failure - but the machine carried on operating!

2.4.2 Operations

Nowadays there is a general acceptance that reciprocating compressors are an integral part of many processes. Try as you may, there are no cost effective alternatives in the majority of cases. Therefore, more energy is now being directed to looking at the process side of the reciprocating compressors. Historically there has been a certain apathy about the condition of the gas going through the compressors and what happens during the compression process. From my experience, this is one of the fundamental keys to the reliable operation of the compressors. Get this wrong and you are in for a lifetime of reliability issues.

One of our key problem areas is hydrogen compression. This is a common duty within a Refinery and is also required on some of the

Petrochemical plants. Often the Hydrogen stream will contain other condensables. These can condense out either in the suction lines to the first stage or after the inter-cooler knockouts. Recips don't work very well as pumps! - although enough people have tried! This has resulted in no end of heart ache for the valve manufacturers, where their valves are blamed for the unreliability of the compressor, even though they are just the symptom of a more fundamental problem.

Picture 2: Damaged suction valve following liquid carry-over.

Another area that has been of concern, was the Operators lack of understanding of how the compressor worked and how to diagnose faults early. The Operators are the first and often the only line of defense for reliable operation of the compressor. If the Operators do not have the knowledge to fulfill this role, then there is a serious risk of major damage being caused to the compressor. This is often not the Operators fault, it has been caused by the way that compressor operations have had to go because of the "Dark Ages".

I have seen a gradual realization that reliable operation of recip compressors is obtained not just by good design and installation but by careful operation by trained operatives. Time invested in training and explaining is well worth the effort. Good instrumentation, operating procedures and operating parameters are also required, as these are the tools the operators require to do the job.

3.0 Present day

Hopefully the first section has given some insight as to how we have arrived at the present state of affairs. Now let us examine what is happening at present, and what our aspirations are for the future.

At Grangemouth we have arrived at a point where the recip compressor is being recognized as a critical component in many of the processes. This has been brought home recently with a series of incidents where recip unreliability has been the root cause of significant lost production. As was mentioned before, a lot of this has been the result of historical unreliability that has not been identified and corrected - this omission has resulted in a costly learning exercise.

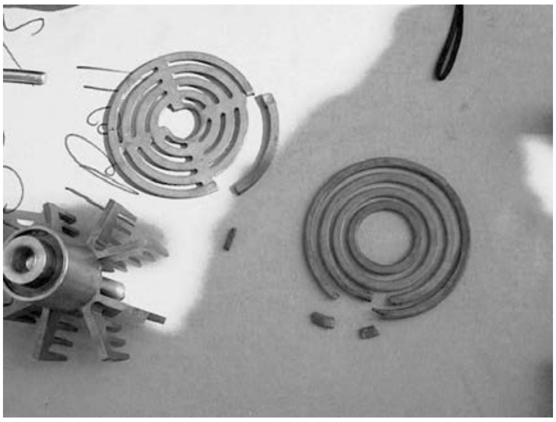
Now, let us examine some of the areas that are causing concern today.

3.1 Non gases going through the compressor.

I have referred to this section as "Non gases" because the problems encountered are not just confined to liquids. First, let us examine the liquid issue. Then we will move on to other "non gasses" that can go through the machine.

<u>3.1.1 Liquid</u>

I am sure that everyone has encountered the liquid issue somewhere. This is a problem that does occur in a number of duties, although the Hydrogen duty is particularly prone to liquid issues. The symptoms are treatable, but it is the root cause that people often miss. The indications of liquid going through a recip compressor are very distinct - i.e. the variable wheezing sound from the valves, the knocking as the machine approaches hydraulic lock, knocking of the motion works, broken valve springs and the characteristic signatures found at TDC and BDC of the condition monitoring traces.



Picture 3: Broken valve plate and rings following extended operation with liquid in gas stream

So what is going wrong? Often it is a combination of circumstances that lead to liquid going through the machine. The one that plagues us most is saturated hydrogen - this problem is quite an issue with us in Scotland because of the wonderful climate that we have. This duty is quite common in the Refinery, and involves make up or recirculation of low purity hydrogen, often to quite high pressures. The knock out of liquid and the maintenance of super-heat in the line between the knock out and the compressor suction are the keys to the problem. Often, this requires little investment, as lagging and trace heating the line to the compressor suction are sufficient. Sometimes, the cylinder is too cool in relation to the incoming gas and this has been known to cause condensation in the cylinder. Again, attention to detail is required, plus a reliable system of controlling the jacket cooling inlet temperature to the cylinders. This either involves a recycle of the plant cooling system or a closed cooling system for the jackets.

Personally, I prefer the closed cooling system. The reason is that we suffer from poor quality cooling water in a number of areas, and the only way to ensure effective cooling water flow and temperature control is with a self contained closed cooling system.

One area of liquid carryover that is not always recognized, is caused by over lubrication. Hydrogen is very poor at carrying entraining liquids, so any liquids tend to build up in the cylinders until they have no choice but to be expelled through the valves. This problem is particularly observable when there is over lubrication. This results in valve stiction and high impact loadings on the valve components. The CM traces will also show a phase shift in the valve event timings. This results in a reliability issue with the valves and can seriously impact production. Correct lubrication is an absolute must on these

types of duties.

3.1.2 Polyethylene

Polyethylene plants have provided some interesting challenges in the area of powder contamination of the gas stream. On certain polyethylene plants, it was not uncommon, until recently, for recip compressors to be taken out by polyethylene (PE) powder break through. This resulted in a beautiful PE coating on the valves, pipework, bottles cylinders etc, as the internal temperature of the compressor exceeded the melting temperature of the PE. The bottom line was that it didn't improve the working of the compressor. There was a high maintenance cost attached to cleaning the system and compressor components. Recently, a new filter system has been installed, and the indications are that plastic coated compressors are now a thing of the past.

3.1.3 Debris

Due to the time constraints involved in plant shutdowns, there isn't often the time to carry out effective pipework / vessel internal cleaning. Following a valve failure or piston ring failure there is going to be a certain amount of resultant debris left in the system. This debris could be metallic or non metallic. Some of it will get washed out by condensate and liquid knockout systems, but some of it will find it's way down the system to the next stage. The probability is that it will pass through a stage without causing problems, but occasionally, debris will get jammed in valves, often between the valve rings / plates and the seats. With the metallic plate valves this is just going to cause overheating, but with plastic ring valves we have seen whole sets of valve rings melt and disappear - producing more debris in the system!

The move to plastic ring valves has nearly eliminated the potential for metallic valve components to get into the cylinder when they fail and cause consequential mechanical damage. This was a big issue in the past, and a number of compressors still bear the scars of broken spring plates etc going through the machine.

Picture 4: Cylinder showing signs of valve debris attack.

3.2 Alignment and Foundations

This is an area that is very often neglected, through lack of understanding of the implications. We now carry out regular cylinder / crosshead alignment checks as part of the routine overhaul of compressors. Sometimes the results can be startling and we have had to carry out a number of cylinder re-alignments recently.

The taking of web deflections is another area that has not been top of the priority list. These are crucial to the monitoring of the motor / compressor alignment, which impacts on the long term health of the machine, and is often relatively easy to correct. These are also now included in all reciprocating compressor major overhauls.

There are a number of machines where the alignment of the cylinder with the crosshead is not correct. This causes problems with vibration and wear. The cause of this is often related to the subsidence in the raft or a lack of strength in the foundations. In Grangemouth, a number of the early installations were built on thin rafts. The soil structure is very soft and there has been a number of instances of significant subsidence of the rafts. New installations are now installed on a piled raft.

The "silent" problem that attacks many compressors installations, is the question of holding down bolts. My experience is that there are a large number of compressors that are inadequately attached to their foundations. This results in excessive movement of the compressor frame. This is damaging to the compressor in the long term and affects the integrity of the small bore pipework around the machine.

So what is the problem? A lot of the problems that we see are on older installations - which covers the majority of the Refinery installations. Often the foundation design is inadequate, with a low density of re-bar and with the re-bar only around the periphery of the block. There are sharp changes of section to the foundations - stress raisers and often the concrete is of poor quality. The older foundations were often designed just to support the static load of the compressor, and not to take account of any dynamic loadings. The resultant vibration caused by this oversight often results in cracking of the foundation block and ingress of oil and water to the concrete.

The issue with the foundations is often the way that the compressor is attached to the foundation block. To many - including some OEM's, this is a black art. There some very good reports on the subject issued by the Gas Machinery Research Council (GMRC), which are well worth a read. The bottom line is that the attachment between compressor and foundations relies on friction. This is the friction between the support faces of the crankcase / distance pieces and the foundation. Often this will be between the compressor and sole plates. The friction generated at these interfaces is controlled by the condition of the faces and the compressive load applied - i.e. the force applied by the holding down bolt stretch.

In the older installations, the holding down bolts were of the "J" type, i.e. a bar with the end bent round. These bolts have been in place for 20 - 30 years and they corrode. This has resulted in a number of cases where the bolts are pulling out. This means that you cannot get the pre-load required and the friction force is not applied to the contact faces. Introduce oil and water again to lubricate the faces and all that is stopping the compressor from walking off the plinth is the physical presence of the bolts! There is very little absorption of the vibration energy by the plinth. Sometimes it is possible to see daylight between the bottom of the crankcase and the foundations that it is supported on during the compression cycle.

This is a very expensive problem to correct. We have reinstalled one compressor so far, although there are plenty more that require attention.

In a number of situations, the resultant vibration from the compressor can be felt 20 - 30 feet from the foundations.

3.3 Maintenance generally

As you may have gathered there are a number of issues that require looking at as far as design and installations are concerned. Associated with these is the need to improve maintenance procedures and recording of overhaul data.

As has been mentioned earlier, there has been a general slippage into "Fire fighting mode" because of circumstances in the past. To progress from fire fighting to proactive maintenance is a big hurdle to overcome. In order to make the move, there has to be integration of the two key types of maintenance, i.e. time based and condition based. The application of these types of maintenance to the reciprocating compressor has to be based upon operating experience, criticality of the components involved, what methods of monitoring there are available and of course the criticality of the machine to the process. During the life of a compressor the mix of these types of maintenance will change as new technology appears on the scene and operating experience of the machine is gained.

Presently, most end-users that I know are working on the overhaul reporting, overhaul procedures and looking at ways to cascade experience to young engineers and operators. The use of a common reporting procedure will aid the assessment of problem areas and help with the evaluation of new technologies and materials.

The wear materials that are in use today are a big improvement on those that were around 10 - 15 years ago. The development of the 3rd generation materials for wearing components will enable us to push to envelope even further. These are helping with the overall operational reliability in the present operating environment and extending the interval between maintenance, but this should not be confused with providing solutions to more fundamental problems, which still have to be tackled separately.

Valves are another area where there have been considerable improvements. Modern valves are becoming more specialized. There are now designs of valves for specific duties, and the understanding and modeling of valves has improved considerably with new computer analysis techniques now available. The level of understanding of how valves operate and what makes them fail is considerably greater than it was say 10 years ago, but the driver for this understanding has not normally been the fault of the valves themselves. It is back to the question of what goes through the valves. The valve failures are often just the symptoms of more fundamental problems.

Condition monitoring is an area that is being utilized more and more. I am sure that most people can come up with examples where reciprocating compressor CM has helped solve problems, identified failing components and identified the source of that infernal knocking sound coming from the compressor - somewhere! With experienced operatives, it is possible to identify quite a number of operational problems and a number of mechanical ones. I have seen significant success with the identification of valve problems and the isolation of motion works problems recently. One of the most impressive successes I was involved in, was the result of combining Recip Trap data with that obtained from the ADRE data collection and analysis system. ADRE was hooked up to the rod drop probes and used a common key phaser to the Recip Trap. From this it was possible to determine that the cylinder in question was rippled and badly worn at the bottom quadrant. This was later confirmed when the cylinder was opened up.

Picture 5: Ring type valve showing signs of liquid in gas stream. This was identified by condition monitoring of the compressor.

In another instance, a critical compressor was kept running long enough to rearrange production plans and bring the plant down in a controlled manner. This was only possible because the machine was closely monitored and the condition of the failing valves were reviewed on a daily basis with the Recip Trap.

Using condition monitoring, advanced materials and keeping a close watch on the process side of things, it is possible for reciprocating compressor to achieved and indeed gone beyond the 2 years between maintenance intervention. These machines are still few and far between, but it can be achieved. I know of a few machines that have achieved the 5 year milstone.

A problem that comes up on a regular basis, and I am open to suggestions on how to solve it, is the removal and reinstallation of compressor valves. This is an easy matter on the hp valves where the valves are mounted on the sides or the top of the cylinder, but has anyone tried to remove or fit 10 inch compressors valves on the bottom of the cylinder when it is freezing cold and wet? It is an important question that needs to be asked when designing a reciprocating compressor. All we ask is that the OEM just stops during the design process and asks “How would I carry out the maintenance to this particular piece of the compressor, especially in the desert heat of the Gulf, or the cold and wet of Scotland”. I know a number of fitters who would appreciate that kind of forethought!

3.4 Operation

We must not forget the operational side of the compressors. This can make or break a machine as much as poor installations. I know of some areas that are presently engaged in what they refer to as a “Basic Care” program. This involves reviews of the operating instructions for machines, operating parameters, check sheets, instrumentation and most importantly, the training of the Technicians who are involved in the operation of the machine.

A number of the process areas that I know of are also in the process of analyzing the process requirements of their compressors. This is to determine where they are presently operating compared to the original specification for the machines. It is a well known fact that most reciprocating compressors do not operate at the originally specified operating conditions, the trick is to find out exactly how far from the original spec they are operating at!

Operations are now starting to take the condition of the gas going through the machines more seriously. This has mostly been the result of condition monitoring taking place. With Reciprocating compressor CM, it is now possible to show the effects of liquid slugging through the machine, it can also highlight the effect of over lubrication with the phase shifting and high impacts of the valves. This is helping to identify areas of operation that require improvement and steps are being taken to put these problems right.

One area of operations that will come under the spot light more and more, is leakage of process gasses to atmosphere. The integrity of the cylinder covers and valve covers is quite easy to obtain, but sometimes a little thought is required. Did you know that Di-ethyl ether will attack Viton? Guess what material is used to seal valve covers? This has resulted in no end of trouble trying to seal the valve covers on one of our processes. This is just one example where everyone has to be vigilant with all aspects of the process interaction with the reciprocating compressor.

Leakage from the pressure packings is another area that requires careful attention. This has always been accepted in the past, but now there are systems available to actively restrict the release of process gas into the distance piece and consequently the atmosphere. The pressure purge system is going to have to become standard, as American style leakage legislation is soon going to catch up with us here in Europe.

One last thought from an operations perspective and this also covers European bureaucracy - please, no more hand operated barring over systems. Manual handling regulations take a dim view of them. Barring over mechanisms should be either electric or pneumatic powered as standard. There is nothing worse than having to bar over a heavy machine in the middle of winter when it is cold and wet. They can also prove to be dangerous if not used correctly.

4.0 The future

The future is green, not Orange, for reciprocating compressors.

Reciprocating compressors have the historical mantle of unreliability, which is influenced more by perception than reality. This will prove difficult to shake off and although we have some notable exceptions on site they can reach a high level of reliability. Reliability depends upon the condition of the gas passing through the machine, the state of the installation and the maintenance regime being carried out. Most screw and centrifugal compressors are not expected to operate under the dire conditions that face some of the reciprocating compressors, so we need to seriously address the conditioning of the gas passing through the compressor.

There are still installations of recips being undertaken - Grangemouth has had 4 in the last 12 months. The recips that will be installed in the future are going to have to be more automated and more closely instrumented - just think of the level of instrumentation on your average centrifugal compressor and compare that to your average reciprocating compressor. On a lot of present installations, the level of instrumentation is positively archaic.

The level of condition monitoring will increase. Compressors will have to be fitted out as standard to accept this. Instrumentation like knock detection, bearing temperature monitoring, pressure packing purge control, and real time piston positioning will have to be installed as standard. Valve temperature monitoring will also be installed as standard.

Material technology will continue to make advances, especially into the 3rd generation materials. The use of these materials is going to have an impact on the time between maintenance, especially with the non lubricated machines. Greater understanding of how these new materials operate and what their limitations are by the end-user is required. Often the material compositions and the limitations of new materials are shrouded in mystery and it is like dealing with medieval alchemy sometimes, i.e. getting information out of some material suppliers is like getting blood out of a stone. Often the new materials prove to be an improvement, or at least on a par, with the previous materials, but occasionally things can go wrong and the end-user can end up looking at a sudden increase in the maintenance expenditure. Therefore a greater openness of what the new materials consist of, what they are trying to achieve and how is required. The end-user also needs to increase his knowledge of the materials involved in order to discuss problems constructively with the materials suppliers. This would benefit both parties.

I believe that valve technology will continue to evolve and new valve designs will be forthcoming to cope with the vagaries of the various processes. Valves are the present day key to improving the time between maintenance intervention, they are the present weak links. A lot will depend upon the way that we, the end-users, operate the compressors. If we continue to abuse these devices, then we will not make progress in our objective to improve availability.

An important fact that needs to be understood by the reciprocating compressor industry, is that most end-users do not want to buy Maintenance or Unreliability!

What I mean is that most OEM's presently will offer a number of maintenance contract proposals with your brand new compressor. These are based upon established and comfortable maintenance intervals, and possibly a condition monitoring element to ensure that they don't get caught out. They get paid for the maintenance and back up they carry out.

This is not what we want in today's environment. The end-user requires a piece of machinery to fulfill a particular process requirement. It should do so reliably and be available when required. The run duration of the plant should be determined by the process, not by the machinery involved - i.e. the tail wagging the dog syndrome.

The alliancing approach that has already been applied to pumps, is based on reimbursement on reducing maintenance costs and increasing reliability and mean time between maintenance. This is often referred to as sharing the gain or the pain. At present most of the parts that break wear out or are just inadequate are replaced by purchasing in replacements. The cost to the supplier is non existent unless there is a warranty issue or there has been the supply of trial components. The cost to the end user can run into hundreds of thousands of pounds if a plant has had to shut down because of the problem. The alliancing process focuses on the development of reliability in machinery and the parties involved and paid on improvements to performance and reliability, not on performing maintenance routines. It is not acceptable to maintain the present status quo.

At the end of the day, the recip compressor is potentially a reliable, low total life cycle cost machine, we have just got to ensure that this happens.

5 Conclusions

Recips are here to stay. In order to maximize their potential, all parties concerned must work towards eliminating potentials for unreliability. This is not just confined to the OEM of the machines, but must include the end-users, valve manufactures and the wear parts manufacturers. There has to be a sharing of information between the various parties involved to allow the development of good operating practices, good installation techniques etc. Training is also another issue. In order to maximize the reliability of these machines, we need to ensure the personnel involved are trained in the relevant knowledge required. This will by necessity involve groups external to the end-user because of the resourcing issue.

Above all, we need to aim to achieve similar reliability to the centrifugal compressors, where a 5 year run is not unusual. This will require the identification of the components, or procedures that are stopping us from achieving our goal, and designing solutions to allow us to overcome these problems.

6 Acknowledgments

I would like to thank the various Asset / Plant Engineers around the Grangemouth site and other BP sites around the world who have worked with me over the years, without whom this paper would not have been possible. I would also like to thank the personnel at Peter Brotherhood, Thomassen, Dresser Rand, Hoerbigger UK, UMP and CPI for their support over the years and encouragement with this paper.

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Durch das in den Muttern (Pos. 7-9) verbliebene Gewindestück wurde ein Feinschliff gelegt.

In der Übersichtsaufnahme (Bild 14) ist die glatte Bruchkante im oberen Bildausschnitt, der minimale Restbruch rechts im Bild an der abgeschrägten Kante erkennbar.

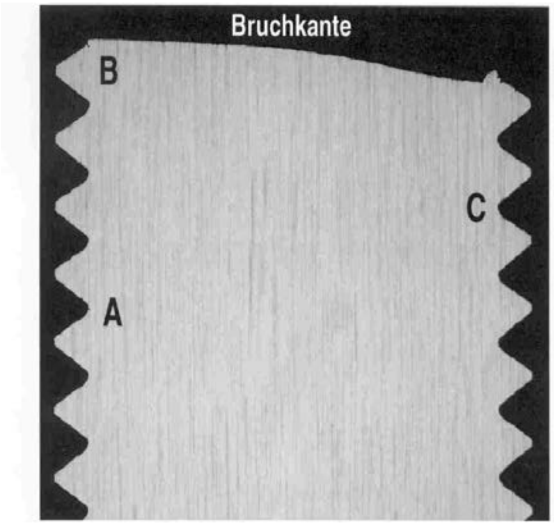

Bild 14 : Feinschliff durch das in den Muttern (Pos. 7-9) verbliebene Gewindestück

An Stelle A wurde eine Detailaufnahme (Bild 15) der Gewindeflanke mit Blickrichtung Gewindegrund aufgenommen.
Auf der Flanke ist Korrosionsangriff (schwarze Punkte) und im Gewindegrund weitere Anrisse erkennbar.

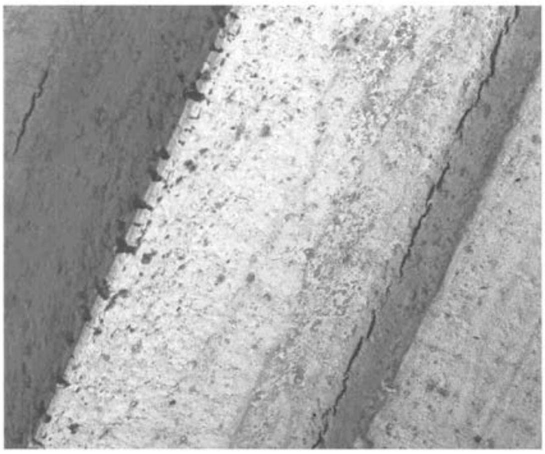

Bild 15: Detailaufnahme (50:1) Anrisse im Gewindegrund

Sowohl im Gewindegrund im Bereich B, als auch unterhalb der Bruchkante im Bereich C sind weitere auskorrodierte Risse zu finden (Bild 16).

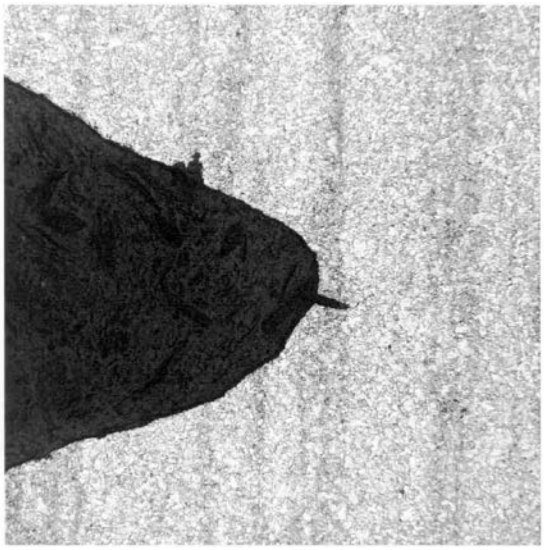

Bild 16 : Stelle B/C (100:1), Anriss im Gewindegrund, Korrosionsangriff im Gewinde

5 Ursachen

Die Schadensbilder und die Ergebnisse der werkstofftechnischen Untersuchungen zeigen, dass der Schaden aus dem Zusammenspiel mehrerer Ereignisse resultiert.

Der Schaden wurde durch einen Schwingungsbruch, ausgehend von dem Gewindegrund der Kolbenstange und einem Schwingungsbruch im Bereich des Einspannbundes ausgelöst.

Beide Bruchstellen haben einen geringen Gewaltbruchanteil. Dies bedeutet, dass nur ein geringes Lastkollektiv wirksam war.

Im Gewinde sind zahlreiche weitere Anrisse und Korrosionsmulden vorhanden. Durch die Wahl des Fertigungsverfahrens - Gewindeschneiden ohne weitere Nachbearbeitung - wird ein Risswachstum begünstigt.

Inwieweit durch die in der Vergangenheit durchgeführte Demontage/Montage der Kolbenmutter das Gewinde möglicherweise geschädigt wurde ist nicht eindeutig feststellbar.

Die Schwingungsbrüche der Kolbenstange im Bereich des Gewindegrundes und der Einspannstelle blieben über lange Zeit ohne nennenswerte Auswirkung.

Ob zuerst das Versagen des vorderen Führungsringes mit nachfolgendem Verklemmen in der Zylinderlaufbüchse zum Abriss der Kolbenmutter führte oder zuerst die Kolbenmutter abriss und dann der vordere Führungsring zerstört wurde ist nicht genau nachweisbar.

Die abgerissene, vor dem Auslasskanal der Druckseite der 6. Stufe mit Führungsringmaterial verklemmte Kolbenmutter bedingte einen rasanten Zylinderinnendruckanstieg.

Aus Abschätzungen und der Interpolation eines vorhandenen p.V-Diagrammes lag zum Versagenszeitpunkt ein Zylinderinnendruck von > 1200 bar vor.

Dieser Druck wirkte auf die Saugventilfläche und führte zum gleichzeitigen Gewaltbruch aller 4 Befestigungsschrauben des Ventildruckstückes.

Die Konstruktion des Ventildruckstückes (Zentrierbund mit zusätzlichen O-Ringen) verhinderte eine vorzeitige Leckage und beeinflusste das Ereignis positiv.

6 Maßnahmen

Zunächst wurde die Konstruktion so geändert, dass der Stufenkolben 6. Stufe aus einem Stück gefertigt wird und die Ringbüchse mit Kolbenmutter entfällt.

Eine Konstruktionsänderung der Einspannstelle der Stufenkolben der 5. und 6. Stufe zur Verbesserung der radialen Beweglichkeit ist im Planungsstadium.

Die Ausrüstung der Maschinen mit einer Kolbenlage- und Schwingungsüberwachung wird vorbereitet um Korrelationen bzw. Maßnahmen mit den bereits elektronisch erfassten Prozessdaten herbeizuführen.

Ergänzend zu den bisher durchgeführten zerstörungsfreien Werkstoffprüfverfahren im Rahmen jeder Maschinenrevision werden an gebauten Stufenkolben oder bei Vorliegen dieses Konstruktionsprinzipes zusätzliche MP- bzw. FE-Prüfungen im Gewinde- und Einspannstellenbereich durchgeführt.

7 Zusammenfassung

Dieses Schadensereignis stellt ohne Zweifel einen Sonderfall dar.

Es wurde aufgezeigt wie das Zusammentreffen mehrerer Ereignisse bedingt durch konstruktive und fertigungstechnische Ausführungen den Schaden auslöste.

Nun gilt es, die aus der engen Zusammenarbeit zwischen Betreiber, Service, Werkstoffprüfung und Hersteller gewonnenen Erkenntnisse auf baugleiche oder bauähnliche Konstruktionsprinzipien konsequent anzuwenden.

Ein Baustein hierbei ist die Anwendung von Werkstoffprüfmethoden (zerstörend, zerstörungsfrei und analytisch) im Rahmen von Neubeschaffungen sowie Triebwerks- und Zylinderrevisionen weiter zu intensivieren und zu verbessern.
So gelingt es frühzeitig Werkstoffschädigungen zu erkennen und Gegenmaßnahmen einzuleiten.

Präzise Spezifikationen, Ausrüstung der Maschinen mit Zustandsüberwachung, Anwendung der geplanten Instandhaltung reduzieren erkennbar die „life cycle- costs“.

Reliability and Economy with Process Controlled Valves (PCV)

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability

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Abstract:

A topical theme in the operation of reciprocating compressors is the reduction in operating costs due to energy saving and increasing the life of the vital components.
These include mainly the compressor valves, which give rise to a high proportion of the energy and maintenance costs.
At the same time it must also be recognised that the principle of the conventional compressor valve, with its closing and damping devices actuated by differences in pressure, has reached the limits of its performance due to the use of new materials and designs.

1 Introduction

The advantages and disadvantages of a valve technology that has been in use for over a hundred years were compiled by systematic analysis, which has led to the principle of the self-acting valve being replaced by a force-actuated valve.
In an age when monitoring systems are used to monitor the condition of reciprocating compressors, it is self-evident that the pressure signals detected

by sensors should be used for a force-actuated control of a valve.
As, with the exception of the flow control of a compressor, the conventional compressor valve is not particularly suitable for this task, a construction was developed that made the implementation of a force-actuated valve possible.
Areas of application such as control and measurement technology and mechatronics have contributed to putting this idea into practice.

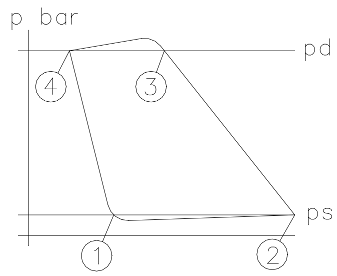
2 Process Controlled Valves (PCV)

2.1 The Principle of a Force-Actuated Valve

Process controlled valves (PCV) differ from conventional valves by the fact that, in contrast to self-actuated control by differences in gas pressure, the valve is subjected to a force-actuated control whereby the data for the control is obtained from the process.
This occurs by using sensors to measure the suction pressure and the discharge pressure as well as the internal cylinder pressure for each compression stage.
By this means, the valve can be operated independently of the position of the crankshaft.

In the suction phase, the differential pressure between the suction pressure and the internal cylinder pressure is calculated until a pressure balance occurs for the first time. This point in time is taken as the opening time for the suction valve, whereby the valve is opened by applying a control signal and the torque of the closing mechanism to be operated is calculated simultaneously.
Subsequently, the opening point is continuously advanced or retarded by a time interval Δt until the torque reaches its minimum value.
The suction valve is closed the last time the pressure balance is reached, whereby the controller operates in the same manner as described above.

In the compression phase, the differential pressure between the discharge pressure and the internal cylinder pressure is calculated until a pressure balance occurs for the first time. This point in time is taken as the opening time for the pressure valve, whereby the valve is opened by applying a control signal and the torque of the closing mechanism to be operated is calculated simultaneously.
Subsequently, the opening point is continuously advanced or retarded by a time interval Δt until the torque reaches its minimum value.
The discharge valve is closed the last time the pressure balance is reached, whereby the controller operates in the same manner as described above.



1 first pressure balance (suction valve)

2 last pressure balance (suction valve)

3 first pressure balance (discharge valve)

4 last pressure balance (discharge valve)

Picture 1: p-V diagram

2.2 The PCV System

2.2.1 Construction of the valve

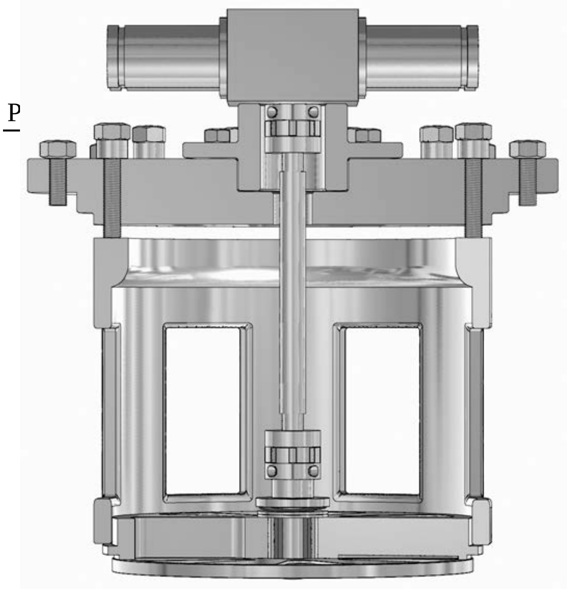
The valve consists of a valve seat with segmented channels, the number of which depends on the speed of the compressor. On the side to be sealed, the channels are equipped with sealing faces, which contribute towards a better seal due to the increased compression surface pressure. Furthermore, contamination from the gas can be wiped off the sealing faces by the rotational movement of the rotary disc.

A rotary disc is used as a closing mechanism with a drive shaft that is guided into the centre of the valve seat. As the valve is subjected to a force-actuated control and the closing mechanism is not subjected to any stroke movement, closing and damping springs as well as the valve guard are no longer required. The rotary disc is moved by a positioning drive, preferably a control cylinder, via a drive shaft.

Picture 2: Suction Valve PCV-System

2.2.2 Choice of material

Essentially, the choice of material is restricted to the valve seat and the rotary disc.



For the seat, a steel is used that meets the requirements dictated by the gas. For the rotary disc, a valve steel is used, which can however be replaced by other materials such as titanium. The use of lighter materials has the advantage of lower masses and thus smaller inertias.

Plastics are rejected as closing elements due to the friction occurring between the valve seat and the rotary valve. This gives rise to the advantage that the rotary disc is not subject to any temperature limitation.

The problems of abrasive media are solved by the use of ceramic coatings, whereby the sealing faces and the rotary disc are subjected to this process. Ceramic materials can be used because the rotary valve is subjected to a rotational movement, unlike in a conventional valve where the closing element is subjected to a stroke movement.

Similar coatings can be used advantageously in the valve seat as anti-sticking effect together with the use of large channels to prevent strong contamination.

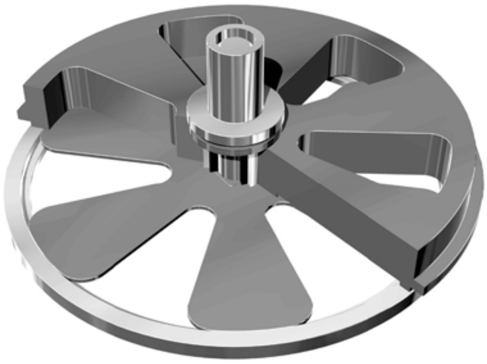
2.2.3 The control of the system

The continuous measurement of the internal cylinder pressure as well as the suction and discharge pressures takes place by means of sensors. These signals are evaluated by a processor for the purpose of controlling the differential pressure. When a pressure balance between the suction pressure and the internal cylinder pressure is reached for the first time, a positioning drive is actuated, which sets the rotary disc of the suction valve into a rotational motion and opens the suction valve. The actuator, consisting of one or more hydraulic control cylinders, is actuated by a fast-switching 3/2 position valve or by a 2/2 piezoelectric valve to release the flow of oil. When a pressure balance is reached for the last time, the positioning drive is actuated once again, as a result of which the suction valve is closed.

The same process is used for regulating the control of the discharge valve as for the control of the suction valve. However, in this case, the discharge pressure and the internal cylinder pressure are used as a basis for the calculation of the pressure balance.

A flow control can be an indispensable component of a compressor. For this purpose, a further programme in the system is activated, which uses the principle of revers flow control. In this regard, after the pressure balance is reached for the last time, the valve is not closed until the part quantity of gas that does not correspond to the delivery volume has been exhausted.

An advantage of using the PCV system is that it is not necessary to apply high operating forces to hold the valve plate open against the force of the flow as is the case with passive, mechanical, revers flow controllers or even with the force-actuated control of conventional suction valves.



Picture. 3: Suction Valve

3 Advantages of a Force-Actuated Valve

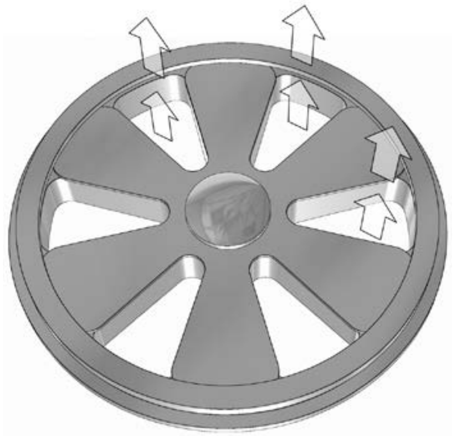
Basically, the PCV system has the advantage that, due to obtaining the control data from the process, every change is detected and can be corrected without any problems via the processor.
By this means, it is possible to operate the valve independently from the molecular weight and pressure of the gas, varying operating conditions and variable speeds.

The operation of the valves at the instant of pressure balance requires little energy.

Due to the construction of the valve, there is a larger effective cross section in comparison with conventional valves. Furthermore, because of the optimal flow geometry, the gas is not deflected but can flow straight past the valve seat, which along with the large flow passages, in the end leads to lower pressure losses. Due to the design of the flow passages, the length of the valve sealing faces is significantly reduced, which in the case of light gasses such as hydrogen leads to lower leakage losses due to the gas reverse flow.

The guardless design is an advantage, which, in the case of suction valves, contributes to a significant reduction in the clearance volume.

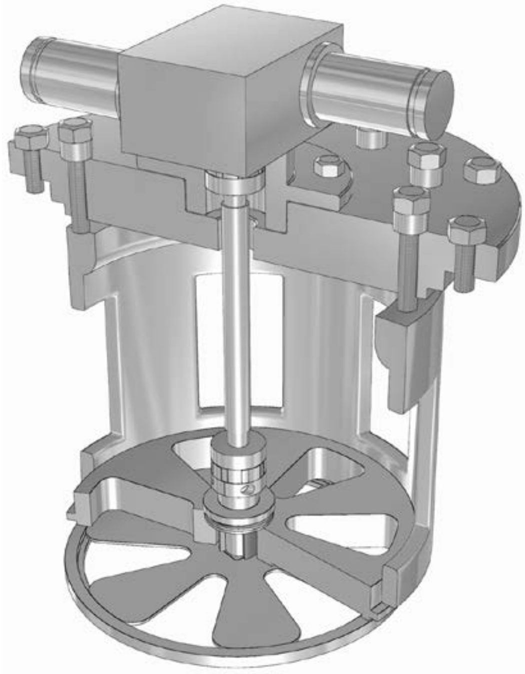
Force-actuated valves does not require the gas to exert any force for the actuation of the valve elements and does not require springs for damping or to support the closing process. Therefore, it is possible to work at low gas speeds even with light gasses as long as no self-cleaning effect is required when using contaminated gasses. The diameter of the valves can be reduced due to the large cross sections and the force-actuated control of the valves.



Picture 4: Suction Valve with optimal flow geometry

Alternatively, valves with effective cross sections could be used for hydrogen compressors, which would be equivalent to a flushing operation with nitrogen and which would not have a detrimental effect on the hydrogen service as a result of the application.

The oil sticking effect is noticeably reduced by the smaller contact surfaces of the valve guard that is no longer present and due to the rotational motion of the rotary disc.



Picture 5: Suction Valve completely installed

4 Possible Applications

Fundamentally, the PCV System can be used in all areas without any restriction on the pressure and type of gas.

The diameters of the valves are limited of approx. 200 mm. In the end, a compromise must be sought between the speed of the compressor and the diameter of the valve.

However, at this point it should be mentioned that PC-Valves with a reduced diameter, in spite of having a smaller effective cross section, exhibit lower pressure losses than conventional valves due to an optimal flow geometry as well as the force-actuation, which makes it advantageous to retrofit them to existing compressors.

Because of the tendency to build compressors with ever-higher speeds, the masses of the moving parts must automatically be reduced. This also has an affect on the valves, which is accommodated by the use of a PCV system.

The PCV system can therefore be used with advantage right from the initial installation.

5 Conclusion

In comparison with a conventional valve, reliability results from the constructional design such as the absence of damping plates and valve springs. The use of metallic materials with ceramic coatings for rotary valves lends advantages over plastic plates with regard to their thermal loading. Furthermore, due to the rotational motion of the rotary disc, a wobble is avoided such as occurs with ring and plate valves.

Economic advantages are achieved with the PCV system on the one hand by the force-actuated control in conjunction with obtaining the control data directly from the process. The design of the passages with optimum flow geometry and the large effective cross section that results from this are in the end a guarantee of better efficiency.

With the concept of the PCV system, we believe we are able to make a contribution to the reliability and economy of reciprocating compressors.

Functional Composits – the Solution for Better Reliability of Compressor Valves

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability

May 17th / 18th, 2001, The Hague

Abstract:

Considering the increasingly high requirements for the efficiency of reciprocating compressor valves one can conclude that there is an increased amount of total strain. Special valve constructions have to be provided that are highly adapted to the conditions in the compressor in order to meet the requirement of increased reliability and service life . Material selection is essential, especially when employing thermoplastic material.

Being the worldwide leading manufacturer of compressor valves , Hoerbiger concentrate continuously on the topic of ’increased reliability’ in key projects, in order to supply valve constructions and materials that are suited for such applications, and on valve design according to the latest state of engineering. The actual potentials for increased reliability and life time derived thereof show real possibilities on the one hand and they also form the basis for new attempts to reach the defined goal of ’increased efficiency and lifetime of reciprocating compressor valves’.

1 Einleitung

Schon seit Jahren finden Kunststoffe für dynamische Dichtelemente in Kompressorarbeitsventilen Anwendung. Aufgrund der hervorragenden Werkstoffeigenschaften setzten sich dabei vor allem kurzfaserverstärkte teilkristalline Thermoplaste gegen metallische Werkstoffe durch. Verstärkte Kunststoffe ermöglichen in diesem Anwendungsbereich die Realisierung einer höheren Kompressoreneffizienz und Zuverlässigkeit [1,2,3] und zeigen gleichzeitig eine Unempfindlichkeit gegen variable Arbeitsbedingungen sowie geringere Lärmemissionen. Aus diesen Gründen werden heute bereits 90% der für die Erstausrüstung vorgesehenen Ventile in der Prozessgas- bzw. Erdgasindustrie mit Kunststoffdichtelementen ausgeliefert.

Je nach Anwendungsfall treten Einsatztemperaturen bis über 200 °C auf, und die Einsatzumgebung kann zwischen Luft und verschiedensten Prozessgasen variieren. Die Beanspruchung resultiert aus der hohen Bewegungsgeschwindigkeit der Ventilplatte oder des Ventilringes zwischen Ventilsitz und Ventilfänger. Das Dichtelement wird bei jedem Ladungswechsel zwischen Sitz und Fänger bewegt, wobei in Abhängigkeit vom Ventilhub in Extremfällen Aufprallgeschwindigkeiten bis zu 5 m/sec gemessen werden können.

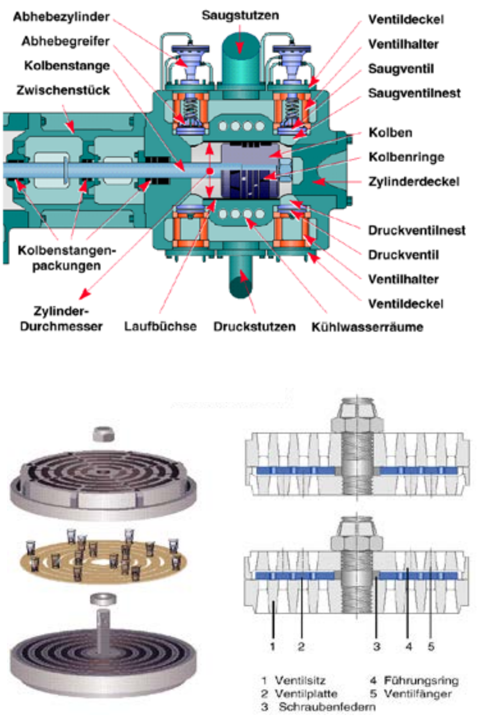


Abbildung 1: Schnitt durch den Zylinder und Ventilaufbau eines Saug- und Druckventiles

Die hohen thermischen, chemischen und mechanischen Anforderungen an den Werkstoff limitieren die Anzahl der einsetzbaren Polymere auf einige kurzfaserverstärkte Hochleistungsthermoplaste. Trotz der hervorragenden Eigenschaftskennwerte der eingesetzten Kunststoffe kann vereinzelt ein vorzeitiges Versagen der Kunststoffdichtelemente beobachtet werden. Eine einfache Abschätzung der Werkstoffbeanspruchung mit Hilfe von analytischen Ansätzen führt zu Spannungen, die nicht für ein Bauteilversagen verantwortlich gemacht werden können. Viele der aufgetretenen Versagensfälle liegen an Stellen, die auf den ersten Blick keiner hohen Belastung ausgesetzt sind. Es ist anzunehmen, dass die Ventildynamik neben den Spannungen im unmittelbaren Kontaktbereich zu hohen Spannungen im Gesamtbauteil führt.

Zur Untersuchung dieser Problematik muss die Belastungssituation in den zyklisch dynamisch belasteten Kunststoffdichtelementen mit Hilfe numerischer Methoden umfassend analysiert werden. Dazu ist es erforderlich, den Aufprall einer Ventilplatte auf den Ventilsitz unter verschiedensten Bedingungen zu simulieren. Bei diesen Untersuchungen dürfen die für diese Werkstoffklasse typischen Materialeigenschaften wie Viskoelastizität und Orthotropie aufgrund der Faserorientierung nicht unberücksichtigt bleiben. Die Forderung nach werkstoffspezifischen Materialdaten für die Simulation setzt eine intensive Untersuchung der verwendeten Werkstoffe in Hinblick auf den Faserorientierungszustand und die Werkstoffeigenschaften voraus. Gleichzeitig muss der Eigenspannungszustand ermittelt und die Auswirkung auf den Belastungszustand in Ventilplatten bewertet werden.

2 Grundlagen

2.1 Bestimmung der Werkstoffkennwerte

Die Eigenschaften kurzfaserverstärkter Werkstoffe werden wesentlich durch die Verstärkungsfasern und deren Orientierung bestimmt. Dies trifft sowohl auf die thermo-elastischen Eigenschaften als auch auf Eigenschaftskennwerte in Hinblick auf die Festigkeit zu. Die Strömungsverhältnisse beim Füllvorgang sind für die typische Faserorientierung in kurzfaserverstärkten Bauteilen verantwortlich. Die resultierende Faserorientierung kann kaum durch Änderung der Verfahrensparameter beeinflusst werden. In Plattenbereichen tritt in der Regel ein dreischichtiger Aufbau auf. In den Randschichten stellt sich eine Faserorientierung in und in der Mittenschicht eine Faserorientierung quer zur Strömungsrichtung ein. Fasern, die in Dickenrichtung orientiert sind, können nur vereinzelt beobachtet werden.

Diesem anisotropen Aufbau und dem damit verbundenen Verhalten muss bei der numerischen Simulation von kurzfaserverstärkten Bauteilen Rechnung getragen werden. Speziell in Hinblick auf eine dynamische Impactsimulation muss das orthotrope Steifigkeitsverhalten Mitberücksichtigung finden.

Der Faserorientierungszustand in den Ventilplatten wird mit Hilfe von Schliffen an mehreren Stellen über den Radius bestimmt. Die Schliffauswertung erfolgt mit Hilfe eines speziell für diesen Anwendungsfall adaptierten Bildanalysesystems.

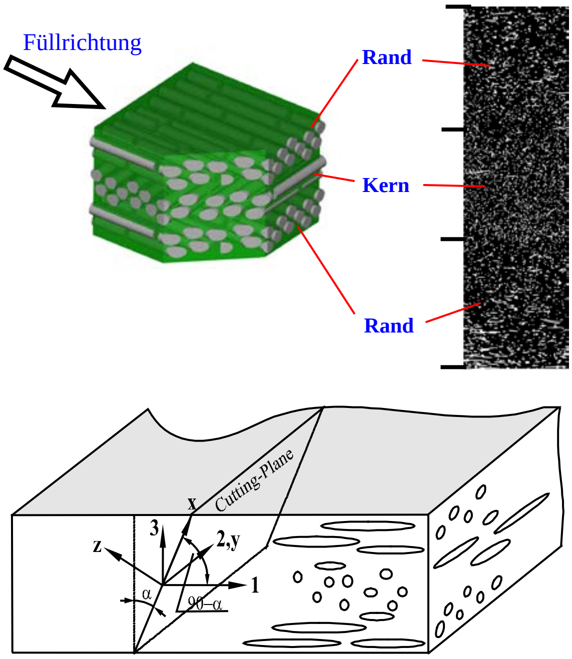


Abbildung 2: Schematische Darstellung der Dreischichtstruktur in kurzfaserverstärkten Bauteilen hergestellt im Spritzgussprozess. REM-Aufnahme eines schrägen Schliffes

Zur Bestimmung der Faserorientierungsverteilung wird die Dicke in acht Schichten unterteilt. Die Beschreibung des gemessenen Faserorientierungszustandes der einzelnen Schichten erfolgt mit Hilfe von sogenannten Orientierungstensoren zweiter und vierter Ordnung [4]. Sie erlauben die Beschreibung der dreidimensionalen Verteilungsfunktion. Die Güte der Beschreibung hängt dabei von der Ordnung des Tensors ab. Die Orientierungstensoren beschreiben die Momente der Orientierungsverteilung. Die Tensoren zweiter und vierter Ordnung ergeben sich wie folgt:

$$\underline{\underline{a}} = a_{ij} = \oint p_i p_j \psi(\underline{p}) d\underline{p}$$

(Gleichung 1)

$$\underline{\underline{\underline{a}}} = a_{ijkl} = \oint p_i p_j p_k p_l \psi(\underline{p}) d\underline{p}$$

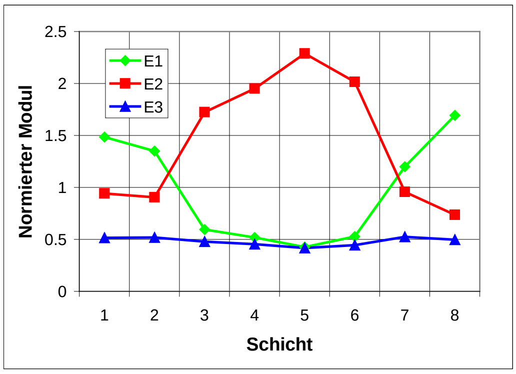
(Gleichung 2)

Mit Hilfe geeigneter Mikromechanikmodelle [4,5,6,7,8] können aus dem Orientierungszustand der Verstärkungsfasern mit Kenntnis der Materialkennwerte der Matrix die anisotropen Eigenschaftskennwerte bestimmt werden. Unter Annahme linear elastischen Materialverhaltens kann die konstitutive Gleichung für das Verbundsystem folgendermaßen angeschrieben werden:

$$\underline{\varepsilon} = \underline{\underline{C}}^V \underline{\sigma} + \underline{\alpha}^V \Delta T$$

(Gleichung 3)

Abbildung 3: Normierter Modul in radialer (E1)



tangentialer (E2) und Dickenrichtung (E3)

In diesem Fall steht C^V für den Nachgiebigkeitstensor vierter Ordnung des Verbundes und ε_{ij} bzw. σ_{ij} für die Verzerrungs- bzw. Spannungstensoren zweiter Ordnung. Als Eingabedaten für das Mikromechanikmodell sind neben den elastischen Materialdaten Faser und Matrix auch Faservolumengehalt, die Fasergeometrie und der Orientierungstensor vierter Ordnung notwendig. Die Grundlagen des Modells basieren auf der Theorie von Maewal und

Dandekar [4] welche in [5] erweitert wurden. Ausgangspunkt dieser Betrachtungen ist die analytische Lösung für eine elliptische Inklusion in einem unendlichen Medium. In Abbildung 3 ist beispielhaft das Ergebnis einer Faserorientierungsmessung und anschließender Mikromechanikmodellbildung einer 6 mm Platte aus dem Werkstoff ‚MT' (POLYAMID + 30%GF) in normierter Form dargestellt.

2.2 Viskoelastisches Materialverhalten

Dynamische Ventildichtkomponenten werden durch den Aufprall auf den Ventilsitz und Fänger belastet. Während des Impactvorganges laufen Spannungs- und Biegewellen durch das Bauteil. Dabei treten hohe Beanspruchungs-geschwindigkeiten auf, womit die Verwendung von belastungsgeschwindigkeitsunabhängigen Material-kennwerten nicht länger zulässig ist. Die Materialien verhalten sich aufgrund der hohen Belastungsfrequenzen deutlich steifer und können nur über die Berücksichtigung ihres viskoelastischen Materialverhaltens entsprechend beschrieben werden.

2.2.1 Allgemein

Kunststoffe unterscheiden sich von den klassischen Werkstoffen hauptsächlich durch ihr spezifisches Deformationsverhalten. Zeigen metallische Werkstoffe bis zu ihrer Fließgrenze meist ein linear-elastisches Materialverhalten, so ist das Verhalten von Kunststoffen auf Grund ihres makromolekularen Aufbaues durch ein hochgradig zeit- und temperaturabhängiges Verhalten geprägt. Um diesem Anspruch Rechnung zu tragen, wurde vor Jahren die technische Mechanik der Polymere entwickelt, die im wesentlichen auf dem Superpositionsprinzip von Boltzmann beruht [9].

$$\gamma(t) = J_0\sigma(t) + \int_{-\infty}^{t} J'(t-\xi)\sigma(\xi)d\xi$$

(Gleichung 4)

Das Verhalten der Kunststoffe ist linear viskoelastisch unter der Voraussetzung von kleinen Deformationen und kleinen Spannungen. Die Grenzen dieser Theorie werden als Linearitätsgrenzen bezeichnet. Als Linearitätsgrenze für PA66 wird von Schwarzl [9] ein Spannungswert von 59 MPa und eine Dehnung von 0.9 % angegeben.

In Hinblick auf die Viskoelastizität ist es wichtig, zwischen amorphen und teilkristallinen Polymeren zu unterscheiden. Im gegebenen Fall kommen aufgrund der Anforderungen nur teilkristalline Kunststoffe zur Anwendung. Leider sind die teilkristallinen Kunststoffe experimentell weniger gut untersucht. Für amorphe Kunststoffe kann man heute eine allgemeine Übersicht über den Zusammenhang zwischen mechanischem Verhalten und Molekularprozessen geben, für die teilkristallinen ist dies wegen ihres komplizierten molekularen Aufbaues noch nicht möglich.

2.2.2 Ermittlung der Relaxationsfunktion aus dem Torsionsschwingversuch

Das viskoelastische Verhalten wird mit Hilfe eines Torsionsschwingversuches an unverstärkten Kunststoffen bestimmt. Als Ergebnis der Messungen steht der Speichermodul G'(w) und der Verlustmodul G''(w) zur Verfügung. Als Eingabe für die Simulation ist im gegebenen Fall der Relaxationsmodul G(t) notwendig. Nach der linear-viskoelasischen Theorie sind die einzelnen Modulverläufe ineinander überführbar wenn bestimmte Voraussetzungen erfüllt werden [9].

Ist der Verlauf des Verlustmoduls als Funktion der Frequenz und der Wert des Speichermoduls bei einer Frequenz bekannt, dann ist es möglich, den Spannungsrelaxationsmodul zu berechnen. Die entsprechende Formel hat die Gestalt: (w = 1/t).

$$G(t) \cong G'(\omega) - aG''(\omega/16) - bG''(\omega/8) - cG''(\omega/4) - dG''(\omega/2) - eG''(\omega) - fG''(2\omega) - gG''(4\omega) - hG''(8\omega)$$

(Gleichung 5)

2.2.3 Zeit-Temperaturverschiebung von mechanischen Eigenschaften

Grundsätzlich ist es kaum möglich den mehrere Zehnerpotenzen umfassenden Zeitbereich, der für die Praxis wichtig ist, mit einem Versuch zu beschreiben. Dazu bedient man sich häufig des Zeit-Temperatur Verschiebungsprinzips.

In der Regel wird dieses Prinzip angewandt, um Aussagen über das Werkstoffverhalten bei Langzeitbelastung zu generieren. Im gegebenen Fall wird die Richtung umgekehrt und von Messungen im Frequenzbereich des Torsionsschwingversuches auf die viskoelastischen Eigenschaften bei sehr hohen Belastungsfrequenzen zu schließen.

2.2.4 Viskoelastische Eigenschaften von Teilchenverbunden

Sämtliche Messungen der viskosen und elastischen Eigenschaften werden an unverstärkten Kunststoffen durchgeführt. Um das viskoelastische Materialverhalten eines orthotropen Werkstoffs meßtechnisch zu bestimmen, wäre es notwendig, alle neun Koeffizienten des Steifigkeitstensors zu messen. Um trotzdem Aufschluß über das viskoelastische Materialverhalten eines orthotropen Materials zu bekommen, wird davon ausgegangen, daß sich nur die Matrix viskoelastisch verhält und die Fasern ein rein elastisches Materialverhalten aufweisen. Der Nachweis für die Zulässigkeit dieser Vorgehensweise wird in [5,10] geführt.

Die charakteristischen Zeiten wie Relaxations- und Retardationszeiten der bestimmenden Materialfunktion des Verbundes sind daher mit denen des Matrixsystems ident. Mit Hilfe der Grenzwerte des Matrixmoduls für t = 0 und t = ∞ kann die Steifigkeitsmatrix für den orthotropen Verbund zu diesen Zeitpunkten bestimmt werden. Die Lage der Relaxationsstufen des Verbundsystems stimmt mit der Position der Relaxationsstufen der Matrix überein. Die Relaxationsstärken für jede Komponente des Steifigkeitstensors des Verbundes werden verhältnismäßig dem Steifigkeitsabfall der Matrix zwischen t = 0 und t = ∞ angepaßt.

2.2.5 Implementierung des Materialmodells in ABAQUS EXPLICIT

Die Größe des zu untersuchenden Modells und die hohe Anzahl an Zeitschritten erfordert die Wahl eines expliziten Berechnungsprogramms zur Analyse des gegebenen Problems. In das Programm ABAQUS EXPLICIT wird ein für den Anwendungsfall maßgeschneidertes orthotrop-viskoelastisches Materialmodell implementiert. Bei der Implementierung muss aufgrund der Modellgröße und Anzahl der Zeitschritte großes Augenmerk auf eine durchlaufzeitoptimierte Programmierung gelegt werden. Die Implementierung eines orthotrop-viskoelastischen Materialmodells in ABAQUS STANDARD wurde schon in [11] durchgeführt. Die mathematischen Beziehungen werden teilweise übernommen und entsprechend den Anforderungen angepasst und erweitert.

Die Orthotropie kann als Spezialfall der allgemeinen Anisotropie betrachtet werden. Nach Tschoegl lässt sich die jeweilige Spannungskomponente mit einem anisotropen Materialgesetz in Integralschreibweise folgendermaßen berechnen [11,12].

$$\sigma_i(\tau) = \sum_{j=1}^{n} \int_0^{\tau} S_{ij}(\tau - \tau') \tfrac{d\varepsilon_j}{d\tau'} d\tau'$$

(Gleichung 6)

Es werden Prony-Serien eingeführt, um die zeitabhängigen Steifigkeiten zu beschreiben. Die Steifigkeiten für t = ∞ und die einzelnen Relaxationsstufen und Relaxationszeiten müssen vorgegeben werden. Damit kann angeschrieben werden:

$$S_{ij} = S_{ij}^{\infty} + \sum_{k=1}^{m} S_{ij}^{k} e^{(\tau' - \tau)/\tau_{ij}^{k}}$$

(Gleichung 7)

Auf eine detaillierte Ausführung der Grundlagen zur Implementierung eines orthotrop-viskoelastischen Materialmodells wird an dieser Stelle verzichtet.

3 FE-Berechnung

Ziel der Analysen ist es, den Belastungsfall „Impact eines Dichtelementes auf den Ventilsitz“ zu simulieren, um Aussagen über die auftretenden Spannungen zu bekommen, entsprechende Einflussgrößen auf das Spannungsniveau zu isolieren, und Konstruktionsrichtlinien sowie Werkstoffspezifikationen für verbesserte Ventile und Ventilwerkstoffe zu ermitteln.

Je nach Ventiltype, Ventilplattenhub und einer Vielzahl von weiteren Einflussfaktoren ergeben sich verschiedenste Auftreffgeschwindigkeiten der Ventilplatte auf den Ventilsitz. Für einen Vergleich wird deshalb standardmäßig eine Aufprallgeschwindigkeit von 3 m/s gewählt.

3.1 Modelle und Materialdaten

Für die Analysen werden zwei verschiedene Modelle verwendet. Ein Querschnittsmodell, welches geeignet ist, die Spannungsverteilung im Sitzbereich zu erfassen und ein Gesamtmodell der Ventilplatte zur Bestimmung der Spannungen in Steg- und Ringsektionen.

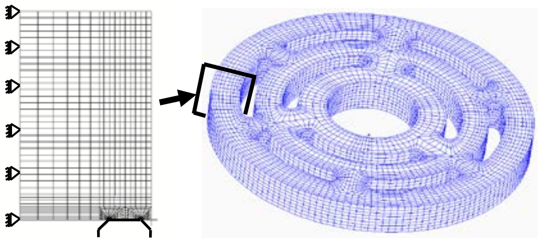


Abbildung 4: Links: Idealisierung des Querschnittmodells unter Ausnützung der Symmetrie. Rechts: Idealisierung der Gesamtventilplatte.

Eine weitere Forderung für die Modellerstellung ergibt sich aus der Tatsache, dass die zu untersuchenden Werkstoffe über die Dicke nicht homogen sind. Kurzfaserverstärkte Werkstoffe zeigen einen für diese Werkstoffe aus der Spritzgussverarbeitung stammende typische Faserorientierung. Um diesen inhomogen Aufbau in die Simulation einfließen zu lassen, wird jedes Modell in Schichten unterteilt. Dabei wird bezugnehmend auf die Materialkennwert-bestimmung eine Schichtenanzahl von acht gewählt. Der Ventilsitz, welcher im Falle der Gesamtventilplatte nicht sichtbar ist, wird als unendlich steif angenommen. Bei der Untersuchung von Kunststoffventilplatten ist diese Vereinfachung ohne weiteres zulässig, wohingegen eine Simulation von Stahlventilplatten unter dieser Voraussetzung zu falschen Aussagen führen würde.

Aus den Torsionschwingungsmessungen im linear-viskoelastischen Bereich bei unterschiedlichen Temperaturen wird unter Voraussetzung der linearviskoelatischen Theorie und thermo-rheologisch einfachem Verhalten, die Relaxationskurve des Schubmoduls errechnet. Diese Kurven werden anschließend entlang der Zeitachse verschoben, um die Masterkurve für eine bestimmte Temperatur zu generieren. Die Verschiebung der einzelnen Kurven erfolgt empirisch ohne Verwendung eines polymerphysikalischen Ansatzes. Die Masterkurve für den Relaxationsmodul bei einer bestimmten Temperatur muss für die Generierung von Eingabedaten für die Simulation durch Prony-Serien ausgedrückt werden. Mit Kenntnis der Faserorientierung und der Schubmoduln G_0 und G_∞ der Matrix kann mit Hilfe der Mikromechanik [4-8] und unter Voraussetzung einer konstanten Querkontraktion über den Temperaturbereich die Steifigkeitsmatrix S_0 und S_∞ berechnet werden. Da die Relaxationszeiten des Verbundes durch die Relaxationszeiten der reinen Matrix [5,10] ersetzt werden können, müssen die Stufenhöhen entsprechend den Relaxationsstufen des unverstärkten Kunststoffes angepasst werden.

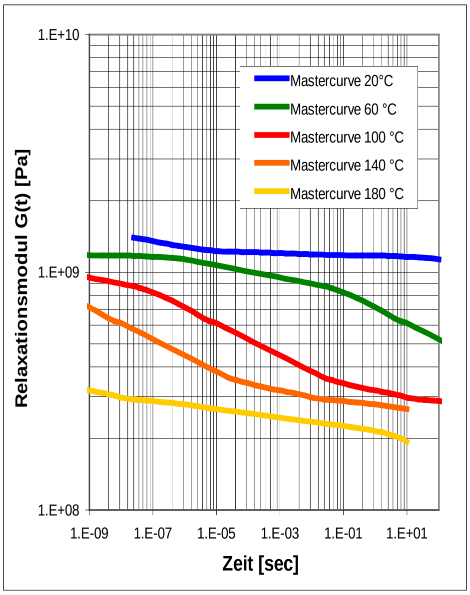


Abbildung 5: Masterkurven für den Relaxationsmodul G(t) von MT im Kurzzeitbereich.

3.2 Ergebnisse

Die Untersuchungen mit dem Querschnittsmodell liefern wichtige Aussagen über die Spannungsverteilungen im Sitzbereich und die Auswirkungen von Einflüssen wie Geometrie, Aufprallgeschwindigkeit und Viskoelastizität. Des weiteren kann eine Optimierung der Sitzleistengeometrie durch ein Anpassung an die Verformung zufolge Impactbelastung durchgeführt werden (siehe Abbildung 6).

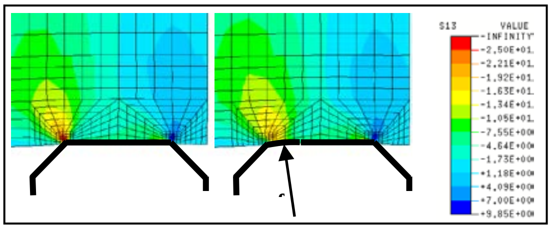


Abbildung 6: Vergleich der auftretenden Schubspannungen beim Impact auf einen Ventilsitz mit Standardgeometrie und optimierter Geometrie. (Platte aus MT, 4 mm, 3 m/s, 20°C)

Mit Hilfe eines Modells der Gesamtventilplatte werden Spannungsverhältnisse in der Ventilplatte außerhalb des Kontaktbereiches analysiert. Neben dem planparallelen Aufprall wird auch der Impact unter kleinen Anstellwinkeln untersucht. Es zeigt sich, dass der planparallele Aufprall auf den Ventilsitz zu keinen nennenswerten Spannungen (ca. 10 MPa) im Steg- und Ringbereich führt. Geht man von einem leicht schrägen Aufprall (0.2° bis 1° Schrägstellung) aus, können wesentlich höhere Spannungen in den kritischen Bereichen ermittelt werden. Für die Untersuchungen wird ein Element aus dem Stegbereich gegenüber der Stelle des ersten Kontaktes gewählt, da in diesem Bereich aufgrund dynamischer Effekte die höchsten Spannungen beobachtet werden können.

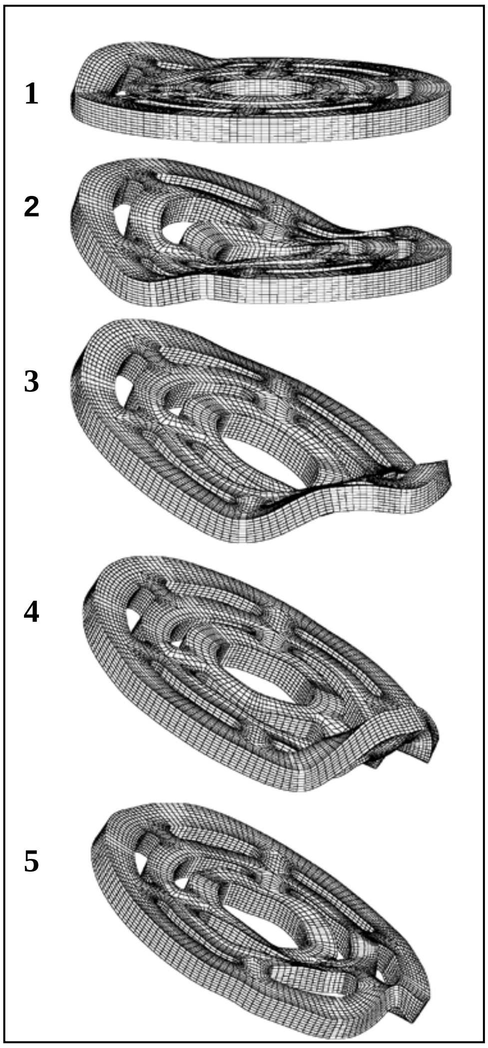


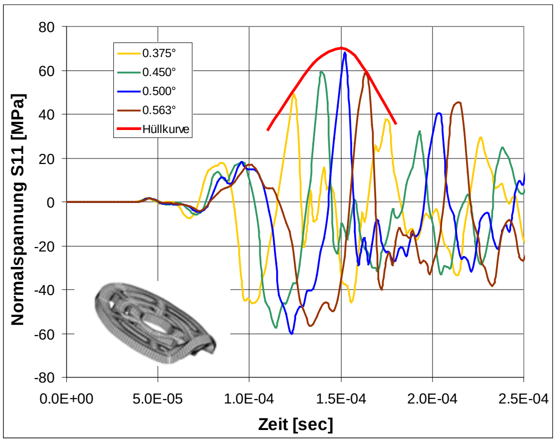
Abbildung 7: Aufprall einer Ventilplatte auf den Sitz (Sitz nicht dargestellt) Verformung 40-fach überzeichnet.

Im Rahmen der durchgeführten Analysen mit unterschiedlichen Aufprallrichtungen kann eine deutliche Abhängigkeit der Spannung im Stegelement vom Anstellwinkel gefunden werden. Unter einem bestimmten Winkel kann bei gleichbleibender Aufprallgeschwindigkeit ein Maximum der Spannung beobachtet werden. Wird ein größerer oder geringerer Winkel gewählt, nimmt die ermittelte Spannung wieder ab (siehe Abbildung 7). Die eventuelle Existenz eines derartigen Effektes wird schon in [13] für Stahlventilplatten überlegt. In diesem Zusammenhang wird die Bezeichnung „Dynamischer Spannungskonzentrationseffekt“ DSKE eingeführt. Es wird postuliert, dass durch die Ventilplatte Spannungswellen mit charakteristischen Geschwindigkeiten laufen. Wenn nun der Kontaktpunkt bei schrägem Aufprall aufgrund der geometrischen Verhältnisse mit der selben Geschwindigkeit voranschreitet wie die Spannungswelle, kommt es zu einem ständigen Energieeintrag in die Wellenfront. Bei sehr kleinen Winkeln schreitet der Kontaktpunkt schneller als die Spannungswelle fort und der DSKE kann sich nicht bilden. Bei größeren Winkeln ergibt sich der Fall, dass die Spannungswelle dem fortschreitenden Kontaktpunkt davoneilt und damit die Bildung des DSKE verhindert. In Abbildung 8 wird die Verformung der Ventilplatte während eines schrägen Aufpralles auf den Ventilsitz unter kritischen Aufprallbedingungen dargestellt. Zur besseren Visualisierung der Problematik wird die Verformung um den Faktor 40 verstärkt dargestellt. Damit erscheint auch der Sitz (nicht dargestellt) um das 40-fache schräger.

Vor dem Kontaktpunkt kann die Bildung einer Biegewelle beobachtet werden. Diese führt zu einer starken Verformung an der gegenüberliegenden Seite des ersten Kontakts. In den Ring- und Stegsektionen der verformten Bereichen können hohe Spannungen beobachtet werden, die in weiterer Folge für Ventilplattenbrüche verantwortlich gemacht werden können. Anschließend werden die ausgelenkten Bereiche gegen den Ventilsitz beschleunigt, wodurch sich mehrfache Aufprallgeschwindigkeiten ergeben können. Dadurch kann es zu einer Schädigung des Werkstoffes an den Kanten der Sitzleiste oder knapp unterhalb der Oberfläche der Ventilplatte kommen. Bei diesem Vorgang wird ein Großteil der elastischen Verformungsenergie in kinetische Energie umgewandelt. Die Auswirkung dieses Effektes ist dermaßen ausgeprägt, dass er sogar in der Energiebilanz für den gesamten Impactvorgang nachgewiesen werden kann.

In Abbildung 8 wird die Vorgehensweise zur Bestimmung der kritischen Aufprallbedingung demonstriert. Es zeigt sich, dass für die gegebene Ventilplatte, Einsatztemperatur und Aufprallgeschwindigkeit unter einem Aufprallwinkel von ca. 0.5° die höchsten Spannungen im Stegelement vorgefunden werden können.

Abbildung 8: Ventilplatte (4 mm) MT bei RT.



Maximale Spannung im Stegbereich in Abhängigkeit von der Zeit unter Variation des Aufprallwinkels.

Eine Abweichung von der kritischen Aufprallbedingung führt jeweils zu einer Verringerung der Beanspruchung. Besonders interessant gestaltet sich die Untersuchung der Einflüsse von Temperatur und Viskoelastizität. Das generelle Spannungsniveau wird stark durch den matrixabhängigen Modul in Dickenrichtung beeinflusst. Die viskoelastische Dämpfung führt speziell bei Temperaturen von ca. 30 bis 70 °C oberhalb der Glasübergangstemperatur zu einer signifikanten Spannungsreduktion.

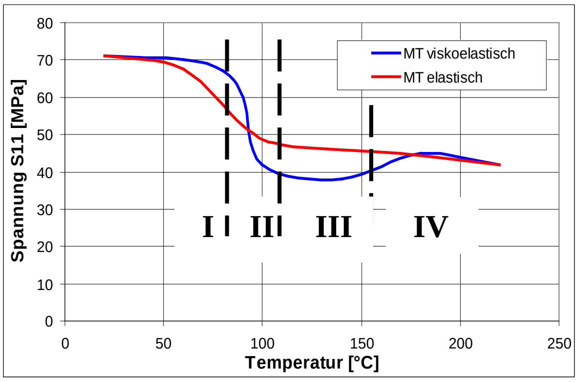


Abbildung 9: Verlauf der maximalen Spannung im Stegbereich beim Aufprall der Ventilplatte unter dem kritischen Winkel und einer Geschwindigkeit von 3 m/sec für MT in Abhängigkeit von der Temperatur.

In Abbbildung 9 werden die Ergebnisse aus der orthotrop-elastischen und der orthotrop-viskoelastischen Simulation gegenübergestellt. Zur besseren Erklärung der Ergebnisse unter Berücksichtigung des viskoelastischen Materialverhaltens wird diese Kurve in vier Abschnitte unterteilt. Abschnitt 1 beschreibt einen Bereich in dem noch keine Dämpfungseffekte wirksam sind. Aufgrund der hohen Belastungsfrequenzen können keine viskoelastischen Vorgänge ablaufen. Erst in Abschnitt 2 kommt man in den Bereich der „Glasübergangstemperatur“ welche durch die hohe Belastungsfrequenz um ca. 20-30 °C nach oben verschoben erscheint. Dabei werden schlagartig starke Dämpfungseffekte wirksam. Dies äußert sich in einem Steilabfall der auftretenden Spannung. Im Abschnitt 3 zeigt sich ein langsames Abklingen der Dämpfungseffekte gepaart mit einer starken Abnahme des Moduls in sämtlichen Richtungen. Damit bleibt das Spannungsniveau niedrig. Erst gegen Ende von Abschnitt 3 und beim Übergang zu Abschnitt 4 klingt die Dämpfungseinwirkung ab und der Modul erreicht annähernd einen konstanten Wert. Dies führt wieder zu einem Anstieg der Spannung. Nach dem Überschreiten eines Maximums kommt es wieder zu einer langsamen Abnahme der Spannung, da der Modul des Werkstoffs vor allem in Dickenrichtung stetig abnimmt und damit das gesamte Spannungsniveau reduziert.

Das differente Verhalten zwischen viskoelastischer und rein elastischer Simulation resultiert aus der Vernachlässigung des Dämpfungsverhaltens und der Nichtberücksichtigung der Steifigkeitserhöhung zufolge der hohen Belastungsfrequenzen. Bei der elastischen Simulation wird im Temperaturbereich von ca. 60 bis 100 °C die Materialsteifigkeit unterschätzt, und im Temperaturbereich von 90 bis 160 °C führt die Vernachlässigung der Dämpfung zu einer Überschätzung des Spannungsniveaus.

4 Zusammenfassung und Ausblick

Kompressorenarbeitsventile werden heute größtenteils mit dynamischen Dichtelementen aus kurzfaserverstärkten Hochleistungsthermoplasten ausgestattet. Trotz der hervorragenden Eigenschaftskennwerte dieser Werkstoffe und der auf den ersten Blick geringen Belastung wird vereinzelt ein vorzeitiges Versagen beobachtet.

Um Aussagen über den wahren Belastungszustand in diesen Bauteilen zu bekommen und gleichzeitig Einflussgrößen auf das Spannungsniveau zu finden, werden die dynamischen Vorgänge beim Aufprall einer Ventilplatte auf den Sitz simuliert. Dazu müssen die orthotropen Materialkennwerte dieser Verbundbauteile bestimmt werden und in die Berechnung Eingang finden. Einsatztemperaturen im Bereich und weit über der Glasstufe und die hohen Beanspruchungsgeschwindigkeiten erfordern die Berücksichtigung des orthotrop-viskoelastischen Materialverhaltens.

Über die Simulation gelingt es erstmalig die Schadensursache von kurzfaserverstärkten Ventilplatten nachzuweisen. Dynamische Biegewelleneffekte beim schrägen Aufprall erzeugen hohe Spannungskonzentrationen in kritischen Bereichen und mehrfache Spannungen gegenüber dem planparallelen Impact. Des weiteren kann die Auswirkung der Viskoelastizität auf das dynamische Verhalten und das Spannungsniveau nachgewiesen werden. Die Vernachlässigung des zeitabhängigen Materialverhaltens führt zu einer Fehleinschätzung der Steifigkeit, und Dämpfung und in weitere Folge zu falschen Aussagen über die Ventildynamik und die auftretenden Spannungen.

Aus den Ergebnissen konnten die Einflussfaktoren und deren Auswirkung auf das Spannungsniveau gefunden, und somit neue Richtlinien in der Werkstoffwahl und Auslegung von Ventildichtelementen aus kurzfaserverstärkten Thermoplasten erstellt und umgesetzt werden. Damit ist ein erster Schritt zur Erhöhung der Zuverlässigkeit von Kompressorventilen gegeben. In weiterer Konsequenz bieten die vorgestellten Methoden und Ergebnisse ein neues Potential zur Modifikationen der Verbundwerkstoffe in der Mikrostruktur sowie zu konstruktiven Änderungen an Ventilen mit dem Ausblick die Zuverlässigkeit von Kompressorventilen einen entscheidenden Schritt in Richtung der Marktanforderungen von zwei Jahren ununterbrochenen Betrieb zu bringen.

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RECIPROCATING COMPRESSORS: RELIABILITY IN DESIGN
A Practical Approach.

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Life Cycle Costs – Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th / 18th, 2001, The Hague

Abstract:

The new European Union Clean Fuels Directive to reduce gasoline and diesel sulphur contents has resulted in the need for increased hydrogen compression capacity at the CONOCO Humber Refinery. Due to the varied and complex process requirements, compression equipment with flexibility, high availability and reliability was a key project objective.

The following technical paper details the practical approach applied to produce a compressor specification that combined end-users experience with new technologies and life cycle cost analysis, to produce a reliable reciprocating compressor installation.

1.0 Introduction

The Humber Refinery is the most complex refinery in Europe (Wood Mackenzie Report-1999), due to the carbon upgrading plant and the interdependency of the process units.

This degree of complexity requires rotating equipment with a high degree of reliability and availability.

In late 1997 the Clean Fuels Program started at the Humber Refinery, to produce low sulphur gasoline and diesel products that comply with the European Union Clean Fuels Directive.

The first phase of this program identified a need for increased hydrogen compression capacity.

Rotating equipment reliability was identified as a key objective for the project and in particular the hydrogen compressor. Increasing the compressor reliability should result in increased plant availability, which would allow the refinery to achieve its vision statement:-

“To be the Safest, Cleanest and Most Profitable European Refinery”.

To meet these requirements a specification was written, aimed at achieving the very ambitious API 618 - 4th Edition target of 3 years uninterrupted service (Ref: 1).

To achieve this goal the specification addressed the following key elements of the compressor design:-

1. RELIABILITY / AVAILABILITY
   The incorporation of the end users operational and maintenance experience, with the lessons learnt from compressor failures and the identification of components that successfully work in the process gas environment, are included in the specification.
2. CONDITION MONITORING / SAFETY
   Machinery protection and health monitoring by the application of proven condition monitoring techniques with new technologies, to produce a safe, reliable and efficient compressor
3. ENERGY / ENVIRONMENTAL
   Capacity control and energy optimisation methods to minimise power consumption, for flexible and efficient plant operation. Environmental control measures to eliminate fugitive emissions, reduce oil usage and contamination.

A reliability based Life Cycle Cost Analysis technique was used for the final selection of the compressor vendor.

The aim of this technical paper is to detail the practical aspects of developing a reliability based specification for a reciprocating compressor. The following text combines operational and maintenance experience with proven and new technologies, to identify the key elements in achieving a reliable compressor installation.

2.0 Reliability Specification

A reciprocating compressor was selected for this application, because it offers the following advantages;-

- The reciprocating compressor provided the necessary flexibility of operating modes that cannot be easily achieved with other types of compressor.
- Individual compression cylinders can be designed for the various process requirements.
- Variations in process conditions can easily be accommodated and have minimal impact on compressor performance.
- There are few aerodynamic limitations to the compressor operating envelope.

In a competitive and cost conscious environment the purchase of a manufacturer’s standard compressor was not considered an option, since it would not incorporate all the reliability features developed from refinery experience.

A detailed, prescriptive reciprocating compressor specification was written based on the American Petroleum Institute Standard 618 - 4th Edition. The specification was compiled using amendments to API 618, which were based on end users experience of the process plants, knowledge of the components which work in that environment and new technologies which improve reliability.

The following parts are key elements in compiling a reliability based machinery specification.

2.1 <u>Process Requirements</u>

The first step before writing a specification is to fully understand the process requirements and the process gas conditions. This is the environment in which the compressor parts must work and knowledge of these conditions is essential in achieving reliability.

The hydrogen compressor process specification detailed the following operating conditions:-

- OPERATING MODES = 5
  Normal, Initial, DHDS, Maximum and Minimum.
- PROCESS REQUIREMENTS = 3
  Booster, Bypass and Recycle Cylinders.
- GAS CONDITIONS = 6
  Normal, High Purity and Low Purity (x 2 Services)

Due to the wide range of operating conditions, it is necessary to identify which of these parameters should be used as the starting point for the compressor design.

Based on this condition the other operating requirements were evaluated, following which any deviations or problems were identified and resolved.

The minimum process requirement and the design condition set the speed range of the compressor. The maximum speed was increased by 25% of the design condition for, the anticipated increased plant throughput, equipment flexibility and the compensation of any deficiency in compressor efficiency; which is an important factor because of high plant integration.

2.2 <u>Gas Composition</u>

The term hydrogen compressor suggests that the compressed gas will have a molecular weight of 2. However hydrogen in a process gas compressor is unlikely to have a high gas purity, consequently it is very important to review the gas molecular weight.
The gas molecular weight for this application during the normal operating mode is 9.14, indicating a complex gas composition. The gas analysis identifies some light ends in the gas stream i.e. ethane, propane and butane.

It is important to review the full gas composition and identify gases, which may have a large influence on the compressor design and material selection. The following are typical examples of process gas problems in a refinery hydrogen system.

Hydrogen (H_2) gas is used in Hydro-Desulphurisation and may contain Hydrogen Sulphide (H_2S), which is an extremely toxic and dangerous material. High levels of H_2S, as defined in NACE Standard MR0175 (Ref. 2) and H_2 may cause stress corrosion and hydrogen induced cracking, which will influence the material selection and fabrication techniques for the compressor and associate process piping and vessels. It should also be considered in the design philosophy for the compressor sealing systems.

Ammonium chlorides are produced in a catalytic reformer when nitrogen in the reformer feed reacts with the hydrogen in the presence of the catalyst to produce ammonia. The ammonia then reacts with the compound used to control the catalyst acid level, to produce Ammonium Chloride (NH_4CL).
The hydrogen off-gas from the catalytic reformer will carry the ammonium chlorides into the hydrogen system.
Ammonium chloride can have a serious affect on plant reliability, because the chloride deposits build up on compressor valves and gas passageways, reducing the flow areas and restricting valve operation.
Chlorides can also have an adverse affect on PEEK based materials used in valves, piston rings, high pressure packing seals etc. As well as compressor valve springs.

Catalytic reformers also produce heavy aromatic compounds due to some hydro cracking. This creates a very small amount of tar, in the reactor effluent. This material, when carried in the gas stream, can settle on compressor valve sealing elements, producing sticktion and premature failure.
Hydro Desulphurisation Units (HDS) can produce Ammonium Sulphide ($NH_4)_2S$ from the combination of ammonia with hydrogen sulphide. This material when mixed with water will disassociate and attack steel.
A liquid separator is fitted in the compressor suction line prior to the compression cylinder, to remove any liquids from the gas stream. The conventional knock-out drum is designed to separate the gas and liquids. The gas will be at its dew point and the liquid will be at its bubble point. Because the gas leaves the separator at its dew point and at a high velocity, a demister is required at the outlet, to prevent any fine moisture particles being carried away in the gas stream.
It is important to prevent any liquid carry over to the compressor, because this can cause severe mechanical damage. The type of problem will depend on the amount of liquid carry over, however typical problems can be compressor valve damage, dilution of cylinder lubricant and with large quantities piston, piston rod and cylinder damage will occur.

2.3 Reliability Factors

Reciprocating compressors are manufactured in many shapes, sizes and speeds. API 618 provides a design profile for the compressor, however to achieve the required level of reliability, specific mechanical features and limitations must be detailed in the specification.
The following are examples of the mechanical parameters, which should be quantified to maximise reliability.

- Piston Velocity Limits.
- Valve Cage and Seat Impact Velocity Limits.
- Materials of Construction.
- Lubricated Cylinders and Packings.
- Piston Rod Coatings.
- Fastener Design and Tightening Method.

Specific vendors and materials which have proven successful at the refinery, should be detailed in the specification, for example:-

- Compressor Valve vendor.
- Piston and Rider Ring vendor.
- High Pressure Packing Box vendor.
- Coupling Vendor

API 618 requires the purchaser to make a number of decisions, highlighted by the bullet points. These decisions were all reviewed against the reliability target for the compressor.

A Failure Modes and Effect Analysis (FMEA) was used with a Cause and Effects diagram to identify potential reliability issues, which have been addressed in this paper.

An assessment of the refinery reciprocating compressor reliability data for the past 20 years, revealed that this information was very similar to the 25,000 hour process compressor survey, as illustrated in figure 1 (Ref: 3). The four main reasons for refinery compressor unscheduled downtime were:-

1. Compressor Valves 36%
2. High Pressure Packings16%
3. Process Problems 14%
4. Piston Rings / Rider Rings. 8%

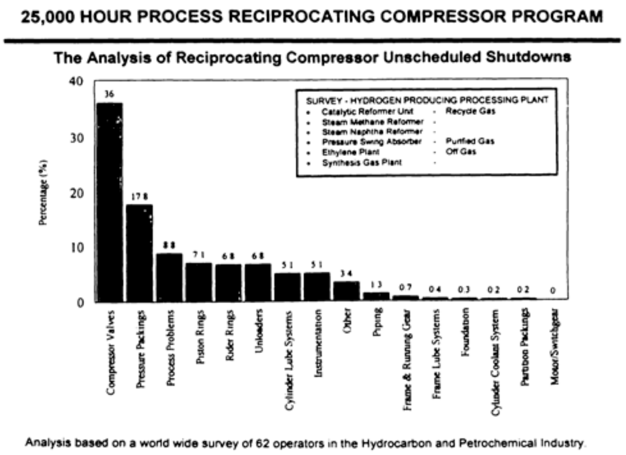

Fig: 1 – Reliability Survey

2.4 Maintainability

One important aspect of mechanical reliability is the maintainability of a machine. Human error in maintenance can account for up to 15% of machine failures. Combining these two aspects produces Availability.

Availability = Reliability x Maintainability

Many failures are not due to component design, but to poor accessibility resulting in incorrect installation or repair.
Consequently the compressor specification must give consideration to the layout and design of individual parts.
If the machine has good access and allows good ergonomics, the result is a safer environment and the probability of incorrect maintenance is minimised.
Areas considered in the maintainability analysis include:-

- Component design which prevents incorrect assembly or fitting.
- Access to lubrication and instrumentation points.
- Sufficient space for a good working posture and maintenance access.
- Labelling and identifying individual components.
- Providing lifting equipment for heavy and large components.
- Detailed procedures for the operation and maintenance of the machine.

3.0 Reliability in Compressor Design

The following section details refinery experiences of compressor failures and improvements, which were used to develop the reliability features, specified for this reciprocating compressor

3.1 Stationary Components

3.1.1 Cylinder Design

A two valve per corner cylinder profile was specified for the compressor. The suction gas passage on this cylinder arrangement allows the suction compressor valve and a port unloader to be fitted in separate housings. If a compressor valve change is required only the suction valve cover has to be removed, which greatly simplifies maintenance and reduces down time. The unloader will not require removal, reinstallation and setting-up.
There are two ports available from the discharge gas passage, however only one discharge valve is fitted per end of cylinder. The remaining port is machined for a valve cover, but not for the additional discharge valve.

Fitting large and heavy discharge valves, which are mounted beneath the cylinder complete with valve cage and cover, can be a difficult task. To ensure that the discharge valve assembly can be fitted correctly first time, a valve fixture was fabricated and tested as part of the factory test procedure. The same fixture was tested onsite to ensure the equipment could still be used in the installed environment, with the additional obstacles of piping, platforms, dampeners etc.
The specification required the compressor valves to be sealed against the cylinder with a valve gasket. This feature greatly reduces any maintenance time and the skill levels required. The alternate design requires the valve to be lapped into the cylinder, to form the seal. Which is a very time consuming and difficult task.

The indirect method of cooling was specified for the cylinder liner and high pressure packing box. Because the liner cooling is through a part of the cylinder wall and not directly on the outside of the liner, this allows easier liner removal. Direct cooling of the liner wall produces deposits, which can build up on the liner, and in the small clearances that locate the liner in the cylinder. This build up creates difficulties, which may require the liner to be removed by machining rather than simple jacking techniques. The deposits may also accelerate corrosion around the sealing areas that may require extensive refurbishment and extended down time during maintenance.
The reasons for the in-direct cooling of the high pressure packing box, are detailed in section 3.4.1.

3.1.2 Cylinder Valve Cage and Covers

There are three important design aspects of the valve cage and covers which should be carefully considered:-

1. Evaluate the valve cover sealing system to ensure a gas tight assembly.
2. Determine the method used to ensure the compressor valve is correctly fitted against the valve gasket.
3. Consider the compressor valve, cover and cage weight, position and method of assembly.

The method specified was an o-ring seal on the combined valve cover and cage. Using this design the compressor valve is tightened onto its gasket, by the cover assembly sliding through the cylinder housing as the cover studs are tightened.
The o-ring seal is used as part of the cylinder helium testing, to check and verify the design.

This method provides a simple design with the minimal amount of components and sources of gas leakage. Alternate designs may use separate cage tightening studs, which require a cap nut and gasket to prevent any process gas leakage. These extra studs are all potential sources of gas release.

3.1.3 Frame

The compressor frame, some times known as the crankcase, is an important element in reliability. The design should eliminate any frame deflection, which would influence the crankshaft alignment and any side wall movement from gas and inertia loading. Sidewall movement could result in cylinder vibration and possible cylinder to frame misalignment.
An analysis of the frame should be considered for the full speed range and loading of the machine.

The frame material of construction was specified for strength and repairability. Previous experience with a damaged crankcase highlighted the need for these two criteria.
Maintenance access to the crankshaft is considered a major requirement following a breakdown, where significant damage was found to a big end bearing journal. The frame should be large enough to allow easy access for main and big end bearing maintenance and allow cleaning inside the frame. Large inspection panels with lifting points are necessary to permit the required access.
The frame explosion relief vents were mounted onto the side of the frame and were fitted with a shroud, to direct any discharge downward.
Positively secure unions and flanges were specified for the lubrication oil piping, to prevent any oil leakage, which may result in a low lubrication oil pressure trip and shutdown.
The frame acts as the lubrication oil reservoir for the machine, the following features are built into the design.

- Oil level sight glass and low oil level alarm.
- A large bore, easily accessible oil filling point.
- Electric Oil Heater removable in situ.

3.1.4 Distance Pieces

The compressor is fitted with Type D compartment distance pieces, as detailed for hydrogen service in API 618 (Ref: 1).
The bolting of the distance pieces to the frame and cylinders was of great concern due to problems with loose and broken bolting on the existing compressors.
In 1996 Rotobolt heads had been retro fitted to a set of the OEM studs and installed to the problem machine.
During the torque loading the Rotobolt indicator cap clearly highlighted that the loading was not uniform and the studs had to be re-tightened a number of times for all the bolts to be at the correct elongation. This method also provided maintenance and operations with a simple but effective bolt tightness monitoring system.

The specification detailed the option of Rotobolt Load Indication or hydraulic bolt tensioning for the distance piece bolting.
Hydraulic bolt tensioning is achieved by stretching the bolt and applying a pre-determined load along the bolt axis. The nut is then screwed down to the opposing face and the external load is released. This leaves a measured tension in the fastener, which is a more accurate, uniform and repeatable bolt tightening method than torque.

Another important feature of the distance piece was the access available through the inspection panels. In the high pressure packing box compartment, the specification required sufficient space to fit the complete packing box through the opening and into its housing, without the need to disassemble of any other part.
This is a very important feature since working in a confined space with a number of parts from a packing box assembly, leads to assembly errors and poor compressor reliability due to unexpected downtime. Because of the importance placed on this requirement, this was specified as a witness test.

3.2 Rotating Components

3.2.1 Piston and Piston Rod

The design and assembly of the piston and the piston rod was carefully evaluated because of maintenance and operational problems. The type of problems had included:-

- Achieving the correct piston nut tightness.
- Achieving assembly concentricity.
- Maintaining assembly tightness during operation.
- Reducing the wear rate of the piston rod.

The original main hydrogen compressors have a conventional single nut and tab washer fastener. The correct torque, detailed by the vendor, was 1960 Nm (1446 lbft.), which requires a very large torque wrench that used a series of lights to indicate when the load was insufficient, correct and excessive. Trials using the torque wrench proved that great care was required when applying a high load to a single nut, since the results were difficult to repeat. Applying torque is very dependent on the friction forces, component materials and thread lubricant.

To prevent problems in achieving the correct piston nut tightness and maintaining this load during operation, the original specification detailed a hydraulic method for securing the piston to rod and piston rod to crosshead. This was intended to use stretch instead of torque as the method of tightening the single nut fastener.
A review of the proposals identified one design, which did not comply with the specification. However after lengthy technical discussions the following method was accepted, because its design achieved the basic requirement while offering additional advantages.

The alternate design accepted was the thread-less piston rod, which as the name implies has no threads on the piston rod for attachment to the piston or crosshead.
In theory this piston rod diameter should be smaller because the stress raisers produced by rolled threads on conventional piston rods, are removed. Analysis of the three proposals, revealed that the thread-less rod design had the largest diameter in relation to the rod loading, resulting in the lowest load per unit area.

The piston is a two-part assembly, with the two halves clamped over the raised section of the rod, using a multi-bolt arrangement, figure: 2.
The piston is tightened together with a series of cap headed screws and secured with a cup shaped tab washer.
The tab washer provides a very positive retention method.
The advantage of the multi-bolt design is that the torque is spread over a larger area than that of a single nut and consequently has lower residual stresses.
The cap screws allow the use of conventional tooling with repeatable loading accuracy.

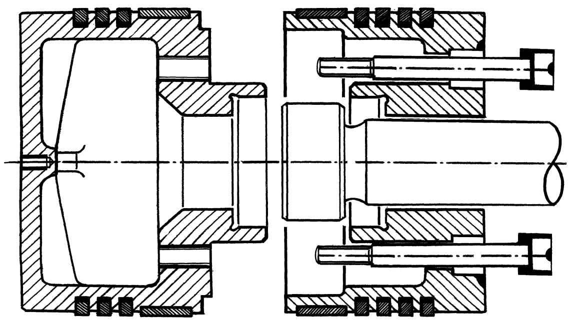

Fig: 2 – Piston Design

The piston rod is secured to the cross head with a solid one piece flange, a split collet and four hydraulically loaded studs, which are secured simultaneously. This method is relatively simple and does not require work in the distance piece with heavy hand tools.
The piston clearance volume is adjusted, by changing the thickness of a washer, which is secured, to the end of the piston rod with a countersunk screw.
The correction of any piston rod runout is detailed in section 3.2.2.

Achieving the run out tolerances of a piston rod assembly has been a major time consuming maintenance problem.
The main difficulty is the concentricity and squareness of the components that make up the rod assembly. When fitting the piston onto the rod, with the spacer and nut, skill, time and effort was required to achieve required tolerances. This problem was compounded by the single nut and tab washer arrangement, which was difficult to tighten and did not provide repeatable measurements.

The design and assembly methods for the piston to the piston rod were reviewed. A specification inspection requirement was to witness the assembly and repeatability of the concentricity.

Piston rods have frequently been replaced during compressor maintenance because of excessive wear, due to a combination of factors including poor lubrication, gas contaminants and the use of hard PEEK based packing materials. To prevent wear and increase the working life of the rod, a wear resistant coating was specified for the rod surface.
The coating and method of application were detailed as a grade of tungsten carbide, applied by the Detonation Spraying method (Praxair D-Gun). Refinery experience has proven this to be a successful combination, with negligible wear after 3-5 years operation.
The surface finish of the piston rod is a critical item for the life of the packings. If the surface finish is too fine this will inhibit the transfer of the packing polymer material to the rod surface. However if the counter surface is too rough this will cause excessive packing wear, very similar to a filing action on the surfaces.

3.2.2 Crosshead

The crosshead converts the rotary motion of the crankshaft to the linear motion required for gas compression. The conventional arrangement is a cross-head that moves in a guide; the guide is part of the distance piece casing. Bolted to the crosshead are white metal coated sliding surfaces, which can be removed for repair or renewal. Fitted between these two surfaces are a series of shims, which are used to adjust the crosshead to guide clearance and correct any piston rod run out.
The removal and adjustment of a crosshead is a difficult task because of the space limitations and access to the components. Care has to be exercised when removing a crosshead from the guide, since the weight and lubricating oil can make this a difficult rigging task.

A review of the compressor exceptions to specification identified one vendor, which had submitted an alternate crosshead design. This proposal was accepted following a design and maintenance assessment, because it complied with the basic requirements while providing additional benefits.
The alternate crosshead has white metal surfaces but they are not removable. This eliminates any possible looseness between the two components. If the crosshead is damaged this can easily be removed through the crankcase. This was checked and verified during the post mechanical test run inspection.

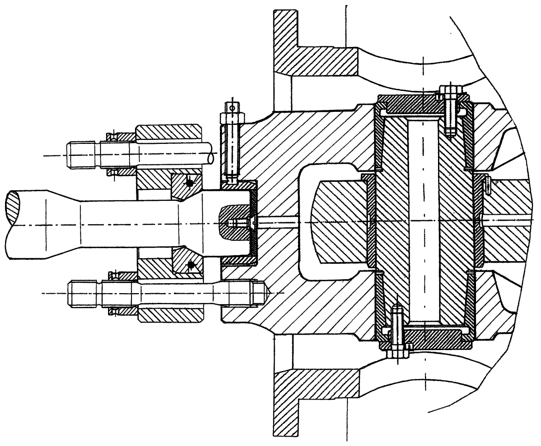

Fig: 3 – Crosshead Design

The end of the piston rod, which is located into the crosshead, is fitted into a steel collar (figure: 3). The piston rod runout is then corrected by moving three equi-spaced set screws on the outside of the crosshead. This is a very simple and effective correction method.
One of the main maintenance concerns has been how to repair a distance piece guide, if it was damaged by the crosshead. One of the features of the alternate crosshead design is a removable tubular crosshead guide. (Fig. 4.), which is secured into the frame by a series of cap screws. In either design the distance piece would require removal but the repair of the tubular crosshead guide is very quick and simple.

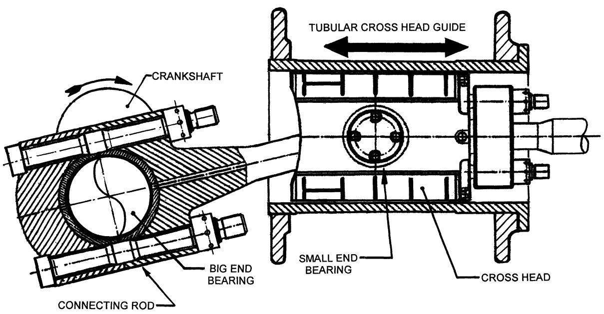

Fig: 4 – Tubular Crosshead Guide

3.2.3 Small End Bearing and Pin

Double acting pistons require a load reversal i.e. from compression to tension, for small end bearing lubrication and reliable operation. A non-reversal condition can be produced during a compressor valve failure. This could result in a rapid bearing failure and seizure of the small end pin.
This type of problem has been identified in many refinery installations. Because this is a very difficult parameter to effectively condition monitor in operation and the machine has a wide operating range, special consideration was given to the small end bearing design and its ability to operate in a non-reversal condition.

It is essential to obtain the correct cylinder loading sequence from the compressor manufacturer, to maintain the correct rod reversal and prevent any dynamic instability.

The removal and re-installation of the small end pin has been a major maintenance problem with the existing refinery compressors. A review of the design proposals identified one design, which was selected, because it offered a number of advantages.
The pin was captive between two plates and secured with a number of set screws and tab washers. Tapered collets locate and position the pin within the connecting rod small end. To remove the pin a simple jacking device is fitted into a slot, machined inside the tapered collet and two set screws are used to push the taper away from the housing and pin. This arrangement was tested following the factory mechanical test run.

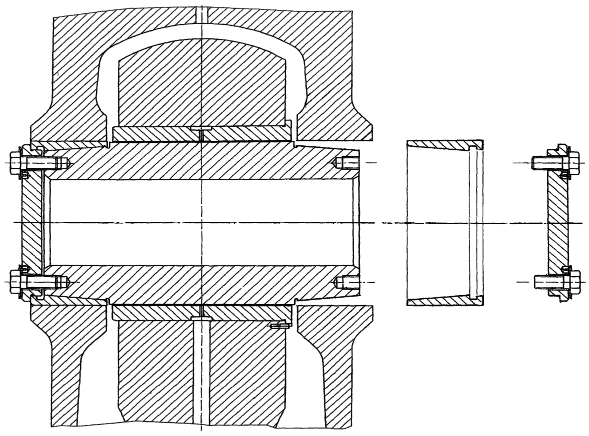

Fig: 5 – Small End Pin

3.3 Compressor Valves and Unloaders

3.3.1 Compressor Valves.

The hydrocarbon processing industry and refinery experience identifies compressor valves as the main reason for poor compressor reliability and plant downtime (Ref: 3).
The refinery hydrogen compressors were originally commissioned in 1969 with plate valves. During the 1980's Ring Type valves with thermoplastic sealing materials were introduced. This change produced a significant improvement in valve reliability, because of improved design and materials technology. However there were still some limitations.

One of the main problems associated with a Ring Type valve is the rapid failure mode. Any problem with a ring was not confined to that area, but soon lead to the whole ring being ineffective. In many instances the valve would fail within minutes of a non identified problem. This leads to very quick, un-planned shutdowns. Because of the plant complexity the consequential losses are very high.

In 1995 Poppet Valves were trialed on a small hydrogen compressor. During the first run the valve life increased from an average of 3 months to 21 months.
The experience with poppet valve failure modes showed that a single poppet would fail and the problem would be isolated to that poppet and not the entire valve. Typically compressor flow would be reduced by 10-20%.
The reduction in flow did not require a rapid plant, shutdown and provided sufficient time for the process units to be adjusted to accommodate a compressor valve change. This is a very important benefit with a highly integrated plant, to minimise the consequential losses.

Fig: 6 – Poppet Valves

Compressor efficiency is a major factor in the valve selection, however this must be balanced against valve reliability. A high efficiency valve that only provides a very limited life in a demanding gas environment will create a very unreliable machine.
The typical poppet valve has a low effective flow area, which increases the pressure drop across the valve, increasing the compression power required and lowering the valve efficiency. To increase the flow area the valve lift must be increased, which can have an adverse affect on reliability.
The compression of hydrogen allows higher gas velocities through the valve, which reduces the required valve lift. As a result the negative effects of a low effective flow area are reduced, increasing the efficiency and reliability of the poppet valve. Because the valve lift has a large influence on valve reliability the guard and seat valve maximum impact velocities were detailed in the specification.
It is important to specify a Dynamic Valve Analysis (DVA) for the compressor valves. The DVA is a method of modelling the valve motion during a full compression cycle and evaluating the valve performance and reliability.
This is of particular importance with the variable speed drive machines, since different compressor speeds will produce different dynamic affects on the valves, which will influence their reliability.

The compressor valve operating environment is the critical factor in valve selection and the main reason for premature valve failure. As detailed in section 2.0, the process gas conditions must be carefully analysed to determine the real gas composition and any other products in the gas stream.
Ammonium chlorides are a particular problem at the Humber Refinery because the high nitrogen content of the naphtha feedstock, from the Thermal Cracker to the catalytic reformer; as detailed in section 2.2.
Examples of this solid deposit and its formation on compressor valves can be seen on figure: 7.

Fig: 7 – Chloride Deposits

The formation of these deposits has a very big impact on valve operational life and machine reliability.

Refinery experience identified that variations in gas temperature have a large influence on valve reliability. The expansion of the sealing ring in a ring type valve should be very similar to the materials used in the manufacture of the seat. Some sealing ring materials have a different coefficient of expansion to the seat, resulting in mis-alignment of the ring and seat during a rapid temperature change. The ring was expanding at a higher rate than the seat, which allowed gas to escape through the narrow gap between the ring inside diameter and the seat. This gas leakage would further increase the local temperature of the seal ring, making the problem worse.
The poppet valve does not suffer with this problem since the poppet expands about its own centreline, without any serious affect on its poppet to seat sealing efficiency.

Valve sticktion arises due to cylinder lubricant passing through the valves, excessive lubricant can result in increased valve impact velocities (Ref: 4).
Refinery experience has shown that while some oil is carried across the valve in the gas stream, cylinder oil drains to the bottom of the cylinder and collects in the discharge valve port of the single valve per corner design.

The free flowing nature of this design should remove the oil accumulation, however the oil tends to accumulate on top of the valve.
The specified two valve per corner cylinder design reduces the affects of this problem, because the angled discharge valves do not allow the accumulation of oil in the valve. Any oil or gas condensate that accumulates, that is forced through the valve will only affect the lower elements of the poppet valve.
The low start up speed of the compressor, because of the motor Variable Frequency Drive, provides a slow sweeping action of the cylinder to displace any liquids that may have collected during the start-up period. The low speed prevents any damage to the compressor internal components.

In a process gas environment there is an expectation that the compressor valves will not last indefinitely. The valve selected has provided increased reliability from previous designs. Because of improvements in component design, materials and technology, it is important to constantly review new compressor valve designs. If new designs are not used there will be no improvement in valve reliability from the present value.

3.3.2 Unloaders

The finger unloaders used at the refinery have provided good service, however there have been problems.
During compressor valve maintenance the finger unloaders must be removed. The re-installation and setting the correct lift etc. has extended the shut down period.
To prevent these problems Port Unloaders were selected for this compressor. This is a simple unloader design, which does not require sensitive adjustment. The unloaders are fitted into their own individual cylinder housing. This removes the need for unloader removal during valve maintenance.

Examination of a number of ring valve failures revealed a problem with the finger unloader. When the unloader is actuated, it pushes the valve ring off its seat. The gas flows in and out of the cylinder through the narrow gap between the ring and seat, creates heat. This raises the temperature of the valve, which can attract “sticky” deposits onto the sealing ring. When the valve is fully loaded the “sticky” deposits cause valve sticktion and premature failure.

Fig: 8 – Ring Valve Failure

The port unloader acts independently of the compressor valve and there is no flow through the valve when the cylinder is unloaded.
Because of the higher flow area created by the port unloader, there is a power saving when the compressor is unloaded.
The unloader is pneumatically actuated, with air to load the cylinder. This method was selected to simplify the unloader design and provide warning off air failure. A manual loading device was included in the event of loading air failure.

3.4 Compressor Wear Parts

3.4.1 High Pressure Packing Box

Industry surveys (Ref: 3) have identified the life of the high pressure packing box i.e. the main sealing element between the compression cylinder and the distance piece, as the second major cause of unscheduled shutdowns for reciprocating compressors.
The Humber Refinery has been using a range of PTFE (Poly-Tetra-Fluoro-Ethylene) and PEEK (Poly-Ether-Ether-Ketone) based packing ring materials for a number of years. The original PTFE material increased packing life to approximately one year. The introduction of PEEK based materials has increased the life of high-pressure packings to 3 - 4 years, in process hydrogen service.
PEEK is a hard material and does not transfer material to the counter surface. It is the fillers within the polymer mix that produce the counter film.
A disadvantage of the PEEK material blends is a high wear rate on the counter surface i.e. piston rod, with little or no significant wear to the packings. Consequently it is important to evaluate the test results and field experience of the packing material to ensure the correct tribological interface. The final material selection should be a compromise to get the best wear from the wear parts, without wearing out the more expensive and difficult to repair components.
A lubricated packing box was specified to increase compressor reliability. The selection of lubricated or non-lubricated packing can be subjective, since both systems can work successfully, but are dependent on a number of

factors. At the refinery lubricated packings have achieved 2.5 times greater life than non-lubricated packing.
The three main reasons for this improvement are:-

1. The lubricant acts as a coolant reducing the packing temperature.
2. The lubrication film reduces the friction between the piston rod and packing rings.
3. The thermo plastic based packings must work in either a wet or dry environment, because a multi phase condition can drastically increases wear. Because the packing is always lubricated, any gas condensate that may enter the box will not produce a multi-phase condition, but only a slight dilution of the oil for a short period.

The reliability of the high pressure packing box is affected by the material selection and the design. The correct installation of the packing box is one of the major factors in achieving reliability.
The distance piece opening and packing box shall be designed to allow the fitting and removal of the high-pressure packing box in one piece. In some existing installations the packing box must be fitted in a number of parts, which has lead to poor reliability due to incorrect assembly.
In-direct annulus cooling was specified for the high pressure packing box. This avoids direct cooling through small inter channel connections, within the packing box, which are prone to blockage. It also removes the need for a series of small o-rings that seal the coolant from the process gas.

The high-pressure packing box is fitted, at the flange end, with two side loaded packings known as WAT rings. These are required for the nitrogen purge system, which provides three functions:-

1. Prevents the leakage of hydrocarbons to the distance piece and through the vent to atmosphere. The reduction of fugitive emissions is an environmental concern.
2. Because the distance piece is purged with nitrogen, this will prevent the formation of gas/oil deposits, which block the atmospheric vent. There have been many occasions when compressors have been removed from service because of high pressures in the distance piece compartment, due to blocked vents.
3. The nitrogen purge system provides a method of monitoring high pressure packing / piston rod wear. As the differential pressure between the vent gas pressure and the nitrogen purge pressure increases, this indicates an increase in packing wear.
   Details of the condition monitoring philosophy of this system are detailed later in section 4.1.4.

3.4.2 Oil Wiper Rings

The oil wiper rings were given special consideration to reduce the amount of crankcase oil used by the compressor. Any oil from the distance piece will be drained to an oily water sewer, reclaimed and fed back to the crude units. Whilst the oil is recycled it is expensive and does require constant monitoring to ensure the correct oil level is maintained.
To reduce crankcase oil usage, which has an impact on the environment, operating costs and reliability, the OT design of oil wiper packing was specified.

3.4.3 Piston Rings and Rider Rings

A PEEK (Poly-Ether-Ether-Ketone) based material was specified for the piston pressure rings and rider rings, due to refinery experience of extended component life using this material. However it is important to assess the material structure to ensure that the counter surface i.e. the cylinder liner, does not suffer from excessive wear due to a poor tribological interface between the two materials. A lubricated cylinder was specified to increase reliability, for the reasons detailed in section 3.4.1.
The introduction of PTFE based materials allowed the fitting of piston rings to be made easy, using a split one piece ring with a scarf joint.
The fitting of a one piece PEEK ring by stretching it over the piston has resulted in a fractured ring because PEEK is less flexible and more brittle than PTFE. Consequently a two piece ring design was used for this application.
However the use of a two piece ring design requires careful consideration, because two ring gaps require sealing. Evaluation of the designs available with the manufacturer identified the lap-joint as the best solution, particularly for the low molecular weight gas in this application..

The need for 3-years uninterrupted service requires a piston ring design that will accommodate a certain amount of wear, while still maintaining compression efficiency. The lap-joint provides two essential features:-

- Accommodate a high degree of radial wear, before the piston ring gap opens completely leading to excessive leakage.
- The sealing efficiency is higher than other joint designs increasing the efficiency of the machine.

The rider ring is again a two-piece design due to the use of a PEEK based material. The rider ring is designed to support the weight of the piston during its stroke. However the forces acting on the piston change during the forward and reverse parts of the stroke. Combining these forces with the weight of the piston, may result in uneven rider ring wear.
The rider ring design must be carefully evaluated to ensure that the ring does not rotate excessively during operation, to prevent damage to the piston grooves. The amount of wear in the grooves will depend on the rider ring material.
Angled pressure relief slots, in the form of a chevron, are machined into the outside diameter of the rider ring, to prevent ring rotation. The slots also prevent any pressure build up at the rider ring, which would cause excessive rider ring wear and also extrusion of the ring from its

groove. The rider rings were not pegged to stop rotation. This was to prevent ring breakage around the peg area, reduce the scarf joint gap and remove the risk of the peg becoming loose in the cylinder.

The position of the pressure and rider rings on the piston should be assessed to maximise reliability and efficiency.
The use of rider rings in the middle of the piston, with the pressure rings at either end does maximise efficiency. However this does need to be evaluated against the weight and size of the piston.

3.5 Compressor Lubrication

3.5.1 Crankcase Lubrication

A separate dedicated lubrication oil skid was specified for the compressor. This was not a fully compliant API 614 system, but a design, which used the main features, for example duplex oil pumps, filters, and coolers.
Traditionally reciprocating compressors have been built with a crankshaft driven lube oil pump and possibly a separate electric motor driven standby pump.
In the event of a main lube oil pump failure, the auxiliary pump will start and maintain the oil supply. However the compressor must be shutdown to repair the crankshaft driven pump. If the main pump is not repaired the compressor is running without an auxiliary pump.
Because of the need to avoid an unnecessary shutdown, to repair a main lube oil pump problem, both pumps are driven by electric motor drives. The electrical supplies for each motor are fed from a separate source.

The lubrication oil skid would use the crankcase as the oil reservoir and not have a separate tank.
An electric oil heater was specified and was designed to allow removal insitu. Electric heating was specified to prevent any possible condensate contamination from steam heating.

Duplex oil coolers were installed, since there had been many problems with the earlier machines, which had only one oil cooler. The coolers were manufactured from stainless steel because of many problems with the original cast iron coolers and corrosion of their cast iron and steel shells, mainly due to the refinery cooling water system.
Duplex oil filters and all lubrication oil pipework were manufactured from stainless steel.

3.5.2 Cylinder and Packing Lubrication

The existing lubricated compressors with metering pumps are all driven from the crankshaft driven via a solid coupling and speed reduction worm box. These items have failed in the past and are subject to regular maintenance checks.
Failure of the drive system or the metering pump would require a compressor shutdown to repair the equipment. Consequently a separate electric motor driven point to point lubrication metering pump system was specified for the compressor. This arrangement allows the electric motor driver and the metering pump to be repaired insitu, without the need for a shutdown.
There is a spare oil distribution point that is connected to a pressure transmitter, with an onward signal to the DCS control system. This activates an alarm should the metering pump fail.

Oil is distributed to the cylinders and high-pressure packing from the metering pump. One major problem with safety implications has been the failure off the non-return valves in the oil feed lines. These failures have resulted in high pressure hydrogen and hydrogen sulphide passing back to the metering pump, resulting in a gas leak.
Because of this problem the metering pump was specified with two non-return valves. One at the metering pump and the other at the end of the feed line, before entering the compressor compartments.
An added feature was an isolation valve, which can only be operated by engaging the drive system with a unique key. This is fitted between the two non-return valves at the metering pump end of the supply line.
Whilst the system has built in redundancy with two non – return valves, if both should fail then the isolation valve can be operated and the non return valve on the metering pump side of the system changed.
The importance of supplying the correct amount of oil to the rings and packings cannot be over emphasised.

3.6 Cooling Water Skid

The correct cooling of a reciprocating compressor is a very important reliability issue.
The cooling water system has to control the cylinder temperature between two main parameters:-

- The cylinder temperature must be maintained above the dew point of the process gas.
- The cylinder temperature should be maintained below a value that would impair compressor performance due to excessive temperature.

A dedicated cooling water system was specified to reduce any fouling in the compressor cooling passages and maintain efficient heat transfer to the coolant, throughout the design operating cycle.
The cooling water skid was similar in design to the lubrication oil skid, except that a separate coolant reservoir was installed. Two centrifugal cooling water pumps were fitted with electric motor drives. The duplex coolers were manufactured from stainless steel as well as the system piping and coolant reservoir. This material was specified to prevent any iron oxide contaminants entering the system and any corrosive affects with the refinery cooling water system.
Because the cylinders can all operate with different gas conditions, it is important to design the system so that each cylinders coolant flow can be regulated to achieve the optimum performance.

An electric heater was specified for the coolant reservoir and was designed to allow removal insitu. The heater is a very important element since the coolant temperature needs to be increased to the minimum temperature requirement i.e. to prevent gas condensation in the cylinders, before starting the compressor.

3.7 Capacity Control

Energy is one of the main operating costs for a refinery. The power required for gas compression must be minimised to reduce gas processing costs.
To reduce operating costs various methods of compressor capacity control were given careful consideration, due to the wide and varied range of operating conditions and gases.

3.7.1 Variable Speed

The main and most efficient method of capacity control is the Electric Motor Variable Frequency Drive (VFD). This form of variable speed drive provides a very flexible operating system, because the speed of rotation is directly proportional to the volumetric flow (Ref: 5).
However because of the different operating scenarios alternate capacity control methods were needed to maintain the correct flow from the various cylinders.

3.7.2 Port Unloaders

Each end of each cylinder was provided with a Port Unloader. This form of capacity control prevents process gas compression at one end of the cylinder, by recycling the gas back to the suction gas passage. It also provides a cylinder with compression steps of 100%, 50% and 0% loading.
Because the gas has not been compressed the amount of energy is reduced. However because there is work done in moving the gas in and out of the cylinder this system is not as energy efficient as speed control.
Section: 3.3.2 details the design of the Port Unloader.

3.7.3 Clearance Pocket.

The fourth cylinder on this compressor is required to meet a number of varied services. Due to the wide variation a Clearance Pocket was added to the head end of the cylinder, which provided an extra 10% reduction in flow.
This method provides a larger clearance volume to reduce the volumetric efficiency of the cylinder. Because of the increased volume, it takes longer for the pressure to increase to discharge pressure. When the valve does open the piston is closer to the end of stroke, resulting in a reduced flow rate.
The clearance pocket actuator was the same as the port unloader. This was for uniformity of spare parts and ease of understanding for machine operation.

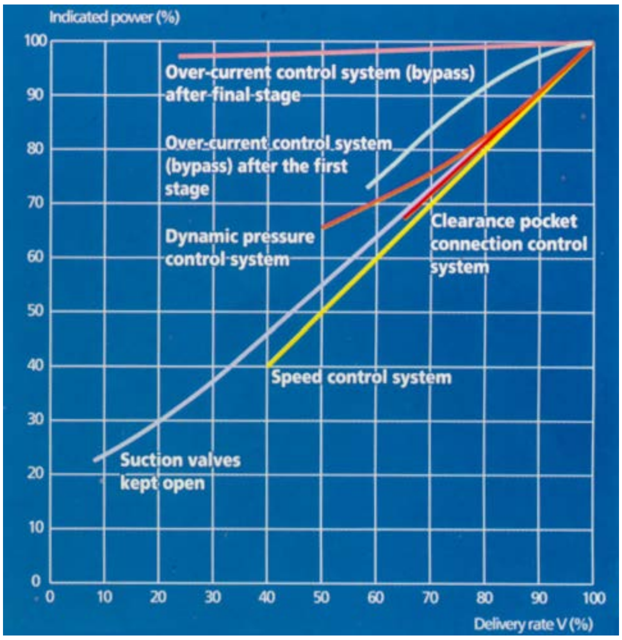

Fig: 9 – Capacity Control Efficiencies

3.7.4 Recycle Lines

In addition to the previous capacity control techniques bypass lines were also fitted around each cylinder to provide gas recycle back to suction.
This is the most inefficient flow control technique and is only used when the alternate methods cannot provide the required conditions.

3.8 Driver Design

3.8.1 Electric Motor Drive

A 3.25 MW induction electric motor with a Variable Frequency Drive (VFD) system was selected for this application, as detailed in section 3.7.1. The alternate synchronous motor was not considered because of the design requirement for slip rings and brushes. These components require frequent maintenance and were not considered suitable for the target of 3 years-uninterrupted service.
The use of a VFD motor drive also removes the need for a speed reducing gear box with i.e. a steam turbine drive. Because a gearbox is not required the system complexity is reduced, increasing reliability; the cycle efficiency is increased because there are no gearbox losses and the space required for the installation is minimised.
A two bearing motor design was required, because a flexible coupling is specified between the compressor and motor. This arrangement has the following advantages: -

- Test run the motor uncoupled from the compressor.
- Electrical isolation from the compressor.
- Facility to adjust the system torsional response.

A two bearing motor is also required because of an unusual phenomena associated with variable frequency driven electric motors. There is the potential for crankshaft damage due to the VFD control system. A flexible coupling is required to isolate the compressor from the driver to remove that potential problem (Ref: 6).

The use of a VFD system has the advantage of no high starting currents, which reduces the temperature rise and stress in the motor windings. A major advantage to this system is that there are no limitations to the number of motor starts. In an integrated process plant, compressor shutdowns due to e.g. high liquid level in a separator may require the compressor to be started quickly. A constant speed direct drive motor is limited to the number of starts permitted within one hour, which can be a major problem.

Horizontal split hydrodynamic white metal bearings were specified for the electric motor. This type of bearing has a high load capacity, ability to handle small axial movement, less sensitive to lubricant contamination and can be removed insitu. An oil ring is used to supply the oil lubricant from the air cooled oil reservoir to the bearing. The oil ring is also split to allow its removal.

The motor was specified for an air / air cooling system because of potential fouling problems using the refinery cooling water system. Using water would have created potential cooler inefficiency and high maintenance costs for an air / water system. The air / air system also reduces the loading on the refinery cooling water circuit.
The condition monitoring package specified for the electric motor, is detailed in section 4.2.

3.8.2 Flexible Coupling

Traditionally reciprocating compressors with an electric motor drive have a single motor bearing arrangement.
The crankshaft drive end bearing carries the load of the motor drive end bearing, through a solid coupling connection to the flywheel.
This configuration simplifies the torsional calculations, reduces the installation space required and reduces the overall cost of the equipment.

In both the upstream and downstream business units there have been torsional vibration problems, with this design of reciprocating compressor installation.
Based on these experiences and the need to avoid damage from the VFD controller, the specification detailed a flexible drive coupling, with the following features:-

- Simple design and assembly.
- Complete coupling assembly to be removable insitu.
- Flexible elements must be removable insitu.
- Adjustable torsional stiffness.
- Fail safe drive system.
- Electrical isolation from the motor drive.

Based on refinery experience the selected coupling flexible elements were designed to operate in compression and not in shear.

Fig: 10 – Coupling Element Damage

This decision was based on damage found to a number of flexible elements, found in various stages of degradation and near to failure after a short period of time. The cause of the coupling element damage was a combination of various factors, including alignment, heat and immediate environment. The type of damage seen is illustrated in figure: 10.

The design and major features of the final coupling selection are illustrated in figure 11. The driven side of the coupling has a spigot location into the flywheel, while the driver side has a spacer section to allow removal of the complete coupling. There are two half covers which permit the flexible elements to be inspected and replaced insitu.

The interlocking design of the two coupling hubs provides the fail safe drive system, in the event of a flexible element failure. This is an essential requirement to avoid a sudden reduction in hydrogen flow, which can cause fires due to the thermal shock of process equipment and the release of hydrocarbons.

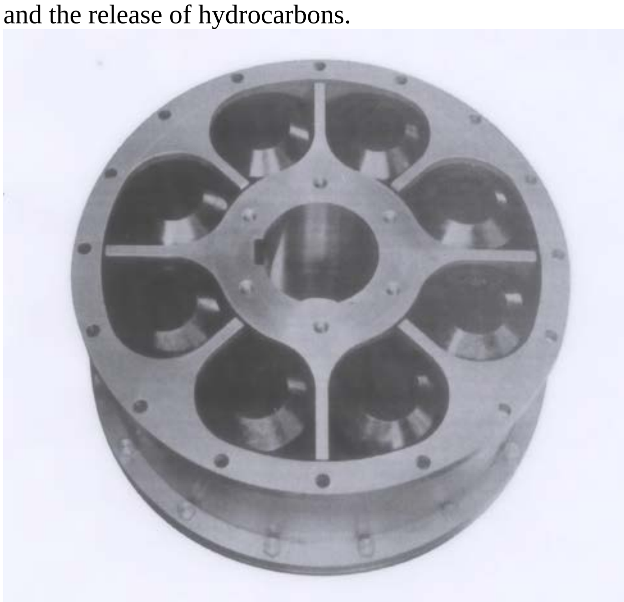

Fig: 11 – Coupling Design

3.9 Accessories

3.9.1 Pulsation Dampeners

Separate suction and discharge pulsation dampeners were specified for each cylinder, because of the various process requirements. A minimum of two access ports was required for internal cleaning and inspection.
A pulsation study was specified in accordance with API 618 Appendix M - Design Approach 3. This is a very detailed analysis and essential with such a wide and varied range of operating conditions and speeds.
Special attention was given to the design and construction of the dampeners, because of the hydrogen, hydrogen sulphide and possible ammonium chloride environment.

3.9.2 Acoustic Study

An acoustic study in accordance with API 618 Appendix N was also specified for the compressor. This study is very important, however it has more emphasis in this application because of the wide and varied range of process conditions and operating speeds.

3.9.3 Noise Control

The compressor train had to comply with the refinery noise standard, of a sound power level no greater than 85 dBA at a distance of one metre from any part of the compressor. Considerable effort was taken to achieve this requirement, to provide a safe and environmentally friendly working area.

3.9.4 Instrumentation

The compressor shutdown systems were kept to a minimum to prevent unnecessary shutdowns. The shutdown systems all used a dual voting arrangement, to further minimise the risk of a false machine shutdown.

4.0. Condition Monitoring

Turbo compressors are generally fitted with a comprehensive package of condition monitoring systems. This comprises of vibration, axial position, temperatures etc. and all usually for one or two shafts.
The condition monitoring for a reciprocating compressor, which has far more moving parts, has in most cases been on a periodic frequency basis. With some continuously monitored parameters i.e. valve temperature measurement.
The difficulty of measuring vibration at the low running speeds has been the interpretation of the signal.

Because of the high criticality rating of the compressor and the need to achieve a 3-year uninterrupted service, a comprehensive condition monitoring package was considered essential for this project.
To provide useful data and real time information, the monitoring systems should be a fixed installation, which checks both the mechanical and thermodynamic performance of the compressor. The analysis of this data should provide detailed information on the health of the machine and used as a basis for maintenance planning and the identification of worn or faulty components.

4.1 Monitoring Systems

There were six separate systems, which were integrated to provide a comprehensive monitoring system.

1. Big End Bearing Temperature Measurements
2. Main Bearing Temperatures Measurements.
3. Suction & Discharge Valve Temperatures.
4. Rod Packing Flow and Temperature.
5. Piston Rod Drop Monitoring.
6. Prognost NT Performance Monitoring

4.1.1 Big End Temperature Monitoring

Following a major failure of a big end bearing and the consequential damage to a main hydrogen compressor crankshaft; a method was required to monitor the condition of the big end bearings during operation. Initially the only systems available were simple reactive indicators i.e. they gave a warning after the problem had started.
Because a continuous monitoring system was required, discussions with a diesel engine instrument manufacturer identified the TB2 Big End Temperature Monitor as the most suitable system for the compressor.
Working with the compressor and instrument vendors the equipment was certified for hazardous area operation and successfully fitted to all four big end bearings.

The TB2 system consists of a double thermistor sensor fitted to the big end bearing. The sensor produces a output which is fed to a moving coil assembly mounted on the bearing cap, which is in the form of a three prong fork. As the crankshaft and big end bearing rotates the moving coil passes the fixed coil of the emitter, which is secured to the crankcase (fig: 13).

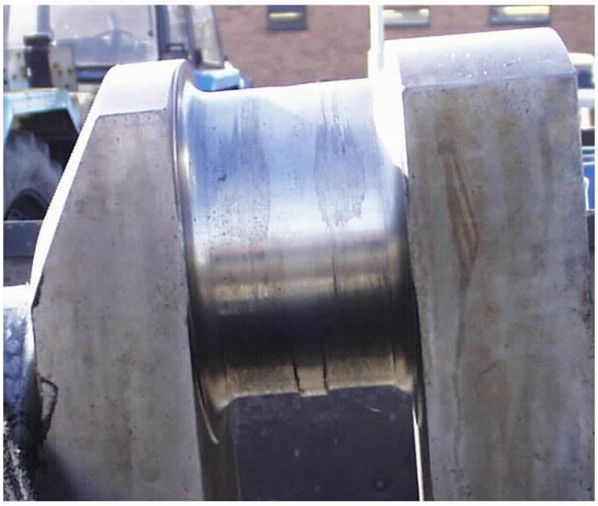

Fig: 12 – Crankshaft Damage

The signal received by the emitter from the moving coil is proportional to the bearing temperature. This signal is then converted and processed in an electronic transducer to provide an on-line temperature measurement. This system does not require any external power supply.
The system was successfully used as part of the factory mechanical run test.

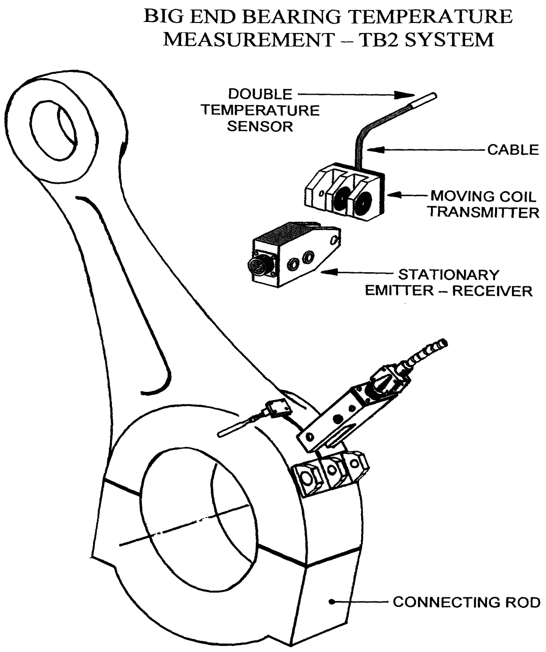

Fig: 13 - Big End Temperature Measurement

4.1.2 Main Bearing Temperature

Due to problems with the main bearings supplied by an OEM, Resistance Temperature Detectors (RTD's) were fitted to all the crankshaft main bearings. This system provides a continuous measurement of the bearing temperatures, which is then used for trend analysis.

4.1.3 Valve Temperature Measurement

Previous experience with suction valve temperature monitoring has been very positive, since any deposits or damage to the valve sealing rings allow gas to pass back to the suction passage, where the thermowell is fitted.
Because the gas is at a higher temperature due to compression and flowing at a high velocity through a narrow gap, the temperature of the gas on the valve suction side will increase. This provides another method if identifying a faulty valve.
This monitoring technique has been extended to both the suction and discharge valves. The Resistance Temperature Detectors (RTD's) are fitted into thermowell's, through the valve covers.

4.1.4 Packing Box – Temperature and Flow

The nitrogen purge system for the piston rod high-pressure packing box, as described in section 3.4.1, provides another condition monitoring technique.
The gas flow and temperature of the packing box leakage, is measured at the high pressure vent. The packing box condition is determined by comparing the nitrogen gas flow into the packing box, with the gas (nitrogen plus process gas) flow going to the vent line. The gas vent line temperature provides another indication, since the process gas leakage will be at a higher temperature than the nitrogen purge gas.

4.1.5 Piston Rod Drop Measurement

A piston rod drop system was specified for each cylinder. The measurement system specified was the proximity probe, which uses an eddy current technology to monitor the difference between the probe and the piston rod. Alternate mechanical and eutectic systems were considered however they are a warning system and do not provide a continuous monitoring output.
The rod drop monitor measures the piston rider ring wear rate and by trending this information can protect against cylinder liner damage, should the piston rub on the liner wall. Excessive rider ring wear can also increase piston rod runout and bending, resulting in high loads on the piston rod to crosshead connection.
Piston rods will flex during operation, this is especially true with long piston rods as used with a Type D distance piece.
The conventional practice would be to take a rod drop measurement at the same point on the piston rod during each crank revolution., which should provide more useful information.
The PROGNOST system uses a segmented analysis technique, where the full analog rod drop signal for one complete stroke is measured and the signal cut into 36 segments each of 10-degree crank angle.
For every segment one average value is calculated and compared with its own related warning threshold.
There is no difference between the "one value" method and the segmented analysis for normal rider ring wear monitoring. However the problems associated with piston

rod scratches, lubricant on the probe or rod are prevented by the segmented analysis technique.
An added advantage to this technique is the possibility of identifying damage to the piston rod to crosshead connection.
It is important to ensure that the rod drop probe is calibrated for the piston rod coating material, to prevent any misleading readings.

4.1.6 Performance Monitoring - PROGNOST

The condition monitoring techniques detailed above, all provide methods of identifying a change and allowing that parameter to be trended. Whilst all these values are important they are not integrated into one system which can provide an on-line performance evaluation.

Periodic performance monitoring had been used at the refinery for a number of years, with limited success. The main problems with the technique were the time required between taking readings and providing a detailed report and accurate interpretation of the results.
Performance monitoring utilising Pressure-Volume diagrams does provide very useful information (fig. 14).
With the target of achieving 3 year uninterrupted service, an On-Line Performance Monitoring system was specified for the compressor. Due to the complexity of the machine dynamics and all the moving parts, the on-line system would receive the inputs from the other condition monitoring sources to provide a comprehensive monitoring system.

The PROGNOST (Prognosis and Glasnost) NT On-Line Performance monitoring system was specified for the compressor, because it provided a very flexible monitoring and data management system for the various monitored parameters.
The basic Prognost-NT analog input system comprises: -

- 8 – Cylinder Pressure Transducers.
- 4 – Cylinder Vibration Accelerometers.
- 4 – Crosshead Vibration Accelerometers.
- 4 – Rod Drop Monitors
- 1 – Once per Revolution Trigger Signal.

One of the key elements for the selection of the PROGNOST system was the continuous measurement of the cylinder internal pressure to produce Pressure Volume (PV) Diagrams. This information combined with the other analog and digital monitored parameters provides a very valuable monitoring system for volumetric and mechanical performance.

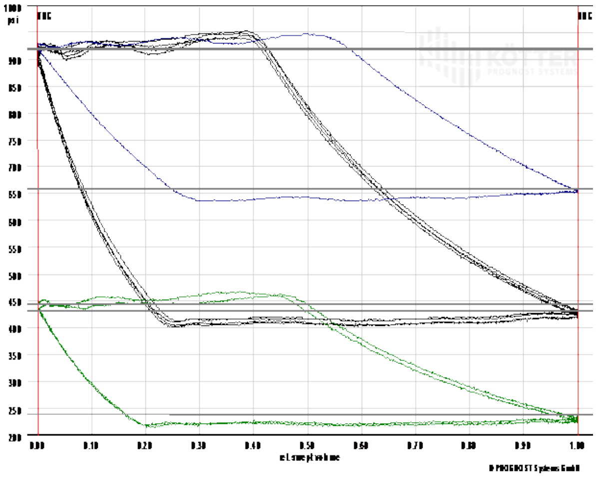

Fig: 14 – PROGNOST P-V Diagram

The PROGNOST- NT software automatically carries out actual data interpretation and analysis. A combination of methods are used to analyse the data, ranging from frequency, time domain, PV and trending analysis.
One example of the way in which the various signals are monitored and analysed, is compressor valve monitoring:-
The basic analog input data produces the PV Diagram, from which the suction and discharge valve losses at the “toe” and “heal” of the curve are calculated.
The digital input signals to the software allow the valve temperatures and cylinder vibration readings to be trended and compared with the analog valve loss calculations.
This technique provides a comprehensive analysis of the machine by using a number of different measurements, to identify the problem area.

An important feature of PROGNOST- NT is the Pattern Recognition software. When an alarm level is reached PROGNOST saves all the instantaneous measured values. When the problem has been identified i.e. piston ring failure, a text message can be added to the measured values.
If another problem occurs the instantaneous measured values are compared with the stored patterns from the database. The patterns consist of different violated warnings and alarm thresholds. The analysis data is continuously stored in the trend and can be compared with the pattern at any time. If the actual and stored patterns are the same a text message will appear identifying the probable failure mode.
As the database increases with experience, a ranking list of failure modes will be displayed on the screen. This form of Expert or Smart technology is ideally suited to this form of process gas compressor.

Reliability of the condition monitoring system is an important consideration, because this data provides the information on the health and general condition of the compressor.

Initial concerns of the expected life of the cylinder pressure transducers were discussed. However after a detailed review this was not considered a problem.
Because a flexible coupling is fitted between the compressor and the driver, a separate once per revolution trigger device was fitted to the compressor crankshaft. This is necessary to ensure that the Pressure-Volume measurements are taken at the correct position of crank rotation. The motor speed output signal was not considered adequate, because of potential changes in angular displacement due to the torque on the flexible coupling elements.
A graphical display of the compressor layout indicates the condition of all the monitoring instruments on the machine, by changing colour from green to red, when they malfunction.

Compressor efficiency has to be maximised to reduce the refinery energy consumption, which has a financial and environmental benefit. The PROGNOST-NT system provides an online power measurement system for the machine and individual cylinders (figure: 15).

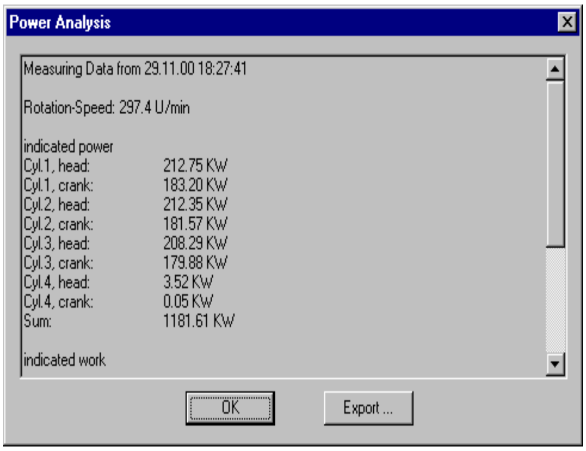

Fig: 15 – Power Display.

4.2 Electric Motor Monitoring

The electric motor drive condition monitoring package consisted of two radial proximity probes at the drive and non-drive end bearings and a once per revolution pulse known as a key phasor.
This system was specified because of the slow machine speed range, excellent diagnostics for plain bearing condition and useful data on the condition of the multi pole rotor design plus the option of shaft crack detection.
The motor bearings were also fitted with a local and remote temperature monitoring system.

5.0 Life Cycle Cost Analysis

The application of a Life Cycle Cost (LCC) analysis to a machine is often subjective and is traditionally based only on power consumption. Because reliability was a primary project objective the following format was developed to apply a LCC analysis to selecting the appropriate compressor vendor.
The life cycle methodology was based on the requirements of API Standard 618 - 4th Edition Section 2.1.1, which states "The equipment (including auxiliaries) covered by this standard shall be designed and constructed for a minimum service life of 20 years and an uninterrupted operation of at least 3 years. It is recognised that this is the system design criterion."
Consequently the analysis was based over a 20 year period, with maintenance frequency of 3 years. The maintenance period of 3 years is in alignment with the refinery turnaround strategy.

The energy consumption of the compressor was based on a 20 year cycle, operating at the "design" conditions.
No time was removed for maintenance periods or changes in compressor operation, in order to normalise the analysis.
The basis for the reciprocating compressor efficiency was evaluated to ensure it was a realistic value.

Lost production costs were based on the consequential losses of the compressor being unavailable for a period outside of its normal planned maintenance period.
The factors which influence the consequential losses for the compressors, were:-

- Availability factor from the compressor vendor.
- Reliability guarantee from the compressor vendor.
- Maintenance schedules from the compressor vendor.
- Component reliability based on the vendor proposal.

The availability and cost of spare parts was the final LCC parameter. If the vendor does not keep some critical spare parts on warehouse stock, this would have to be held by the refinery. Keeping high cost spare parts at the refinery incurs a high cost, not only in initial cost but also non available capital, plus warehouse space and management.

6.0 Conclusions

The reliability of individual compressor components should be reviewed as a system and not in isolation.
It is important to examine how the parts interact with the gas conditions and other components.

Evaluate all technical proposals even if they do not comply with the current standards. The proposal may be a better technical solution – keep an open mind!

Specify the mandatory components, designs and vendors that you know work in your the process gas environment.

Team work and good communication between the end user, manufacturer and if applicable the engineering contractor, is essential.

Attention to detail and a simple but robust machine design produces a reliable compressor.

Ensure that all the “Lessons Learnt” from compressor operation and maintenance are recorded, correctly evaluated and detailed in the specification.

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Shell Global Solutions

Reliability of Reciprocating Compressors in the Petrochemicals Industry

by:

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability

May 17th / 18th, 2001, The Hague

Abstract:

Shell's world-wide operations employ reciprocating compressors in numbers that are conservatively estimated at 5000. The perception of the operators and plant designers is that they are unreliable and maintenance intensive. A review of reliability data obtained from all types of activity suggests an overall MTBF of between 12 and 24 months, but probably nearer to 12. In this respect, reciprocating compressors fall far short of the standards set by other equipment types. Sparing will be required in continuous production units for the forseeable future.

Shell is supportive of new developments and understanding that will help to improve the reliability of these machines.

Introduction

Shell operates reciprocating compressors in every aspect of its operations, including Exploration and Production, Refining, Chemicals, Natural Gas, LNG and others. Machines cover the complete spectrum of size, absorbed power and pressure, from small machines operating at less than 1 bar g discharge pressure, through natural gas reinjection machines with discharge pressures of 400 bar, to hyper machines discharging plastics at 3000 bar. Driver powers from 5 kW to over 12 MW is used. The total number of machines operating on Shell sites is impossible to gauge, but when it is considered that a large refinery may possess 150, chemicals operations perhaps 30 and many more at gas, LNG and E&P sites, it is possible to make a conservative estimate of at least 5000 in Group service.

To an observer from outside the Shell organisation, it might be imagined that the provision of data from Group companies, for the purposes of world-wide benchmarking, is merely a question of asking the relevant engineer. In fact this is far from being the case. Unlike some other petrochemicals companies, Shell is not a single entity but a group of companies, each with its own identity and autonomy. In many cases there are differences in operation, policy, manning levels and equipment that prohibit the gathering of data on specific classes of equipment, or in some cases of any. Computerisation has not led, as might be imagined, to the promised situation in which information can be shared readily between operators, but instead we are faced with the real world of incompatibilities between software, programs installed without a full appreciation of its requirements and incomplete understanding on behalf of its operators. All of these conspire to prevent a true, world-wide comparison of compressor performance.

A further complication is the variation between standards and definitions that exist between the various organisations responsible for the benchmarking terms. Until recently, Group companies in USA, UK, The Netherlands, France, Singapore and South Africa all reported reliability statistics that varied slightly in one or more of their terms. Agreement has been reached that future reporting will adopt the standards of the Process Industry Practices Machinery standard REEE02, Benchmarking of Reliability Indicators for Rotating Machinery, although even in this case, European refineries have been forced to rename the term from Mean Time Between Repair to Mean Time Between Failure in order to avoid conflict with the now almost universal Maintenance Management software, SAP.

Plant reliability

The situation is far more robust when it comes to plant or unit Key Performance Indicators. Detailed records are kept of every refinery unit's performance, enabling lost days' production to be attributed to the responsible equipment category. Failures of rotating equipment are found to contribute 12.3% of overall unplanned lost production. This data is further analysed and broken down to specific rotating equipment machine types, as shown in Table 1.

Table 1. World-wide days per year lost due to unplanned downtime, sorted by equipment category

Description	CCU	CDU	HCU	PFU	TCU	HVU	HDS	HDT	ISOM	LUBE
Pumps	1	43	51	54	85	45	26	74	10	84
Expanders	42		5	3				1	4	3
Compressors	40	36	44	136	38	4	199	39	12	93
Blowers/Fans	39	24	4	22	9	19		8		1
Ejectors					1	1	7			

Further detailed analysis requires additional input, firstly because reciprocating and centrifugal compressors are not differentiated, and because the table takes no account of spared and unspared equipment.

As noted earlier, many locations do not separate compressor types in their Maintenance Management systems, but an estimated proportion of the downtime caused by compressor types is estimated in Table 2 below.

Table 2. Estimated proportion of downtime by equipment type

Description	CCU	CDU	HCU	PFU	TCU	HVU	HDS	HDT	ISOM	LUBE
Compressors	40	36	44	136	38	4	199	39	12	93
Centrifugal %	100	50	70	80	100		80	100	100	80
Reciprocating %	0	50	30	20	0		20	0	0	20

In all of the plants shown, all reciprocating compressors are spared, whereas it is most unusual to spare centrifugal machines. This arrangement has been proved to give acceptable reliability, except in the situation of older refineries in which throughput has been increased over a number of years. De-bottlenecking and re-rates may then allow increased throughput for the centrifugal machine, whereas the low-cost option of using both the running and spare reciprocating machines simultaneously has been adopted. In several cases the result has been very poor plant reliability, with maintenance only possible at a cost of lost production throughput.

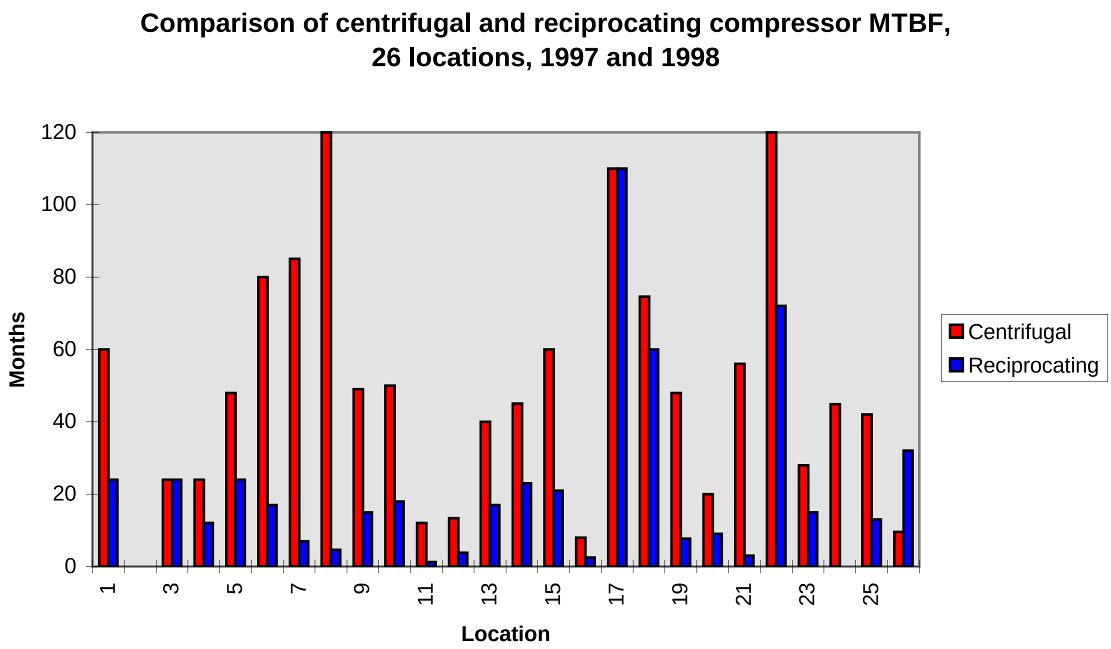
Reciprocating compressor standards

The perception of many Rotating Equipment engineers within the group is that there has been little improvement in the reliability of reciprocating compressors in the past 20 years. Studies were carried out in 1975 and 1997 in which engineers were asked to state their life expectation of refinery reciprocating compressors. In each case the answer was 6 - 12 months. The universal response of project engineers responsible for the construction of new plant has been to achieve reliability by the installation of spare capacity.   The Shell engineering standards, or DEPs, are sometimes criticised for placing additional requirement over and above manufacturers' and industry standards, sometimes with considerable effect upon the purchase cost of the equipment. There is good reason for this. The DEPs are based on life-time costs, in an industry in which even a single failure can wipe out an installation's profit for the year. Purchase cost is not unimportant, as all projects run to strict budgets, but relatively small improvements in the standard product can make dramatic improvements when total cost of ownership is taken into account. The DEP 31.29.40.31 places application boundaries on life limiting components in the machine over and above the requirements of API 617, including speed, both rotational and piston, temperature, cooling, lubrication, pressure and differential pressure.   The selection DEP advises the preferred machine for different duties, in terms of vertical or horizontal, lubricated or dry, installation details, etc. Every clause in the DEPs represents a response to an operational unreliability that has occurred with a machine in service.

Key Performance Indicators

Outside USA there is no regular cross-location comparison of reciprocating compressor reliability within the Shell Group. Indeed, very few sites monitor reciprocating compressor reliability data on a regular basis. In the past few years, data has been gathered by the Rotating Equipment Department in The Hague in preparation for its triennial meetings of RE engineers. This data is currently the best available. In the interests of confidentiality, no locations have been named.

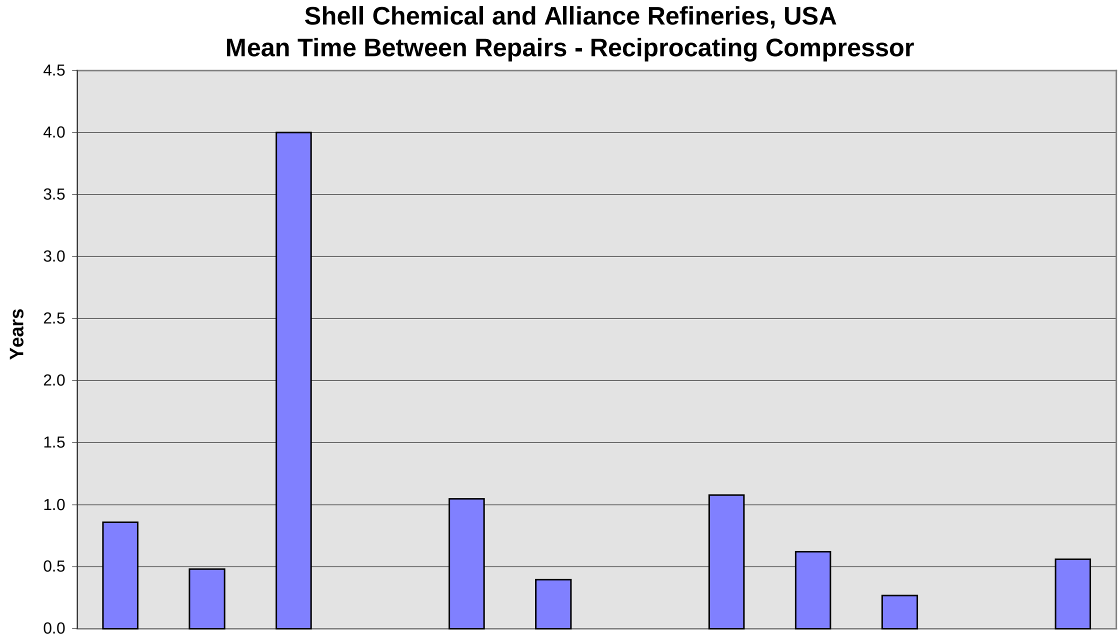
Chart 1. Comparison of Downstream MTBFs, compressors



A number of records need explanation to assist interpretation of Chart 1. No Exploration and Production sites are included. One USA refinery is included. Locations number 7 and 8 are LNG plants, number 24 is a gas plant. Location number 2 is a new refinery for which records are not yet available. Locations 3 and 17 have reported identical figures for centrifugal and reciprocating machines, not because this is the true situation, but because their Maintenance Management system does not differentiate between the two types.

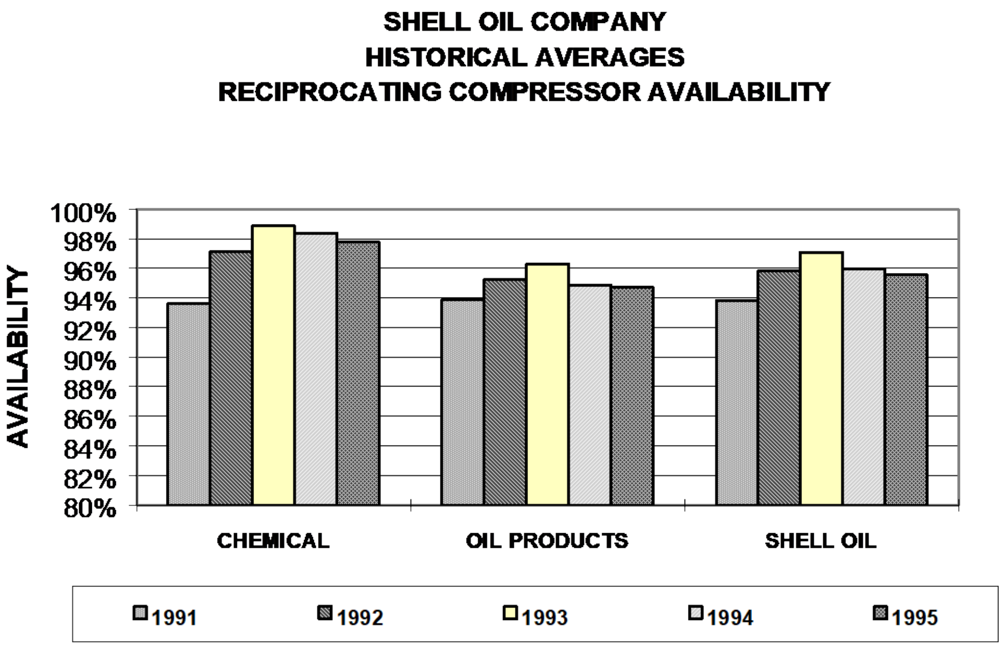
The two outstanding figures for reciprocating compressors are locations 18 and 22. The likely explanation for number 18 is that there is an abnormally high proportion of installed spares at that location. Their reliability records for centrifugal pumps are also the best in the Shell Group, for the same reason. Location 22 almost certainly measures MTBF in a different way from the standard definition, a likely explanation being that planned shutdowns are not included. We are aware that plant shutdowns at a frequency greater than every 6 years are still unusual and the likelihood that the average centrifugal compressors ran for 10 years without stopping is low.

Chart 2. Mean Time Between Repairs, Shell Oil refineries, 1999



Reciprocating compressor reliability is monitored annually in Shell Oil companies in USA. Chart 2 shows Mean Time Between Repairs, measured in years, for 12 locations for 1999. Shell Oil is in alliance with other operators, including Texaco and Aramco, whose data is also included here.

Chart 3. Availability data, reciprocating compressors, Shell Oil, 1991 - 1995



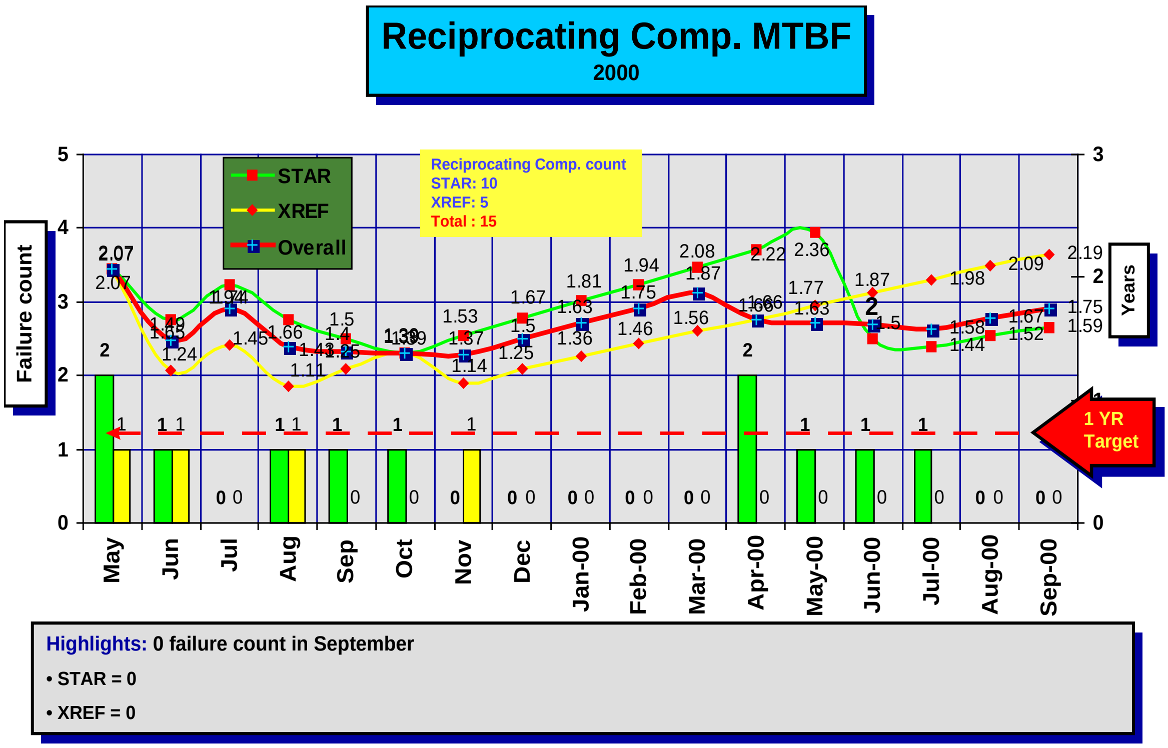
Trended data by year is expressed more commonly in Shell Oil USA refineries in terms of machine availability, where:

$$\text{Availability (Percent)} = \frac{\text{Budgeted Operating Days} - \text{Unavailable Days}}{\text{Budgeted Operating Days}} \times 100\%$$

Again, reciprocating compressors are normally spared, meaning that any availability less than 100% implies that both machines are not functional. For guidance, an availability of 94% implies 22 days of lost production and 99% availability would imply 4 days lost.

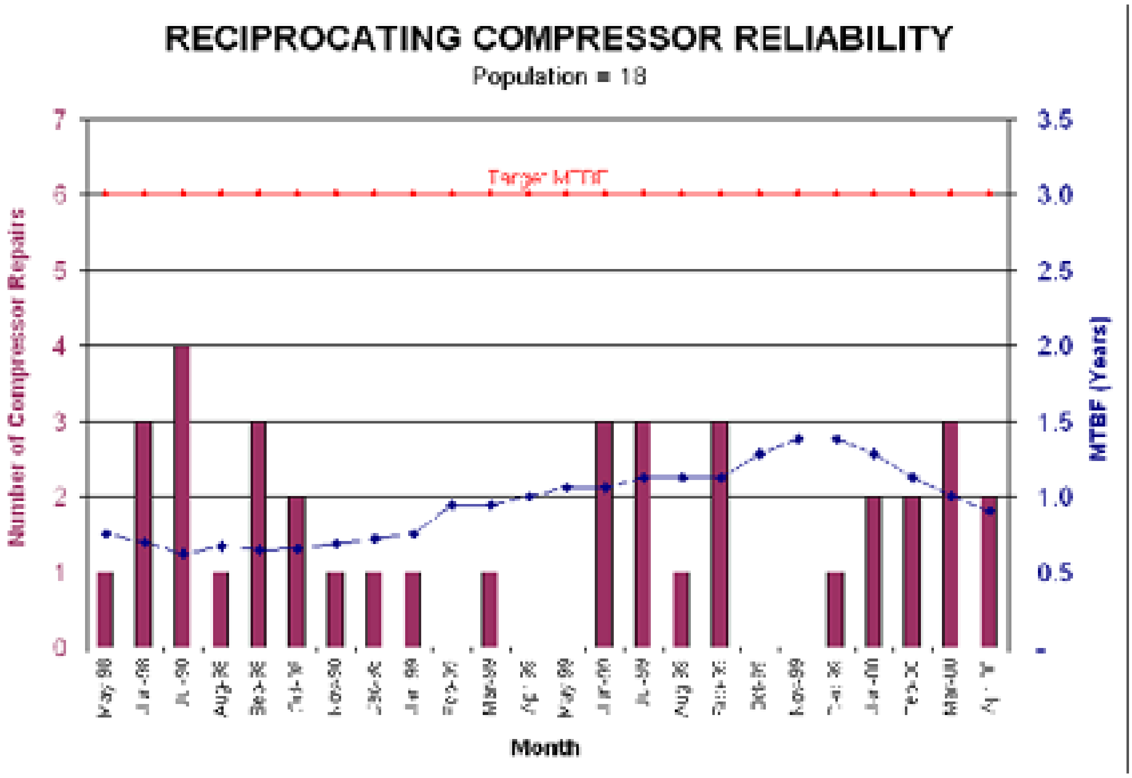
In another location, in the Far East, MTBF figures are superimposed on the total failure numbers for the year 2000. A peak MTBF of 2.07 years in May 1999 has not been sustained, 1.87 being the maximum in the 15 months since then, with an average figure closer to 1.5 years. Shown in Chart 4.

Chart 4. 15 month MTBF and failure rates, Far East refinery



There is little doubt that a major cause of reciprocating compressor unreliability is the quality of the product passing through it. It could therefore be predicted that compression of a very clean gas would lead to noticeable reliability improvements. Liquefied Natural Gas contains neither solids nor aggressive chemicals, so is virtually the purest gas compressed by any petro-chemicals operations. Unfortunately, the reliability of the machine type in this sector is not noticeably better, as demonstrated in Chart 5.

Chart 5. Reciprocating compressor reliability data, LNG plant, East Zone



Very few Exploration and Production companies record reliability data for reciprocating compressors. Some large reinjection and export machines have achieved notoriety due to persistent problems and in these cases their reliability has sometimes been measured in days or weeks, rather than months or years. In most of these cases concentrated, and expensive, effort has been applied to the causes, in terms of design, lubrication, component materials and maintenance methodology, resulting in significant improvement. However, it is a rare Exploration and Production machine that achieves a run life longer than one year.

Maintenance activities

It has been traditional to base reciprocating compressor maintenance on regular inspection intervals, established by experience with the specific machine and the manufacturer's recommendation. In the past it has been quite acceptable to programme inspection intervals annually, for most petro-chemicals plant. The reliability now enjoyed by the majority of rotating and static components in all types of plant, coupled with modern maintenance methodologies, has consigned the annual shutdown to ancient history. Offshore oil platforms now shut down for little more than one week every three years, gas-turbine driven LNG plant life is limited by turbine inspections to a six year life, whilst leading refineries are now planning a shut down only once in eight years. Shell's experience is that reciprocating compressors cannot compete in this arena and sparing must continue for the foreseeable future.

However, the story for recips is not entirely bleak.

The most frequent activity in reciprocating compressor maintenance is, and always has been, replacement or repair of suction and discharge valves. It is no secret that the key to increased valve life is the elimination of liquid and solid contaminants from the gas stream, by the efficient separation and/or filtration of the gas. Lifetime reductions of more than 80% can be expected where contamination is not controlled. The importance of suction piping and dampener cleanliness cannot be overemphasised, not only for new installations, but also for older plant, where corrosion and fouling produced in more than 10 years of operation may have destroyed the spotlessly clean internal piping surface that was created by the chemical cleaning process during pre-commissioning.

Premature valve failure can sometimes be attributed to common suction and inter-stage knock-out vessels and coolers, where piping lay-outs have created layouts which are prone to liquid and/or dirt accumulation. In other cases, maximum utilisation of plant equipment has often caused overloading of existing separation facilities, resulting in liquid carry-over and a consequent dramatic increase in valve failures. The results of these abuses have sometimes been a continuous battle by the maintenance functions to keep machines in operation. Valve lifetimes of 1000 hrs and shorter have been quite common and improvement has generally only been obtained by elimination of the cause of the problem.

Two notable success stories in this battle against contaminants have been the plastic valve plate and, more recently, the split plate design. Plastic plates are now almost universally employed and, with no other change, machine life can often be quadrupled. In a particularly troublesome refinery flare gas compressor, valve life has improved from six or eight weeks to ten months by the substitution of a split ring design, despite being fed molecular weights varying from little over one to 40 or 50, water vapour and significant levels of solids.

Piston rings, rider rings and rod packings, all designed to have a life limited by wear, have been made in a variety of materials during the last 30 years. In both lubricated and non-lubricated services, the life time of these parts depends on a number of parameters, some of which may difficult to predict or understand. It is definitely the case that the most spectacularly short lives of reciprocating compressors in Shell service have been caused by failures of these components, whereas valve failures are generally a persistent but perhaps less frequent occurrence. In many cases the inability of reputable manufacturers to accurately predict the service life of rings and packings, and to provide them to a consistent quality, has been an operational nuisance and a maintenance burden.

During the past 20 years a gradual change has been made from metallic to non-metallic materials. Most early products were based on PTFE materials, with a variety of fillers, for which various manufacturers tried to create a successful all-purpose material. Given design gas properties, counter-face surface roughness and other conditions, standard carbon-reinforced PTFE materials can experience lifetimes in excess of 24,000 hours. Indeed, three machines compressing hydrogen in one refinery are known to have run six years without replacement of piston rings or packings.

Many reported problems are due to variations of operating conditions outside the acceptable envelope. However, causes of excessive wear and premature failure are too often attributed to excessively soft materials, incorrect liner finish, non-uniform liner/ring contact and these result in extreme local temperatures, condensation in cylinders and excessive pressure loading. In several problematic situations, compressor manufacturers have shown insufficient basic knowledge of the prevailing wear mechanism to solve the problem. Specialist suppliers are now the primary source of trouble-shooting services in this field.

Engineering plastics are playing an increasing role in the extension of life in difficult operating conditions. Unfortunately for the user there appears to be little co-ordination or co-operation between the various suppliers of parts in these materials and it remains the responsibility of the operator to offer his production machine as a test bed in the hope that better reliability will result. Shell’s most recent experience is that selection of the optimum plastic, running against a counterface of the ideal material and surface roughness in non-lubricated machines, has given packing and ring life in excess of 34,000 hours. Considerable engineering investment and attention has been required to achieve these results.

A recent trend, much welcomed by Shell Rotating Equipment Department, is investment by the reciprocating compressor industry in a scientific approach to the tribology of the piston ring and pressure packing. Only when the interaction between static parts, wearing parts, gas, contaminants and lubricant is fully understood will solutions be available to the long-running and enduring problems of these machines.

The floating piston concept is innovative and, after some teething problems, is now overcoming several previous instances of poor piston ring and rider band reliability.

Summary

Reciprocating compressors are widely used in every facet of Shell’s operations, from the upstream activities in Exploration and Production, through Refining to the downstream activities of Chemicals and Natural Gas. The reputation of these machines in all sectors of the petro-chemical industry is of unreliability and high maintenance cost. This presentation has shown the true situation to be somewhat better than suggested by Shell RE engineers, but a good distance short of that of the reciprocating compressor manufacturing industry.

Mean Time Between Failures across the whole industry lies somewhere between 12 and 24 months, but probably nearer the lower figure. Higher reliability has sometimes been achieved for critical machines, but always at high cost. Sparing for continuous processes will continue for the forseeable future.

Encouraging developments are taking place in the areas of valves, materials and cylinder assembly design. Of greater importance, it is evident that attempts are being made by the industry to understand the complex mechanisms that exist inside the compressor. Shell supports these initiatives to the fullest extent

Prevention of Pulsation and Vibration Problems in ethylene hypercompressor systems

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EFRC - Life Cycle Costs
Reciprocating Compressors in the Focus of
Function, Economics and Reliability
17th-18th May 2001 - The Hague, The Netherlands

Abstract:

Safety and reliability are important aspects in the design of high-pressure pipe systems for polyethylene plants. The reciprocating hyper-compressors, applied in this kind of processes, often consist of primary and secondary compressors delivering the ethylene to the reactor at pressures up to 300 MPa. Allowable pulsation levels at these conditions are beyond the scope of API 618, which specifies limits for allowable pulsations up to a pressure of 20 MPa. Compressor manufacturers and operators often apply an allowable pulsation level of 10 % peak-to-peak, independent of line size or frequency. In some cases lower levels are specified by operators for specific parts of the layout or in the final discharge stage in the reactor. The API 618 [1] further states that: ‘for systems with a static pressure above this limit the corresponding cyclic stresses should be carefully evaluated’.

According to DSM experience, described in this paper, vibration measurements on-site show that sometimes the higher harmonics in pulsation forces result in vibration levels, which deviate considerably from the levels calculated in a mechanical response study by the compressor manufacturer. For this reason a pulsation analysis should include higher harmonics of compressor speed, up to at least the 25th harmonic, as sufficient pulsation damping is not always possible to prevent strong excitation of mechanical resonances. It is also important that support details including the underlying supporting structure is included in the mechanical response analysis to prevent unrealistic assumptions of infinitely stiff supports for the frequencies concerned.

We emphasize that a review of the piping and support layout in an early stage of the design of the plant is preferable and prevents the necessity of expensive modifications in a later stage. TNO TPD has been involved in hypercompressor pulsation analysis for various operators like DSM, DOW, Shell, Exxon and Basell. This paper highlights some cases in which pulsation and vibration measurements on site have been performed and compared to simulation data. Pulsation and/or a mechanical response analyses are used to find the optimum layout within the constraints of the plant, also in existing systems. Measures to dampen pulsations and shift acoustic resonances are limited, as thick-walled pulsation dampers are not feasible at these high pressures. A number of alternative solutions will be discussed in this paper and will be illustrated via these case studies.

1. Introduction

The presentation of this paper is a joint initiative of TNO TPD Delft as a consultant and DSM Geleen as an operator. We share our experience and common interest in the field of prevention of pulsation and vibration problems in high-pressure ethylene plants obtained from different points of view. DSM will show their experience in LDPE plants and in continuation TNOTPD will give some general remarks obtained from working together in preventing and solving practical problems with various operators. Finally we summarize a number of alternatives to solve pulsation and vibration problems by means of ‘tailor-made solutions’, well fitted to the problem concerned.

An example of the size of a hypercompressor for low-density polyethylene production is shown in fig. 1.1. It is the world’s largest reciprocating compressor with respect to power consumption 21 MW, located at Basell Aubette in France. Typical is also the layout of the large radius bends without pulsation damper volumes on either side of the compressor. The capacity of the compressor is 116.000 kg/h compressing from 28 MPa to 310 MPa discharge pressure.

Fig. 1.1 Ethylene Hypercompressor 21 MW at Basell Aubette

2. DSM LDPE Plants

2.1 Introduction

DSM Service DSM-Polyethylenes, part of the business group DSM Petrochemicals produces a variety of polyethylene using several technologies.
The polymerization of ethylene for low-density polyethylene production is achieved by compressing the ethylene gas to a pressure ranging between 200 and 300 MPa.

This pressure is reached by the use of reciprocating hyper compressors that often consist of a primary compressor (final pressure 20 MPa) and a secondary compressor (final pressure 200- 300 MPa). This high pressure reciprocating compressors require special design features to achieve reliable and safe operation.

2.2 DSM recommendations in relation to pulsations and vibrations

Vibration and pulsation phenomena are typical for reciprocating equipment and related piping, also under normal operating conditions. During the design and construction stage a lot of care is taken to limit vibrations and pulsations occurring during operation.
For these systems a pulsation study and a mechanical response study according API618, design approach 3 is carried out in order to predict pulsation and vibration levels i.e. stress levels for all operating conditions. For an evaluation of the calculated pulsation levels in the piping API 618 requires that the peak-to-peak pulsation level of each individual pulsation component is limited to that calculated by equation (1)

$$P_1(\%) = \frac{397.1}{\sqrt{P_l . ID . f}} \qquad \text{(Eq. 1)}$$

Where:
$P_1(\%)$ = maximum allowable peak-to-peak level of individual pulsation components as a percentage of average absolute line pressure.
P_1 = average absolute line pressure in bar
ID = inside diameter of line pipe in millimeters
f = pulsation frequency in Hz

However this equation is only applicable to systems at absolute line pressures between 0.35 and 20 MPa. For systems operating at higher pressures, such as secondary compressor systems, pulsation levels will be higher because the absence of pulsation dampers. For these cases API 618 requires a careful evaluation of the calculated cyclic stresses.
For secondary compressors DSM apply additional recommendations in relation to vibrations and pulsations. Criteria are used that are based on experiences from the past with other high-pressure systems. DSM defined criteria for different parts of the piping system, i.e. for suction, interstage and discharge piping. These deviations from the allowable values calculated with eq. 1 can be tolerated. The values calculated with equation 1 can

be multiplied with a factor according to column 3 in the table 2.2.1 below. Indicative values based on DSM experience are shown in column 2 of the table.

Location	Puls. [%]	Factor
Manifold before first stage	8	5
Suction pipe before first stage	10	5
Discharge pipe after first stage	10	1.5
Interstage piping	5	5
Suction pipe before second stage	7	1.5
Discharge pipe after second stage	4	4
Manifold after second stage	2	3
In the reactor	1	1.5

*Table 2.2.1. Pulsation levels (DSM experience) and Factor*Equation 1 (API 618) for different parts*

The allowable mechanical vibrations are specified in terms of dynamic displacement amplitude (μm 0 to peak). The vibration must preferably be lower than $800/\sqrt{f}$ (f = frequency of the vibration)

With respect to the pulsation-induced stresses DSM has the experiences that calculated dynamic stress levels lower than 10 MPa peak-peak are achievable.

2.3 Experimental evaluation

In general simulation studies give a good insight in pulsation and vibration problems that can be expected. It is rather easy to study the effects of for instance the insertion of an orifice on the pulsation level in the pipe system or the effects of an additional support on vibrations and stress levels etc

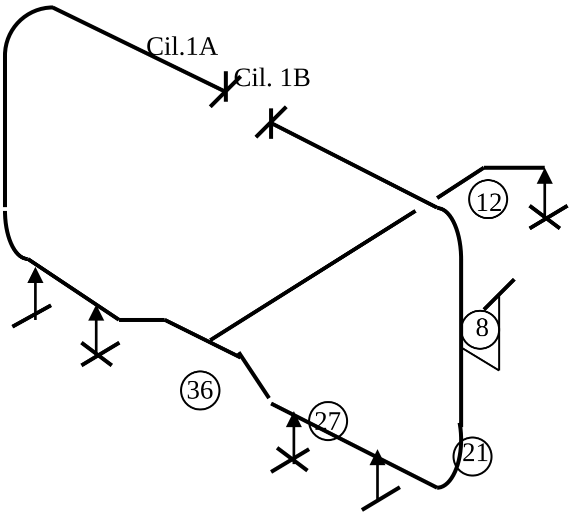


Fig.2.3.1. Measurement locations for a part of the discharge line first stage secondary compressor

However in practice it appeared that these studies do not guarantee that no vibration problems are met at plant start up or during plant operation. To get an insight in the predictability of problems by simulation studies an experimental evaluation of these studies were made. For a number of pipe spans of a secondary compressor with rotational speed of 180 RPM, vibration amplitudes were measured at locations for which these amplitudes were calculated. Displacement amplitudes were measured in a frequency range from 2 to 200 Hz. A part of the measured system is depicted in Fig. 2.3.1.

In the graphic presentation of Fig.2.3.2 a summary is given of the measurement results. In this graph the measured and the calculated dynamic displacement amplitude is presented for all measured locations. In Fig.2.2.3 the results are given in more detail for the measurement locations as indicated in Fig.2.3.1.

In analyzing the vibration data, we concluded that, at many locations of the pipe system, the measured vibration levels significantly exceeded the calculated levels. Analyzing the measured vibration data and the preconditions of the calculations a number of reasons could be indicated for the differences between measured and calculated data:

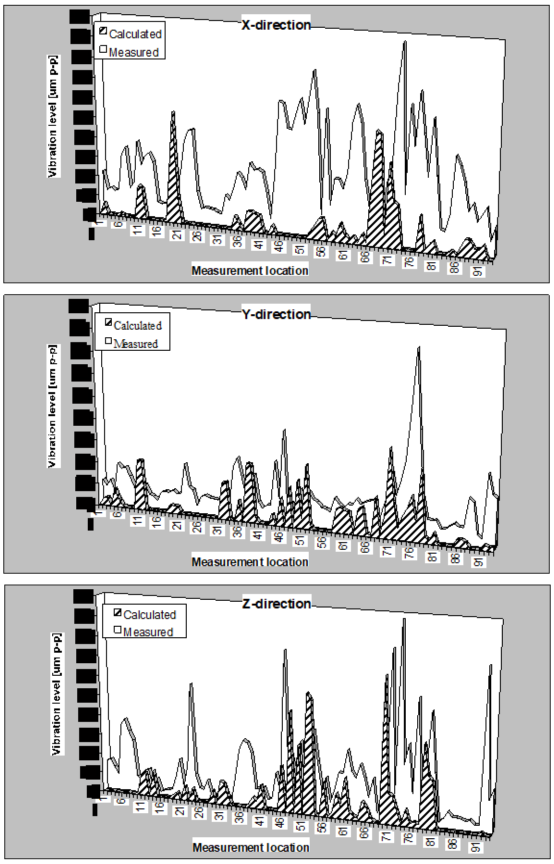


Fig.2.3.2. Calculated (shaded) and measured vibration levels for all measurement locations

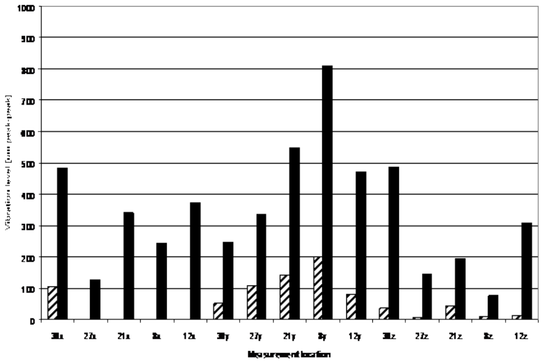


Fig.2.3.3 *Calculated (shaded) and measured (dark) vibration levels for the measurement locations indicated in Fig.2.3.1*

An important reason is the stiffness of the supports that of course mainly determine the vibration levels caused by shaking forces. In practice it appears that the actual stiffness deviates significantly from the stiffness that was assumed as a precondition in the model.
This means that either the actual stiffness of the support is not included or the design of the support is imperfect.
In the studies examined supports are often defined, as points were all degrees of freedom are blocked i.e. the stiffness in all directions is infinite. In practice such a support does not exist.

It appears that at some locations less attention is paid on the stiffness of the foundation or base plate. In some cases the foundation was even less stiff then the stiffness of the support itself.
In the investigated pulsation studies the calculations were carried out for a frequency range from 2-12 times rotational speed. In general the assumption was made that higher frequencies had no dynamic load on the piping.
For a typical rotational speed of 180 RPM, this means a frequency range of 3 to 36 Hz. In practice however also higher frequencies occur in the pulsation forces, which have an impact on the piping. This applies especially to stiff pipe spans for which the mechanical natural frequencies are high.
In Fig.2.3.4 the frequency spectra in x-direction for the indicated locations in Fig.2.3.1 are presented. These spectra illustrate that dominant frequencies at these locations mainly occur in a frequency range from 40 to 100 Hz. In this case the pulsation and mechanical response study are too restricted in frequency range to make a good prediction of the vibration levels.

This also means that the actual stress levels will deviate from the calculated stress levels. This is the reason, that DSM still applies the conservative recommendation of maximum allowable stress levels of 10 MPa.

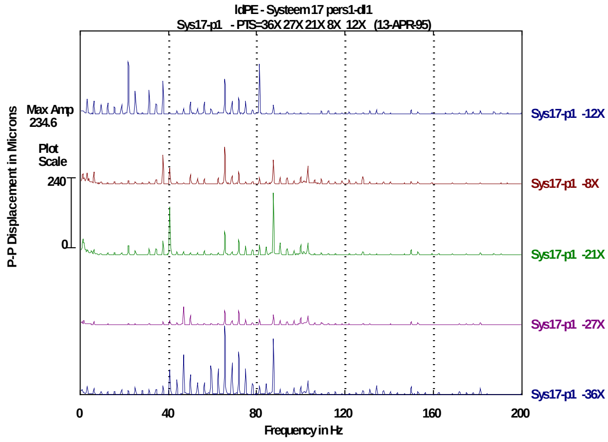


Fig.2.3.4 Vibration spectra for the measurement locations indicated in Fig.2.3.1.

General conclusions from the evaluated studies are that these studies give a good insight in pulsation and vibration problems.
However the calculated vibration levels often give a too optimistic image because the preconditions of support stiffness and frequency range of the calculation are not sufficiently defined.

3. PULSIM approach in the design of new-build hypercompresor plants

3.1 Development of PULSIM

TNO TPD has been in the field of simulation of flow dynamics in fluid machinery and pipe systems since the early seventies.
Originally by means of simulation on an electronic analog computer and since 1983 by means of digital simulation, based on the PULSIM software.
The software package, a development of TNO TPD software engineers, has proven to be an effective tool in the design of piping systems free from pulsation and vibration problems [2]. Nowadays some 40 projects a year are performed for compressor manufacturers, engineering contractors and operators.
The PULSIM package is not only used in the design of new build systems, but is also deployed to analyze systems for trouble shooting in combination with on-site measurements. The possibilities are not restricted to reciprocating machinery but also involve other sources like turbo

machinery, centrifugal pumps, flow-induced pulsations and transients.

The combination of pulsation and mechanical response analyses offers a powerful tool to prevent or solve pulsation and vibration problems [3, 4]
Nowadays our PULSIM software is also available for compressor manufacturers and others, who regularly deal with pulsation analyses. Licenses are available for limited and extended calculations. Currently several compressor manufacturers in Europe use our PULSIM software.
We welcome opportunities to compare calculated levels with on-site measurements, including pulsation and vibration levels. In fact our first activities in the simulation of high-pressure ethylene systems were for DSM. The results proved that also in high-pressure ethylene systems the calculated pulsation levels correspond well to the pulsations measured on-site by the operator.

3.2 Our role as a co-designer

A preview of the design of the piping system with respect to pulsation excitation can be of great value to prevent expensive and time-consuming modifications in a later stage.

Once the basic design is known and the properties of the process like flows, pressures and temperatures are available a review of the original layout is essential. Minimizing pulsation forces on pipe sections by symmetric layout and through combination of cylinders is an effective means, which can be used in such a preview without extensive calculations.
Appropriate supporting of pipe sections and heavy components like recycle or relief-valves are essential in preventing unacceptable vibrations and cyclic stresses.
Supporting of small diameter side-branches to instrumentation or valves is often overlooked, though very important to prevent excitation of local mechanical resonances in these side branches, Appropriate supporting to the main piping will result in an increase in stiffness and reduce the possibility of fatigue failure.

In our on-site troubleshooting we often observe poor engineering, mainly in details through lack of knowledge. A preview study makes engineering contractors and operators aware of potentially danger areas and pitfalls.
In a regular study a pre-check of the pulsation damper and/or the piping between the cylinders, close to the compressor, is also an effective means to reduce pulsation levels at the source in an early stage.

We stress the need of such an evaluation by engineers, who are well aware of potential pulsation and vibration problems.

3.3 Special problems in hypercompressor systems

Pulsation and mechanical response analyses in high-pressure ethylene systems involve a number of special problems. In the pulsation study following aspects complicate the analysis:

- Low compressibility of the gas and consequently a high velocity of sound with a considerable uncertainty margin [5]
- High harmonics of compressor speed are not damped sufficiently and thus high frequencies occur in pulsation and vibration levels
- Pulsation damper volumes are not feasible at the hypercompressor
- Possible interaction between primary and secondary compressor at different speed
- Uncertainty in boundary conditions for pulsations on the high pressure reactor side

An example of the layout of a LDPE plant is shown in fig. 3.3.1 indicating primary and secondary compressor, vessels, heat exchangers and reactor sections.

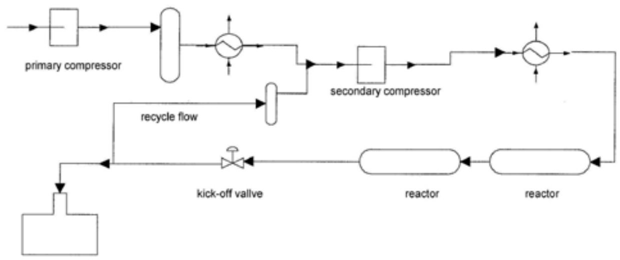


Fig.3.3.1 Schematic layout of hypercompressor piping in a LDPE plant

The pulsation and mechanical response analyses for the multi-stage **primary** compressor can be performed according to API 618 [1] as the mean pressures are still below 20 MPa. An exception is the final discharge stage of the primary compressor, which is compressing the ethylene to the suction 1st stage of the hypercompressor at approx. 25 MPa.
The recycle flow from the reactor is returned to the 1st stage of the hypercompressor. Approximately 30 % of the ethylene is converted to polyethylene in the reactor at a pressure between 250 and 300 MPa and a temperature of ca. 80 degrees C. At the kick-off valve the pressure is relieved from 250 to 25 MPa.

The polymerization process in the reactor, consisting of some 600 meters of pipe sections, results in an uncertainty in the boundary as the effective volume and corresponding reactor area is not known precisely. Therefore a variation in the acoustic reflection in the reactor should be included to investigate the impact on pulsation levels and forces.
In the pulsation analyses performed on hypercompressor systems we recommend the following approach:

- Calculations up to at least the 25th harmonic of compressor speed
- A safety margin of plus and minus 20 % around the nominal velocity of sound to account for deviations in temperature, compressibility and pipe geometry
- A variation in reflection coefficient from reactor on discharge side

The interaction between primary and secondary compressor is included in the pulsation analyses by simulating the complete piping from primary to secondary compressor. The pulsation levels and forces caused by each compressor are calculated independently. The maximum of the resulting beating pulsation can be found by adding the maximum level caused by each compressor individually.
Feed back from on-site pulsation measurements shows that the calculated pulsation levels are well in line with the measurement results (chapter 4.1)

The next step, following the pulsation study, is the mechanical response analysis. In this mechanical response analysis we first calculate natural frequencies and corresponding mode shapes for a given support layout. A number of natural frequencies are coinciding with or will be find close to pulsation frequencies. In this case the pulsation forces can excite the mode shapes, which can result in unacceptable vibration levels and/or cyclic stresses. Additional supporting or stiffening of the structure is necessary to increase natural frequencies.
Also in the mechanical response analyses one should take into account the correct properties of the boundary conditions such as:

- The compressor cylinders are not infinitely stiff
- The structure on which the support is mounted is sometimes even less stiff than the support itself
- Damping properties of materials are not always known precisely

With the experiences from practice by various operators we are improving our models constantly:

- We can include the compressor cylinder stiffness or even compressor manifold if necessary [6]
- We can include the support details and, if necessary, the structure on which the support is mounted in the calculations [7]
- We are able to determine support stiffness and damping by on-site measurements in existing installations
- We include nozzle flexibility on dampers, vessels and heat-exchangers

4. Some examples of a pulsation and mechanical response analysis for hyper-compressors

4.1. The optimum location of orifice plates

Pulsation analyses for a number of interstage lines have been performed for DSM. It concerns the interstage piping of a hypercompressor at a mean pressure of 116 MPa from discharge 1st to suction 2nd stage. The problem concerns optimization of the location of orifice plates to dampen pulsations in the interstage piping.
The layout of one of the interstage branches is shown schematically in fig.4.1.1.

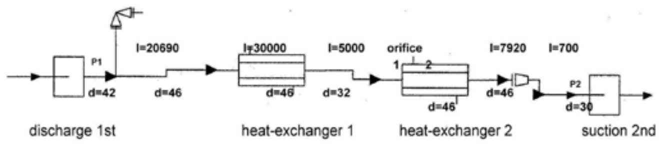


Fig.4.1.1 Layout of the interstage piping of a DSM hypercompressor – length 93 m

In practice the installation consists of four parallel lines between the single acting cylinders of different length and layout. PULSIM was asked to optimize the location of orifice plates in this existing polyethylene plant. DSM has performed on-site pulsation measurement with and without an orifice plate at location 1 or 2 at heat-exchanger 2.
The results of the PULSIM calculation can be presented in several ways. The flow and pressure pulsations along the piping show a standing wave pattern of half a wavelength for the 2nd harmonic.
The cylinders act as an acoustic boundary condition with reflection factor close to 100%.
In this case the compressor speed f=180 rpm or 3 Hz and the speed of sound c=1125 m/s. This corresponds to a wavelength $\lambda = c/f = 375$ m for the 1st harmonic or 187.5 m for the 2nd harmonic. The

total pipe length of 93 m between discharge 1st and suction 2nd stage corresponds thus to almost exactly half the wavelength for the 2nd harmonic at nominal conditions.
The pressure pulsation is in anti-phase with the flow pulsation as shown in fig.4.2.1: a maximum in pressure pulsation near the cylinders corresponds with a minimum in the flow pulsation.

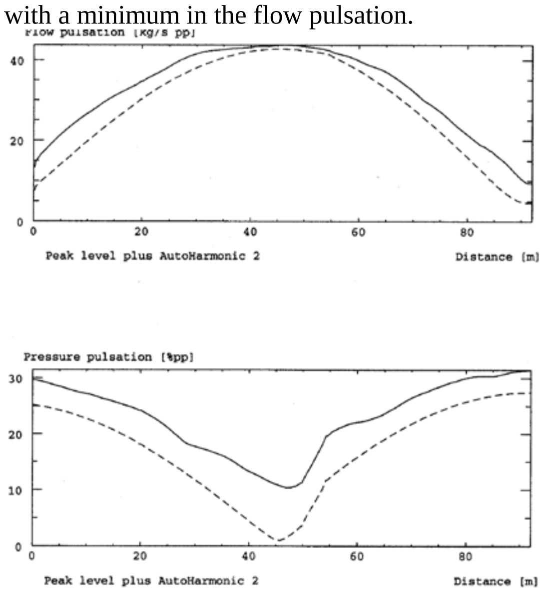


Fig.4.1.2 Waveplot at 180 RPM (0 % deviation) for flow (upper trace) and pressure (lower trace) pulsation

Another way is to present pressure pulsation against time for a given location. These levels determine the actual excitation of the piping at each change of flow direction or bend. An example of the pressure pulsation as calculated at P1 is shown in fig. 4.1.3

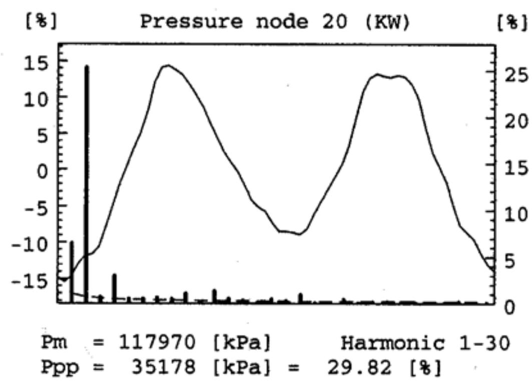


Fig.4.1.3 Result of the PULSIM calculation at the discharge flange P1

An uncertainty is included in the speed of sound, which is determined by compressibility, pressure and temperature. We therefore prefer to investigate these systems within a range of 20 % around nominal speed of sound to account for these deviations.

The result of this calculation shown in fig.4.1.4 shows the overall pressure pulsation and the dominant harmonic as a function of the deviation.
The maximum in pressure pulsation at both P1 and P2 is approx. 30 % peak-to-peak close to the nominal condition (0% deviation)

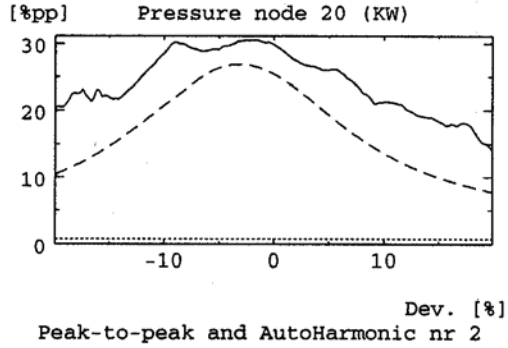


Fig.4.1.4 Pressure pulsation for P1 as a function of the deviation of nominal condition: Dev [%]

The simulation results show that pressure pulsations are far above the 10 % pp limit, so that prevention of resonance and damping of pulsations is required.
In the actual situation an orifice with a bore of 11.5 mm has been installed (pressure loss 1.3 %) in the middle (see orifice location 1 in fig 4.1.1), which is very effective for the 2nd harmonic resonance.
The PULSIM calculations shows a reduction from 30 MPa pp to approx. 16.5 MPa or 14 % pp at P1, which again corresponds well with the result of the on-site measurement.
An overview of the results of the PULSIM calculation compared to on-site measurements by DSM is presented in the table below.

Interstage piping	Friction	Location	PULSIM	DSM
I-93 m	Pipe Friction	Dis1 P1	29.8%pp	27.4%pp
		Suc2 P2	31.0%pp	29.0%pp
I-93m	Orifice 1 11.5mm	Dis1 P1	12.5%pp	13.4%pp
		Suc2 P2	11.0%pp	15.1%pp
II-211m	Pipe Friction	Dis1 P1	17.2%pp	-
		Suc2 P2	18.1%pp	-
II-211m	Orifice 11.5 mm	Dis1 P1	9.9%pp	11.1%pp
		Suc2 P2	9.4%pp	11.0%pp

Table.4.1.5 Comparison between calculated (PULSIM) and measured levels (DSM) for the interstage piping in a LDPE Plant

The calculated and also the measured pulsation levels are still above the 10% pp limit as shown in table 4.1.5
At a resonant condition of –9 % deviation a level of 14.8 % pp is calculated (see Fig.4.1.6) as a result of a standing wave at 4th and the 10th harmonic,

which are now important components. The flow pulsation along the piping is shown in Fig.4.1.7 and Fig. 4.1.8 for respectively the 10 the harmonic, at –9 % dev, and for the 4th harmonic at 0% deviation.

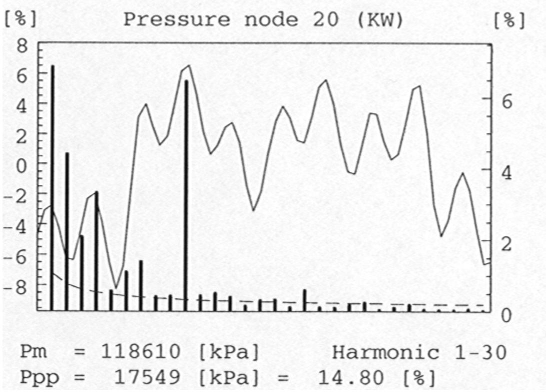


Fig.4.1.6 Result of the simulation for discharge P1(t) at –9% deviation of nominal conditions

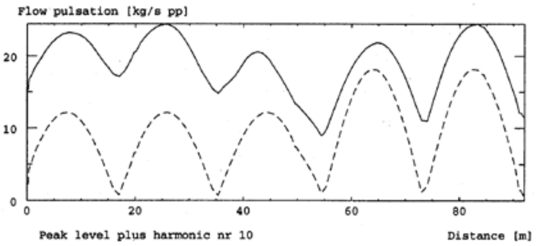


Fig.4.1.7. Waveplot at –9% dev (10th harmonic)

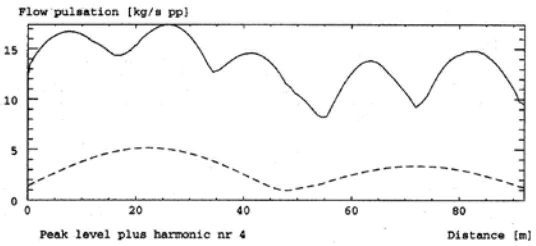


Fig. 4.1.8 Waveplot at 0% dev (4th harmonic)

Based on the results of the simulation a location for an additional orifice plate (bore 15 mm with DP=0.4 %) is recommended at 25 m from the discharge flange. This is a location with an anti-node in the flow pulsation both for 4th and 10th harmonic and thus optimum for additional damping of pulsations. With the combination of these two orifices a reduction in pulsation levels from 30 to 11 % peak-to-peak could be achieved. In this example the analyses are restricted to a pulsation simulation without considering the mechanical response of the pipe system.

4.2 A combination of pulsation and mechanical response analysis to avoid unacceptable vibrations and cyclic stresses

As a rule TNO TPD recommends a combination of pulsation and mechanical response analysis according to design approach 3 of the API 618 Standard. Even in case of acceptable pulsations according to API-618 limit, pulsation induced forces can result in unacceptable vibration levels and/or cyclic stresses.

In case of a hypercompressor, for pressures above 20 MPa, API 618 pulsation limits are no longer applicable and compressor manufacturers and operators often specify 10 % pp. However sometimes stricter limits on pulsations are specified by the operator (see DSM table 2.2.1) for different parts of the system.

According to API 618 and also based on our experience a mechanical response analysis should always be included for hypercompressor piping regarding critical aspects in such a plant. The extent of the analysis and the details of the supporting structure to be included depend on the excitation forces. Higher harmonics in the forces can be considerably as possibilities for damping are restricted: pulsation dampers are not feasible and possible modifications in layout are limited. The following example shows the approach for the suction 1st stage of a hypercompressor.

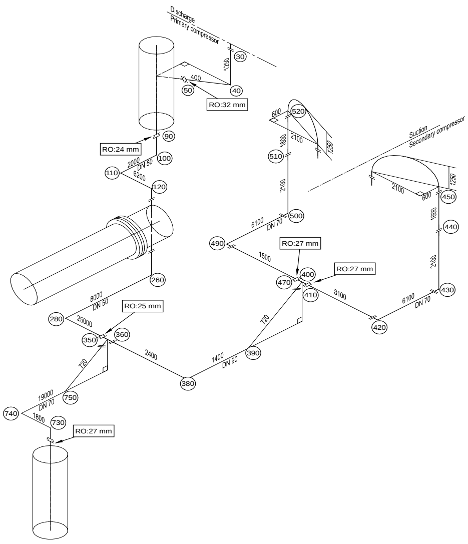
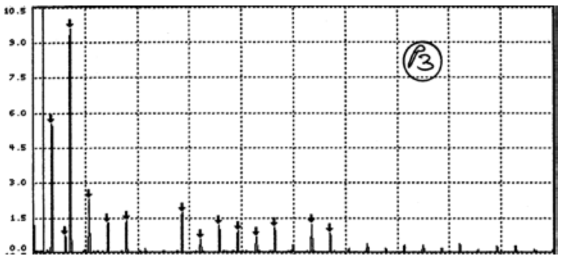


Fig.4.2.1 Isometric layout of suction piping for a hypercompressor in a LDPE plant

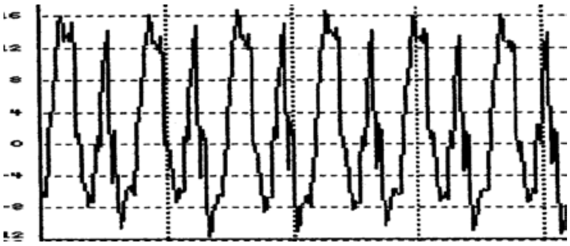
Fig.4.2.2 Detail of layout for hypercompresor suction piping

The compressor manufacturer has performed a pulsation analysis and recommended orifice plates at suitable locations. The maximum pulsation levels in this suction pipe system are approx. 14 % pp at a mean pressure of 27 MPa, which results in considerable vibration forces over 10.000 Newton pp. on DN70 sections close to the compressor.

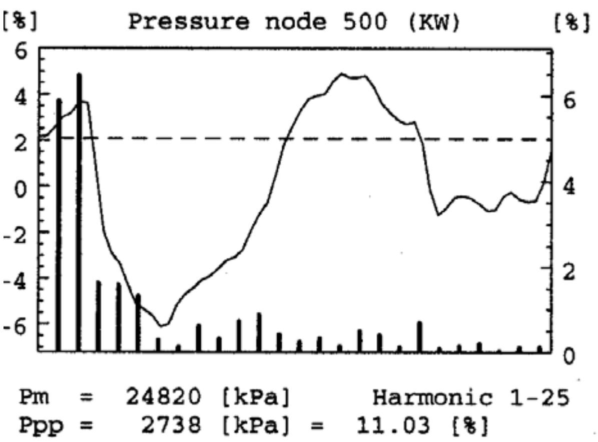
The pulsation levels are confirmed by on-site measurements on various locations.



Spectrum of measured pulsation levels in bar pp.



Measured pressure pulsation in bar pp. versus time; overall pulsation level 28 bar pp. = 11.5 % pp.



Calculated pulsation level at P3 suction 27.3 bar pp.

Fig.4.2.2 Comparison between measured and calculated pressure pulsation level for suction piping hypercompressor

The measured vibration levels are rather high as we normally judge 15 mm/s rms as an allowable level whilst here levels up to 40 mm/s rms are measured. A beginning of fatigue failure has been noticed at a threaded connection. We have recommended a combined pulsation and mechanical response analysis to decrease vibration levels. The calculation for this model is an illustration that assuming infinitely stiffness of supports results in a too optimistic result with respect to vibration levels. The mechanical model for the inlet piping of the hypercompressor including the supporting structure is shown in Fig. 4.2.3.

The first calculation is carried out with infinitely stiff supports at most of the locations as assumed.

The lowest natural frequency for this model is found at 21 Hz. If we excite the piping with the pulsation forces as calculated we find that the resulting vibration are below the limit of 15 mm/s rms with a maximum of 5 mm/s rms. The resulting cyclic stresses are less than 2 MPa over the complete frequency range as investigated up to the 25th harmonic of compressor speed.

If we follow the more realistic approach and include the support construction and details we calculate a lowest natural frequency of 15 Hz as shown in Fig.4.2.4.

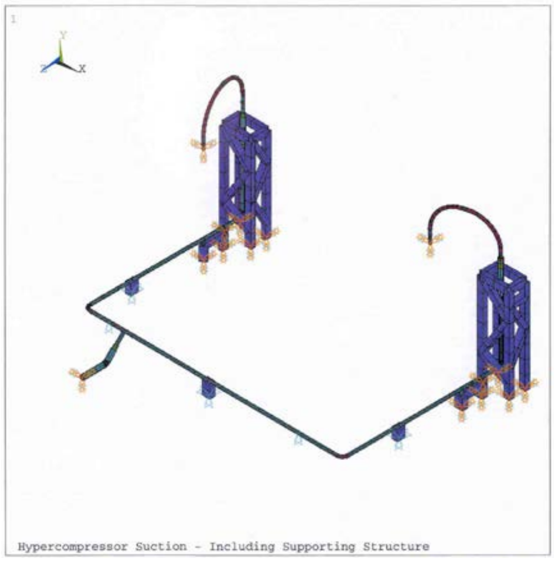


Fig. 4.2.3 FEM model of suction piping with supporting structure

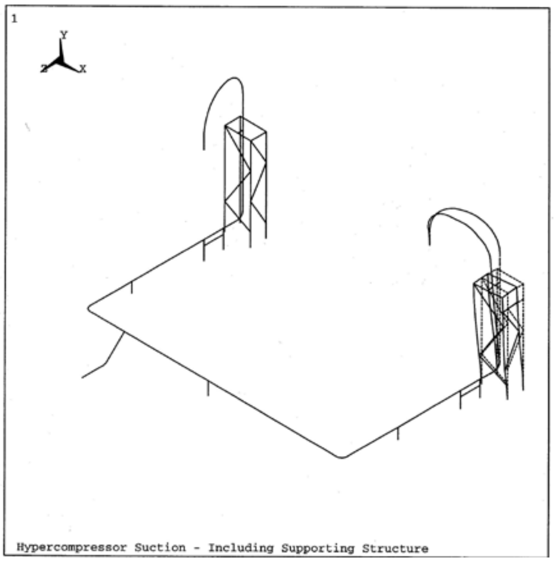


Fig. 4.2.4 Mode shape at 15 Hz including supporting structure

The calculated vibration levels are maximum 70 mm/s rms, as measured on the DN70 piping, whilst the calculated cyclic stress is maximum 10 MPa for this model. In practice the vibration levels measured are lower (45 mm/s rms). It is to be expected that calculations will show a higher vibration level as 'worst case' conditions are calculated by exiting at the mechanical resonance frequency.

5. Conclusions and recommendations

In the experience of TNO TPD on-site measurements on several LDPE-plants have proven that pulsation levels and forces in high-pressure ethylene systems are calculated accurately by means of pulsation analysis. Deviations from calculated levels are within a margin of approx. 20 % of the peak-to-peak value for a well-defined system.

Also higher harmonics are well predicted, assuming that the frequency range in the calculations is at least up to 25th harmonic. DSM experience has shown that higher harmonics are underestimated if the pulsation analyses are restricted to the lower harmonics.

As pulsation dampers are not feasible at the hyper-compressor optimization of orifice plates and/or pipe modification in an early stage of the design is essential. It can prevent expensive and time-consuming modifications during start up.

A pulsation analysis should be followed by a mechanical response analysis, in which vibration levels and cyclic stresses are evaluated, as recommended in the API 618 Standard. Nozzle flexibility, pipe supports and supporting structure should be included in the mechanical models to obtain accurate results. Assumptions on infinite pipe support stiffness, as sometimes assumed, result in too optimistic conclusions on vibration levels and cyclic stresses.

As standards do not specify allowable pulsation and vibration levels in high pressures systems manufacturers, operators and knowledge providers should co-operate to define these standards.

We summarize recommendations to prevent pulsation and vibration problems in hypercompressor LDPE plants:

- Pulsation analyses should include the higher harmonics up to 25 times compressor speed
- Special attention should be given to possible ways of pulsation damping by orifice plates and piping modifications as damper volumes are not feasible
- A wide range of 20 % around the nominal velocity of sound should be investigated to account for deviations in compressibility, temperature and geometry
- A pulsation analysis should be combined with a mechanical response analysis including the supporting structure taking into account critical stress levels as mentioned in API 618 for high pressure systems
- Manufacturers and operators should agree on allowable pulsation levels in hypercompressor pipe systems. We recommend 10 % pp. as a maximum allowable level. Further differentiation with regard to the piping and location may be considered
- A guide line for acceptable vibration levels measured in practice is 15 mm/s rms independent of frequency, though for frequencies below 10 Hz a displacement criterion is sometimes used (0.5 mm peak-to-peak) in hypercompressor piping
- The allowable cyclic stress can be defined as 179 MPa (for low-carbon steel with an operating temperature below 371°C) as stated in API 618, assuming all stress concentration factors are included in the calculation and all other stresses within applicable code limits

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A New Design of Non-Cooled Pressure Packing for Improved Life and Reliability

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability May 17th / 18th, 2001, The Hague

Abstract:

Increasing demands are being placed on reciprocating compressors for longer lifetime between services and improved operating reliability. The piston rod pressure packing is a highly stressed and vitally important seal. Many of the current packing designs are becoming marginal in performance terms and there is a requirement for improvements over the existing industry standard designs.

It is desirable to eliminate the traditional water cooling jacket from the pressure packing assembly. The water jacket can be the cause of problems and shutdown failures mainly due to water leakage contaminating the process gas and also water channel blockage leading to cooling problems and short lifetimes. To overcome the problems two new components have been designed, these are the "Thermosleeve" a non-water cooled pressure packing assembly and the "BOT" packing ring element.

The Thermosleeve packing achieves efficient removal of the frictional heat by using high conductivity solid materials with a special sleeve to reject heat to the compressor cylinder cooling system.

The BOT packing ring has been developed to provide much lower frictional heating than other packing ring designs. The reduction in heat generation has lead to reduced wear and increased life and reliability

1 Introduction

The function of the piston rod pressure packing is to provide a good seal between the gas pressure in the cylinder and atmospheric pressure outside the compressor. This is achieved by a contacting, sliding seal between the stationary packing rings and the moving piston rod.

The contacting packing ring seal is simple and effective but unfortunately the work done against sliding friction generates heat in the seal assembly.

The amount of frictional heat generated depends on a number of parameters such as; coefficient of friction between packing rings and piston rod, the magnitude of the cylinder pressure which is being sealed and the speed of the compressor. There are many factors influencing how quickly the frictional heat can be removed, including, thermal conductivity of the materials for the rings, rod and packing housing, effectiveness of the compressor as a heat sink and presence of any specific cooling arrangements such as water jackets, air blast cooling etc.

The frictional heat which is generated in the packing causes a rise in temperature of the seal assembly especially the packing rings and the piston rod. For successful long term operation of the packing the frictional heat must be removed from the packing at the same rate as it is generated without any great increase in temperature.

If the frictional heat is generated at a greater rate than it is removed from the seal the temperature will continue to rise, leading to an increase in friction coefficient and excessive wear. The increase in friction causes yet further heating and a thermal run-away can occur leading to complete packing failure and often damage to piston rods.

The decision whether cooling is necessary is based upon design guidelines which have been established over a long period of time. Due to the many factors involved it is difficult to calculate cooling requirements exactly, hence the use of the well established empirical rules

Packings which operate with lubrication have a much lower coefficient of friction than non-lubricated (“non-lube”) packings. The coefficient of friction in non-lube is in the order of 0.2, whereas in fully lubricated conditions it may be 1/10 of this value.

The differential pressure, discharge temperature and mean piston speed of the compressor are the main factors to determine the cooling requirements of a packing for a non-lubricated compressor. A conservative rule is to use a water cooling jacket above a discharge pressure of 20 bar g.

This means that for non-lube compressors, packing cooling must be considered on all but the lowest pressure machines.

1.1 Water Cooling

Where cooling is deemed to be necessary, packings have traditionally been cooled using a cooling jacket comprising of drillings and slots machined into the packing housing. These provide a good heat exchange to fluid fed into the packing under pressure. Water is often used, but glycol or process fluids may be used if water is incompatible with the process gas.

Cooling by water is very effective at removing heat and it is a good system when working well, however, it suffers some disadvantages including; leaks which can contaminate the gas, gas leaking into the cooling system, corrosion and blockage of cooling jacket and the extra cost and complexity of external cooling systems.

In theory there should be no leakage to, or from the cooling jacket and packings are leak tested when they are new. However, during the lifetime of the compressor and if packing maintenance or reconditioning has been poor there is a high chance of leakage from the water jacket, particularly from the ‘O’ rings which are needed to seal between each cooled container. ‘O’ ring seals in packing cooling jackets are such a problem that the larger seals around the piston rod have been banned by the API618 standard.

Although water is an excellent and cheap cooling medium it can lead to problems if it leaks into a compressor, leading to corrosion of cylinders and internal components and contamination of process gases.

With tightening of legislation in industrial and process plants many water cooling systems for compressors have changed from constant loss systems using mains water supplies, to closed circuit systems involving header tanks, pumps and cooling towers. These systems bring an extra cost and complexity to the installation. They are particularly susceptible to gas leakage from the compressor water jackets because this causes the systems to pressurise. The closed circuit water systems sometimes suffer from bacterial contamination which can cause water jackets to “furr-up” or worse could contaminate sensitive clean processes like PET bottle blowing air.

It can be seen that there a number of reasons why it is preferable to eliminate the water cooling in a packing. Before taking the decision to remove the cooling it is necessary to analyse the specific cooling requirements of each compressor.

2 Thermal Analysis of Piston Rod

The piston rod is heated from two sources. Heat can be conducted along the rod from the hot piston and the packing generates heat from friction between the packing rings and rod.

The piston is heated due to compression of the gas. Gas laws can be used to give the theoretical adiabatic temperature after compression. The temperatures achieved in practice are very similar to the theoretical and so the adiabatic temperature calculations are often used if there is no actual data.

Figure 1 shows the distribution of actual suction and discharge temperatures, for data obtained from several thousand compressors in a database.

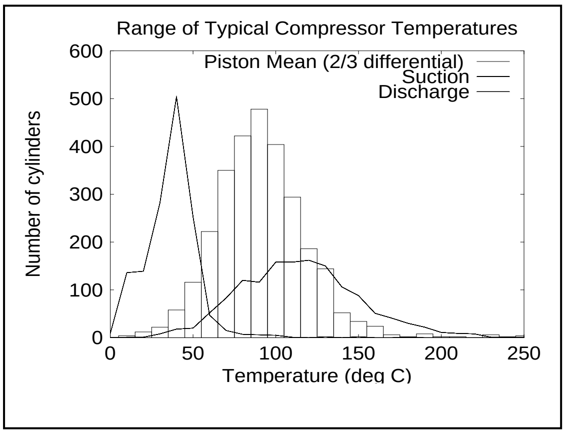

Figure 1: Compressor temperatures from database

The accepted way of determining the piston design temperature is to use the empirical rule

$$T_{piston} = T_s + 2/3\,(T_d - T_s) \tag{1}$$

Years of experience in designing piston and rider rings to this rule shows that it is sufficiently accurate. If the cyclic mean adiabatic temperature of the gas was calculated it would be lower than this value, in-fact less than ½ of the difference between suction and discharge, however, the discharge temperature dominates the piston temperature because the gas is denser after compression and so there is more heat exchange to the piston from the gas at discharge conditions.

Therefore from figure 1 it can be seen that an average compressor may have a piston temperature of about 90 deg C.

A machine operating at higher than average temperatures, i.e. a PET air compressor, may have a final stage suction temperature of 40 deg C and a discharge of 220 deg C. By the 2/3 rule this puts the mean piston design temperature at 160 deg C.

Figure 2 shows a schematic representation of the heat flows and temperature gradient for the piston rod. There may be some cooling effect on the rod due to the cool suction gas being drawn into the cylinder. If the cylinder has the common arrangement with multiple suction valves arranged symmetrically there will, however, be little cooling effect from the suction gas because the gas flow will be mainly parallel to the rod, moving with almost the same velocity. Therefore there will be very little convective heat transfer.

It can be assumed that at the cylinder end the piston rod will have virtually the same temperature as the piston.

The cross-head provides a reasonably good heat sink at the crankcase end of the piston rod. The cross-head is normally well supplied with cooled lubricating oil and there is considerable windage of oil and air in the crankcase, which will ensure good heat transfer. Typical crankcase bulk temperatures are in the order of 70 - 80 deg C.

If there was no frictional heating from the packing, the rod might have a temperature of about 80 deg C for a slow speed process machine and the rod of a high speed machine at high pressure ratio might have a temperature gradient from 150 deg C at the piston to 80 - 90 deg C at the crankcase, depending on exact compressor conditions. The heat input from the packing friction is then superimposed upon these temperatures.

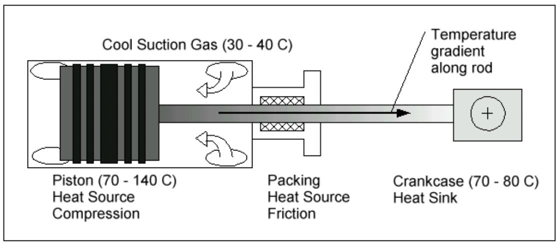

Figure 2: Thermal gradient along piston rod

The packing itself is the greatest heat source for the rod. Depending on the configuration of the compressor, distance piece layout etc., the pressure packing could be 1/3 to ½ of the way along the rod. This will be the area where the frictional heat input and hence rod temperature is greatest. Figure 2 shows that even before considering the effect of frictional heating the rod temperature near mid-stroke may be between 80 and 100 deg C.

2.1 Frictional heating by packing

Figure 3 shows a representation of a packing ring pair sealing pressure between P1 at the higher pressure side and P2 at the lower pressure side. We can estimate the frictional force and hence the heat generation on the rod if we make some assumptions as follows:

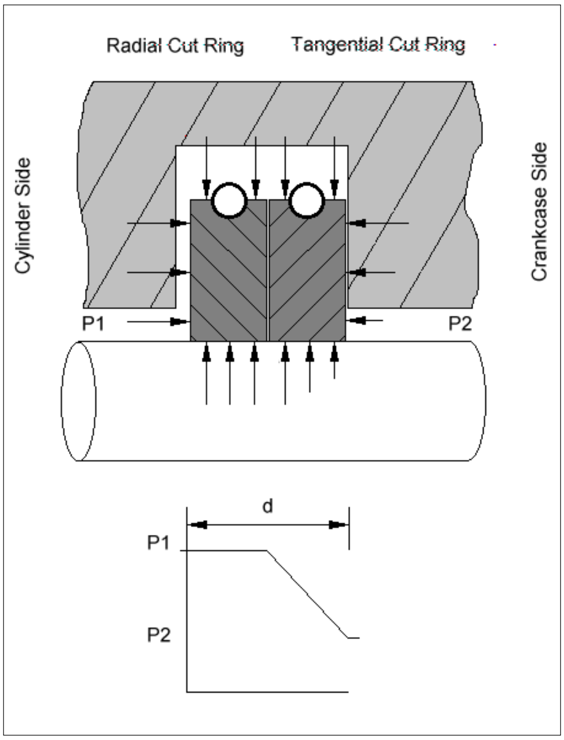

Figure 3: Cross section through conventional packing ring pair

- Figure 3 assumes that the pressure is broken down mainly by the tangential cut ring, at the right hand side in figure 3. When new, the radial cut ring at the high pressure side has open gaps which allow the pressure to pass through un-restricted. By observing used worn rings it is obvious that the majority of the pressure breakdown occurs over the tangent ring. Figure 6 shows a pair of used rings showing this pattern of wear. Therefore it is assumed that the pressure drop occurs over the length d/2.
- The radial cut ring does not add any significant friction. Again this is confirmed in practice, where the used radial cut rings often show very little wear.
- The radial force (and friction) due to the garter spring is insignificant compared to the gas loading.
- The nett radial pressure on the tangent cut ring is (P1-P2) / 2 (N/m^2)
- Assume that there is only 1 ring pair in the packing, which is sealing the whole pressure.
- Coefficient of friction in packing is $\mu = 0.2$
- Consider a rod with diameter = D (m)
- Container depth = d (m)
- Piston means speed is u (m/s)
- Piston Stroke = s (m)

The inwards radial force on the tangent ring due to pressure load

$$= d/2 \ \pi \ D \ (P1 - P2) / 2$$

And:

P2 = atmospheric pressure,

P1 = cyclic mean of suction and discharge gauge pressures , (approx) = $(P_{dis} + P_{suc}) / 2$

So total radial force from tangent ring

$$= d \ \pi D (P_{dis} + P_{suc}) / 8$$

Total Axial force from tangent ring

$$= \mu \, d \, \pi \, D (P_{dis} + P_{suc}) / 8$$

Work against friction = force x distance

Power = work per second

$$= u \ \mu \, d \, \pi \, D (P_{dis} + P_{suc}) / 8 \qquad (2)$$

Where: u = stroke x 2 x rpm / 60

Example:

For piston rod,	75 mm diameter
Suction pressure,	20 bar g
Discharge pressure,	60 bar g
Container depth,	d = 16mm
Speed,	u = 4 m/s (mean)

so heat generated

$= 4 \times 0.2 \times 0.016 \times \pi \times 0.075 \times (60 + 20) \times 10^5 / 8$

= 3.0 kW.

The example assumed just 1 pair of rings was sealing all of the pressure. In fact there is no difference to the total heat generated if more than 1 set of rings is considered. If the pressure was shared equally by say 6 rings then each pair only seals 1/6 of the differential pressure, so the frictional heating from each set is only 500W (1/6 of the total), but the sum from all 6 is still the same.

2.2 Factors Affecting Generation of Frictional Heat

The frictional heat generated by the packing is found to be proportional to:

- Speed of compressor
- Suction and discharge pressure
- Coefficient of friction of ring material
- Width of packing rings
- Diameter of piston rod.

2.3 Heat Flow Paths in the Packing Ring

Figure 4 shows a simple model of the heat flow paths in the packing ring contact area for a conventional water cooled packing. The tangent ring is considered to be the dominant heat source.

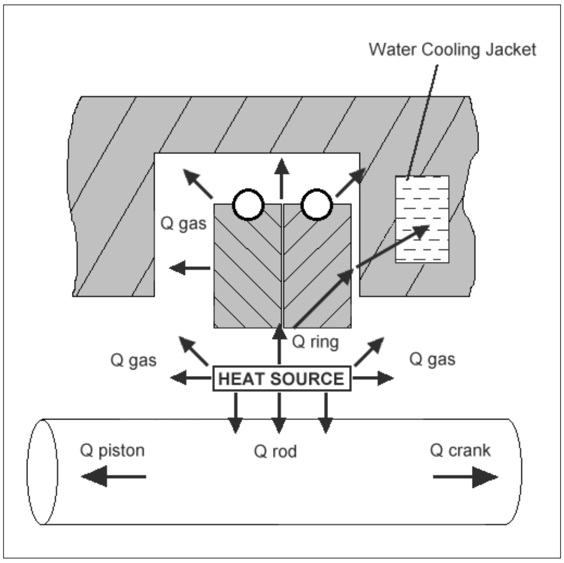

Figure 4: Model of the heat flow in the packing ring contact

The frictional heat generated in the contact Q_{FRIC}, can be estimated by equation (2). By conduction some of this heat will flow into the piston rod, Q_{ROD} and some into the packing ring, Q_{RING}. There will also be heat loss to the surrounding gas, this will occur both from the rod and from the packing ring, it is considered as one heat flow path, Q_{GAS}.

2.4 One - dimensional heat flow model

The conduction of heat away from the frictional heat source was modeled as simple one-dimensional conduction according to Fourier's Law.

It is easy to calculate the heat transfer by conduction, but more difficult to find the convective heat transfer because it depends on many parameters relating to the gas flow, such as density, viscosity, flow velocity and turbulence etc. The conduction is solved by making an estimate of the piston rod bulk temperature in the area of the packing, the remaining heat flow in the thermal balance is then due to convection.

Operating experience shows that a typical rod temperature may be 120 deg C when the mean piston temperature is 80 deg C. Working on this assumption we can calculate example values for the heat flow Q_{ROD} and Q_{RING} in the example 75mm rod. It is assumed that at mid-stroke the packing is approximately 1m away from the piston and cross-head.

By Fourier's Law:

$$Q = \frac{-k\,A}{x}\,(T_2 - T_1)$$

$$Q_{ROD} = -50 \times \pi \times (0.0375)^2 \times (80 - 120) / 1$$

$$Q_{ROD} = 8.8 \text{ Watts}$$

Note that for a stainless steel the thermal conductivity coefficient is only 25 (W/m/K) compared to 50 for a carbon or low alloy steel. Typical values for PTFE based packing ring materials are 0.6 (W/m/K).

The heat flow through the ring was considered as a network comprising the ring, the packing case material and the flow through to the water jacket. As can be seen from the results the boundary layer losses across the solid / water interface can be neglected.

$$T_{water} - T_{rod} = \frac{-Q_{ring}}{A}\left[(x/k)_{ring} + (x/k)_{cont}\right]$$

The x lengths are as shown in figure 5, for a stainless steel container k = 25 W/m/K. The contacting area is consider to be an annulus of rod circumference x ring radial thickness (in this case 10mm). Assuming that water jacket is at 20 deg C and neglecting any transfer losses at the contact between ring and packing gives approximately

$$Q_{ring} = 28 \text{ W.}$$

Working back to find the intermediate temperature between ring side face and container shows that the temperature drop across the metallic container is only about 5 deg C.

The frictional heat generation was previously calculated as 500 W and it can be seen that direct conduction only accounts for about 40W of this heat. The rest of the heat must be rejected to the gas which is constantly flowing in and out of the packing containers due to the varying cylinder pressure.

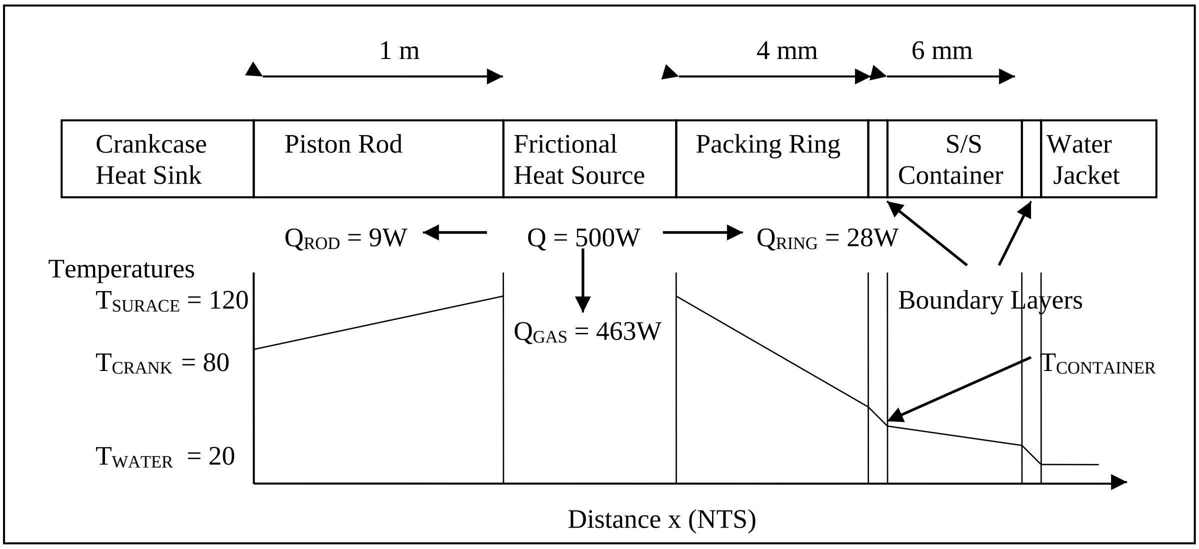

Figure 5: One dimensional heat transfer model showing thermal gradients

Although the analysis which has been conducted was approximate and used many assumptions and estimates, it shows how important convection is in cooling the packing rings. The polymer materials often used for non-lube packing rings have very poor thermal conductivities and therefore there is very little cooling directly by conduction. As a comparison a bronze ring can transfer over 800W in the same configuration.

The thermal analysis of the packing provided a useful tool showing the critical areas which need to be considered when designing a non-cooled packing.

The first step in designing a non-cooled packing was to reduce the heat generation at source, thereby reducing the cooling requirements.

3 Development of BOT Packing Ring

The key to a successful non-cooled packing is minimal frictional heat generation. This has been achieved with the development of a new design of pressure packing ring, the BOT ring. The name describes the functionality of the ring - it is a pressure Balanced, Overlapping Tangent ring.

The design was developed methodically, the starting point being the current industry standard of the radial cut and tangential cut ring pair. The development of the design aimed to reduce loading on the ring, therefore reducing frictional heating and also reducing wear, leading to longer ring life.

Figure 6 shows some conventional rings which have reached the end of their useful life. As previously stated most of the wear and loading is taken by the tangent ring, as seen in the figure.

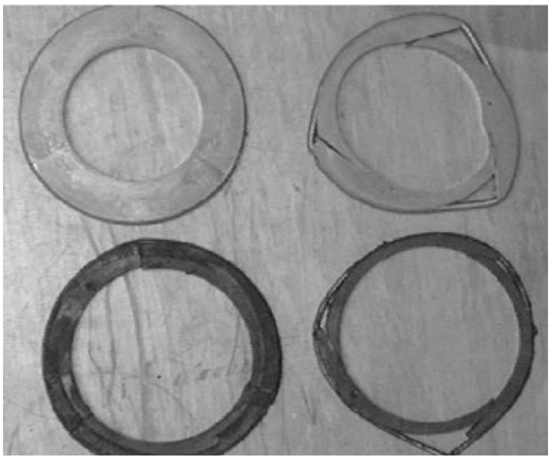

Figure 6: Conventional design rings showing the wear biased to the tangent ring

3.1 Features of the new BOT design

The tangent sealing ring normally occupies half of the cup width. The analysis shows that the frictional work is related to the width of the sealing ring (eqn 2). Therefore the width of the tangent ring was reduced as much as possible without compromising sealing efficiency or ring strength. If the ring is made too thin it will not be able to support the axial pressure load and may extrude.

The radial ring does not normally wear excessively. The new design therefore utilises the radial ring to provide a bearing support to the tangent ring. It achieves this by using an over-lapping step section, see figure 7.

The step allows some of the radial pressure loading on the tangent ring to be shared with the radial ring. The radial ring is wider than normal and has a large surface area to support this extra load. In addition the radial cut ring is heavily pressure relieved so that it acts mainly just as a support bearing, rather like a “rider ring” on a piston.

The thermal analysis showed the importance of convective heat transfer to remove heat from the rings and rod. To aid convective flow the radial cut ring is extensively pressure balanced and features large flow areas for the reverse flow direction even when the rings are worn. There are many wide and sharp flow paths to encourage as much turbulence and mixing as possible.

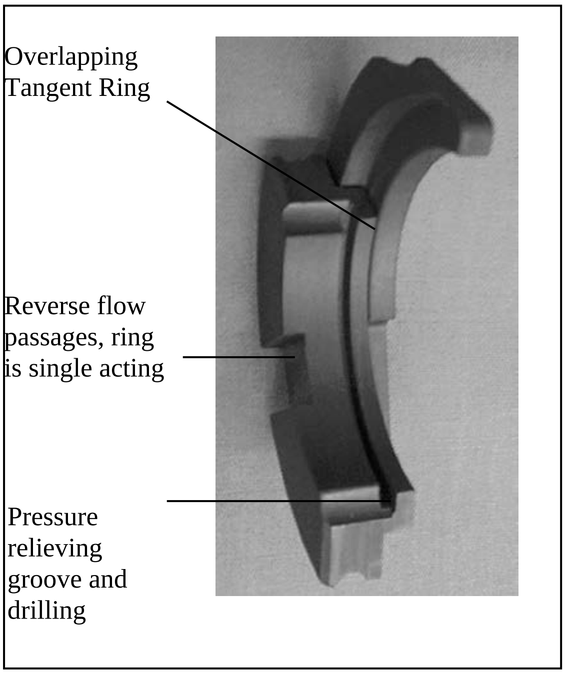

Figure 7: BOT Packing Ring segments

3.2 Testing the BOT ring design

During development the BOT ring was tested using a purpose built transparent pressure packing and the Hoerbiger Rings & Packings “Shell” Test Rig. A clear acrylic housing was used for the test packing so that the dynamic action of the rings could be observed. The packing was also fitted with capacitance transducers in the side face on the packing to sense when the rings had lifted in the axial direction.

The test packing is shown in figure 9. A computer controls valves for air entry and exhaust to simulate cylinder PV conditions, synchronised to crankshaft position. The pressure in the simulated “cylinder” and in each container was measured using pressure transducers recorded by a fast PC based ADC. The packing comprised 3 sets of rings and any leakage after the last set of rings was measured by a vent line and flow meter.

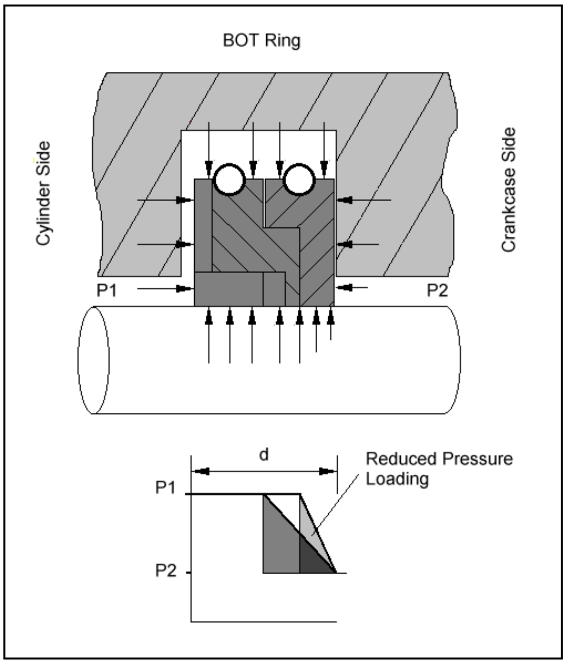

Figure 8: Section through BOT Rings

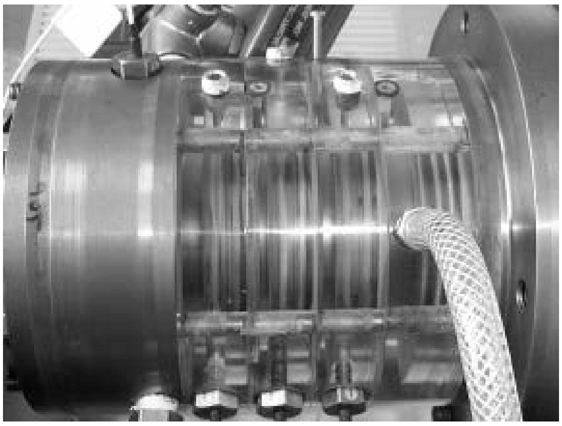

Figure 9: Acrylic test packing

The conventional radial and tangential cut style of rings were tested first. The pressure distribution measured is shown in figure 10.

The pressure results from testing the existing design shows that most of the pressure is sealed by just one pair of rings. In figure 10 this was the first set of rings, sealing most of the pressure between cup 1 and cup2.

3.3 Ring Dynamics

Figure 10 shows the pressure in cup 2 increasing with time, this indicates a slight leakage past the first pair of rings. When the pressure in the 2nd cup exceeds that in the first the ring is forced to move axially towards the cylinder side and the side face seal is broken. The effect is exaggerated due to the rod friction which is also acting back towards the cylinder in the second half of the stroke. The dynamic forces acting on the rings cause them to "shuttle" backwards and forwards in the working clearance space. Movement of the packing rings can lead to high packing leakage if the rings lift before the re-expansion of the cylinder gas has occurred. The rings are more prone to shuttling if the rod friction is high and if there are too many packing ring sets for the operating pressure. The dynamics of the rings where easily observed using the clear test packing.

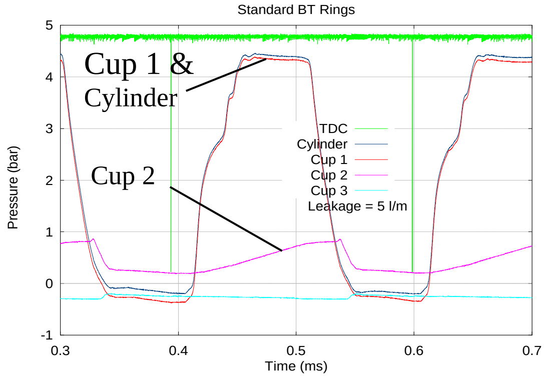

Figure 10: Test results for the standard radial and tangential cut rings

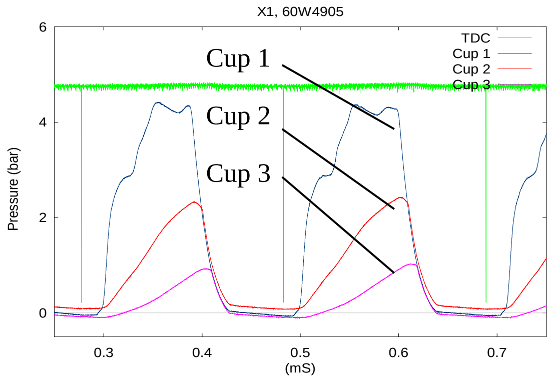

Figure 11: Pressure results for the new BOT ring

The pressure test on the BOT rings show a more even breakdown in pressure for each ring set. This arises because the BOT rings do not seal as tightly as the standard rings. The higher leakage rate can be seen by the increased slope on the graph in figure 11. The BOT rings do not seal so tightly around the rod because they are narrower and also pressure balanced so there is a smaller radial force pressing them against the rod.

Although each BOT ring lets more gas flow past the ring, the total packing leakage was found to be the same as the standard rings. This is due to the fact that the BOT rings have less friction force against the piston rod and so on the reverse rod stroke they shuttle less, reducing the dynamic leakage.

4 The "Thermosleeve" Non-Cooled Packing Assembly

Having developed the low friction BOT ring it was apparent that the ring could provide the low friction, and low heat generation which would be necessary for a non-cooled packing.

The thermal analysis work showed that convective heat transfer is very significant in the packing. The extra flow through the BOT rings helps to aid heat transfer from the packing rings and rod. Because the gas is the significant cooling mechanism in the packing the new Thermosleeve packing was designed using high conductivity materials, so that heat is removed efficiently from all surfaces where the gas contacts the inside of the housing.

Figure 12 shows a picture of the new packing assembly. High tensile bronze was used in the prototype for maximum conductivity, aluminium alloys can be used where copper based alloys are not acceptable due to H_2S or similar aggressive gases.

Using a high conductivity material ensures that the heat transfer through the whole of the packing assembly is good, but once the water jacket system has been removed the heat has no where to go. It is normal practice to make pressure packing assemblies approximately 0.5 - 1.0mm smaller in diameter than the stuffing box of the compressor. This means that there is an air gap around the outside of the packing. The thin layer of air forms a very effective thermal barrier, preventing heat escaping from the packing.

The Thermosleeve design concept addresses the problem by using a special tapered expanding sleeve to make a good solid conduction path between the packing and the compressor. It is normal for water cooled compressors to be cooled in the area of the stuffing box so the Thermosleeve is able to conduct frictional heat directly into this area. Figure 13 shows a typical layout.

The sleeve is activated after the packing has been fitted into the stuffing box using jacking screws accessed from the distance piece.

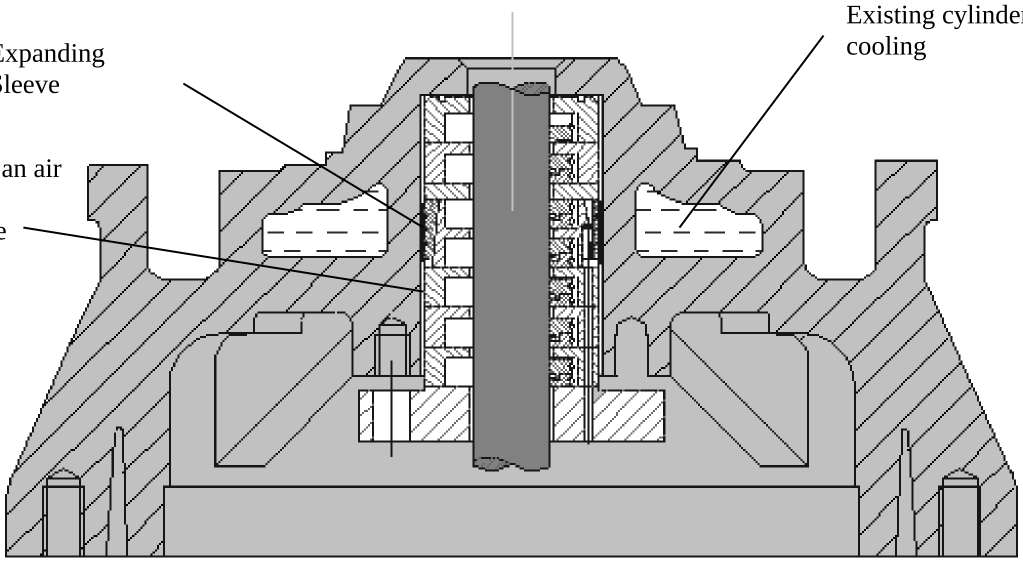

Figure 13: Thermosleeve Packing installed in a compressor

Figure 12: High Conductivity Thermosleeve Packing

5 Field trials on the BOT rings and Thermosleeve Packing

BOT rings have been field tested both in Thermosleeve packings and in conventional packings with and without water cooling. Many trials have been made and all have proved successful in delivering longer life, reduced piston rod temperatures and in conjunction with the Thermosleeve, elimination of water cooling.

A selection of some of the field trials includes:

- Natural Gas booster compressor, the BOT ring and Thermosleeve packing, eliminated problems with gas leakage into cooling system, changed after 12 months, showing normal wear.

- 3rd stage PET air compression, packing removed after 4,000 hrs, predicted life > 8,000 hrs. The rings are shown in figure 13. It can be seen that the principle of transferring force between the rings has worked correctly from the angled wear which has taken place.

- 3rd stage PET air compressor, France, packing installed and now running for over 2 years. Piston rod temperature 105 deg C, standard competitors packings 120 deg C.

- PET air compressors, Austria. Problem of very high rod temperatures with new design of high speed machine, standard packings (non-cooled) raise rod to 260 deg C. BOT and

Thermosleeve reduced rod temperature to 160 deg C.

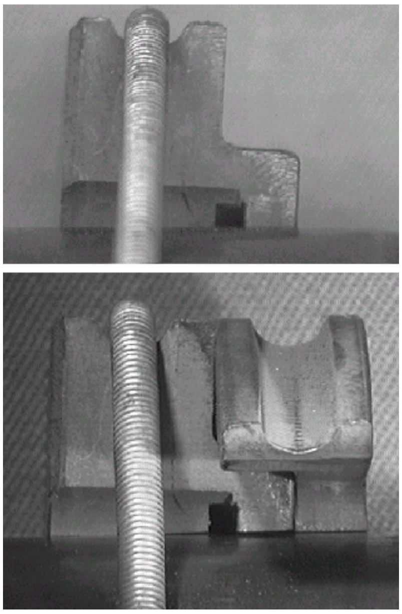

Figure 14: BOT rings after trial in 3rd stage PET air compressor

6 Conclusions

- Thermal analysis shows that the majority of heat is removed from the packing rings by convection to the gas flowing in the packing

- Compressors which run un-loaded for long periods of time, or which run at high speeds with low pressure ratios may not be suitable for non-cooled packings.

- The detail design of the water channels and the amount of water flowing is not the limiting factor for a non-cooled packing with polymer rings. Due to the poor conductivity of polymer ring materials compared to container materials, (PTFE factor up to 100 times worse than stainless steel) the non-metallic ring will always be the limiting factor

- The BOT ring has been seen to significantly reduce the generation of frictional heat in the packing seal and it is ideal for use in a non-cooled packing.

- The lower friction of the BOT rings will reduce energy consumption and increase the efficiency of the compressor. The analysis estimates approximately 50% friction reduction. For the example presented here that equates to an energy saving of approximately 1.5 kW per cylinder.

- The reduced width of the BOT sealing contact reduces the frictional heating.

- The sealing ability of the individual BOT ring is slightly reduced compared to the standard ring but this creates a favourable gas flow which aids cooling. The overall sealing ability of the whole packing assembly is not influenced because the BOT ring dynamics are improved due to lower rod friction.

- The Thermosleeve non-cooled packing design was found to work very well on field trials, particularly when used with the BOT rings.

- The new packing has eliminated problems caused by water cooling circuits, such as contamination and ‘O’ ring leakage.

- The expanding heat conduction sleeve ensures a good fit and good heat transfer into the main compressor cooling system.

- The high conductivity material also helps to remove heat from the gas and dissipate it efficiently to the cooling system.

- Applying the combination of BOT rings and Thermosleeve has significantly reduced piston rod temperatures and packing wear problems in the test compressors, leading to greater operational reliability and longer service intervals.

- The simplicity of a non cooled packing has many attractive advantages for the user.

The CART Test Rig – Measuring Wear and Friction in the Cylinder-Ring Contact

by:

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability

May 17th / 18th, 2001, The Hague

Abstract:

In a joint effort of TNO Industrial Technology and industrial partners a test rig has been designed and build to measure wear and friction in the cylinder liner and piston ring contact. The CART (Cylinder And Ring Tribometer) was developed to reduce the number of time-consuming engine or compressor tests. The CART is capable of testing under different environments from ambient temperature till 500 °C. The tribometer is based on a square block on which two or four cylinder segments are mounted on opposite sites of the block. The stroke of the block with its segments is obtained with an advanced hydraulic actuator. In the CART test rig the ring segments are the stationary samples. The ring segments are fixed in their holders and as a consequence only the cylinder-ring contact is tested. In this paper the CART test rig will be presented and results will be shown.

1 Introduction

For improvement of performance of tribological contacts as used in industrial practise a good simulation on a test rig as simple as possible is required. With such a test rig the influence of an alteration in a single contact parameter can be measured. In pin-on-ring tests the original materials with their specific surface conditions, e.g. obtained by a honing process, cannot be used. In addition it should be possible to study the influence of environmental conditions.

The cylinder liner – piston ring contact is one of the most important contacts in a reciprocating compressor or an engine. In a compressor the time span between revisions is directly determined by wear in this contact. In an engine wear and friction in this contact should be reduced for less emissions and increased fuel efficiency. Therefore, in a joint effort with industrial partners, Thomassen Compression Systems, Grasso Products and DAF Trucks, TNO has developed a test-rig for simulation of the cylinder – ring contact. In the CART test rig (CART = Cylinder And Ring Tribometer) segments of original cylinder liners and piston rings can be used. The CART-test-rig is shown in Figure 1.

Figure 1 The TNO CART test rig.

The aim of the CART-project was the development of a test rig that can be used for both engines and compressors to test piston ring – cylinder liner - lubricant material combinations. With the test rig wear and friction can be measured. Friction is measured using a high-speed data acquisition system. A single test consisting of 2 or 4 experiments simultaneously, takes a relatively short period of time, i.e. approximately 20 hours.

Much effort has been made to simulate the tribological contact between ring and cylinder. And indeed the wear mechanisms obtained in the CART are in agreement with industrial practise. By making a ranking of material combinations and appropriate surface conditions the CART test rig will reduce significantly the number of compressor and engine tests.

2 The CART test rig

2.1 Principle

In the test rig the piston ring segments are the stationary samples and the cylinder segments are the moving samples. The cylinder segments are mounted on a square block, which is fixed to a hydraulic actuator on top of the test rig, clearly visible in Figure 1. On each side of the block a segment can be mounted. The hydraulic actuator gives the reciprocating stroke to the block with the cylinder segments. In Figure 2 a schematic representation of the reciprocating displacement in the CART is given.

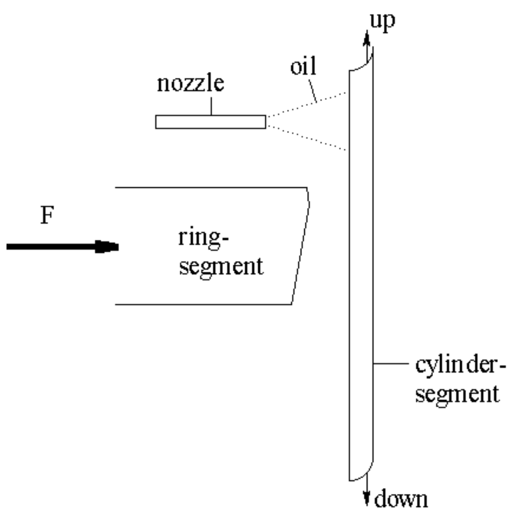

Figure 2 Schematic representation of the reciprocating displacement in the CART. The ring segment is the stationary test specimen.

In Figure 3 a photo of the ring and cylinder segment mounted in the test rig is shown. On the right hand side of the picture the block (marked 1), with the cylinder segment (2) mounted on it, is shown. The piston ring segment is marked 3. The normal force to the four ring segments is applied with the aid of bellows, one of the four bellows is visible (marked 5). The friction forces are measured as bending forces by using strain gauges, one of the strain gauges is visible (marked 4).

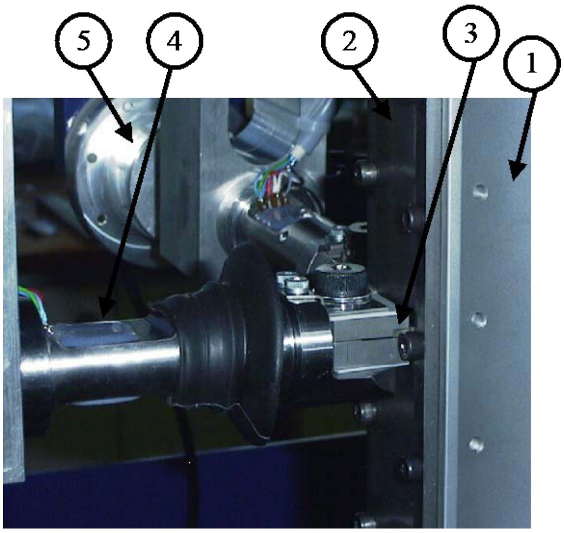

Figure 3 The cylinder-ring contact in the CART test- rig

The piston ring segments are fixed in their holders. By fixing the ring segments the influence of the interaction between ring and cylinder is measured only, not the interaction between piston ring groove and piston ring. In a compressor or an engine the gas pressure on the inner-diameter of the ring and the prestress of the ring supplies the normal force of the ring on the cylinder. In the test rig these normal forces are replaced by one external force applied by the bellows. In engines and compressors the outer diameter of the rings is often larger than the inner diameter of the cylinder resulting in a prestress. To prevent the segments of carrying a non-uniform load, segments are only taken near the slot in the rings, where the radius of the ring is identical to the inner-radius of the cylinder.

If a test is performed with lubrication, the ring segments and cylinder segments are mounted upside down, since in practise the oil will be applied from the lower side of the ring. In the test rig oil is being applied on the upper side. For this reason the ring in Figure 2 is drawn upside down.

2.2 Specifications

In Table 1 the specifications of the CART are listed. Two or four tests can be done simultaneously with loads in the range of 5 - 1000 N which give contact pressures in agreement with actual operation conditions.

Since a hydraulic actuator is used, different types of strokes can be made, the maximum stroke length is 150 mm. Usually a sinusoidal stroke is used, but since an advanced hydraulic actuator is built in, it is also possible to test under conditions of constant velocity with the triangular type of stroke.

Table 1 Specifications of the CART

Simultaneous tests	2 or 4
Load range	5 – 1000 N
Maximum length of stroke	150 mm
Max. frequency	100 Hz
Types of stroke	sinus, triangular, square
Temperature	ambient – 500 °C
Lubrication	4 nozzles, flow rate controlled
Environmental conditions	various types of gasses by using an environmental chamber.
Sample holders can be custom made for a given ring geometry.	

The test-rig was designed to simulate the contact temperatures in a diesel engine, therefore a cylinder temperature up to 500°C can be obtained. The temperature is controlled up to 1°C and heating is programmable e.g. step-by-step or linear.

Inside the block (marked 1 in Figure 3) a heating element provides the heating energy.

The temperature is controlled with the aid of a thermocouple under the surface of one of the cylinder segments, i.e. between a cylinder segment and the block.

If required lubrication can be provided. A nozzle is available for each cylinder – ring contact. The amount of lubricant and the interval between lubricant pulses can be adjusted. In this way the lubrication is well controlled and reproducible.

An environmental chamber (Figure 4) is available to provide for different types of environmental gasses. In order to perform tests with corroding gasses this chamber is made of stainless steel. Lubrication is possible within this chamber.

Figure 4 The environmental chamber.

3 Measurements

3.1 Single measurement

In Figure 5 an example of a measured friction signal is shown. In this figure the measured friction forces of two simultaneous tests are shown, as well as the sinusoidal stroke of the cylinder.

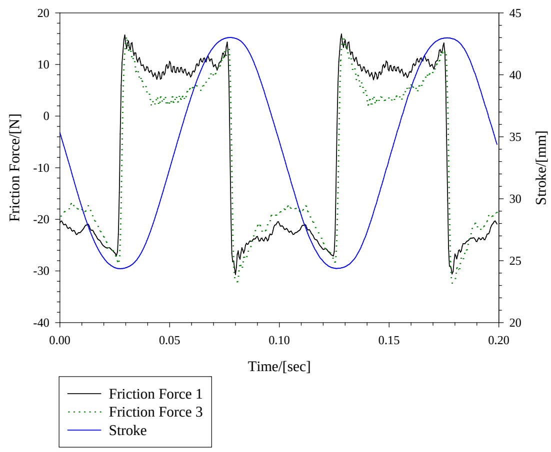

Figure 5 Example of a friction measurement, in the present case two simultaneous friction signals are shown, on the positions labelled in the test rig 1 and 3.

The rings in the case shown are tapered and lubrication is being supplied. The lubrication causes the ''bathtub'' effect in the friction signal. At high differential velocities the friction forces have their minimum as can be explained by elasto-hydrodynamical effects in the Stribeck curve [1].

Data acquisition with a sampling rate of about 1 kHz enables a detailed registration of the friction signals in the ms range. Analysis of the signals shows that details in the signals remain on the same position in a stroke during a 20-hour test, although in general the larger features in the signal become less pronounced during a test. Furthermore, a detailed look at the signals in Figure 5 reveals that the signal in the upward direction is different from the signal in the downward direction.

3.2 20-hour test

In Figure 6 the result of a 20-hour CART test is shown. The average friction of a single stroke, up as well as down, of two ring-cylinder combinations is plotted as a function of time. For this purpose every 30 seconds the signals of two full strokes of the block were recorded. The top dead centre and the bottom dead centre of the ring relative to the cylinder were located first, then the mean of a stroke was calculated, recorded and plotted.

In Figure 6 the running-in phase can be discerned clearly, this phase takes about 2 hours, afterwards minor changes in friction occur. The difference in the coefficient of friction between the upward and downward stroke of the ring can be explained from the fact that the rings are tapered and lubrication is provided at the upper side of the ring only.

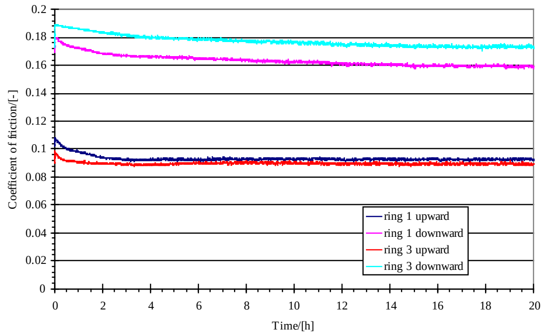

Figure 6 Average friction in a 20-hour lubricated test.

3.3 Wear

Wear has been measured in three ways:

- Mass measurement: the mass of the ring segments can be measured before and after the test.
- Roughness profiles: before and after a test profiles are measured using an optical roughness measurement apparatus. From the difference in profiles the volume worn off is calculated.

- In the engine industry Abbott curves are used to characterise a honed surface. Bearing numbers e.g. Reduced peak height (Rpk), Reduced valley depth (Rvk) and the material core depth (Rk) are obtained from these curves and are often used to quantify wear. These parameters are specified in DIN 4776. Other roughness numbers can be used to quantify wear of a surface with a honed pattern as well.

If the wear of rings is large enough, mass measurement is the preferred method. For cylinder surfaces mass measurement is not accurate enough and roughness measurements will be necessary. A promising method for measurement of very small wear volumes of relatively heavy specimen is differential interferometry [2].

In Figure 7 the results of wear and friction measurements on three different compressor ring materials are shown. Experiments have been done without lubrication and the cylinder material was not changed. In general high friction leads to high wear rates. However, it can be concluded that for the tested compressor ring materials, a decrease in friction results in an increase of wear. This can be explained by transfer of PTFE of the ring to the cylinder. More transfer leads to more wear and less friction. In this case wear was measured by mass measurement.

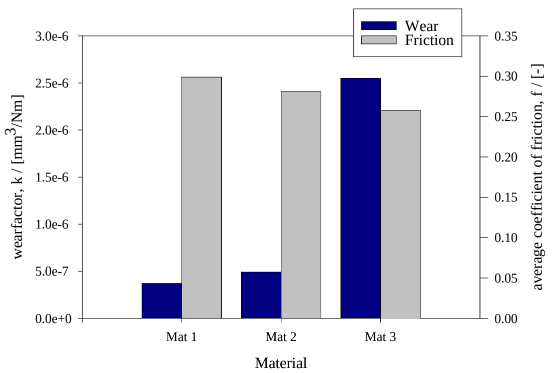

Figure 7 Wear & Friction of Compressor rings made of different types of PTFE. Results are obtained in an unlubricated test.

4 Discussion

The design and first results of a new test rig for tribological measurements on cylinder-ring-lubricant combinations have been described. Friction can be established online by a high-speed data-acquisition system and wear can be measured as well. The test-rig has a robust high stiffness construction and can handle reasonably large forces, frequencies and accelerations.

In industrial compressors or engines the average speed of a piston is very high, however often the wear rate is high in the upper region of a stroke and in this area the piston speed is in the range (0.5 - 2.5 m/s) used in this experimental approach. In fact the critical conditions in the upper and lower region of a stroke are applied in the whole stroke in the CART test rig.

In comparison with standard tribological pin-on-ring test methods the CART test rig offers a lot of advantages as described above. The main advantage is, that parts of original oil rings and cylinders can be used for tests that can be done in a large variety of conditions. If one compares the CART test rig with standard reciprocating wear machines, see e.g. [3], the CART test rig offers some important advantages:

- capability to dose a lubricant in the contact area
- large variety of environmental conditions
- capability for four tests simultaneously
- large stroke and relatively high maximum frequency
- high speed data acquisition in both directions

Therefore the CART test-rig is a unique test facility that enables an improved simulation in comparison with existing test-rigs. As a result the test rig will be used for further R&D programmes with the objective to select or develop material combinations for ring and cylinder surfaces with enhanced tribological performance.

5 Acknowledgements

The authors would like to thank their industrial partners, Thomassen Compression Systems, Grasso Products and DAF Trucks, for their co-operation during the project. We would also like to express our appreciation for their kind permission to publish this article.

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Research of high Pressure Piston Rod Packings

by:
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Life Cycle Costs – Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th / 18th, 2001, The Hague

Abstract:

High pressure rod packings will be investigated up to a pressure of 380 bar. The sealing occurs by means of a combination of special metallic and non-metallic one-piece sealing rings. Sealing effect, pressure distribution and strain of sealing combinations will be compared. Conclusions regarding to further development of seal rings will be made. The experiments are sponsored by MAN Turbomaschinen AG GHH Borsig.

1 Introduction

High pressure piston rod packings designed as shown in *fig. 1* are used up to 500 bar discharge pressure. In this pressure range piston rod packings are lubricated and cooled. On the high pressure side the seal rings consist of sintered bronze with graphite. In cases where bronze is not allowed to be used they are made of sintered iron with graphite. On the low pressure side seal rings of PTFE are used.

The investigations were carried out by means of a BORSIG vertical compressor with piston rod packings on crank side and head side. By combination with a 4-stage pre-compressor a discharge pressure up to 380 bar could be reached.

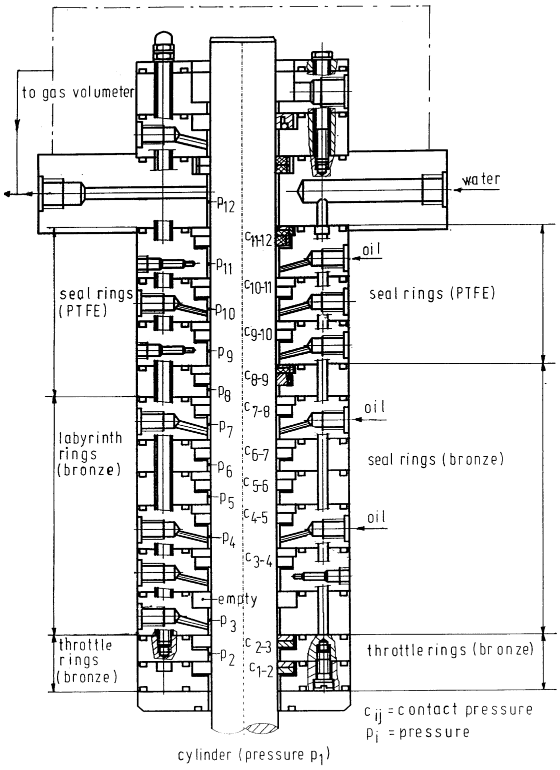

Figure 1. Piston rod packing design

Fig. 1 shows the design of the piston rod packing with two variations of arrangement of the sealing elements: on the left labyrinth rings without gap, on the right seal rings with gap.

The dynamical pressure oscillation is recorded in every chamber between two sealing elements. From the pressure oscillation of two chambers the mean contact pressure of the sealing element between the chambers is calculated.

The pressure p_8 can be adjusted by means of a needle valve. The leakage gas flow rate is measured behind the needle valve as well as behind the flange at the packing end by means of gas volumeters.

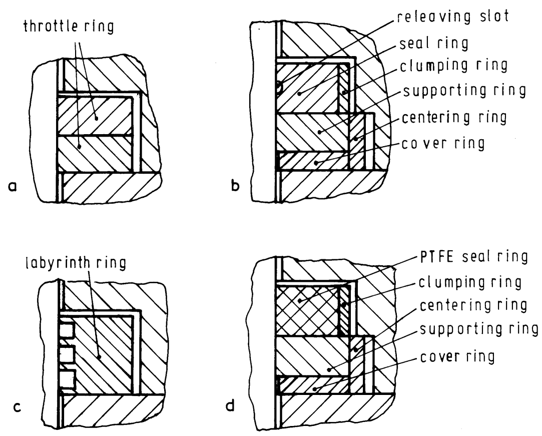

Figure 2. Ring assemblies of investigated piston rod packing

Fig. 2 shows the cross-section of the used sealing elements:
The first and second chamber on the high pressure side contains two throttle rings each (*fig. 2a*). These are bronze rings with a broken gap. They are manufactured with an inside diameter 0,002 mm smaller than the piston rod. Therfore the gap is opened during montage on the piston rod. After some hours operation the gap is closed and the throttle ring has no clearance towards the piston rod.

The following five chambers in the variation of *fig. 1* on the left contain labyrinth rings according to *fig. 2c*. These are bronze rings without gaps with an inside diameter 0,02 mm wider then the piston rod. It was to be tested whether they can lower the pressure down to the following PTFE seal rings clearly in order to releave them.

In the variation of *fig. 1* on the right the two chambers with throttle rings are followed by six chambers with sealing elements according to *fig. 2b*. The bronze seal ring has a radial gap and a release groove on the sliding surface. The supporting ring covers the gap of the seal ring and corresponds to the throttle ring of *fig. 2a*. The covering ring of iron is centered to the supporting ring and supports the inside diameter of the supporting ring. The last chambers (*fig. 1*, left) respectively the last three chambers (*fig. 1*, right) contain seal rings of PTFE instead of bronze.

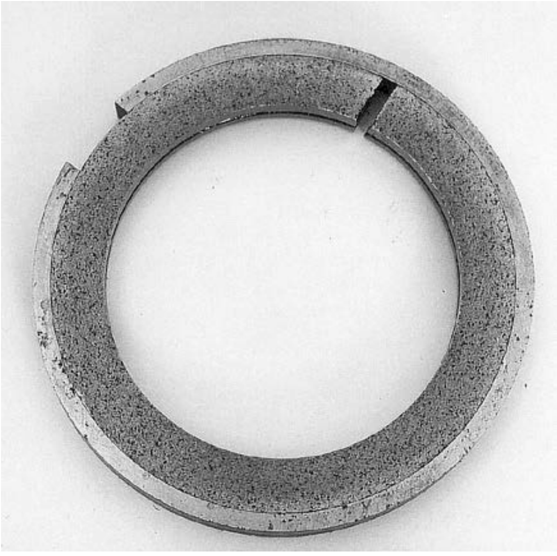

Figure 3. Seal ring (bronze) with clamping ring

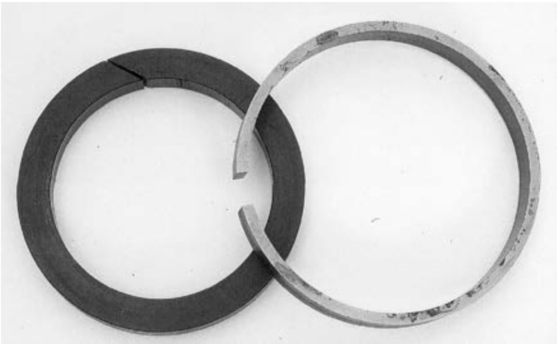

Figure 4. Seal ring (PTFE) with clamping ring

Figure 5. Cover ring, supporting ring and centering ring

Fig. 3, 4 and *5* show the construction of the rings.

2 Ring wear

The piston rod packing had already a longer running time before beginning of the first experiments. After that the fully functioning packing has been dismantled and wear on the throttle rings of the first chamber was confirmed.

Figure 6. Radial wear beside the gap of throttle ring

Figure 7. Radial wear opposite the gap of throttle ring

Fig. 6 and *7* show the wear at the inside diameter of the first throttle ring. The wear appears at the gap and opposite only.This can be declared by the fact that the ring has been manufactured with a smaller inside diameter and retains the smaller curve radius beside the gap. At the opposite side the ring supports itself.

Figure 8. Axial wear of seal ring

Fig. 8 shows wear on the axial surface of the ring as well. It is caused by the radial oscillation of the piston rod together with the ring. This wear has no influence on the tightening effect of the ring.

3 Pressure distribution and contact pressure in the piston rod packing

3.1 Contact seal rings

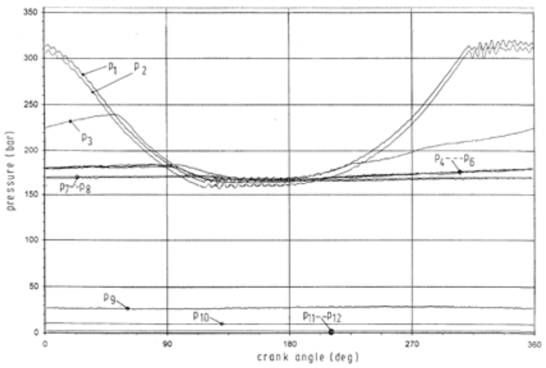

Figure 9. Pressure distribution in the packing at 170 / 320 bar and 400 min^{-1}, using contact seal rings

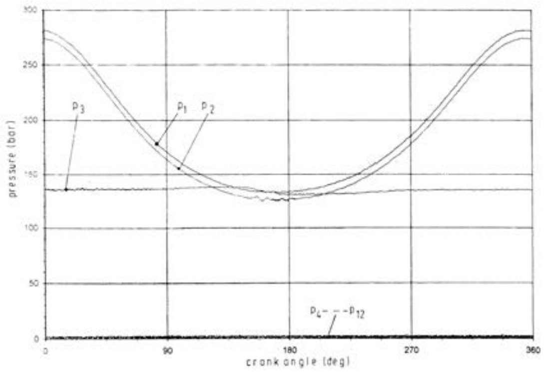

Figure 10. Pressure distribution in the packing at 140 / 280 bar and 600 min^{-1}, using contact seal rings

Fig. 9 and *10* show pressure distribution in the piston rod packing with bronze seal ring according to *fig. 1* on the right:
Only one pair of seal rings takes the most part of pressure range. In *fig. 9* the last pair of bronze seal rings causes a pressure drop of 140 bar which corresponds to a contact pressure of 7 N/mm^2 (c_{89} in *fig. 1*). In *fig. 10* the first pair of bronze seal rings causes a pressure drop of 132 bar which corresponds to a contact pressure of 6,6 N/mm^2 (c_{34} in *fig. 1*). There is no reproduceable dependency of the pressure distribution on the mean cylinder pressure or the speed. During later measurings an exchange of the distribution of pressure drop could be stated.

The throttle elements doop the dynamical cylinder pressure.After installation of new throttle rings the first pair tightened completely the pressure difference (*fig. 13*).

The PTFE seal rings in any case evidently are relieved by the throttle elements and the bronze seal rings. The contact pressure of the PTFE seal rings was lower then 1 N/mm^2.

3.2 Labyrinth rings

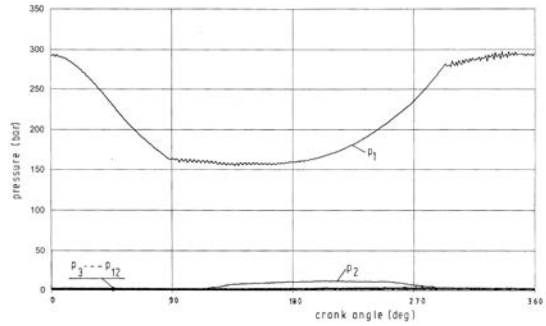

Figure 11. Pressure distribution in the packing, partially using labyrinth rings. P= 160 / 290 bar, n=226 min^{-1}, external pressure p_8 = 0 bar

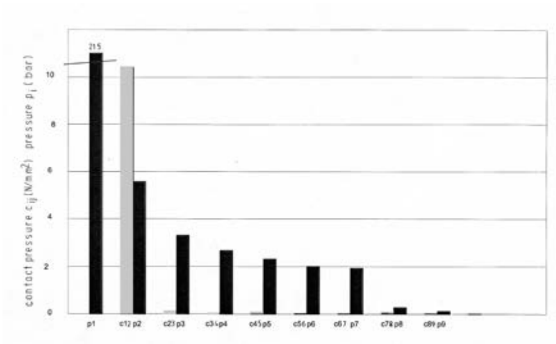

Figure 12. Mean pressure and contact pressure in the packing corresponding to fig. 11

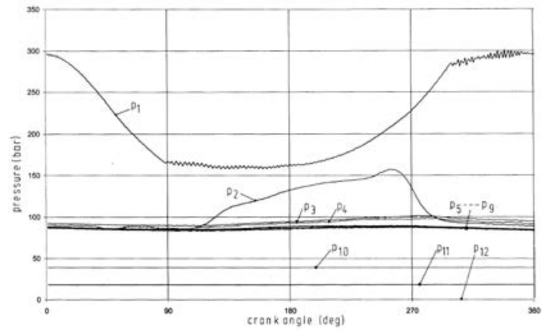

Figure 13. Pressure distribution in the packing, partially using labyrinth rings. P= 160 / 290 bar, n=227 min^{-1}, external pressure p_8=80 bar

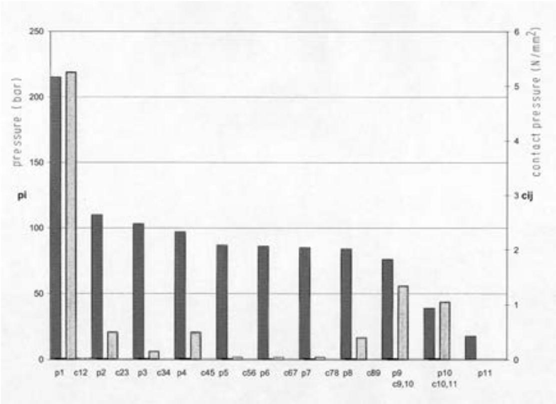

Figure 14. Mean pressure and contact pressure in the packing corresponding to fig. 13

As an alternation to metallic contact seal rings between throttle rings and PTFE seal rings labyrinth rings (*fig. 2a*) were installed according to *fig. 1* on the left. The pressure p_8 behind the last labyrinth ring respectively in front of the first PTFE seal ring could be changed by means of an external needle valve.

The contactless working labyrinth rings were expected to have better live time then the metallic contact seal rings. The question was, wether they can drop the pressure evidently in order to releave the PTFE seal rings.

At a pressure $p_8 = 0$ bar the PTFE seal rings are releaved completely. The leakage flow rate behind the needle valve was 5,6 m^3/h. According to *fig. 11* and *fig. 12* the first throttle element effected with a pressure drop of 208 bar nearly the complete tightening. The contact pressure in this case is 10,4 N/mm^2. The labyrinth rings are not active.

At a pressure $p_8 = 80$ bar the leak rate was 25,5 m^3/h. Again the first throttle element effected with a pressure drop of 104 bar the maximum figure. The contact pressure here is c_{12} = 5,2 N/mm^2, *fig. 13* and *fig. 14*. The labyrinth rings are nearly not active. With $p_8 = 80$ bar of course the PTFE seal rings are strained, the contact pressure here is $c_{10,11}$ = 1,3 N/mm^2.

4 Sealing effect of metallic contact seals

Fundamental research of the sealing effect of metallic contact seals was carried out by BARTMANN /1/. For laminar flow in contact seals, Bartmann proved the validity of the theoretical law of flow resistance with the resistance number 94 / Re. Hence it follows the connection between mass flow rate $\dot{m}$, clearance h and the acting pressure p_i and p_j.

With u = circumference of clearance, R = gas constant, Re = Reynolds number, T_1 = gas temperature, η = dynamical viscosity, l = length of clearance in direction of flow, the resistance number λ equals:

$$\lambda = 2u^2 h^3 / (RT_1 l \dot{m}^2)(p_1^2 - p_2^2)$$

$$\lambda = 94 / \mathrm{Re} = 94 u \eta / (2\dot{m})$$

Using these references for the present sealing case, one can calculate the equivalent clearance of a sealing ring from the measured leakage and the acting pressure difference. The clearance depends from mean contact pressure $(p_i - p_j) / 2$ of the sealing set.

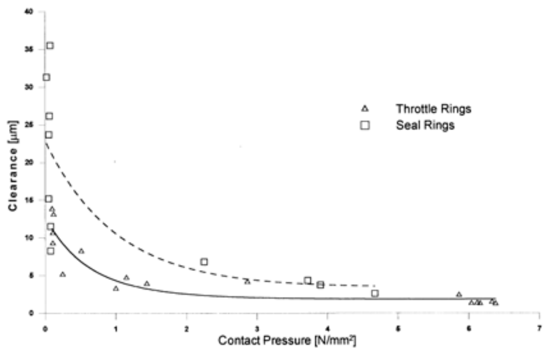

Figure 15. Clearance as function of contact pressure for throttle rings and bronze seal rings

Fig. 15 shows the calculated clearances for the throttle rings and bronze seal rings. The clearance is comparable to the roughness of the machined surface. A seal ring combination the better, the smaller the clearance at given contact pressure is. According to *fig. 15* the throttle ring tighten better then the bronze seal rings. The reason might be the release groove and the wider gap of the latter. With throttle rings an effective tightening already is possible at a contact pressure of 3 N/mm^2.

The results shown in *fig. 15* were obtained at the compressor test bed. The tests will be continued at a static test bed. Here also the pressure distribution in axial and circumference direction of a seal ring can be tested.

Further model tests were executed with a pin – ring tribometer. The contact pressure range was between 6 N/mm^2 and 9 N/mm^2. This range is equal to the figures gained from the compressor. The lowest wear coefficient was $3{,}5 \cdot 10^{-8}$ mm^3/Nm, gained from a PEEK-compound pin against a WC-Co coated ring. These tests were executed with air and dry nitrogene. At time tests with ethylene are going on.

5 Conclusions

From the test conclusions can be stated:

- An absolutely tight seal ring for high pressure piston rod packings would be disadvantageous. It would have to suffer the complete pressure difference and very high contact pressure.

- With increasing piston rod run-out a sudden and short change of loading and sealing effect of the rings occurs. Further single rings of a set can rotate against one another. Therefore an additional piston rod guide would be helpful.
- Measures for pressure release should be taken for a contact pressure between 5 N/mm^2 and 7 N/mm^2 at metallic seal rings and less then 2,5 N/mm^2 at PTFE-seal rings.
- With contactless labyrinth rings no improvement of the sealing effect can be obtained.
- Because of the simple design the one-piece seal rings will be favorized in future instead of segmental ring design.
- Even the one-piece bronze seal ring can be improved. The clamping ring can be left out and finding a tougher material for the supporting ring, the cover ring and the can be left out.

6 References

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On the Strength of Built Pistons

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th / 18th, 2001, The Hague

Abstract:

Low-pressure pistons, mainly those used in multi-stage reciprocating compressors, often have to be optimised in terms of lightweight construction thereby taking into account the controllable inertia forces. For this purpose, welded or screwed semi finished products or cast blanks are applied. The pistons are rotationally symmetrical or designed with ribs. The strength behaviour of these pistons is calculated with FEM programs, which allow the examination of the obtained geometry without considerable simplification. As the strength is practically tested, attention should be paid to the fact that the time-consuming calculation is focussed on those crank angles which are subjected to extreme load variations and that the calculated spatial stress distribution is reliably assessed including the actual material parameters. Experience shows that the critical crank angles can be predicted relatively reliably as the bending stress, which results from pressure and inertia forces, is obtained by a one-dimensional estimation at the point where the piston head is fixed. A local deflection stress can be used to evaluate the stress distribution. The comparison of this deflection stress with a deflection strength, which is obtained from the non-proportional elongation with due regard to the influences of the mean stress, the temperature and the surface, gives an explanation of the damages observed. The latter is particularly vital for making reliable statements.

1 Introduction

Longstanding operational experience gathered with piston compressors that are used in the most various application areas also include piston failures. Such failures are particularly significant as they often lead to major subsequent failures.

An analysis of the piston damages and their causes reveals the following picture:

One failure group can be explained by defects occurring during manufacture and assembly and also by operational conditions that were not duly known. These are for example

- ruptures in pistons as a result of loose or detached connecting bolts,
- loose cores moving outwards in the cast pistons
- broken or loose connection to the piston rod
- faulty burrs and weld seams in the piston
- seized pistons due to thermal extensions
- damages due to insufficient end play or radial play of the piston

Figure 1 shows the components of a jammed two-part piston. This defect was caused by intense stretching of the piston rings, which made the connecting bolts and the piston rods come off.

Figure 1: Damaged piston parts

Experience shows that corrosive gases have an effect on the surface, in particular in areas of high stress concentrations, in the form of fractures. The corrosive gas components may be present in the pumping medium only temporarily. As a result, they were not mentioned when they were ordered and they were not being paid due attention as the machine was designed.

Another failure group results from the insufficient strength of the piston itself. Mostly, it becomes apparent by fatigue fractures in the piston head or bottom. These fractures are caused by notch-sensitive spots and can result in the separation of the piston skirt.

The fact that it is inevitable to approach the strength limits in piston design, in particular for low pressure stages in multi-stage reciprocating compressors, is due to the efforts that are made to make inertia forces manageable. It is known that the oscillating inertia forces in multi-crank piston engines, whose crank shafts and cylinder design fulfil certain symmetry conditions, can be counteracted to a great extent. This requires a homogeneous engine design, i.e. oscillating masses of the same size at each crank. In contrast to the multi-cylinder combustion engine, the multi-stage reciprocating compressor is originally not a homogeneous engine. Designing pistons of the same stroke but with greatly varying diameters for different stages and making their oscillating masses uniform calls for sophisticated structural designs in the field of lightweight constructions.

Figure 2 shows three design alternatives of double-acting low-pressure pistons which in case a) exclusively consist of rotationally symmetrical components, and which in case b) have also periodically arranged ribs. In both cases there is only one central strut between the piston and the piston rod. In case c) the piston consists of two parts and is additionally held by screws that are evenly arranged along the periphery.

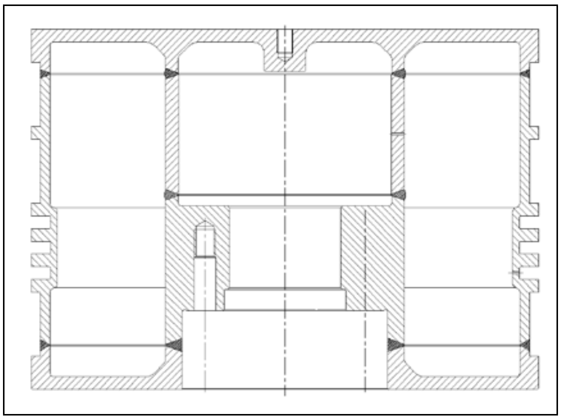


Figure 2a

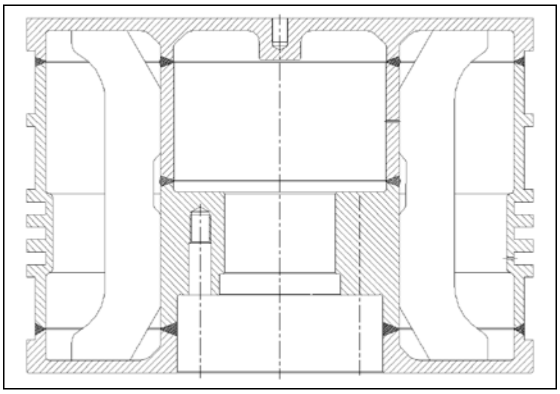


Figure 2b

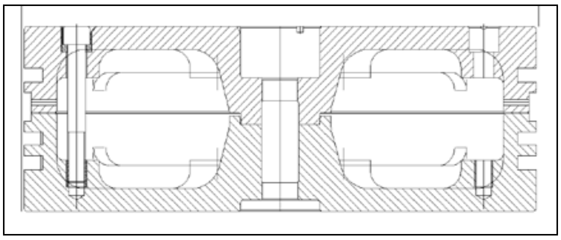


Figure 2c

Figure 2: Various designs of low-pressure pistons
a) rotationally symmetrical design
b) rib design
c) screwed piston consisting of two parts

As a result of the periodically changing pressure in the adjacent working areas and its approximately harmonic motion, the piston is subjected to alternating stress that implies periodical deformations and stress distributions.

The proof of strength requires an FEM calculation that reflects the piston geometry and the piston stress as accurately as possible. It is necessary to compare the temporal stress behaviour, which has thus been determined in certain exposed points, with the alternating stress paying due attention to all strength-reducing factors.

2 Load assumptions

During the constructional phase of the piston, the stage pressure limits, the valve pressure losses and the damage area ratios of the working areas have to be known so that the indicator diagrams can be reliably calculated in advance. Thus, the pressure load at the piston head and bottom is given.

The hollow space within the piston is indirectly connected with the adjoining working areas by compensating holes that lead to the piston ring grooves. As a result of the small cross sections of the connection holes, we can start from the fact that the maximum and minimum pressures at both sides of the piston will produce an average pressure value inside the piston.

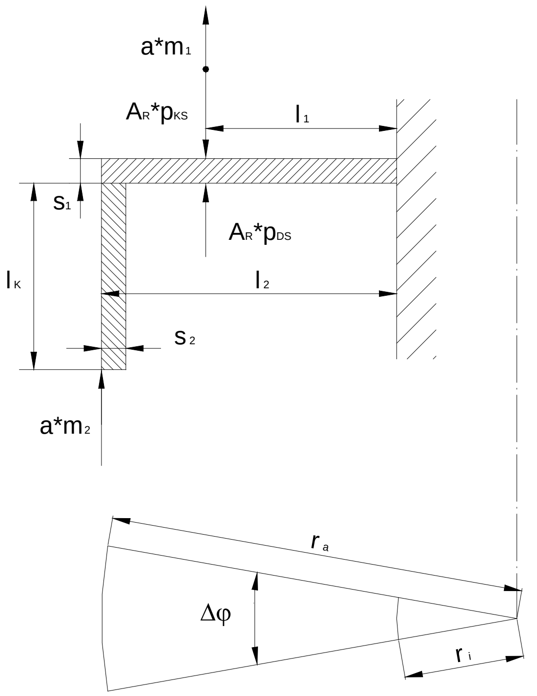
Owing to the time-variable piston acceleration a, which can be determined from the kinematics of the driving mechanism, the piston is subjected to inertia forces that can – in fast running machines – obtain the same dimensions as the gas force load.

While for rotationally symmetrical pistons the two-dimensional FEM calculation is not time-consuming at all so that the piston load can be determined by a quasi-continuous calculation with small angle steps, the three-dimensional FEM calculation for ribbed pistons requires so extensive resources that it should focus just on the extreme load situations.

Since pressure and inertia loads normally lead to opposite types of stress a sufficiently exact estimation of the crank angle of extreme overall load has to be carried out before the FEM calculation.

Figure 3a shows the schematic assumptions applied to assess the extreme load time [2]. Previous calculations indicate that extreme load may occur at the place where the piston bottom and the piston hub meet. The bending moment of the gas and inertia forces at this point can be used as a good approximation of the measure of the time-controlled overall load.

A differential sector of the piston is considered to determine the moment. The gas force and the inertia force of the piston bottom have the lever arm l_1 opposite the critical point where the piston is fixed, while the inertia force of the piston skirt has the lever arm l_2. Figure 3b shows the moment curve calculated on these assumptions. It can clearly be seen that the maximum-load crank angle differs from the dead-centre position by about 80 to 100 degrees for the piston considered.



Determination of moments

$$A_R = \Delta\varphi\left(\frac{r_a{}^2 - r_i{}^2}{2}\right)$$

$$l_1 = \frac{2}{3}\,\frac{r_a{}^3 - r_i{}^3}{r_a{}^2 - r_i{}^2} - r_i$$

$$l_2 = r_a - r_i$$

$$m_1 = A_R * s_1 * \rho$$

$$m_2 = r_a \Delta\varphi * s_2 * l_K * \rho$$

$$M = A_R l_1 \left(p_{DS} - p_{KS}\right) - a\left(2 m_1 l_1 + m_2 l_2\right)$$

Figure 3a

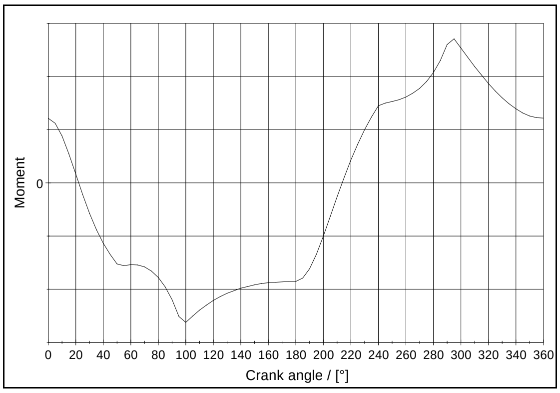


Figure 3b

Figure 3: Determination of the crank angle for extreme loads
a) Diagram of the moment calculation
b) Moment as a function of the crank angle

3 FEM calculation

The actual calculation of strength was done using the finite element program ANSYS, namely versions 5.3 and 5.4. For the rotationally symmetrical design, the piston was approximated with about 4,000 elements [1], while the meshing density was increased in areas of narrow curves (Figure 4).

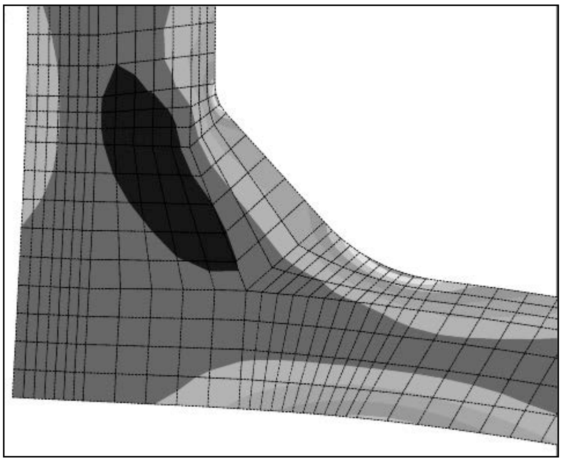


Figure 4: Two-dimensional networking of the piston according to Figure 2a

Figure 5 demonstrates the reduced stress against the time at the places of very heavy loads.

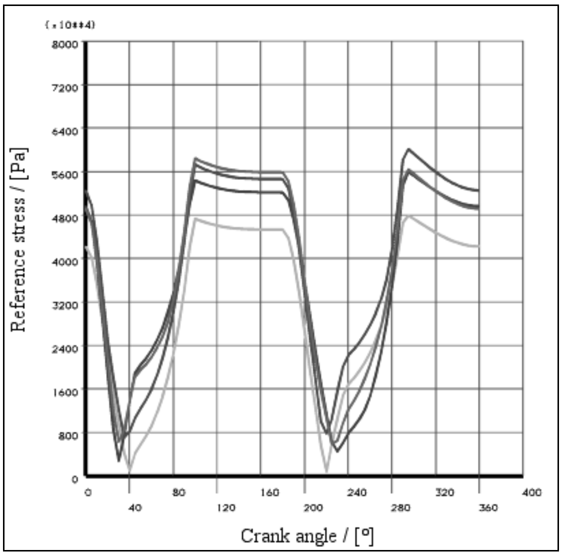


Figure 5: Reduced stress of pistons versus time according to Figure 2a

The sector symmetry of the ribbed pistons made it possible [4] to obtain a rather small calculation area. Eight-node elements (SOLID45) are employed for modelling. For piston diameters between 700 and 800 mm, the element dimensions have been chosen such that the edges were between 5 and 10 mm long. For the rib weld, the meshing was performed so finely that no additional stress concentration factors are necessary.

Figure 6 shows the geometric model with the boundary conditions. The frictional forces that act on the piston rings have been neglected.

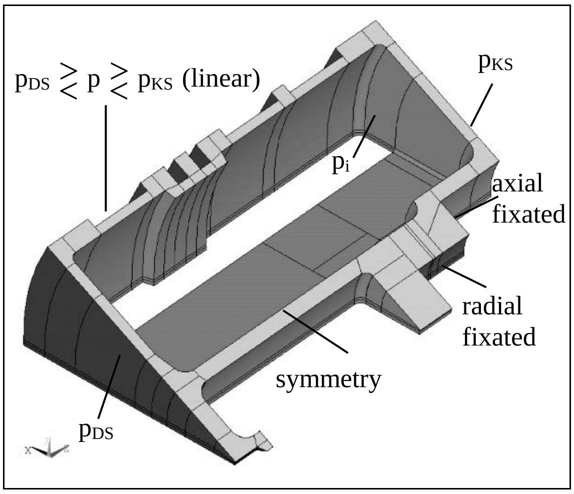


Figure 6: Spatial sector model for pistons according to Figure 2b with boundary conditions

The calculation produces the displacement vector and the stress vector in each node. The axial displacement mainly occurs in the outer piston area due to the strengthening effect of the ribs. However, strong gradients occur at the bottom near the rib end (Figure 7a).

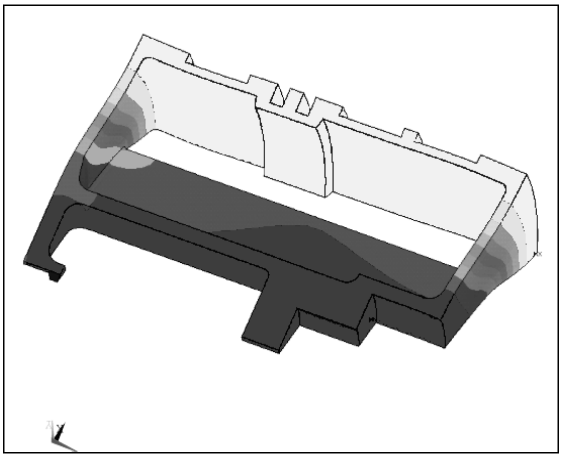


Figure 7a: Calculation results for pistons according to Figure 2b - axial displacement

As it was expected, this is also the place of the maximum reduced stress (Figure 7b).

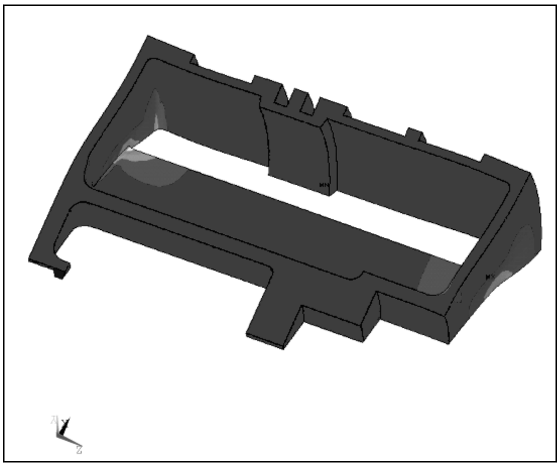


Figure 7b: Calculation results for pistons according to Figure 2b - reduced stress

The state of deformation and stress is the same for the two engine arrangements, which are displaced by approximately 180 ° crank angle, with extreme load, while the absolute values of the vector quantities are about the same in some places, the direction is, however, opposite.

4 Stress evaluation

For the assessment of actual loads, i.e. the estimation of the **existing** safety compared to fatigue fractures, it is necessary to determine the **actual** reduced stress range amplitude and also the reduced stress range amplitude that is **permissible** when all strength-reducing influences are taken into account. This can be done using the instructions given in AD-Merkblatt S2 [6], which – strictly speaking – applies only to pressurised containers and ductile iron materials.

The actual reduced stress range amplitude has to be determined from the time vs. reduced stress curve in the area considered, where the reduced stress range amplitude (double deflection stress) can be equated with the difference between maximum and minimum value during one cycle.

The permissible reduced stress range amplitude can be derived from the permissible stress range amplitude for unnotched specimens in case of pure alternating stress that is shown in [6] as the function of tensile strength and endurance.

Appropriate adjustment factors are incorporated to take into account the variation of the permissible stress range amplitude due to the effects produced by the surface, the wall thickness, the mean stress, the temperature and the weld (Table 1).

quantity	value
tensile strength R_m in MPa (for St 44-2)	600
stress range of the unnotched specimen $2\sigma_a$ in MPa (for $N > 10^8$)	236 (without reduction for the weld as the geometric simulation is sufficiently good)
surface factor f_O (for Rz = 12,5 µm)	0.82
wall thickness factor f_d (for s < 25 mm)	1
temperature factor f_T (for T < 100 ^{0}C)	1
mean stress factor f_M	1 (in spite of $\sigma_{vm} > 0$ not $f_M > 1$ due to altering stress direction)
permissible reduced stress range $2\sigma_{azul}$ in Mpa	194
actual reduced stress range $2\sigma_{va}$ in MPa	120
existing fatigue strength S	1.61

Table 1: Determination of existing fatigue strength

Assuming that the FE calculation has been sufficiently solved in the weld area thus allowing for the notch effect of the weld, the weld is not adjusted in this evaluation.

The ascertained fatigue strength of S = 1.73 proved to be sufficient. Assuming maximum permissible pressure parameters, the same procedure produced a safety value of 0.89 for a damaged piston (according to Figure 1a).

5 Optimised piston design

As the local stress peaks that are critical for the fatigue strength can be attributed to the strong deformation gradients occurring at the rib end, it was obviously desirable to increase the strength by varying the rib design. The geometry alterations were supposed to cause a more gradual stiffness transition in the critical area. For this purpose, rib height H, connection width A and rib thickness T (Figure 8, Table 2) were altered. A ribless variant was examined as well [4].

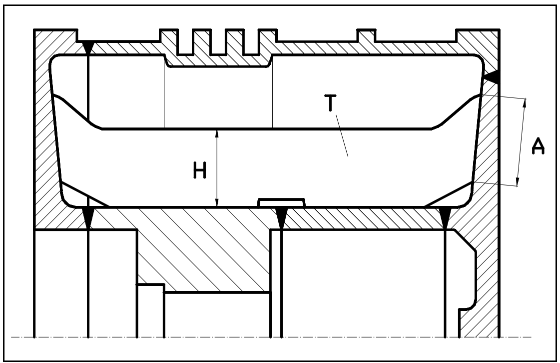


Figure 8: Variation parameter of the rib for pistons according to Figure 2b

variant	rib height	connection width	rib thickness	max. axial displacement	max. reduced stress
	H/mm	A/mm	T/mm	U_z/mm	σ_v/MPa
0	80	80	10	0.16	120
1	75	80	10	0.14	116
2	85	80	10	0.13	123
3	85	90	10	0.12	103
4	85	95	10	0.12	100
5	85	95	12	0.11	91
6	85	95	8	0.13	118
7	ribless design			0.34	107

Table 2: Overview of the parameter variation made for the rib design

The variant comparison allows the following conclusions:

- As the rib height increases, the maximum reduced stress grows in the area where the rib is connected to the internal cylinder (variants 0 to 2)
- As the connection width increases, the maximum reduced stress decreases (variants 2 to 4)
- Thicker ribs also reduce the maximum load (variants 4 to 6)
- If we compare the initial variant 0 with variant 5, the maximum reduced stress is reduced by 23 %.
- With the ribless variant 7 the load can also be reduced by 20 % if compared to variant 0. However, it involves a considerably greater axial displacement, which would call for thicker piston bottoms.

The realisation of the improvements presented here is limited due to technological and design constraints – in particular due to the distance between the ribs at the internal diameter, which is necessary for welding.

6 Evaluation of aluminium pistons consisting of two parts

Strength calculations have not only been carried through for welded steel pistons but also for built aluminium pistons (Figure 2c).

To reduce the FEM calculations to a feasible extent, the screws arranged along the piston periphery were not modelled in detail. Instead they were approximated by a substitute force derived from preliminary theoretical investigations. These examinations did not include the central piston fixation either, which was therefore integrally incorporated using the screw connection design [5].

Figure 9 shows the calculated distribution of the reduced stress which reaches its absolute maximum on the transition radius towards the cast-in rib. In contrast to this, the local maximum at the working point is considerably smaller than the screw replacement force.

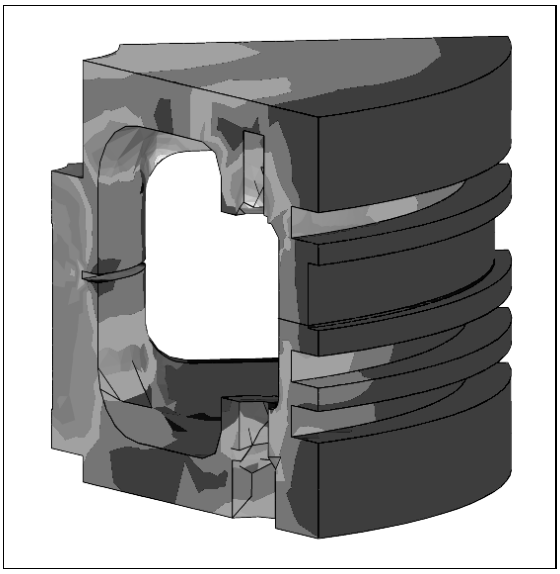


Figure 9: Calculated reduced stress for pistons according to Figure 2c

Special publications dealing with the strength of aluminium [7, 8] could be used to evaluate occurring loads.

The procedure proposed in [7, 8] is principally the same as the one given in [6]. There are quantitative differences concerning the calculation of the correction values for the effects of temperature, roughness and mean stress.

The check up of the initial variant produced safe results for the critical point at the end of the ribs only in the range 0.99 to 1.73. An increase of the curvature radius lead to the necessary safety level.

7 Conclusions

It is vital to avoid piston breakage because of the dangerous subsequent failures involved.

Built steel pistons for low pressure stages require a small wall thickness for the sake of lightweight construction. To guarantee the necessary inherent stability internal ribs are often added alternatively. However, their connection points are frequently subjected to notch stress and cause fatigue fracture in case of insufficient design.

A good alternative to welded steel pistons are die-cast aluminium pistons that consist of two shells. As the small hardness of the ring groove material increases wear and tear, the grooves should be protected by cast-in steel inserts or by local hard coatings or should be strengthened by anodic oxidation.

A reliable bolt connection is vital for divided pistons. In particular, the prestress applied to the peripheral screwed connection must not be reduced unduly by tightening the centre screw.

FEM strength calculations should include comparisons between the maximum values of the local reduced stress variations for the least favourable operating point with an acceptable deflection stress which contains all negative influences. The calculated safety values for failed pistons were just below one.

8 References

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THOMASSEN
COMPRESSION
SYSTEMS

Capacity control by means of Hydraulic Operated Variable Clearance Pocket

by:
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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th-18th, 2001, The Hague

Abstact:

Many Reciprocating compressors must be able to operate under a variety of process conditions including variable flow. Several means of flow control have been devised and used in the past.

The simplest and most flexible of all, the by-pass, has the drawback of energy wastage and the associated high costs. Stepped control by means of unloading suction valves has the disadvantage of relatively large capacity steps, as well as energy wastage from valve losses associated with reverse flow in the valves. While the manually adjustable variable clearance pocket does not suffer from these disadvantages, capacity adjustment can only be performed on an inoperative compressor.

Thomassen Compression Systems, wishing to eliminate these disadvantages, and seeking to reduce the Total Cost of Ownership (TCO), has devised an elegant solution for this market requirement in the form of a Hydraulically Operated Variable Clearance Pocket.

The pocket is simple to install on both new and existing compressors. Remotely controlled stepless capacity control, without the described disadvantages, is now possible, and compression power requirements become directly proportional to the flow.

This paper describes:

- Capacity control for reciprocating compressors.
- HCVOP technology.
- Comparison with available systems

1. Introduction

One of the main features of the reciprocating compressor is the possibility of loss free capacity control. This keeps energy cost low at varying demand.

Many Reciprocating compressors must be able to operate under a variety of process conditions. One of the variable process conditions is the requirement for variable flow. On the one hand variable flow may be required due to process design flexibility. The compressor might be suitable to handle more flow than is needed from the process design. On the other hand variable flow might be determined by process gas availability, i.e. natural gas lift.

Several means of flow control have been devised and used in the past. Depending on the requirement for variable flow and the frequency of adjustment of the delivered capacity, optimal flow control devices are selected. Devices can be either stepped or steppless. Next to this capacity adjustment may be required on an operative compressor or not. Maximum flow control flexibility may be obtained by use of steppless control on running compressors. One main disadvantage of current steppless flow control on operative compressors is energy wastage and the associated high costs.

Thomassen Compression Systems wanted to eliminate this disadvantage and that of less maximum designed flow control means. Seeking to also reduce the Total Cost of Ownership (TCO), Thomassen has devised an elegant solution for this market requirement in the form of a Hydraulically Operated Variable Clearance Pocket.

The pocket is simple to install on both new and existing compressors. Remotely controlled stepless capacity control, without the described disadvantages, is now possible, without energy wastage.

2. Capacity control

Capacity control can be obtained in two ways. On the one hand part of the delivered capacity at discharge pressure can be reduced in pressure to suction pressure level and re-routed to the suction side of the cylinder, the so called by-pass method. When looking at a PV diagram (figure 1), the outer PV graph indicates the work and flow delivered by the compressor. On the other hand capacity can be reduced by means of (virtual) increase of clearance volume. The actual delivered capacity is herewith reduced, in the PV diagram indicated by the inner PV graph.

Thomassen reciprocating compressors can be equipped with stepped or stepless capacity control systems. Several control means are applied to reciprocating compressors. A brief explanation of available control means, divided in both ways of control has been given below.

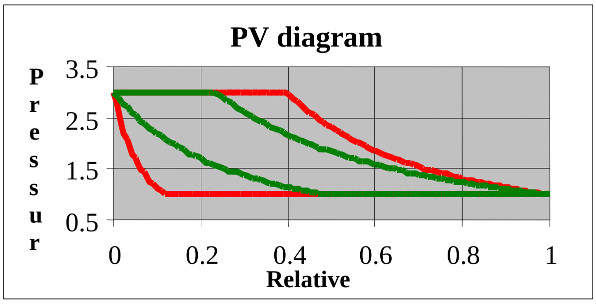


Figure 1: PV diagram

2.1 Stepped control

2.1.1 Cylinder Unloading

Stepped capacity control can be realised by unloading one or two cylinder sides by means of valve unloaders. In this manner a three step capacity control, for one cylinder, can be accomplished: 0-50-100%. More steps can be obtained with more cylinders operating in parallel.

Because of its simplicity and reliability, stepped control is the most commonly used system of capacity control. However, stepped control by means of unloading suction valves has the disadvantage of relatively large capacity steps, as well as energy wastage from valve losses associated with reverse flow in the valves. Further, temperature rise as a result of reversed flow might cause unfavourable environmental conditions for plastic valve plates or piston and rider rings. Lifetime of these materials can be reduced when exposed to high temperatures for longer time.

2.1.2 Volumetric Control

Intermediate load steps can be realised by providing additional clearance volume. A pocket with fixed volume that can either be opened or closed can obtain the additional clearance volume. Adding this clearance volume results in a reduction of the volumetric efficiency, which directly controls the mass flow through the cylinder. In this manner the capacity can easily be controlled since the compressed flow is directly proportional to the volumetric efficiency. Disadvantage of this method is the inflexible control capacity.

2.2 Stepless control

2.2.1 By-pass control

The simplest and most flexible of all means of capacity control is the by-pass control. By use of by-pass control, compressed flow will be redirected to the suction side of a stage or the compressor. The main drawback is continuous and substantial energy wastage and herewith the associated high costs.

2.2.2 Volumetric control manually operated

Stepless capacity control can also be realised by providing additional adjustable clearance volume. A pocket with variable volume to obtain additional clearance volume controls the volume. Adding this clearance volume results in a reduction of the volumetric efficiency, which directly controls the mass flow through the cylinder. In this manner the capacity can easily be controlled since the compressed flow is directly proportional to the volumetric efficiency. Disadvantage of this method is that volume can only be adjusted on inoperative compressors.

2.2.3 Volumetric control remote operated

In opposite to the manually operated volumetric control, Thomassen has developed the Hydraulic Operated Variable Clearance Pocket (HOVCP). This design eliminates the disadvantage of the manual pocket. By means of HOVCP design the clearance volume can be adjusted on operative compressors.

2.2.4 Speed control (VSDS)

By means of speed control the RPM of the compressor can be changed and herewith the RPM and capacity of the compressor. RPM and capacity of the compressor are directly related. Application of VSDS requires the installation of a substantially higher priced motor and speed control unit.

2.2.5 Reverse flow control

Reverse flow control is obtained by controlling the opening time of the suction valves. The volumetric efficiency is hereby influenced. Though capacity can be controlled in a large range, reversed flow trough the valves increase flow losses, increasing the power requirement. Further valve lifetime is affected due to higher gas discharge temperatures and extra loads on the suction valve plates when in operation.

2.3 Comparison of capacity control devices

All capacity control devices have their specific properties resulting in advantages and disadvantages. An overview is listed in below table. The table is made for the most common capacity control range between 50 and 100%.

	Energy wastage	Increased valve load	Flexibility	Economics
Unloading	+	- -	-	+
Fixed pocket	+ +	+ +	- -	+ +
By-pass	- -	+ +	+ +	- -
Volumetric control manual	+ +	+ +	+	+
Volumetric control remote	+ +	+ +	+ +	+ +
VSDS	+	+ +	+	-
Reverse flow	+	-	+ +	+

Table1: Comparison of capacity control devices

3. Hydraulic Operated Variable Clearance Pocket

Thomassen recently introduced the Hydraulically Operated Variable Clearance Pocket (HOVCP). The HOVCP is of the volumetric control remote operated type. The design consists of the following main items; the custom sized clearance pocket, hydraulic cylinder and piston and the power pack assembly. The capacity is controlled by the position of the piston in the pocket. The piston is displaced by means of a remotely controlled hydraulic cylinder. Herewith capacity can be controlled within a customised range.

3.1 Hydraulic cylinder and pocket

Figure 2: Cylinder with mounted pocket

A pocket with a fixed volume is mounted on the head-end side of the cylinder. The actual length of the pocket guide bush, located between the cylinder end and actuator base flange, is custom sized to provide the required capacity.

A conical shaped piston equipped with a riderring is attached to a hydraulic actuator. The riderring has a combined key function, it not only centres and supports the conical piston, but provides a gastight sealing as well.

Figure 3: Conical piston and hydraulic cylinder

The piston nut, which attaches the piston assembly to the hydraulic cylinder, is positively locked as prescribed in the API 618 4th Edition.

The hydraulic actuator, pocket cover and pocket guide bush are attached to the cylinder end by means of studs and nuts.

3.2 Power pack assembly

Figure 4: Power pack

A hydraulic power pack provides operating functions for pocket capacity control. It is connected to the pocket assembly by means of external stainless steel hydraulic hoses and valves.

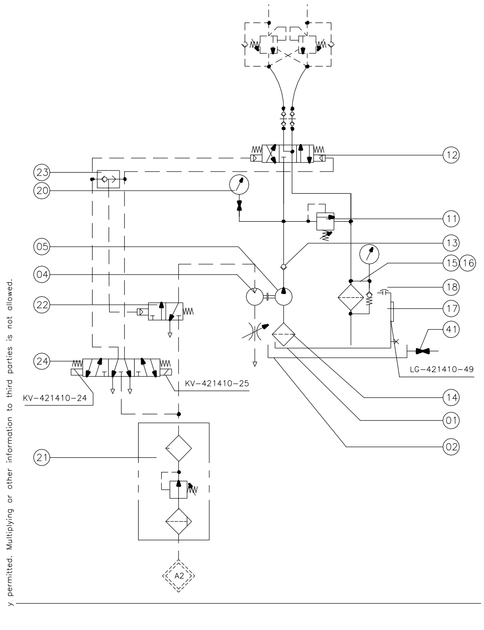
The power pack has been designed for stringent conditions, which implies that this system can be used outdoors and in a continuous hazardous area. It is designed to be mounted directly against the concrete cylinder support structure.

The power pack, consists of a hydraulic reservoir, a filter, non-return valves, electrically operated valves and a pneumatic feed pump. Only air supply is needed for to drive the motor. A hydraulic block valve onto the hydraulic cylinder holds the piston in position after it has been moved. Therefore it is not necessary for the motor to rum continuously.

3.3 Remote control

The advantage of the system is that it can be operated directly from the control room by means of a selector switch. An electrical position transducer has been incorporated, onto the hydraulic actuator, generating a 4-20mA signal in order to monitor the position of the piston.

Optionally this 4-20mA signal can be visualised on a LCD screen. This digital screen indicates the position and visualises the movement of the clearance pocket piston each time the



selector switch is operated from the control room.

Figure 5: System logic

The above scheme shows the system logic for the combination of the hydraulic cylinder and the power pack.

3.4 System advantages

The HOVCP is a high efficiency loss-free capacity control system and economic in use. The design has no negative effects on other compressor parts such as suction valves.

Due to its design, the HOVCP is available to replace existing manually operated versions. The system allows easy and simple field installation, resulting in a minimum of down time. For maintenance purposes, the pocket can be easily disconnected. The construction is simple and has a related high reliability.

Further the hydraulically operated variable clearance pocket can be remote controlled. The control is accurate, as required capacity volume can be determined precisely. It is also possible to record the clearance pocket piston position in a historic database.

No electrical cables and/or provisions are required for installation. Only air supply is necessary. The system can also be controlled locally in case no signal cable is available.

SULZER BURCKHARDT

Comparative Measurements of the Piston Rod Surface Temperature During the Operation of a Dry-Running Crosshead Compressor

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th / 18th, 2001, The Hague

Abstract:

The temperature of the counter surface exerts a considerable effect on the tribological quantities friction and wear and thus influences the operating conditions and especially the life of sealing systems. Hence the counter surface temperature represents an important indicator for the operating conditions of dry-running sealing systems and should be monitored during the operation of the compressor. With packings of crosshead compressors, common methods of monitoring their temperatures are realized indirectly by measuring the gas leakage and cooling water outlet temperatures. The aim of the investigations was to find a suitable method for measuring the temperature of the piston rod surface directly during operation. Three different methods of temperature measurement were tested in a specially prepared packing test stand based on a dry-running nitrogen compressor, and their advantages have been discussed. Additional tests were carried out in a dry-running hydrogen compressor to prove the usability of piston rod temperature measurement as a diagnostic tool. Some of the parameters which significantly influence the piston rod temperature were chosen, and by their variation the clarity and reliability of the information contained in the measured temperatures analysed. The tests showed that an infrared sensor – in spite of the fact that it is not capable to measure piston rod temperatures very precisely due to the varying emission coefficient - represents a valuable tool for the condition monitoring of dry-running packings.

1 Introduction

Temperature has a well-known, high influence on the physical and mechanical properties of plastic materials. With dry-running sealing systems, it also exerts a considerable effect on the tribological quantities friction and wear and thus influences the operating conditions and especially the life of the sealing systems. According to the literature[1-5], an increase of the counter surface temperature results in a significant rise of the wear rate of the plastic sealing elements, that may even show an exponential relationship. Counter surface temperatures should therefore be kept as low as possible to enable a long sealing life. In addition, if the friction heat resulting from the load exceeds a critical level, temperatures may rise quickly and uncontrollably until the sealing system fails completely.

Hence the counter surface temperature can be rated as an important indicator for the operating state of a dry-running sealing system and should be monitored during the operation of the compressor. With packings of crosshead compressors, common methods of monitoring their temperatures are realized indirectly by measuring the gas leakage and cooling water outlet temperatures. The aim of this investigations was to find a suitable method for measuring the temperature directly on the piston rod surface during operation.

2 Methodology and experimental set-up

Comparative tests with various methods of measuring the piston rod temperature in a special prepared packing test stand based on a dry-running nitrogen compressor were planned to show their suitability as a tool for monitoring. The criteria were accuracy, reliability, robustness and cost. However, among the various methods of measuring the rod temperature during operation only those have been taken into account that fulfil the essential demands concerning robustness and reliability to survive the ever increasing service intervals of crosshead compressors.

With a dry-running hydrogen compressor a number of tests under realistic operating conditions were carried out to prove the suitability of rod temperature measurement as a diagnostic tool. For this purpose variations of some of the parameters that mainly influence the rod temperature should show the accuracy, clarity and reliability of the measured information.

2.1 Methods for measuring piston rod temperatures during operation

A very simple and robust method of measuring the piston rod temperature can be realized by a kind of sliding contact between the sensor and the rod. In order to keep the friction heat produced by the sensor itself as low as possible, it has to be pressed against the counter surface with the lowest possible force and should have excellent dry-running properties. Plastic materials based on PTFE offer outstandingly good dry-running properties, but unfortunately they typically have only a poor heat conductivity that may cause a high deviation between the measured and true temperatures.

In order to avoid producing friction heat while measuring the temperature, contactless methods are necessary. Also a very simple and robust solution can be realized by using a thermocouple with a minimum gap between sensor tip and rod surface, hoping that the measured temperature of the ambient atmosphere is as close as possible to the rod temperature.

Another method of non-contact temperature measurement can be achieved by using an infrared temperature transducer (pyrometer). With infrared sensors the emissivity is a major factor in temperature measurement. Among the multitude of parameters that may affect the emission coefficient of a piston rod, its geometry (concave surface!), surface quality (polished!) and especially the thin layer of plastic as a characteristic of dry-running systems may cause considerable difficulties.

2.2 Test stand and sensor arrangement

The experimental investigations were carried out with process gas compressors at Sulzer-Burckhardt's test facility. For the comparison of different temperature measuring methods a dry-running nitrogen compressor of the balanced-opposed type equipped with a special tail-rod design has been used (Fig. 1). In addition, a hydrogen compressor of the same type has been used to clarify the suitability of rod temperature measurement as a diagnostic tool for dry-running packings by long-term tests.

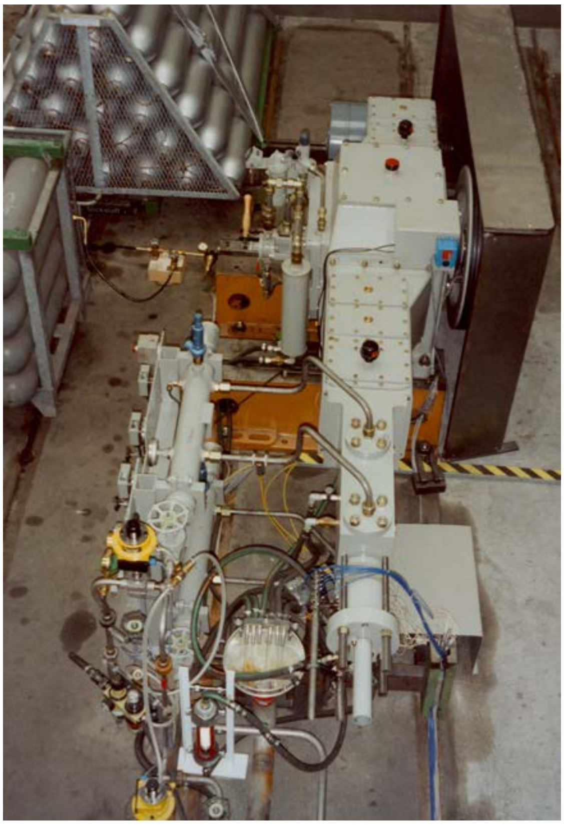

Fig. 1: General view of the packing test stand.

The tail-rod test packing involves the ability to measure the pressure and temperature distribution within a packing (Fig. 2). For the measurement of the temperature in each sealing element chamber, thermocouples were used with a maximum gap of 0.5 mm between sensor tip and piston rod.

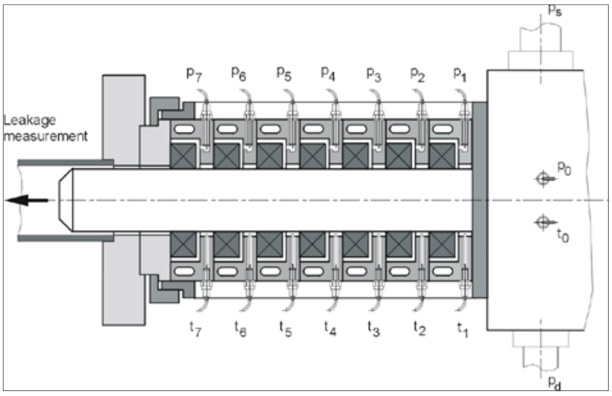

Fig. 2: Cross section of the tail-rod test packing with pressure (p0 - p7) and temperature sensors (t0 - t7).

A special device for carrying up to three thermocouples with an angle of 120 degrees between each was designed to be fixed directly onto the packing flange, as close as possible to the sealing elements (Fig. 3). The thermocouples were used in combination with spring-loaded sliding shoes made of filled PTFE. The sliding shoes were used as a connecting piece between the sensor tip and the piston rod, enabling a direct contact between them. Various filled PTFE grades were tested in order to find a proper compromise between a satisfying heat conductivity and an as low as possible coefficient of friction. Figure 5 shows the device in a special test position, not fixed to the packing flange.

Fig. 3: Measurement of rod temperature by sliding contact of three insulated thermocouples.

For contactless temperature measurement outside the packing, an infrared sensor for potentially explosive atmospheres (intrinsic safety EExib IIC) was used and directly mounted on the packing flange (Fig. 4). Based on its specifications presented in Table 1, it was necessary to have a distance of 75 mm between its lens and the rod surface to achieve the admissible target spot diameter.

Fig. 4: Infrared sensor for potentially explosive atmospheres.

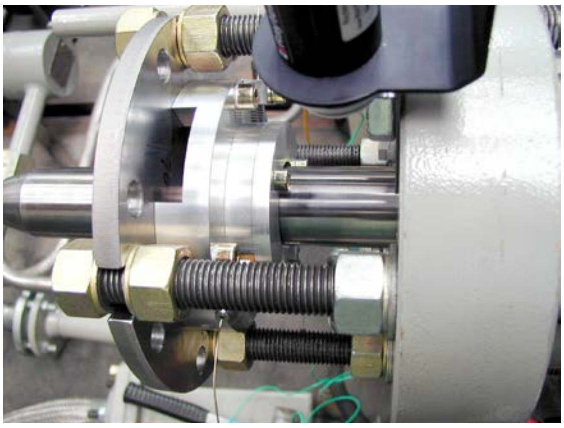

Fig. 5: Tail-rod test packing prepared for temperature measurement (infrared sensor on top).

Specification	Unit	Value
Accuracy	[%] / [°C]	± 1.0 or ± 1.4
Repeatability	[%] / [°C]	± 0.5 or ± 0.7
Emissivity	[-]	0.10 … 1.00
Spectral response	[μm]	8 … 14
Temperature range	[°C]	-18 … 500
Target spot diameter	[mm]	2.5

Table 1: Specifications of the infrared temperature sensor.

3 Comparison of the test results

For the comparison of the various temperature measuring methods, only one dry-running sealing element was used, located in the last ring chamber of the packing. All sealing elements used were of the "tangential to the rod" design. Out of each of the three main groups of dry-running materials – filled PTFE, polymer blend, high-temperature polymer - a typical grade was chosen to test the influence of material on the rod temperature (Table 2).

Material group	Composition of test material
Filled PTFE grades	PTFE, carbon/graphite powder
Polymer blends	PTFE, PPS, carbon fibres, carbon/graphite powder
Modified high-temperature polymers	PEEK, PTFE, carbon fibres, graphite powder

Table 2: Composition of the test materials.

All tests have been carried out with piston rods of 50 mm diameter, made of nitrited steel with a surface roughness of 0.12 μm ≤ R_a ≤ 0.25.

For the test load of the sealing element only a static pressure difference was used, that is $p_s = p_d$. After the start of a test it took about 1 - 2 hours to reach steady-state temperatures on the piston rod. Figure 6 shows a typical example of the differences between the temperatures measured by the various measuring methods during the warming-up period. While the infrared sensor shows temperatures of more than 60 °C, the contactless measured temperatures in the sealing element chamber and those measured by the sliding contact have significantly lower values.

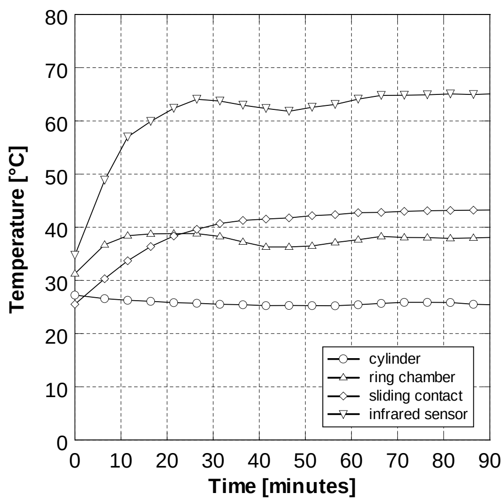

Fig. 6: Temperatures measured by various sensors during operation (p_s = p_d = 20 bar, c_m = 3.19 m/s, carbon/graphite-filled PTFE).

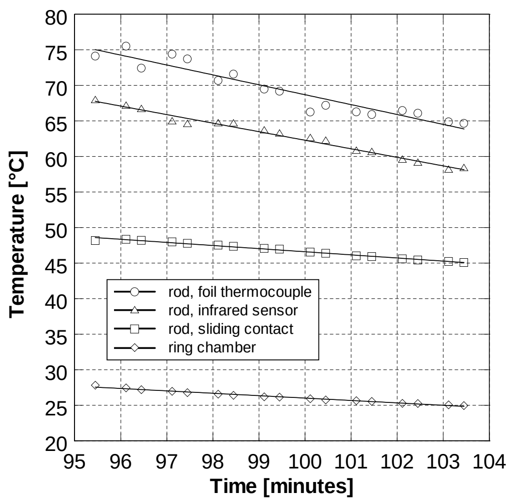

Fig. 7: Temperatures after shut-down of the compressor.

After the warming-up period the compressor was stopped and a foil thermocouple was fixed to the piston rod as a reference for its cooling (Fig. 7). In addition, the rod temperature, which typically decreased by about one degree per minute, was measured by a hand-held device. Thus a deviation of about three degrees was found for the foil thermocouple. Based on the values recorded during the cooling period their deviations from the reference temperature were calculated. Typical values are represented in Fig. 8. A high deviation of 38 °C was determined for the contactless thermocouple located in the ring chamber. As expected, the sliding contact, too, has a high deviation of about 30 °C.

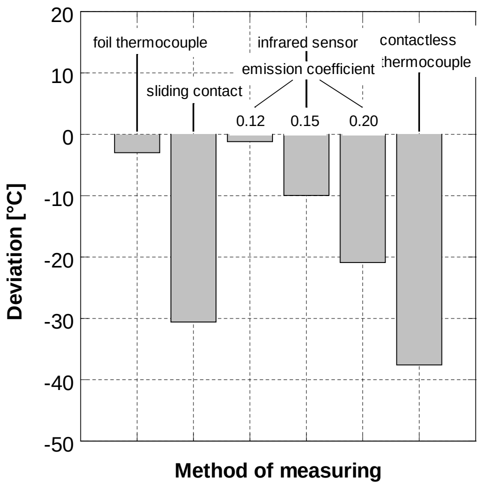

Fig. 8: Deviations of the various sensors compared with the piston rod temperature measured by a hand-held device.

In principle, the temperature deviations of the contactless thermocouple and the sliding shoe can be corrected by the aid of a simplified model for the heat transport between rod and sensor tip. However, with the infrared sensor, deviations can be compensated in a smarter way by adjusting the coefficient of emission. According to the supplier's recommendations the emission coefficient of steel is between 0.1 and 0.8. Figure 8 shows that the best result was obtained with an emission coefficient of 0.12. Unfortunately, this value is only valid for the fully established, dark transfer film of the carbon/graphite-filled PTFE on the piston rod, while for the much lighter transfer films of the polymer blend and the modified PEEK a value of about 0.1 gave better results. In the new state of the rod, without any transfer film, the emission coefficient is below 0.1. Since this value represents the lower limit of the used infrared sensor, tests with new piston rod surfaces resulted in a maximum deviation of up to 30 °C. The difficulties with the emission coefficient are additionally increased by the fact that the infrared sensor sees both parts of the rod, i.e. with and without transfer film.

Regardless of the difficulties with determining the proper emission coefficient, the infrared sensor offers more advantages over the competing temperature measurement methods. For the use as a diagnostic tool a quick response to varying rod temperatures is required. An infrared sensor fulfils this criterion in an excellent way. As an example, Fig. 9 shows the temperatures measured during the shut-down of the compressor. With decelerating speed it becomes visible that the rod temperatures vary along the stroke of 160 mm between a minimum of about 75 °C and a maximum of about 125 °C. Thus the temperature measured during operation represents the average value of the temperature distribution. Finally, another considerable advantage of the infrared sensor is the fact that it can easily be installed into the distance piece of a compressor.

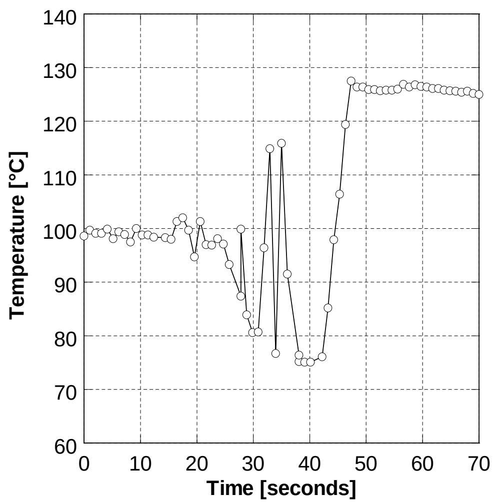

Fig. 9: Piston rod temperature measured by an infrared sensor during shut-down.

4 Using piston rod temperature as a diagnostic tool

The failure of an overloaded dry-running packing typically starts with a quick and steep increase of temperatures within the sealing system. As seen before, an infrared sensor offers a fast reaction to temperature variations, but the accuracy of the measured values depends highly on the correctness of the chosen emission coefficient, whose determination is fairly difficult and, in addition, which does not remain constant during operation. Consequently, the usability of piston rod temperature measurement by infrared sensor as a diagnostic tool was to be checked by a number of tests under realistic operating conditions. Some of the main parameters influencing the rod temperature were chosen, and by their variation the clarity and reliability of the information contained in the measured temperatures analysed.

4.1 Influence of load

In order to have a rough idea of how the pressure difference influences the piston rod temperature, tests with only one sealing element made of various dry-running materials were carried out. As with the tests described before, a stepwise increasing static pressure load ensured that only the friction heat caused the temperature rise of the piston rod. The accuracy of the infrared sensor used to measure the rod temperatures during operation was checked by a foil thermocouple after each test. The results for a carbon/graphite-filled PTFE, a polymer blend and a modified PEEK material are shown in Fig. 10. For the given test conditions in nitrogen of a dew point of about –60°C, lowest temperatures were measured for the modified PEEK, while the polymer blend and especially the filled PTFE caused significantly higher values. Up to a critical pressure load of about 40 to 50 bar, a linear relationship between pressure difference and rod temperature was observed. A further increase of the pressure difference beyond the critical value led particularly with the filled PTFE and the modified PEEK to a steep drop of temperature, which was caused by a sudden increase of the gas leakage. During the test series with the various temperature sensors it was not possible to measure the gas leakage, but their rise was clearly indicated by an increased sound of escaping gas.

The leakage of dry-running friction seals is never really constant and may even change its magnitude to a significantly different level within a short time. Due to the high influence of the packing leakage its variations may also alter the piston rod temperature considerably. Fig. 11 shows rod temperatures measured during a test period, which was characterized by frequent and steep changes of the leakage and thus led to temperature variations of more than 10 °C within a short time.

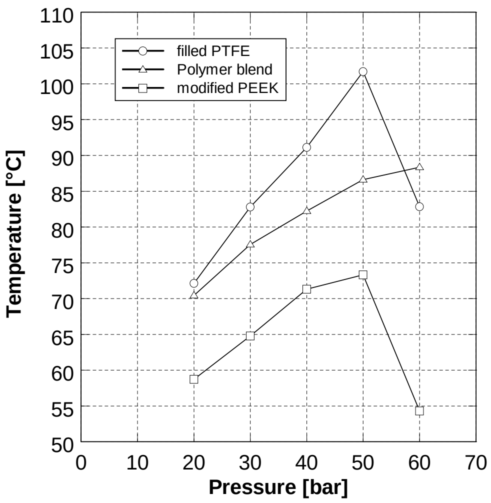

Fig. 10: Piston rod temperature versus pressure difference for various dry-running materials ($p_s = p_d$, c_m = 3.19 m/s).

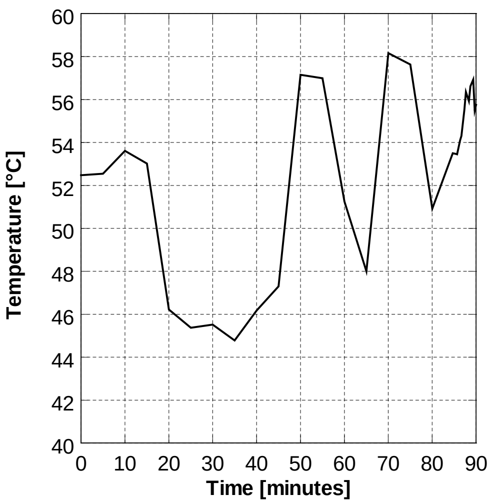

Fig. 11: Varying piston rod temperature caused by an unsteady gas leakage.

So far all tests have been carried out with only one packing sealing element loaded with a static pressure difference. Fig. 12 shows the results gained by tests with two sealing elements made of polymer blend, which were loaded either without any pressure difference, or with only a static pressure, or with a combination of a static and a dynamic pressure component. It is interesting to see that if the sealing elements are not loaded with a pressure difference the force of the garter springs of about 5 N still causes a relatively high rod temperature of 53 °C which is not significantly below the temperature of 68 °C resulting from a static pressure load of 40 bar. On the other hand, the additional heat resulting from the gas compression led to a considerable rise of the piston rod temperature of up to 91 °C.

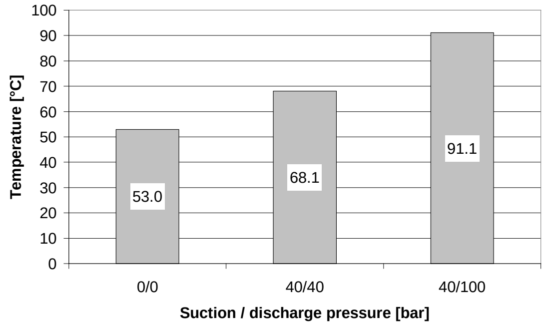

Fig. 12: Piston rod temperatures caused by various compositions of the pressure difference.

4.2 Influence of packing cooling

For a cooling efficiency that would be comparable with the cylinder, the cooling channels of the packing should be placed directly into the piston rod to enable the same short and efficient transport of the heat out of the friction contact into the cooling liquid. In practice, however, they are typically located in the packing ring chambers with the consequence of a considerably deteriorated heat transport mainly due to the plastic material of the sealing elements, which can be regarded as an insulator. Nevertheless, depending on the packing design and load, it is absolutely necessary to have such an indirect cooling - in spite of its limited efficiency. Figure 13 shows the temperatures measured by contactless thermocouples within the sealing ring chambers of a packing consisting of six sealing elements made of polymer blend. After the packing cooling was stopped, temperatures quickly rose up to a maximum value of 160 °C although the load (p_s = 6.4 bar, p_d = 16 bar, c_m = 2.56 m/s) was relatively low.

So far all tests with dry-running packings have been carried out with a cooling water flow rate of about 150 ltr/h. Parameter studies should show whether a variation in the cooling water flow rate also causes a corresponding change of the piston rod temperature. The tests were carried out with a dry-running hydrogen compressor, the second stage packing of which consisted of six sealing elements made of polymer blend and was loaded by a suction pressure of 16 bar, a discharge pressure of 40 bar and an average piston velocity of 3.19 m/s. The piston rod temperatures, measured by the infrared sensor, are represented in Fig. 14. Each measuring point represents the average temperature of an individual test with a duration of 24 hours. Due to the fact that the temperatures gained by this procedure vary within a range of approximately 10 °C, a clear tendency cannot be identified easily. Only for the cooling water outlet temperature a reduction was observed when the cooling water flow rate was increased from 50 to 350 ltr/h. Within the same range the rod temperatures showed no influence of the flow rate. Only below a value of 50 ltr/h a distinct rise of the rod temperature was observed, which obviously was caused by an occasionally interrupted water flow through the cooling channels.

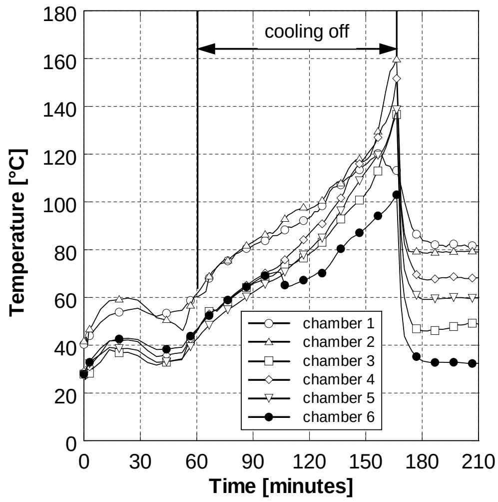

Fig. 13: Temperatures without packing cooling.

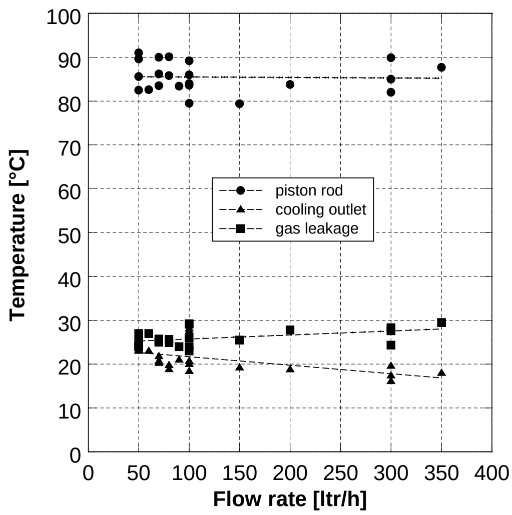

Fig. 14: Influence of the cooling water flow rate on packing temperatures.

As a consequence, the cooling water flow rate of a dry-running packing should not fall below a critical value, depending on packing design and load. However, an additional increase of the flow rate beyond the critical value does not lead to a further reduction of the piston rod temperature and is therefore useless. In particular, a high flow rate does not help if the load exceeds the maximal admissible value of a dry-running packing as can be seen in Fig. 15. Immediately after the start of this test the infrared sensor measured a steep rise of the piston rod temperature up to an unusually high level of about 160 °C. When the temperature finally exceeded a value of 180 °C the test was stopped – just in time to prevent the complete failure of the packing. It is interesting to see that the temperatures of the cooling water at outlet and the gas leakage were in the range of normal operation and did not indicate the forthcoming failure of the packing. Thus an infrared sensor – in spite of the fact that it is not capable to measure piston rod temperatures very accurately due to the varying emission coefficient - represents a valuable tool to protect from a complete breakdown of the packing and the consequences resulting therefrom.

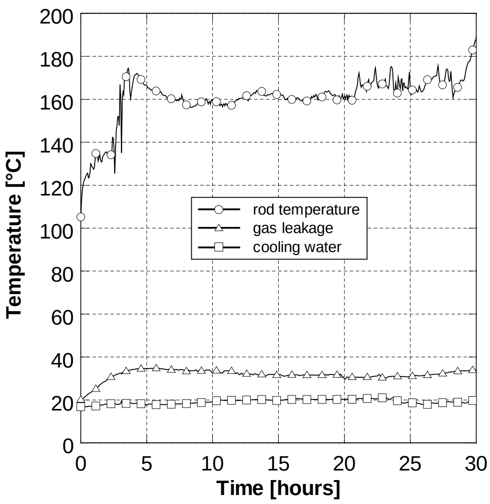

Fig.15: Comparison of the temperatures of piston rod, gas leakage and packing cooling water at outlet.

5 Conclusions

In order to find a suitable tool for monitoring piston rod temperatures during the operation of crosshead compressors out of various measuring methods only those were selected which fulfil the essential demands concerning robustness and reliability to survive the ever increasing service intervals. Besides a contact method by a kind of sliding shoe, contactless methods realized on the one hand as a thermocouple with a minimum gap between sensor tip and rod, and on the other hand as a temperature measurement by an infrared sensor were tested.

Based on the test results only the infrared sensor offers a fast reaction to temperature variations, which represents an important characteristic for the usage as a monitoring tool. In addition, it can easily be installed into the distance piece of a compressor. On the other hand, its accuracy depends highly on the emission coefficient, whose precise determination is fairly difficult and, in addition, which does not remain constant during operation due to a varying transfer film.

Tests with the infrared sensor carried out to prove the reliability and clarity of the information contained in the measured values showed that the gas leakage besides the load and the dry-running material exerts a considerable influence on the piston rod temperature. Since the leakage of dry-running friction seals is never really constant and may even change its magnitude to a significantly different level within a short time, variations of the rod temperature of more than 10 °C were observed during the tests. If loaded with only a static pressure difference of dry nitrogen ($p_s = p_d$), lowest piston rod temperatures were measured for a modified PEEK material while a polymer blend and particularly a carbon/graphite-filled PTFE caused significantly higher values. Tests with packing rings not loaded with a pressure difference showed that even a low garter spring force of about 5 N may cause a relatively high counter surface temperature.

Parameter variations with the cooling water flow rate showed that depending on the packing design and load the flow rate should not fall below a critical value to avoid a quick and steep rise of the temperatures within the sealing system. However, an additional increase of the flow rate beyond the critical value does not lead to a further reduction of the piston rod temperature and is therefore useless. Tests also showed that if the load exceeds the maximum admissible value of a dry-running packing the infrared sensor is capable of detecting the temperature rise resulting therefrom quickly enough to enable a safe shut-down of the compressor. At the same time the temperatures of the cooling water at outlet and the gas leakage were still in the range of normal operating condition and did not indicate the forthcoming failure of the packing. Measurement of the piston rod temperature by an infrared sensor thus represents a valuable tool for condition monitoring of dry-running packings.

Notation

p_s	suction pressure
p_d	discharge pressure
c_m	average piston velocity
PTFE	**p**oly **t**etra **f**luoro **e**thylene
PEEK	**p**oly **e**ther **e**ther **k**etone
PPS	**p**oly **p**henylene **s**ulphide

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Energy Management Services

ENALYSIS - Software for Accurate Real Time Compressor Analysis

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability May 17th-18th, 2001, The Hague

Abstract:

Diagnostic software that continuously monitors and immediately analyzes reciprocating compressor performance has been developed and tested in a variety of field sites. Operating results have shown improved reliability and performance, such as higher flow rates over time and lower maintenance and life cycle costs. The software applies modified algorithms to actual current operating data and accurately predicts compressor performance. The software instantaneously displays horsepower and cylinder utilization, abnormal temperatures, engine overloads, high turbo boost, minimum degrees of rod reversal, high rod loads, volumetric efficiency, minimum net ratios, as well as high valve and piston ring blow-by. The operator is immediately apprised of actions to take in order to enhance gas flow, reduce maintenance costs and prevent failure. This paper discusses the software performance, reviews the results from various applications, relates the results for operators, mechanics, engineers and financial managers, and discusses system improvements.

1 Introduction

The natural gas reservoirs in North America, specifically Western Canada, are slowly declining in pressure. As a result, the multistage Reciprocating Compressor is becoming an important asset to the gas industry.

The gas industry has accepted levels of load efficiency ranging from 75% to 90% as the norm. However, with the development of ENALYSIS software, the reciprocating compressors utilizing the program have been taken to a new load level. This monitoring tool has the ability to take the entire industry up 5% to 10% in production.

The sizes of compressor fleets are on the rise in North America and around the world. Consequently, there is a drive to monitor and optimize these assets to their greatest potential.

As energy prices and maintenance costs continue to climb, compressor performance data, as well as condition monitoring and energy efficiency factors, can provide valuable insights into the operation of individual machines and the overall compressor fleet.

Our company's experience has shown that regular monitoring of a compressor's performance can identify and warn operators of significant situations that would otherwise not be detected. Frequently, compressor deterioration remains hidden until a costly secondary failure occurs.

Many of the commercial monitoring systems available today, while sophisticated, are costly to install and maintain or require expensive manpower to collect and process performance data from pressure indicator cards.

The purpose of this presentation paper is to relate Energy Management Services' successful experience in the development of the ENALYSIS software for the specific purpose of improving the function, economics and reliability of reciprocating compressor fleets.

2 Software Development History

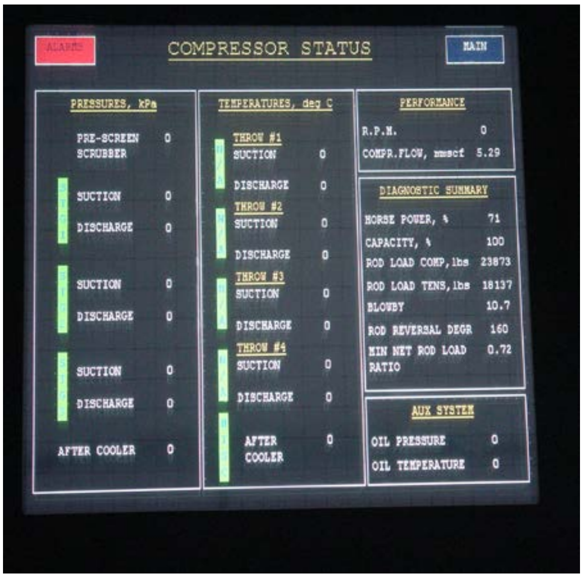
2.1 The Need

There has always been a need to analyze and optimize compressors. This need has intensified as capital and operating costs have continued to rise. Smaller, high-speed separable machines, have replaced the large, slow-speed integrals. New materials have been developed to contend with the higher speeds and higher temperatures associated with the new machinery. With the advent of the PLC, PC and most recently with the integration of the Internet, a vast amount of data is being collected and stored in database formats everywhere. Now, the question is, "How can we convert this abundance of data into useful information?"

ENALYSIS was developed in response to this question.

The standard adiabatic equations and algorithms that are published in the engineering textbooks on compression, still form the basis for design and analysis. However, these equations, alone, do not match actual field performance and often leave the engineer or technician rationalizing the raw data that has been collected. This is a time consuming and costly process. This type of inconsistent analysis and the numerous analysing inefficiencies led to the development of ENALYSIS.

ENALYSIS is the result of 25 years of development and refinement. Since 1993, the software has been used in various commercial applications including direct input, report output and in real time PLC panel applications.



Picture 2.1: *PLC panel **Sentry Pro*** by Prologic Controls of Calgary, Alberta, Canada, using the ENALYSIS Diagnostic Software.*

ENALYSIS eliminates the time required to rationalize raw data and conditions reported from the field, and removes uncertainty in the analysis. The ENALYSIS software has allowed the fleet management team to analyze vast amounts of data efficiently and accurately.

2.2 The Objectives

Process upsets or changes can be detected by measuring the inlet and discharge, pressures and temperatures, of each cylinder or stage. However, interpreting these parameters can become difficult when a multistage machine is operating under varying loads.

ENALYSIS was developed with the intention of meeting a number of objectives based on clients' needs. The program is now:

(a) Providing immediate focus on those machines that need attention through the use of a ***"cash flow at risk"*** ranking system.

(b) Warning the operator of impending mechanical failure.

(c) Warning the operator of operating conditions that could lead to failure.

(d) Alerting the operator of improper machine set up which could lead to failure.

(e) Alerting the operator of proper machine set up to optimize throughput.

(f) Providing data for energy consumption conservation.

(g) Providing historical databases containing mechanical and process data for easy interactive reference.

2.3 The Algorithm Development

The software is unique in that it uses modified, proprietary algorithms to accurately predict horsepower, temperature, volumetrics, compression and mechanical efficiencies.

2.3.1 Volumetric Calculation

The volumetric calculation of a new cylinder with known clearance is based upon the typical volumetric efficiency formula as shown:

$$VE = 100 - R - C\left[\left(R^{(1/K)}\right) - 1\right] \qquad ..1$$

Our experience has shown that this formula must be modified to account for normal valve and piston ring losses, molecular weight, compression ratio and compressibility. The modified algorithm results in good correlation between actual metered flow and the calculated flow of a new, well-defined machine. This calculation is important as it provides the basis for the wear factors and production loss factors.

2.3.2 Accurate Temperature Prediction

The cylinder's expected discharge temperatures are based on the adiabatic compression temperature formula as shown:

$$T_D = T_S\left[R^{(K-1)/K}\right] \qquad ...1$$

Experience has shown that this formula must be modified to account for cylinder size (on water jacketed cylinders), valve and passage loss factors, molecular weight, valve velocity and piston to valve area ratio, to match new machine temperature performance.

The deviations between measured and calculated cylinder discharge temperatures are the basis for the valve leakage and piston ring blow-by percentage factor, which represents the condition of the valves and/or piston rings.

3 Benefits of ENALYSIS

With ENALYSIS' **"Cash Flow at Risk"** ranking system, focus is always placed on those machines which are contributing the most cash flow to the company.

3.1 Increases Cash Flow

The major benefit of the compressor monitoring software is that it identifies those machines that should be reconfigured to increase throughput and, in turn, it maximizes cash flow.

Figure 3.1, *page 4*, shows the significant cash flow increments per year for a 1% improvement to a fleet of varying capacities; assuming $210/$e^3m^3$. For example, a 1% improvement on a fleet producing 2000 e^3m^3 /day, results in an annual increase in cash flow of over $1.5 million dollars.

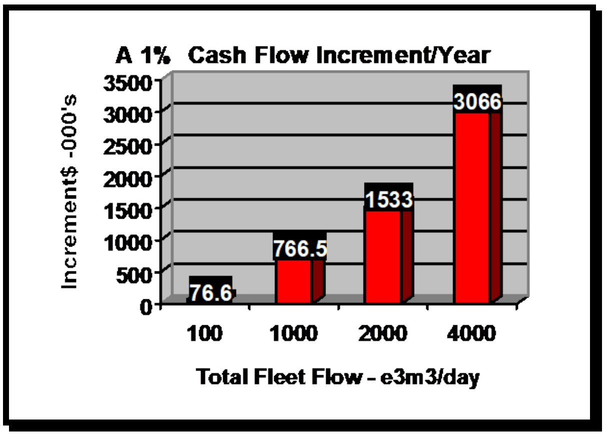


Figure 3.1: *Cash Flow Increment per Year*

What is a 1% increment?…………………..

(a) A compressor running at 10 to 12 RPM less than a maximum RPM of 1000 to 1200.

(b) Small amounts of added clearance, such as a variable pocket that is open a small fraction of a centimetre or leaking O-rings on the pocket piston.

(c) Forgotten high clearance assembly under valves.

(d) Blow-by due to leaking valves, rings, or packing.

(e) Bypass leaking or set point set too high.

(f) Modification to the piston that would create more end clearance.

It has been our experience that compressors on average are being operated at 75% to 90% of their maximum potential. When utilizing ENALYSIS, companies have easily obtained fleet averages from 92% to 97% based on a "once a month" review. Individual compressors within a fleet have achieved 100% utilization month after month.

3.2 Increases Asset Utilization

By monitoring each unit regularly within a fleet, the operator or manager is continually reminded of the under-utilized assets within that fleet. Action can then be taken at the appropriate time to place an old asset into a new project to maximize throughput and use of horsepower. The continual monitoring and reporting, increases equipment awareness throughout a company, so an asset is never overlooked. The current information and the historical data generated by ENALYSIS is easily integrated into the programs and plans of other departments, to facilitate communication and coordination.

3.3 Highlights Mechanical and Maintenance Issues

By highlighting mechanical and maintenance issues on a regular basis, as part of the monitoring program, any problem can be detected early, preventing serious damage.

3.4 Improves Data Rationalization

Because large amounts of data are collected by hand, PLC, PC and the Internet, it becomes imperative that a meaningful analysis report be generated to justify the data gathering in the first place. When using the standard OEM design programs, the user spends valuable time rationalizing this data on a unit-by-unit basis. The rationalization performed by ENALYSIS, allows the company operator to spend time effecting the recommendations that flow from the program rather than running a program that does half the job.

3.5 Provides Common Focus

The ENALYSIS monitoring program, performed at regular time intervals and distributed throughout a company, provides a common focus among all departments from first line operators, and mechanics, to engineers, managers, and financial managers. This ensures that no opportunities are overlooked.

4 The Process

4.1 Module Set Up

The monitoring process begins with a detailed physical definition of each machine, including any modifications that may have been made over the years. Information required here includes:

(a) Compressor name and unique identifiers.

(b) Location and elevation.

(c) A precise gas analysis, gas price and working interest.

(d) Motor or engine definition.

(e) Compressor frame definition.

(f) Cylinder definition including MAWP, diameter, rod diameter, number and type of valves, lift area, minimum and maximum clearances, connecting rod length and reciprocating weights.

(g) Aerial cooler definition.

4.2 Field Process Data Collection

To generate a meaningful diagnostic report, the collection of accurate field process data is essential. The operator should collect data only when the pressures, temperatures and RPM have stabilized. The most accurate method of hand gathering data is through the use of infrared temperature guns and digital dead weight pressure gauges. PLC data gathering is adequate using pressure transducers and thermocouples.

Field information can be sent by fax to EMS and inputted by hand or inputted by a client user via direct dial or Internet, or collected via PLC/PC and sent by Internet into a database. Reports can be generated on a timed interval or upon client request.

The minimum information that is required for the report to be generated includes:

(a) Identification for machine verification.

(b) Actual metered flow.

(c) Engine boost or vacuum, or volts and amps.

(d) Cylinder action whether it is DA, SAHE, SACE or NA.

(e) Variable clearance pocket settings and fixed clearance additions.

(f) Pressures and temperatures of each throw’s suction and discharge condition.

(g) The hours since last overhaul on each of the compressor and driver, scheduled downtime hours and unscheduled downtime hours.

4.3 Review the Report

The last step in the process is to receive and review the diagnostic report and take the appropriate action as recommended.

This simple “three step” process of basic information gathering and analysis, can easily be carried out by existing personnel. The rationalization and repetitious calculations are being performed by the software package, ENALYSIS.

It should be noted that any of the module set up data, field process data and the reports generated are available in a database which allows for easy access to specifics and trending on the ENALYSIS website.

5 The Report

The report form contains five major areas. They will be discussed in order of importance:

(a) Ranking

(b) Key Indicators

(c) Performance Flags

(d) Performance Calculations

(e) Identification

5.1 Ranking

ENALYSIS, in its calculations, ranks every compressor based on performance flags, key indicators, gas price and working interest. Attention is focussed on machines with the highest ranking, as this directly relates to **“Cash Flow at Risk”**. This ranking number is displayed on the report in large bold lettering directly below the machine identifier in large bold lettering, shown in Figure 5.1.

ENALYSIS calculates the **“Cash Flow at Risk”** in four areas:

(a) **“R1” – Damage:** is the cash flow at risk due to high rod loads, non-reversal and low net ratio. As a result, one week of potential down time production loss becomes the cash flow at risk.

(b) **"R2" – Production:** is the cash flow at risk due to the lost incremental production calculated in the key indicator section.

(c) **"R3" – Overhaul:** is the cash flow at risk if an operator pushes a machine past overhaul time intervals. These overhaul hour milestones are made adjustable based on horsepower utilization. The cash flow at risk is one week's production if a major breakdown occurs.

(d) **"R4"- Downtime:** is the cash flow lost due to non-scheduled downtime hours. This alerts the operator and managers to nuisance outages and production interruptions.

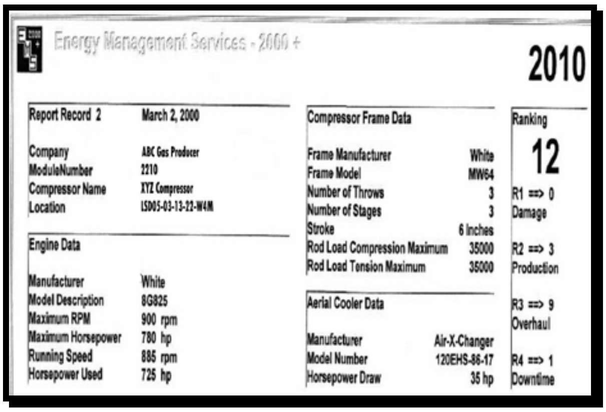


***Figure 5.1**: Ranking Section, upper right*

These four ranking numbers are totalled (rounding does occur) and makes up the **"Overall Ranking Number"**. This "**Overall Ranking Number"** when multiplied by $10,000, represents the total dollar value of the cash flow at risk. With this tool, the operator and/or manager can easily and quickly decide where to expend his resources.

5.2 Key Indicators

The Key Indicators allow the operator to instantly determine if he is utilizing the compressor asset to its fullest potential. The two key indicators are horsepower utilization and first stage cylinder utilization percentages. If both of these key indicators are less than 100%, an incremental production value can be calculated. If either the cylinder or horsepower utilization is at 100%, then an incremental production value cannot be calculated. (See Figure 5.3 below).

5.3 Performance Flags

The purpose of the performance flags section is to provide a quick, summarized warning display system of potential damage or actual damage that is occurring to the compressor.

Performance flagging is done at the conclusion of the performance calculation section and is shown in Figure 5.3.

For example, if the internal, external or net rod loads are above 95% of the manufacturer's rating, the word "YES" will appear below the throw involved.

Likewise, should a compressor throw have low minimum degrees of reversal of less than 70 degrees or have a low net ratio of less than 35%, again, a "YES" will appear in the appropriate boxes below the throw involved.

Cylinder blow-by (i.e. valve or piston ring gas slippage) is calculated and if it is greater than 7%, it is flagged in a similar fashion.

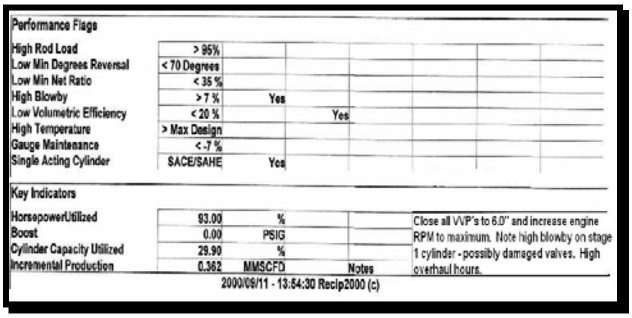


***Figure 5.3:** Performance Flags & Key Indicators*

Low volumetric efficiency often occurs when there are large clearance values in conjunction with high compression ratios. The resulting low flow reduces cooling, heat can build and damage may occur to valves, rings and packing. With ENALYSIS, volumetric efficiency is calculated and flagged if the volumetric efficiency drops below 20%.

The high temperature flag is a reminder of material constraints for vessels and piping, or the materials in valves, rings and packing.

If the field data is collected incorrectly, ENALYSIS will acknowledge the out of bound relationship as gauge maintenance.

The calculations done by ENALYSIS automatically identify specific remedial action, if required. The diagnostic results may pinpoint one or more of the thirty different notes of advice provided by the system. These notes are displayed in order of importance in the lower right hand corner of the report.

5.4 Performance Calculations

The mid section of the report, shown in Figure 5.4 includes the cylinder data from the module database, collected field data and the computer generated calculations.

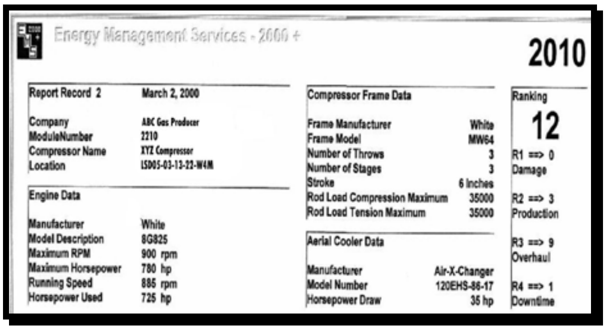
Performance			Flow Total 5.166	MMSCFD
Stage Number		1	2	3
Throw Number		4	3	1
Cylinder Model		135DD	140CD	37AD
Cylinder MAWP	PSIG	330	902	1375
Lube Status		Yes	Yes	Yes
Cylinder Action		SAHE	DA	DA
Cylinder Diameter	Inches	17.5	10.25	6.5
Rod Diameter	Inches	2.25	2.25	2.25
TemperatureSuction	oF	49.0	89.0	108.0
Temperature Discharge	oF	139.0	224.0	232.0
Pressure Suction	PSIG	136.0	209.0	490.0
Pressure Discharge	PSIG	212.0	500.0	1073.0
Compression Ratio		1.508	2.307	2.158
Pocket Adjustment	Inches	7.000	7.000	7.000
Head End Spacers Added	Qty	0	0	0
Crank End Spacers Added	Qty	0	0	0
Running Clearance Head End	%	115.329	81.399	65.839
Running Clearance Crank End	%	17.360	15.780	21.110
Volume Efficiency Head End	%	52.781	19.418	40.148
Volume Efficiency Crank End	%	88.698	79.394	76.389
Mechanical Efficiency	%	90.455	93.183	92.959
Compression Efficiency	%	72.318	85.548	85.881
Blowby	%	19.695	1.584	1.683
Internal Rod Load				
Rod Load Compression	Lbs	22531	28506	24132
Rod Load Tension	Lbs	2252	25525	17711
Total Internal Rod Load	Lbs	24783	54031	41843
External Rod Load				
Rod Load Compression	Lbs	18882	24907	21357
Rod Load Tension	Lbs	-595	21979	15032
Net Rod Load				
Rod Load Compression	Lbs	18202	19525	20097
Rod Load Tension	Lbs	7943	12724	9164
Min Degrees Reversal	Degree	104	145	165
Min Rod Load Net Ratio	%	44	65	46
Horsepower / Cylinder	Hp/Cyl	167.8	269.9	252.7
Cylinder Flow	MMSCFD	5.982	5.116	5.242

Figure 5.4: *Performance Calculations*

5.5 Identification

The top of the report, shown in Figure 5.5 displays a number of unique identifiers including company name, the unique identifier of the machine, a database control number of the machine, the report record number, date of the field data, date of the report generation, location, engine data, compressor frame data and aerial cooler data.

The unique identifier of the machine is presented in the top right hand corner of the report in large bold lettering for easy sorting and identification.


Energy Management Services - 2000 +

2010

Report Record 2 March 2, 2000

Company ABC Gas Producer
ModuleNumber 2210
Compressor Name XYZ Compressor
Location LSD05-03-13-22-W4M

Engine Data

Manufacturer White
Model Description 8G825
Maximum RPM 900 rpm
Maximum Horsepower 780 hp
Running Speed 885 rpm
Horsepower Used 725 hp

Compressor Frame Data

Frame Manufacturer White
Frame Model MW64
Number of Throws 3
Number of Stages 3
Stroke 6 Inches
Rod Load Compression Maximum 35000
Rod Load Tension Maximum 35000

Aerial Cooler Data

Manufacturer Air-X-Changer
Model Number 120EHS-86-17
Horsepower Draw 35 hp

Ranking
12
R1 ==> 0
Damage
R2 ==> 3
Production
R3 ==> 9
Overhaul
R4 ==> 1
Downtime

Figure 5.5: *Top of Report*

6 Actual Monitoring Examples and the Benefits

Energy Management Services has applied the program to improve the efficiency and increase cash flow for a number of gas producers in Western Canada. Below are examples of three companies and the benefit ENALYSIS provided.

6.1 Company A

A medium sized company in Western Canada had twelve Waukesha L7042GL engines coupled to four throw three stage Ariel JGK4 compressors that were experiencing engine difficulties (i.e. admission valve and spark plug life). As a result, they unloaded all the machines in an effort to prolong turnaround intervals. EMS was contacted to apply ENALYSIS in order to enhance compressor performance, minimize downtime and reduce premature failures. These twelve units were set up on ENALYSIS.

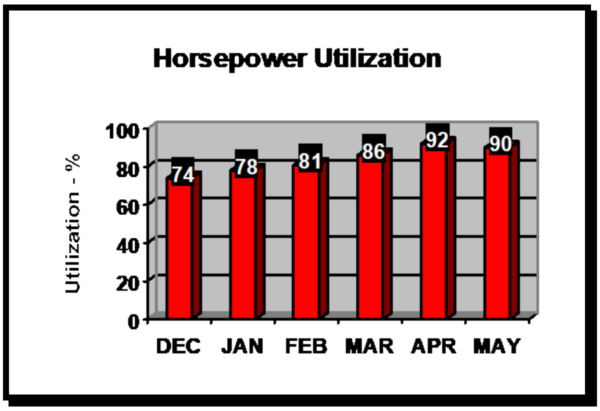


Figure 6.1: *Horsepower Utilization*

The first month's runs established the baseline performance (throughput and horsepower) of each machine. As shown in Figure 6.1, the average horsepower was initially 74% for the twelve machines. The company's target was to have all machines running at 90% of book horsepower and to see if better run times could be achieved. After five months the goal was attained and the client was experiencing excellent run times with the units.

Figure 6.2 shows the capacity improvement with the corresponding load adjustments to the engines. The twelve units began with an average of 172 e^3m^3 / day each. Five months later they were averaging 206 e^3m^3 /day each. At today's price of \$210/ e^3m^3, this revenue increase equates to \$2,570,000 per month for the twelve units.

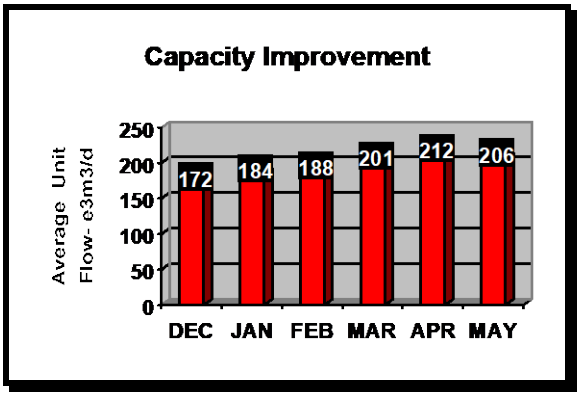


Figure 6.2: Capacity Improvement

6.2 Company B

The next example is of a small gas company that had forty machines with a throughput of 3250 e^3m^3/day. EMS' task was to improve the overall fleet performance.

The most effective method of tracking performance was to look at the utilization factors on individual machines. By totalling the potential incremental flows and dividing that sum by the actual throughput, an overall weighted average utilization factor of the fleet was determined.

The first month review had an 84% utilization for the entire fleet. From the monthly ENALYSIS reports, the company's operators and supervisors increased their utilization to 92% (see Figure 6.3). This resulted in a monthly cash flow increase of $1,758,000 at $210/ e^3m^3.

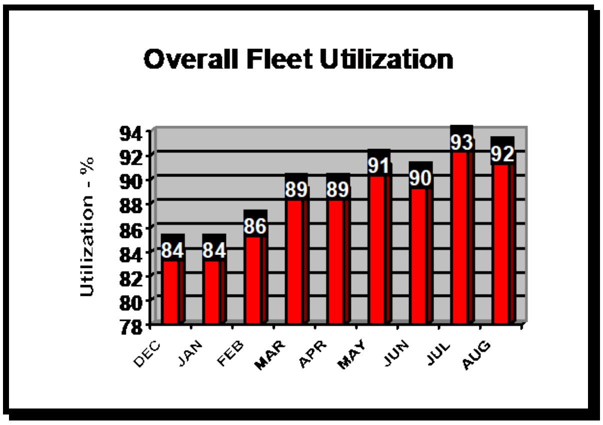


Figure 6.3: Overall Fleet Utilization -%

6.3 Company C

Company C, a large oil and gas company in Western Canada, wished to try this program on two specific gas fields, involving thirty-three machines. This company was already tracking each machine's throughput performance on a monthly basis. Their intention was to see if they could benefit from using the ENALYSIS software.

Figure 6.4 shows the improvement gained on a Waukesha H24GL coupled to an Ariel JGJ2 two throw single stage. Closing pockets, increasing RPM and adjusting the bypass set point, resulted in a cash flow increase of $85,000 per month.

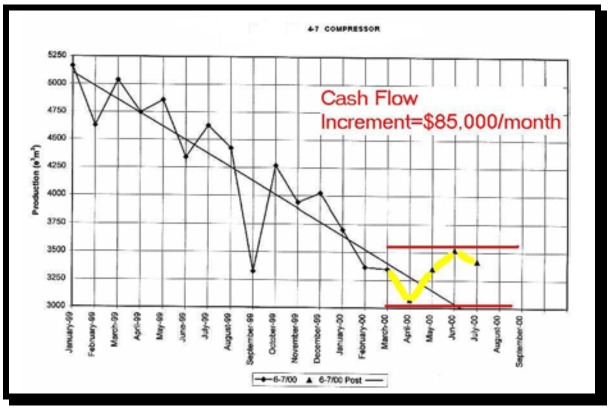


Figure 6.4: Waukesha H24GL/Ariel JGJ2

Figure 6.5 shows the improvement gained on an electric Westinghouse motor (800 hp) coupled to an Ariel JGE2 two throw single stage. Improvement was seen after convincing company operators to close the bypass. They thought they were close to rod load, which was incorrect. This increase in throughput was achieved with the same horsepower and resulted in a cash flow increase of $360,000 per month.

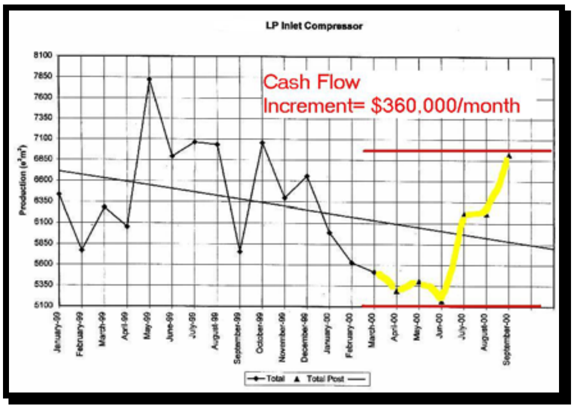


Figure 6.5: Westinghouse (800 hp)/Ariel JGE2

The maintenance supervisors of this large company were impressed with the software's ability to detect valve and ring wear before company personnel was able to. The savings alone in finding a deteriorating valve and repairing it before it became a more serious problem, justified the program, without consideration of the dramatic improvement in production and cash flow. By picking up valve problems early, the maintenance personnel could simply repair the valve and not wait for major failure. In the end, the overall life of the valve seat was prolonged. Therefore, major lapping to take imperfections out of the seats, did not have to be done as often.

7 Conclusion

We have proven, through extensive client results over a number of years, that the overall function, economics and reliability of Reciprocating Compressors can be dramatically enhanced. This breakthrough in production capability and cost management has created a new perspective from which gas producers can view the role of reciprocating compressors as productive assets.

ENALYSIS is truly a window into the heart of the Reciprocating Compressor. Regular monitoring of the fleet with ENALYSIS, will result in;

- Confident management of the entire fleet, as well as certainty in the operation of each compressor
- Increased throughput
- Identification of unseen production potential within the fleet
- Enhanced and effective preventative maintenance programs
- Reduced overall maintenance costs
- Avoidance of catastrophic failure and downtime
- Reduced operating costs, in both manpower and fuel

After detailed testing and implementation in Western Canada, this proprietary tool is now available to gas producers throughout the world. It is easily set up with minimal cost and impact on equipment and is equally effective on fleets of ten machines as it is on fleets of a thousand.

Detailed information is available from DE*TECH*TION Technologies at www.Detechtion.com.

8 Acknowledgements

I would like to thank my colleagues at EMS and DE*TECH*TION Technologies for their support, and the EFRC committee, for this opportunity to participate in the conference.

References

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Reciprocating Compressors Coupled to Gas Turbine Drivers in Gas Industry of the Russian Federation

by:

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th / 18th, 2001, The Hague

Abstract:

Unique economic conditions of the Russian gas industry: low cost of gas, country's geography and well developed turbine manufacturing capabilities (mostly the remnants of the old USSR Industrial-Military- Complex) make gas turbines a natural choice as prime movers for reciprocating compressors. In the West gas turbines rarely considered to be suitable for this application. However in Russian Federation the absence of nationally manufactured gas engines and the abundance of inexpensive Russian made turbines promise to become the decisive factor in selection of this type of driver for various compressor packages in a variety of applications throughout the country

1 Introduction

23% of the world gas output is produced by GAZPROM. Gas output of the Russian Federation in the year 2000 was about 570 billion cubic meters. GAZPROM pumped around 94% of that volume. To maintain and expand its infrastructure GAZPROM needs vast numbers of compressor units. GAZPROM general development plan for the next 5 years calls for modernisation of 114 compressor stations with total power about 15 million kW. By the year 2015 there must be built 240 new compressor stations.

The transition of highly centralised USSR economy to the market economy and subsequent cost consideration in selection of compressor equipment made reciprocating compressors by far more attractive than in the past. Over the last seven years reciprocating gas compressor units underwent remarkable metamorphosis from the pariah of the Russian gas industry to a viable and economically justified choice. The advancement of reciprocating compressor technology in the USA and other Western countries became the catalyst for this transformation. The ascent of the new generation separable gas compressor units (GCU) and the decline of old integrals GCU proved to be a potent combination for a market-oriented search to meet the demands of Russian gas industry. The specificity of present economic conditions in Russia makes it attractive to use turbine driven separable gas compressor units. Below we shall analyse in detail the peculiarities and applications as well as future potential for this type of units in Russian Federation.

Chapter 1

GCU fleet of GAZPROM: trends and present conditions.

Gas industry of the former Soviet Union begins with the wide use of integral compressor units. In 40-s and 50-s integral reciprocating compressors were universally used in oil and gas industry world - wide. Imported from the USA GMV-10 by Cooper-Bessemer with 746 kW rated power (1000 hp) became the sole compressor unit on every compressor station of the first major Soviet Union gas pipe line: Saratov – Moscow (1946). Imported from the USA Clark and Ingersoll-Rand integral compressor units were delivered to the first Gas Treatment Plant (GTP) – Moscow GTP.

In 1948 1949 in the city of Gorky (now Nizhny Novgorod) at the Dvigatel Revolutsyii (now RUMO) plant began the production of the first Soviet made integral compressors, the so called GMK 8GK with rated power of 300 hp and GMK 10GK (the exact replica of Cooper-Bessemer GMV-10) with rated power of 1000 hp. GMK 10GK became the mass produced Soviet integral compressor. Every compressor station built in the Soviet Union from 1949 to 1955 was equipped with that unit.

By the middle of 1950 in Leningrad (now St. Peterburg) the compressor plant Nevsky Zavod Lenina (NZL) began production of gas turbines with centrifugal compressors rated at 4 MW. From that moment on all compressor stations working on large diameter gas pipelines were equipped only with this type of compressor units. The golden age of integrals had come to an end.

However Dvigatel Revolutsyii plant continued to produce integral reciprocating compressor units. In 1962 – 1964 the plant completed the development of the turbocharged version of the GMK 10GK model. It was renamed GMK 10GKN and was rated at 1500 hp (1100kW). This model without any major modifications was manufactured for almost 30 years. Machines made in Nizhny Novgorod were sold to underground storage facilities, small compressor stations in USSR and also in Poland, Romania, Hungary, Bulgaria and Afghanistan. In 1971 the plant began to produce GMK MK-8 rated at 2800 hp (2060 kW). The design of that model was based on Clark TLA-8 model. In 1973 – 1974 the same plant started production of GMK DR-12 rated at 7500 hp (5500 kW) under the license from Cooper-Bessemer.

We would like to mention that in 1978 – 1980 in the USSR there was designed and manufactured first separable compressor unit (GPA-5000) with an impressive rated power of 3600 kW. It consisted of pre-chamber combustion gas engine and directly coupled opposed 375 RPM reciprocating compressor. All in all 13 such units were manufactured and were installed on 3 compressor stations in Ukraine. Due to low reliability of these units the production was shortly stopped.

By the time of the Soviet Union collapse in 1991 the aggregate power of Gazprom compressor stations was equal to 46.6 million kW. Gas turbine driven centrifugal compressors made up 38.6 million kW, electric motor driven centrifugal compressors made up 7.0 million kW and reciprocating (mainly integral) compressor units made up 1.0 million kW. By the year 2000 the aggregate power of GAZPROM compressor stations was equal to 41.3 million kW. Gas turbine driven centrifugal compressors made up 35.6 million kW, electric motor driven centrifugal compressors made up 5.3 million kW and reciprocating (mainly integral) compressor units made up 0.4 million kW. The majority of integral compressor units are working in gas storage facilities. They have very low efficiency, very high oil consumption, very high pollution level and very low reliability.

Chapter 2

Reciprocating GCU with the Gas Turbine Drive – applications and design.

2.1 Applications of centrifugal and reciprocating compressor units

At present maximum rated power of a reciprocating compressor unit is about 8 MW. All data given below is related to units with no more than 7 MW – 8 MW rated power. While selecting compressor unit for the compressor station two types are usually considered – GCU with gas engine drive and a reciprocating compressor and GCU with gas turbine drive and a centrifugal compressor. It is a well-known fact that the advantages of the reciprocating compressor vis-à-vis centrifugal compressor are:

- ❖ Ability to work with high compression ratios in single stage compression (up to 3 and more): one stage compression of the reciprocating compressor can do the work of 9 stages in a centrifugal compressor.
- ❖ High efficiency level – adiabatic efficiency of a reciprocating unit is about 0.81 – 0.87 compared with 0.76 – 0.81 of a centrifugal unit
- ❖ Ability to achieve high efficiency levels in broad pressure ranges
- ❖ Simplicity of fine tuning of the capacity of the unit, including the absence of a complicated anti-surge systems
- ❖ Low sensitivity to temperature and gas density deviations at suction

The most suitable driver for the reciprocating unit is a reciprocating gas engine. Nowdays there are plenty of manufacturers of gas engines which makes it easy to select the engine with the RPM equal that of the reciprocating compressor. If the speed of engine changes the reciprocating unit maintains required torque, on the other hand gas engines are capable to keep continious torque within the broad range of speed. The most advanced gas engines working in a compressor shop have 0.39 – 0.43 efficiency while their centrifugal «brothers» do not exceed 0.27 – 0.32. As a result the fuel consumption of the reciprocating unit while compressing the same amount of gas is 40% to 90% less than in centrifugal unit. The main advantage of the centrifugal compressor is the ability to compress large volumes of gas with relatively small weight and size units. Since most modern gas turbines have high power output combined with low weight and small size, and as a rule have the same speed as a centrifugal compressor they are effectively used at major compressor stations servicing primarily gas pipelines of large diameter.

2.2 Reciprocating Compressor Units with Gas Turbine Drive

The design of the Reciprocating Compressor Units with Gas Turbine Drive has some peculiarities. Vast majority of the turbines in our category have nominal speed of no less than 4000 RPM. The most modern reciprocating compressors suitable for GCU have no more than 1800 RPM. That is why the combination of the Gas Turbine Drive and a reciprocating compressor must have gear reducer to bring the speed of two shafts in a conformity. It also must have an additinal coupling to connect turbine shaft with the gear reducer. The extension of the shaft line leads to increased torsional vibration forces which requires special attention during design phase. Coupling between reciprocating compressor and gear reducer must prevent gear reducer from torques overloads – it must serve as torque limitor and must have special torque limitation device. The necessity to have gear reducer and a coupling makes the design of the GCU somewhat more complicated. It also increases the length of the unit.

Reciprocating Compressor with Gas Turbine Drive allows to combine in a single unit the advantages of a reciprocating compressor and a turbine. The unit is best suited for applications where the shortfalls of the reciprocating compressor and a turbine are not important. It is best suited in cases listed below:

- ❖ Low cost of fuel gas
- ❖ Size considerations, minimum dimensions and foundation limitations
- ❖ Quality of fuel gas
- ❖ Availability of the driver, its cost, delivery time etc.

Let us look at each case in details.

Low cost of fuel gas can exist under two conditions – if the price of gas is low and if small amounts of gas are used. The price of gas in Russian

Federation in 2001 is about $10 - $15 US Dollars per 1000 cubic meters, which is approximately 10 – 15 times lower than in Western Europe and the USA. GAZPROM is determined to increase gas prices 2,5 – 3 times compared with present prices. But even if the price of gas in upcoming years will be increased several folds it still will remain below the world prices. The main avenue to save fuel gas is to increase efficiency of the driver. The numbers for turbine and gas engine efficiency had been given above. Another way to use less fuel gas is when GCU not working continiously during a given year. It is typical for GCU working in undeground storage facilities where the units are engaged only during the summer injection season (3000 to 4000 hours per year), for mobile compressor units to evacuate gas lines before the maintenance work, to pressurise pipe line sections (no more than 2,500 hours in a year) and etc.

It is always more advantageous (under the same given conditions) to use GCU with less weight and dimensions and lower foundation loads.. These conditions are especially important for mobile compressor units, for units working on offshore platforms, for units destined to work on compressor stations built in hard to reach and swampy areas, for compressor stations in remote Northern regions. The nominal weight diagram is given below:

Reciprocating and Turbine Driver Nominal Weight Chart, kg/kW

Power range, MW	Gas Engine	Gas turbine
1.5 – 2.5	14 – 10	6 – 5
3.5 – 4.5	9.5 – 8.4	4 – 3
5.5 – 7.2	9.5 – 7.8	3 – 2

In some cases the quality of fuel gas can determine the selection of the driver. In particular presence of agressive gases and gas low Btu. All modern gas engines can tolerate no more than 0.1% (vol.) of hydrogen sulfide. Gas turbines on the other hand can use gas with 1% to 2% (vol.) of H_2S.

In the USA and Western Europe up to 7 MW gas engines and gas turbines are mass produced. In Russian Federation several plants manufacture gas turbines of the same power rating but gas engines rated over 1 MW are not produced at all.

Below given some examples of the Turbine Recip Compressor Packages (TRCP).

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At the Lisbon Recycle Plant near Moab, Utah (property of Union Oil Co. of California) recycled gas had to be injected to the reservoir. Suction pressures varied from 85 Bar G to 99 Bar G and discharge pressures varied from 191 Bar G to 219 Bar G. The only fuel readily available was untreated casing head gas containing 40% inerts (CO_2 and N_2) and well over one percent of hydrogen sulfide. The high sulfur content and low Btu (20.6 to 22.3 MJ/m^3) was not acceptable fuel gas for gas engines or integrals. That is why Solar Saturn T-1000 two shaft mechanical drive package rated 810 kW at 22300 RPM was selected to drive Joy Manufacturing Co. Model WBH 74H reciprocating compressors. Maximum speed of the compressor shaft was 700 RPM. Turbine and compressor shafts were connected via double stage gear reducer. A lot of attention during design phase was given to the effects of torsional vibration. Compressor coupling and flywheel selection were based on satisfactory torsional analysis. The goal for the design of the units was a speed range of 80% to 100%. Operation of two machines resulted in an operational availability of better than 98%. Weight of the unit with structural steel skids was 27.2 t., length – 6.0 meters, height - 1.8 meters and width – 3 meters.

****

Seven TRCP's are used in gas lift service on a platform offshore Louisiana coast. They consist of 4 cylinder Ingersoll-Rand 4RDS-3 reciprocating compressors connected via gear reducer and flexible coupling with Solar Saturn Mark II, M.D.-1200 mechanical drive turbine package. Three stage compressor is designed to compress gas from 5 – 8 Bar to 75 – 92 Bar. Two shaft mechanical drive turbine package with double stage gear reducer rated 773 kW at 948 RPM. The coupling connecting reciprocating compressor with turbine output shaft also protects turbine from torsional overload by means of pre-loaded brake shoe arrangement to allow slippage between the drive and driven half of the coupling.

****

While expanding Ruhrgas cavern storage facility at Epe near Gronau, between Germany and Netherlands, new gas compressor units were needed to compress gas from 40 – 70 Bar to 70 – 198.5 Bar (pressure ratios between 1.6 and 5.0). The comprehensive feasibility study was made to compare integrals, 2 casing centrifugal (low pressure section consisted of 9 stages and high pressure section consisted of 5 stages) and TRCP compressor units. The same gas turbine with 4250 kW power output was considered as a possible driver for centrifugal compressor and TRCP. A detailed study indicated that a single stage reciprocating compressor coupled to a gas turbine by a two-stage gear reducer (TRCP) would be the most advantageous solution both from technical and economical point of view. The gear reducer stepped down gas turbine shaft speed from 10290 RPM to 333 RPM. The high speed side was equipped with an oil lubricated gear type coupling. The low side was equipped with a flexible coupling with tangential spring in order to provide a high degree of flexibility and resistance to torsion. A mechanical torque limiter was installed between the low speed coupling and the gear reducer to protect gear reducer against high torque. Two such units were commissioned in 1988.

****

The most popular application of TRCP is on a mobile compressor units (MCU) which are used to evacuate gas from the pipeline prior to maintenance work. During the last 5 years Ariel authorised distributors/packagers and first of all Enerflex Manufacturing of Enerflex Systems Ltd. assembled 15 MCU's with Ariel compressors. Single trailer mounted Ariel JGJ/6 compressor driven by 1044 kW Solar Saturn T1400 M.D turbine unit designed in co-operation with TransCanada Pipelines Co became the most successful MCU manufactured by Enerflex. This TRCP is used for gas pipeline evacuation (from 64 Bar G to 7 Bar G) and for pipeline pulldown to a parallel 64 Bar G line. To provide efficient work of the MCU in wide pressure range the unit features totally automatic stepped load control utilizing six head end unloaders on compressor cylinders and single/two stage configuration valving. Torsionally soft low speed coupling is used between reciprocating compressor and gear reducer. On the same 16.2 meter long trailer also mounted auxilliary 45 kW generator set, air conditioned control room with controls by Solar Turbotronics, gas coolers, and etc. According to TransCanada Pipeline estimation this MCU allows to save 49 million cubic meters of gas per year which is a significant economical and ecological consideration

Chapter 3

Manufacturing of Gas Turbines and Reciprocating Compressors in Russian Federation.

In Russian Federation there are many plants capable to manufacture gas turbines, especially aviation type turbines. Some Russian plants make gas turbine drive packages for the gas industry. These gas turbine drive packages can be used in TRCP. The most promising are packages made by OAO Permskyie Motory. Based on a Д-30ЭУ-1 gas turbine OAO Permskyie Motory offers 2.5 MW (ГТУ-2,5П) model, 4.0 MW (ГТУ-4П) model and 6.0 MW (ГТУ-6П) model. These models are equipped with the gear reducer to bring
down the shaft speed to «compressor friendly» 1000 RPM – 2000 RPM.

Models offered by OAO Permskyie Motory have relatively low fuel efficiency, however in Russian Federation and especially if not used continiously this shortfall is not very important. OAO Permskyie Motory and its daughter companies are successfuly cooperating with GAZPROM and produce for GAZPROM units with centrifugal compressors and also turbine driven generator sets.
Reciprocating compressors with rated power over 1 MW which could be used by GAZPROM are not produced in Russian Federation at this time. RUMO plant in Nizhny Novgorod is trying to initiate production of such compressors.

Major Performance Data for Permskyie Motory Gas Turbine Drive Packages

Performance Data	ГТУ-2,5П	ГТУ-4П	ГТУ-6П
Power output at gear reducer shaft, MW	2.5	4.12	6.1
Efficiency,%	21.4	24	27
Nominal speed, RPM	5500	5520	7000
Reducer speed, RPM	1800(1500)	1000	1000
Lubrication oil loss, kg/h (not more than)	0.4	0.5	0.55
Gearbox lubrication oil loss, kg/h	0.2	0.2	0.2
Dimensions(L-H-W), m	6.8–2.0–2.2	6.8–2.0–2.2	7.0–2.0–2.2
Emissions (maximum): NOx, mg/m^3 CO_2 , mg/m^3	 57 50	 50 50	 50 50
Major overhaul interval, hours	25 000	25 000	25 000
Availability (not less than), %	98	98	98
Operational use (not less than), %	95	95	95

Chapter 4

Particulars of TRCP development

Based on the above data we can clearly see that in Russian Federation it would be economically justifiable to manufacture TRCP with power ratings from 1 MW to 6 MW. This course of events in turn will open new niche for reciprocating compressor market. Ariel Corporation (USA) manufactures compressors with power ratings up to 7 MW. These compressors could be used in TRCP's. Ariel has an experience in this area. 21 such units had been manufactured over the past 10 years. At present Ariel in co-operation with GAZPROM works on practical development of TRCP in Russian Federation.

At this time based on GAZPROM's request OAO Permskyie Motory and Ariel have completed preliminary design and technical work for two types of TRCP – model rated 4 MW for use in underground storage and 2.5 MW model for use in mobile compressor unit.

NPO ISKRA had completed project design work for a salt dome storage which will use Ariel JGC/6 compressor driven by ГТУ-4П. Storage conditions call for gas compression from 46 Bar A to 53 – 215 Bar A (pressure ratios between 1.15 and 4.67). To achieve such wide pressure range NPO ISKRA used Ariel compressor with single/two stage configuration valving. Each unit will have capacity within 1.7 to 3.3 million nm^3/day.

Compressor unit consists of several modules (sections). Main compressor module houses reduction gear mounted integrally with the turbine and a compressor. This section has an enclosure with a separation wall, which divides it into turbine driver housing area and a compressor area. The exhaust stack is insulated for silencing and emission passage. Exhaust stack is mounted above the turbine-housing compartment of the enclosure. In addition, exhaust stack has air pollution and noise reduction device. Modular lubrication system for turbine driver, gear reducer and a compressor and modular fuel/start up gas filter systems are located on the side of the turbine housing area. Control module includes compressor unit control system, fire alarm system and motor control center. It can be located at any side of the main compressor module, which gives additional flexibility while planning the layout of the site.

Turbine, gear reducer and a compressor oil after-cooling module as well as cooling fans mounted on top of the roof above the turbine driver compartment of the main module. All modules are bolted to a cement foundation.

OAO AVIADVIGATEL with Ariel assistance had designed a mobile compressor unit (MCU) which will be used for the evacuation of natural gas from pipelines during maintenance procedures by GAZPROM.

Permskyie Motory ГТУ-2,5П powered mobile compressor unit uses Ariel JGE/6 frame. In accordance with GAZPROM technical requirements this MCU is designed to guarantee gas transfer from the 1420 mm diameter 30 km long gas pipeline sections, lowering pressure from 75 BarA to no more than 10 - 13 BarA. Discharge pressure must be no less than 78 BarA and pulldown time no more than 62 hours.

Ariel makes several compressor models capable to meet these requirements. Ariel JGE/6 compressor was selected because compressor size (especially width) was an important consideration. Just like in underground storage applications this TRCP must handle wide pressure range and that is why again single/two stage configuration valving was used.

MCU consists of two modular sections mounted on two trailers. Installed on the first trailer is a six throw, reciprocating, Ariel model JGE compressor, driven by a ГТУ-2,5П mechanical drive package.

Inlet and interstage gas scrubbers, control system and oil cooler are also located on the first trailer. The trailer is totally enclosed. As per Russian Federation Safety requirements (ПУЭ), the turbine is separated inside the trailer from the compressor by a vapor tight partition.

The second trailer contains gas radiators, generator set, control room and motor control center. Storage space is provided under the gas radiator for storage during transport of the flexible piping and the electrical and control cable inter-connects. The MCU is suitable for highways and pipeline right of way travel. First prototype of this MCU scheduled to be completed in 2001.

In our opinion TRCP can be also succesfuly used on booster compressor stations to maintain constant pressure flow in major pipelines inlet when the gas field pressure drop is experienced.

Variable and high pressure ratio requirements do not favour centrifugal compression. Relatively cheap wellhead gas and mostly remote and hard to get locations better suited for gas turbine engines. Thus TRCP can be a viable alternative. Of course each case must be considered individually.

Mobile High Pressure Nitrogen Generation Systems / Compound Design

by:

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability

May 17th / 18th, 2001, The Hague

Abstract:

Due to its properties being similar to those of an inert gas, nitrogen is used for a variety of purposes; in the production and storage of oil and natural gas, as well as in the chemical and petrochemical industries. Separating nitrogen from the air by membrane technology not only provides gaseous nitrogen on site, but also offers considerable economic advantages as well, compared to the use of cryogenic nitrogen. The purpose of this article will be to introduce a compact, mobile, containerised unit that can generate nitrogen up to a purity of 99%, and then compress it to up to 350 bar (5.000 psi), at a delivery rate of up to 40 Nm³/min (1.450 scfm). Included in the container are not only the compressor units themselves, but also the conditioning and separating equipment, the control system and the panel visualising the process parameters.

Stickstoff wird wegen seiner Eigenschaften als Inertgas für eine Reihe von Arbeiten in der Gewinnung, Förderung, Lagerung und Verarbeitung von Produkten der Erdöl- und Erdgasindustrie eingesetzt. Neben der Beistellung von Stickstoff in flüssiger Form gewinnt die Vorort-Erzeugung mittels Trennung aus Luft, nicht zuletzt wegen ökonomischer Gesichtspunkte, immer mehr an Bedeutung.In diesem Beitrag wird eine kompakte, mobile Einheit zur Erzeugung von Stickstoff mit einer Reinheit von bis zu 99 % mit anschließender Verdichtung auf bis zu 350 bar und einer Kapazität von bis zu 40 Nm³/min vorgestellt. Neben den Aggregaten zur Verdichtung sind im Container auch die Komponenten zur Luftaufbereitung und Gastrennung, sowie die Prozeßsteuerung samt Visualisierung installiert.

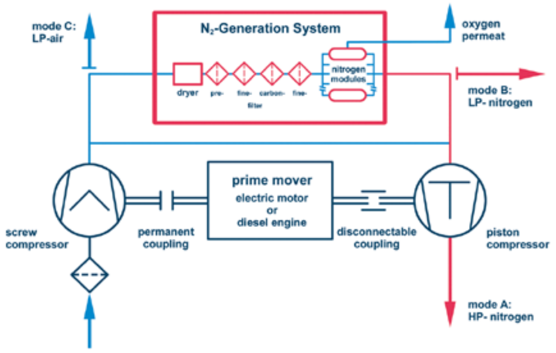
1 Introduction

The earth's atmosphere consists, to more than 75%, of nitrogen. This colourless, odourless and tasteless gas behaves similarly to the inert gases, helium, neon, etc., meaning that it normally does not react with other gases or with the material of piping, pressure vessels, etc. This convenient attribute is, in a wide range of cases, independent of the temperature and pressure of the nitrogen gas. Explosion protection is of course of the utmost importance during the production and storage of oil and natural gas, and in the chemical and petrochemical industries. It is therefore essential to use an inert gas in pipeline testing, well testing, loosening blocked drilling equipment, and for well stimulation and EOR – enhanced oil recovery. Because of its rust-preventing properties, nitrogen is also used for the long-term conservation of facilities and pipelines, as well as for maintenance and repair work on machines.
Nitrogen is therefore ideally suited to many industrial purposes and, thanks to membrane technology, it can be efficiently generated on site, by simply separating nitrogen and oxygen from the surrounding atmosphere. This procedure is normally carried out within a pressure range of between 5 and 15 bar (70 and 215 psi) and within a temperature range of between 10 and 50°C (50 - 120 F). The pre-compressed, pre-conditioned air is then split into nitrogen (at a purity of up to 99%) and oxygen-enriched air (so-called permeate). Depending on what it is to be used for, the nitrogen thus generated can be further compressed, if required, by a booster compressor.

The use of such a nitrogen generator on site avoids the numerous difficulties (and expenses) involved in transporting liquid nitrogen. Furthermore the generator will, after a short warm-up period, ensure a practically unlimited supply of nitrogen. Its output can be adjusted to provide any working pressure up to 350 bar (5.000 psi).

2 Description of the System

The Leobersdorfer Maschinenfabrik AG, called LMF, has developed a nitrogen generator for both stationary and mobile use on site, designed on the same well-proven principles as the widely used LMF compound air compressors.



Picture 1: LMF Nitrogen Generation System, Flow Diagram

The above flow diagram shows the pre-compression, by an oil-flooded screw compressor, of atmospheric air to 13 bar (185 psi). After the pre-compressed air has been cleaned of remnants of oil and condensate, it is filtered and dried and then routed to the membrane modules. In the membranes, the nitrogen is separated out, to a purity of max. 99%, from the oxygen-enriched air, which is also called permeate. The purity of the nitrogen produced can, depending on the application, be adjusted as required by an optimised control system.

The nitrogen emerging from the generator can be further compressed up to 350 bar (5.000 psi) by a booster compressor, designed with up to three stages. The maximum output capacity of such an LMF unit is 2.350 Nm³/min (1.450 scfm).

In addition to generating high pressure nitrogen, this same unit can alternatively produce low pressure air, low pressure nitrogen and high pressure air. Following the design of the well-proven LMF compound units, both compressors (pre-compressor and booster) of the LMF nitrogen generators are powered by a gearless prime mover (either electric motor or diesel engine) situated between them. In cases where the unit is used to produce either air or nitrogen at low pressure, the high pressure reciprocating compressor (booster) can simply be mechanically uncoupled.

The whole unit, together with its intercoolers and heat exchanger, is mounted on a self-supporting steel frame, designed to fit in the container. The scope of supply also includes insulated steel doors, pre-cooling systems for cooling water, heaters for both the oil and the container to cope with ambient temperatures from -40°C to +40°C (-40 to 105 F). The dimensions (length, width and height) of the container, which can be supplied as a separate entity or can be mounted on a truck (see Picture 2), are within the limits laid down by the ISO regulations.

Picture 2: Containerised Nitrogen Generation System (Truck Mounted)

All auxiliaries, including the hydraulic drive for the cooling fans, are installed inside the container. Where the prime mover is a diesel engine, diesel is the only source of energy required, (i.e. no electric current, compressed air etc., is needed).

3 Electric and Electronic Control Unit

The control cabinet, housing the control system and the power distribution equipment, is mounted inside one section of the containerised unit, surrounded and protected by the walls and the roof of the container. The cabinet itself is a steel sheet enclosure, water and dust-proof (IP54), mounted on shock absorbers and preheated/ventilated as required by the temperature inside the container.

Functions :

- Distribution of the main power and control voltage to all sub-units of the package. The main power is produced by two generators.

- Two rectifiers for the generator AC voltages (to supply the 24V DC bar), with a battery backup system are installed. This can also be used as a start-up battery for the water heaters and the auxiliary diesel engine.

- Control system for the screw compressor, the diesel engine, the piston compressor, the cooling systems (water and air circuits), the hydraulic system, the nitrogen membrane system and for all instruments in the container. The control system is equipped with a programmable logic control system, which is a modular design system with various plug-in cards for power supply, CPU, memory, process inputs and outputs and interface modules for connecting the display and the monitoring devices. The programmable logic control system is suitable for outdoor installation, certified for operation down to -25°C (-13 F), in part down to -40°C (-40 F).

The main hardware information, such as generator currents and voltages, common failure lamp, common alarm lamp and basic operational status lamps, are displayed on the front door of the control cabinet.

The operator display panel of truck mounted units is fitted in the driver's cab. All alarms and shutdowns of the nitrogen compressor unit are displayed on the text lines of the LCD panel. Manual operations, such as compressor starts/stops, adjustments of the condensate drain, manual starts/stops of pumps, reset function, lamp test, etc., can be performed using the function keys on the panel. There are also separate lamps for common alarm and common shut down.

The industrial personal computer is also mounted in the driver's cab, on the back wall behind the passenger seat (which can be turned around to face it). Proven, off-the-shelf hardware is used for the personal computer, together with a robust system, to operate in the demanding conditions down to -40°C (-40 F).

A process visualisation system is used for all the different screen diagrams of the nitrogen compressor unit. It is possible to select detailed screen pictures of all parts of the unit (i.e. the screw compressor, the membrane system, the piston compressor, the cooling system, the hydraulic system, the diesel engine, etc.). All switches and transmitters are displayed, as well as the status and the momentary readings. In addition, all alarms are stored in a separate list and are displayed with higher priority than the normal screen diagrams. A history file may be programmed internally, to monitor any value of the whole unit, with an adjustable sample rate. This file is stored on the hard disk drive and may be used for data logging, as described above.

The whole personal computer system can be operated by one person from the driver's cab. The language of all texts and diagrams can be selected. All the values of the entire package are calibrated in "SI" units. For service and display purposes a telecommunication modem with mobile telephone or Internet connection may be installed.

4 Advantages of the LMF Nitrogen Systems

Besides the ready availability of nitrogen produced on site, the LMF nitrogen generators offer the added advantages of extreme compactness (by using the “V” compressor design) together with very high output capacities for their footprint and weight. Despite their compactness however, the importance of easy access to the main components has been fully respected. The advantage of having the electronic control system and personal computer integrated in the driver’s cab on truck mounted units, provides not only more space in the container, but also the greatest possible safety and comfort for the operator. The design of these units is a highly successful further development on the basis of the well-proven LMF compound compressor units, hundreds of which are at present in use around the globe.

THOMASSEN
COMPRESSION
SYSTEMS

Performance Monitoring on Reciprocating Compressors, a Rational Extend on Condition Monitoring

by:
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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th / 18th, 2001, The Hague

Abstract:

Performance Monitoring can be defined as the online calculation of the compressor performance based on actual process data, and the comparison of the theoretical compressor performance with the actual process performance. The compressor is regarded as a “Black Box”. Condition Monitoring supplies information on the condition of the machine irrespective of the changes in the process.
In the light of the total maintenance and total care contracts the next step in machine condition monitoring should be Performance Monitoring.
Changing process parameters will change the flow, required power and temperatures of the compressor output. Performance Monitoring can prevent misinterpretation of data.
Expert system’s such as the B&K Advisor, will generate extensive data bases. These data bases will be used to give a recommendation on the decision for machine overhaul.
In the presentation an overview will be given of the options of the Performance monitoring system, the required input and the produced output.

1. Introduction

The compressor is often the most critical component in a process gas system. The condition, good operation, and response to changes in the process conditions are therefore of great importance. The condition of the equipment and therewith the need for an overhaul or repair is indicated by a Condition Monitoring System or the experience of an operator coupled to deviations from the normal readings of the instrumentation.
The condition is derived from vibration, temperature and pressure transmitters or gauges. In case of a Condition Monitoring System the readings are compared to pre-set values and an alarm and sometimes a shutdown can be activated.
The pre-set alarm and shutdown values may come from the user experience, and from the equipment manufacturer. They are based on normal operating conditions and the maximum permissible loading of the equipment. Vibration settings for new machines are often based upon operative values.
Once the values are set, they will usually not be changed, unless they lead to false alarms which is naturally an undesirable situation.

For compressors that are used in processes where the operating conditions can change, the pre-set alarm and shutdown values should be far from the normal operating conditions, to avoid the unnecessary disturbances of the operation. Examples of such processes are the natural gas gaslift, depletion and booster compressors, and Recycle compressors in Hydrogen applications.

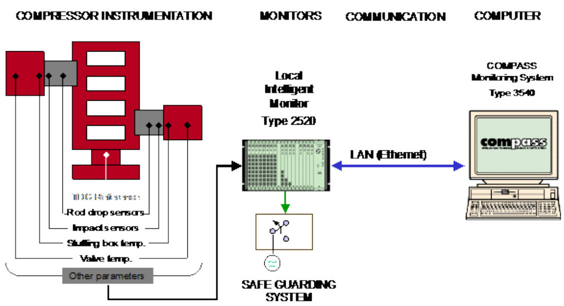
In those cases where process flexibility is required, the choice for a reciprocating compressor is the preferred option.

Calculating the compressor performance and output conditions can then be the tool for making the warnings “floating”. This means that the settings can be determined from calculated values that depend on the actual process conditions.
Unnecessary alarms based on non-comparable data can be avoided. Comparison of the calculated values to the measured data using statistical techniques can be a useful tool to indicate the condition of the equipment.
This means that a condition monitoring system must be converted in an “intelligent” system. Software must be implemented in order to perform the required calculations.

Brüel & Kjaer, Schenck Condition Monitoring Systems, in co-operation with Thomassen Compression Systems have developed a method to implement the software in the B&K Schenck CMS Compass system.

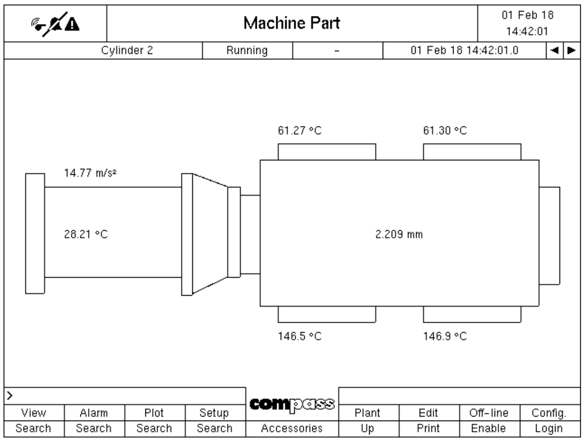
2. Description of the System.

The Compass system reads the required process data via the user’s DCS, an existing monitoring system, or directly from the instrumentation.
The formulae and set-ups of the performance monitoring system are fully transparent. The user can tailor the calculations to his own requirements. Formulae can be changed (when the user is authorised) and new extensions or additional functions can be added.



Process gas calculations require an extensive database of gases from which the properties can be calculated. Where the performance of reciprocating compressors is calculated the correct determination of the gas-properties is essential. In Compass an extensive gas property database is available. This database contains data of almost all the gases which are usually present in petrochemical or refinery service. The database can be updated to include specific or additional user’s requirements. The input of the gas composition can be through the keyboard, or can be treated as an input signal from a measuring device.

figure 1: Example of a Compass screen showing cylinder instrumentation



3. Required input data.

The number of signals to be read depends on the desired output. The gas composition is of course the major component in the calculations.
Pressures and temperatures in process lines and in the compressor itself are required to compare the

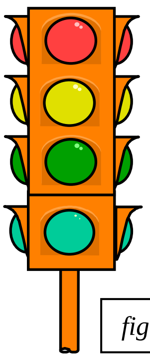
calculated and the actual data. This means that the instrumentation should have a reasonable grade of accuracy and should be calibrated regularly. This applies especially to flow measurements that are subject to a number of uncertainties, of which pulsating flows and pressures in the measuring device are the most important ones. (references to several reports of TPD-TNO and South West Research Institute on measuring errors in flow meters)

4. Calculation Sequence

The objective of the performance calculations is to check the proper operation of the machine. The output of the calculations may for example be:

- Discharge temperatures
- Flow
- Power
- Gasloads
- Rodload
- Pinload

The more extensive the calculations become, the more detailed information of the machine must be. The accuracy of the calculations is a direct function of the accuracy of the measurement devices.
When the measurement results are in error, they may be caused by failure of an instrument, or by the built-in read-out error. Calculations will then not reveal the real situation. The Compass system checks on system errors, such as failure of an instrument. Deviations of the measuring equipment have been overcome by including factors that have been included in the formula's for matching the calculations with the measured values when the machine is new or after overhaul. Trending the results will then show the decrease or increase of the deviation between the measurements and the calculation results. . In case of an failure or unreliable read-out, the Compass system will induce a warning.



RED: danger state

YELLOW: alert state

GREEN: normal

BLUE: system alarm

fig. 2: Compass alarm handling

4.1 Temperature calculations:

The final discharge temperature (Td) results from : suction temperature(Ts), pressure ratio (pd/ps) between suction and discharge and Cp/Cv (k) ratio. The discharge temperature is corrected for valve losses and other flow losses by a constant Ct. The suction temperature and pressures can be measured, the gasconstant has to be calculated for the current gasconditions.

$$Td = Ts * \left(\frac{pd}{ps}\right)^{K-1/K} * Ct$$

4.2 Flow calculations:

The displacement of the compressor (Qs) depend on the stroke volume (SV) of the respective cylinders the volumetric efficiency (VE), and the clearance volume (CV). The volumetric efficiency changes with the pressure ratio and the gas properties.

$$VE = 1 - \frac{CV}{100} * \left[\left(\frac{pd}{ps}\right)^{1/K'} - 1\right]$$

$$Qs = SV * VE * \frac{SPEED}{60} * density$$

4.3 Power calculations:

The indicated power (Ni) requirement is a function of the displaced volume (Qs), the compression ratio and the gas properties.
Factors Ct and Cn are added to make corrections for mechanical and valve losses.

$$Ni = Qs * FACTOR * Ct * Cn$$

$$FACTOR = Zs * Rs * Ts * \frac{K'}{K'-1} * \left[\left(\frac{pd}{ps}\right)^{\frac{K'-1}{K'}} - 1\right]$$

Power calculations are particularly suited to determine internal leakage's, such as piston ring leakage.

4.4 Gasloads:

The gas load on the piston is a function of the actual pressure difference over the piston. It should be calculated on the basis of the actual cylinder pressures at Head-end and Crank-end sides, and for one complete revolution. The effect of valve losses and other leakage effects are then automatically included. Part load conditions achieved by suction valve depressing or clearance pockets, may have a major influence on the loading of the compressor parts. The loads must not exceed the manufacturers maximum allowable.

4.5 Rodload:

When the gasload has been determined and the reciprocating masses are available, the Rodload, Pinload, and the Rodload reversal, can be calculated. The values have also to be calculated for one revolution, and results from the combination of gasload and the reciprocating mass forces. The

Pinload is of major importance for the life of the crosshead pin bearing.

5. Instrumentation

The performance calculations are based on the values produced by measurements. As a minimum the following input is required:

- Pressure measurements at suction and discharge side. The pressures should be measured near the cylinders in order to avoid deviations from the real cylinder pressures caused by line and or orifice pressure drops.
- Suction and discharge temperatures.
  The most effective location for temperature measurements is in the valve cavities of the compressor valves. These values are highly influenced by the load condition of the cylinder (suction valve unloading). This means that an additional signal is required enabling the operation mode of the compressor. This function can be provided by:
- Load indicators:
  The load indicator can be a signal from DCS or from a hardwired indicator .
- Flow measurements:
  Actual flow can be calculated from the operative gas conditions and the machine dimensions. It is be compared to the output of the flow measurement devices. The flow itself is not a direct indicator of the condition of the machine. Other parameters will yield earlier indications of possible malfunctioning of parts inside the machine. The flow comparison however, can give indications on process conditions, such as process control and/or leakage of recycle lines by means of flow comparisons.

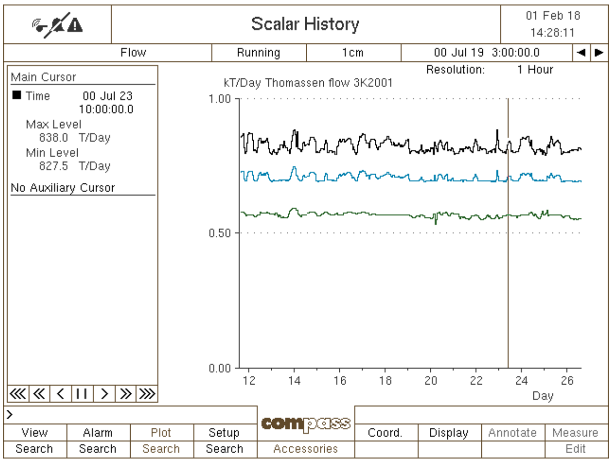


fig. 3: Compass screen showing flow in several pipeline sections

6. Case history

The figures below show an example of a one stage two-cylinder natural gas compressor. Figure 5 shows a change of flow in time, calculated and measured. It is evident that the change of flow is mainly caused by changes in suction pressure. This is illustrated in figure 6.

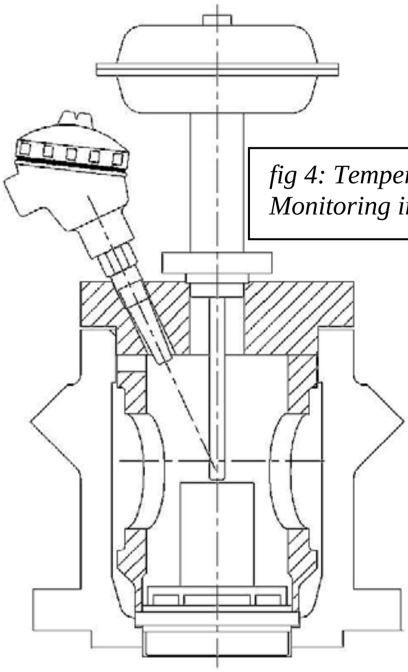


fig 4: Temperature Monitoring in valve cavities

7. Conclusions:

- Performance Monitoring is an efficient tool for analysing both compression process and machine condition simultaneously, and in a highly efficient manner. It is able to reveal machine deficiencies, that may otherwise be masked by non machine related process deviations.
- Expert system’s, such as the Advisor™ in the Compass system, allows incorporation of performance monitoring software.
- Performance Monitoring can be added as an extension to an existing Condition Monitoring system. When a system is to be extended, the abilities of a Performance Monitoring System are dependent on the abilities of the exiting Monitoring System.
- The accuracy of the instrumentation is of major importance for the accuracy of performance calculations.
- The gas composition and the changes in the gas composition can be of major importance for the flow and temperature calculations. Automatic gas analysing devices are available on the market. The output can be directly read in the system.

figure 5: flow versus time (absolute values)

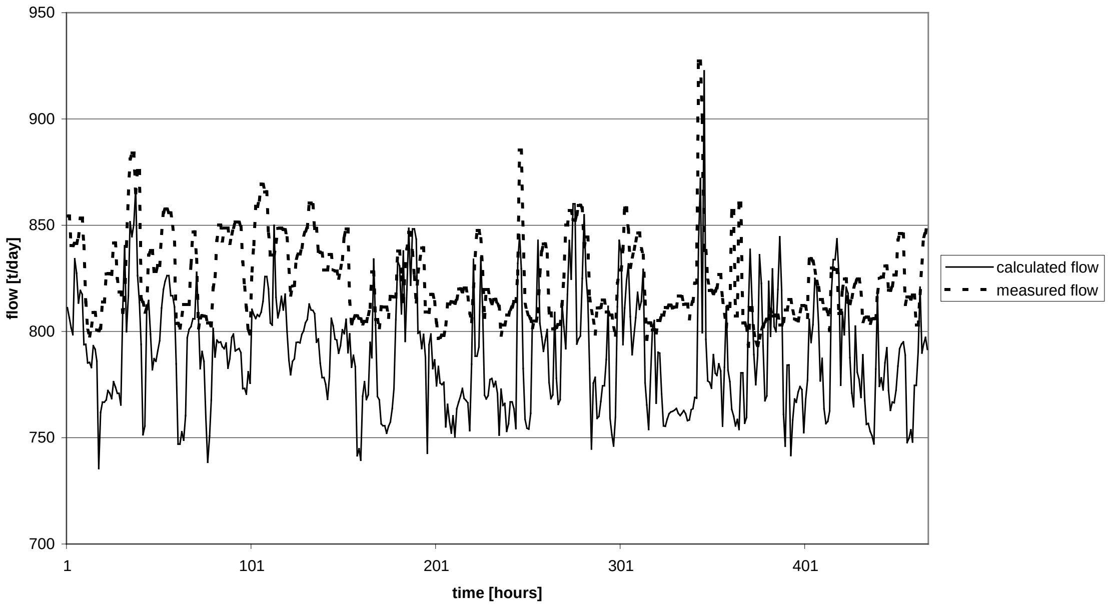
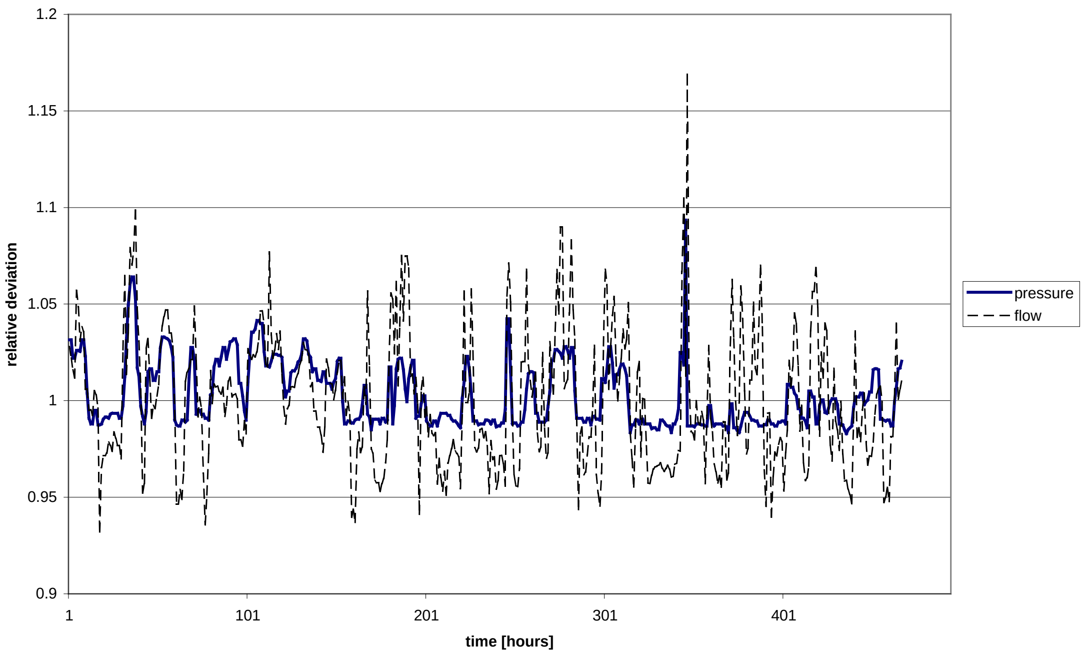


fig 6: relationship between suction pressure and flow (measured values)



Condition Monitoring
for
Reciprocating Compressors -
State of the Art

by:

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
17/18 May 2001, The Hague

Abstract:

As measurement equipment has allowed the collection of data as a basis for the evaluation of the condition of reciprocating compressors, acquired data - together with operator experience - has been used to identify problems as well as a basis for maintenance and operation decisions. The development of electronic sensor technology along with increased performance of computer equipment has led to electronic measurement and digital data storage. Although threshold monitoring for trend data has become standard in the past for numerous applications, experience shows that a higher level of analysis and data interpretation is necessary to gain "information" instead of "data".

To reach this level the acquisition and evaluation strategy for different measurements is as essential as the "intelligent" interpretation of the data. When comparing turbo machines in the development of condition monitoring systems, reciprocating engines need a substantially different strategy. This paper describes the state of the art reached with monitoring strategies and techniques relating to reciprocating equipment.

1 Introduction

Over the years electronic measurement instruments and analysers have been increasingly used in condition monitoring in the maintenance of machines in industrial installations. The screwdriver behind the fitter's ear was replaced by the piezo-electric accelerometer and the digital FFT analyser.

With simple data collection with manual assessment initially playing the leading role, in the course of rapid technical progress in the field of sensor technology and electronic data capture many scientific analytical methods could be made available to a wide circle of users involved in maintenance. A range of well-known procedures for condition analysis, e.g. Fast Fourier Transformation (FFT) were integrated in order to make it easier for the user to find a diagnosis.

For economic reasons the developers focused initially on the large market for rotary machinery, e.g. turbines and centrifugal compressors. For condition assessment and monitoring of oscillating machines, such as reciprocating compressors, special segmented analyses were developed later in order to meet the special requirements in terms of the characteristics of reciprocating engines.

The ever increasing demands made on the availability of machines resulted in a change from periodic (off-line) to continuous (on-line) condition monitoring. With the major expansion in digital process control systems and LANs networkable integrated analysis systems are needed today which collect electronically measured data over the whole compressor and present it to a group of maintenance engineers via a central visualisation monitor. Those responsible for maintenance can use their workplace computers to access the monitoring system directly through a networkable visual display. Modern possibilities in telecommunications and the increasing automation of business mean that remote monitoring is fast becoming the standard in condition monitoring.

Against a background of proven ability condition monitoring for reciprocating compressors has established a firm place for itself in modern maintenance engineering.

2 Condition Monitoring for reciprocating compressors

2.1 Maintenance strategies

The technical requirements on modern condition monitoring are essentially prescribed by three technical and economic objectives:

1. Machine monitoring to ensure the safety of the plant.

2. Avoidance of production downtime from unscheduled shutdowns, targeted planning of maintenance and shutdown times.

3. Optimal use of the wear potential of engine parts (condition-oriented maintenance).

Depending on the machine's production environment the three aims mentioned above are weighted very differently in order of priority. In the case of a 2.5 MW hydrogen compressor operational safety clearly plays a more important role than in the case of a 150 kW air compressor, where reducing the maintenance costs to a minimum can possibly be defined as the most important aim.

1. Machine monitoring to ensure the safety of the plant

To provide continuous operational condition monitoring, data on a machine's condition must be constantly collected and assessed. Of especial interest here is data which indicate a rapid change in the condition of the machine, e.g. vibrations. This gives early warning of disaster and minimises possible consequential damage, which is often very expensive to put right.

2. Avoidance of production downtime from unscheduled shutdowns.

Along with the direct expense of a maintenance procedure, e.g. material and labour costs, stoppage of the machine causes production downtime which is often several times more costly than the simple maintenance costs. This is the case in particular if there are no machines in reserve.

Total cost of maintenance measure =
material costs + labour costs
+ costs of production downtime

With carefully targeted planning of maintenance works the frequency and duration of shutdowns can be reduced thereby resulting in a decrease in costs caused by production downtime. To plan such work information on machine condition is necessary, e.g. the number of fitters, the type and quantity of spare parts and the duration of the work must be determined.

(weaknesses) must be analysed in terms of maintenance.
A survey conducted in 1997 by Dresser-Rand of 200 operators and designers on the causes of unscheduled shutdowns of reciprocating compressors shows clearly the main points in terms of maintenance technology (see Fig. 1).

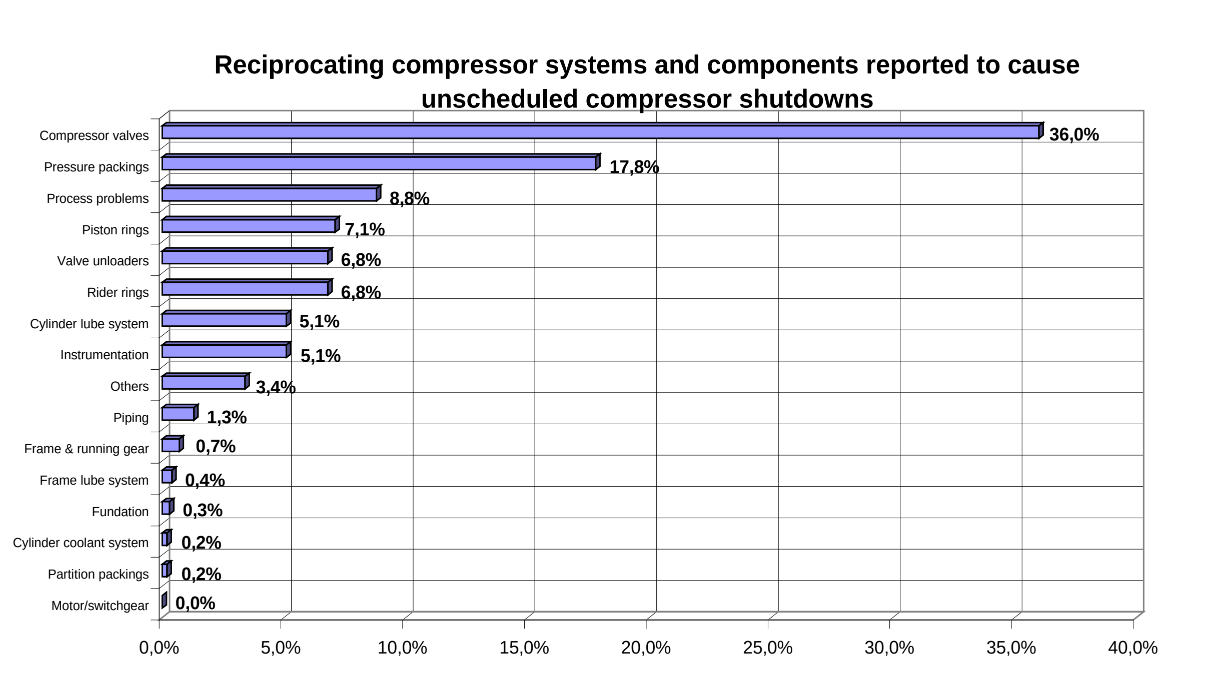

Fig. 1: Primary causes of unscheduled reciprocating compressor shutdowns

3. Optimum exploitation of the wear potential of engine parts (condition-oriented maintenance)

In condition-oriented maintenance machines are only shut down if their condition demands it. Parts are only changed if a damage criterion is reached. In this way the total (wear) potential of the parts in terms of service life and wear reserves should be exploited and thereby above all a reduction in material costs is achieved. This may only be possible if one has a precise knowledge of the machine condition. First and foremost the existing wear potential in “traditional” wear parts, e.g. piston rings or packing, can be exploited in a focused way, thereby optimising machine operating life.[i]

2.2 Technical requirements

The objectives described in Section 2.1 are general and are not specific to reciprocating compressors; they apply to numerous industrial machines. In order to design an effective monitoring strategy for condition monitoring systems especially for reciprocating compressors the mechanical features

The statistics show that eight component groups were responsible for 94% of unscheduled shutdowns. First and foremost defects in the suction/discharge valves are obviously responsible for unscheduled maintenance. More recent experience indicates similar results, even though the absolute involvement ratio of valves has dropped slightly due to the use of new materials.[ii].

Today there are monitoring systems available on the market for every component that features in the statistics which were partly developed for rotary machinery. When choosing a system to install, care should be taken that the specific requirements of a reciprocating compressor as a piece of reciprocating machinery are taken into account, for example, by segmented analyses or operating point-dependent threshold checks.

3 Monitoring of suction and discharge valves

Given the frequency of valve damage a range of monitoring methods is employed today. The three most important methods are:

- Measurement of the valve pocket temperature
- Vibration analysis
- p-V Diagram analysis

3.1 Valve pocket temperatures

Measuring the gas temperatures in the valve pocket is the most simple and most cost effective method of monitoring valve condition. Using a temperature probe the gas temperature can be measured in the valve chamber: the up-flow of the suction valve and down-flow of the discharge valve. If there is an obvious increase in temperature at one valve one can assume that there is damage, e.g. a leak, at this point (see Fig. 2).

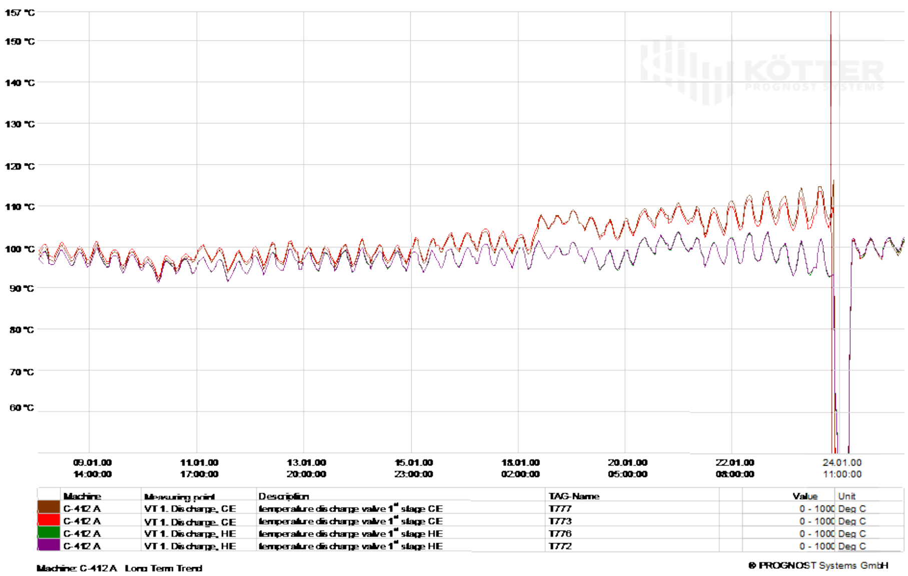

Fig. 2: Head and crank end discharge valve pocket temperatures showing a distortion caused by a broken valve plate on a crank end valve

This method has the advantage of being cheap to realise and simple to install in almost all types of valves. Generally speaking, the measuring points are integrated directly to the process control system and therefore can be linked into larger monitoring systems. The disadvantages first and foremost is the in part difficult interpretation of the measurement results in multi-stage machines or where there are varying pressures on the intake and discharge sides.

But these problems can be countered by adopting operating condition-dependent threshold monitoring. With moderate technical outlay valve pocket temperatures do in many instances provide important information on the seal condition of the valves.

3.2 P-V Diagram Analysis

In p-V diagram analysis the dynamic march of pressure inside the cylinder is measured. For this special pressure sensors with a frequency sensitivity of about 5 kHz are needed which are installed in a bore leading directly into the cylinder. A particular requirement on the pressure sensor when used in process gas compressors is often the composition of the hydrogen-bearing gases.

The p-V diagram analysis is one of the most important methods in valve condition monitoring. The analysis of a recorded signal allows conclusions to be drawn on conditions of the seal elements in the cylinder area. Damage to valves that result in leaks cause characteristic changes in the measured march of pressure. The measured pressure flow is converted into a pressure - volume diagram for which characteristic values are calculated at certain fixed points. These values, e.g. valve losses, polytropic exponents or crank angle at which the suction pressure is reached, undergo automatic threshold monitoring in order to identify from the excess values occurring whether the suction or discharge valve is defective.

Cylinder pressure is a very good condition indicator as it reflects the real situation inside the cylinder. The user not only obtains clear local and function-oriented information on condition but can also identify the precise effects on the compression process of the machine, for example the extent of a reduction in capacity caused by damage. This information is again a basis for a decision whether servicing in mechanical and production engineering terms in respect of any production commitments is economical.

P-V diagram analysis with automatic data formation does, however, require almost as sine-qua-non operating point-dependent monitoring in order to take account of the various compressor operations and to differentiate between operational-dependent and condition-dependent changes in data. In particular this applies to machines with valve unloaders to regulate capacity. The energy used and power for the compression process can be read directly from the area of the measured p-V diagram, making it possible to directly determine the efficiency.

The p-V analysis is one of the central and most comprehensive methods for assessing changes in condition in the area of the cylinder seals. Given the mechanical and measurement procedures required, fitting to older machines can be relatively expensive. But given the quality of information and its comprehensive nature these costs can as a rule be justified.

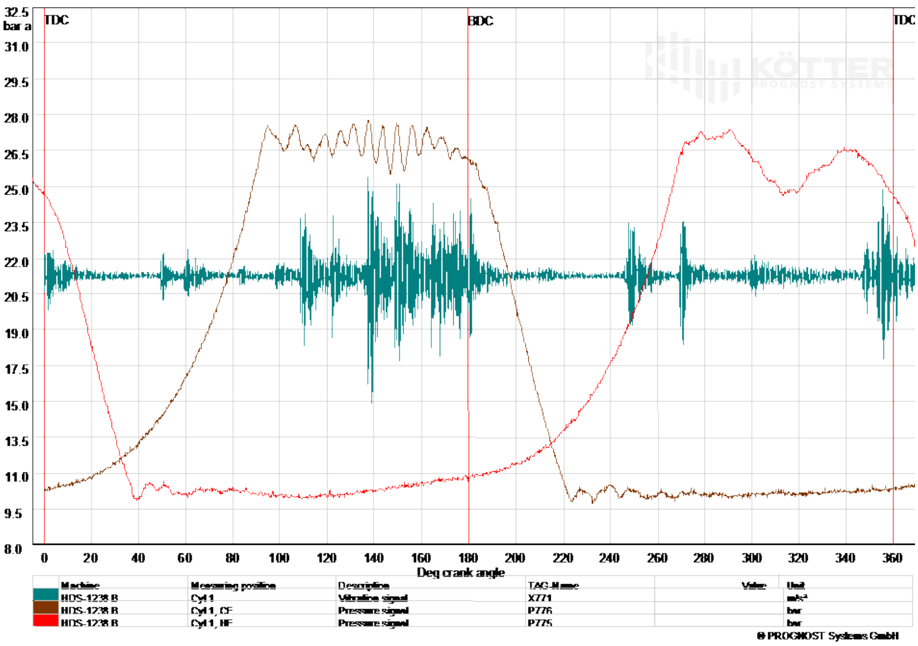

Fig. 3: Cylinder vibration and indicated pressure course measured with valve flutter occurring at the crank end discharge valve

3.3 Vibration analysis

Additional information on valve condition is provided by the vibration accelerations measured at the cylinder. This involves installing accelerometers on the cylinders which cover a frequency range up to 30 kHz in order to record the mechanical vibrations caused by the opening and closing of the valve. When installing the sensor devices the locations must be selected in such a way that vibrations from as many valves as possible can be measured.

This method provides above all information on the valve opening and closing processes where large vibration peaks occur. If what is called “valve flutter” occurs with multiple opening and shutting of the valves, this can be identified from vibration analysis. “Valve flutter” often results in dramatic reductions in valve endurance and frequently occurs in speed regulation machines or where the stage pressure has been altered. Figure 3 illustrates a typical vibration process with occurring valve flutter.

As the opening and closing of the suction/discharge valves only takes place at certain intervals at different points on the crank revolution, the vibrations measured at the cylinder - apart from the peaks of valve action - also show wide areas with relatively low amplitudes. In order to provide effective vibration monitoring and to be able to separate the low vibration areas and the vibration peaks caused by valve action and to monitor them separately, a special analysis is required. Here the vibration signal of a crank revolution of 360° is split into 36 segments each of 10°. Characteristic values are calculated for each segment, e.g. for peak and RMS values. Each segment is allocated its own threshold value (see Figure 4, next page).

The particular quality of segmented vibration analysis lies in the integrated diagnosis aspects which generate condition information directly from the measured signal. As each segment is allocated one phase of the work cycle such as compression or suction, when a threshold is exceeded reference is made directly to functions and components such as valves.

Vibration analysis on the cylinder is above all effective in combination with pressure monitoring so as to prioritise the interpretation of threshold violations more rapidly. When installing the accelerometer the mounting points must be checked for signal quality to ensure that peaks when valves open and closed are transmitted. Usually only one sensor is mounted per cylinder so that the installation costs are slightly less than for p-V diagram analysis.

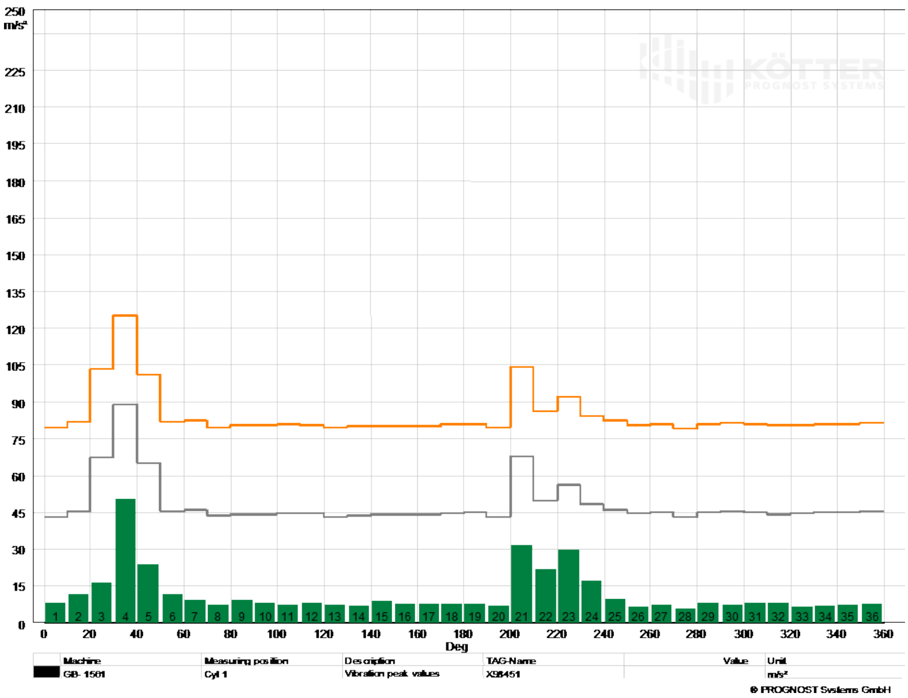

Fig. 4: Segmented vibration signal with two-staged threshold setting

4 Packings

The second most frequent reason for unscheduled shutdowns, after the inlet and outlet valves, is problems with packing. The packing seals the cylinder along the outside of the piston rods. Currently the following procedures are commonly used to monitor wear:

- Measurement of leakage gas flow and temperature
- P-V diagram analysis

When measuring leakage gas flow and temperature the leakage gas removed by the packing is studied as an indicator for the wear condition of the packing. To this end either a temperature probe or flow gauge is installed in the leakage gas pipe. If there is a clear increase in the quantity of leakage gas or of temperature it can be assumed that there is a leak in the packing.

Leakage gas quantity monitoring is a widely used method which in new compressors frequently forms part of the basic kit. Given the lower susceptibility to distortion, temperature measurement is preferred to quantity measurement in most cases. However, an advantage of quantity measurement is the indication of the quantity of gas lost which allows an economic assessment of any exchange work.

Conducting the measurement of leakage gas quantity and temperature requires preparation of the packing or the distance piece on the cylinder in order to facilitate the appropriate input and removal of the gas. For leakage gas quantity and temperature measurements the packing must be appropriately prepared in structural terms which may require a change of the component used. The measuring points are either directly linked to the monitoring system or via the digital process control system.

The p-V diagram analysis also fulfils an important function in analysing the condition of the packing. Apart from the valves the packing is also a cylinder seal component. Increased leakages lead to characteristic changes in the indicated pressure flow. Figure 5 (see next page) shows the pressure course on the crank end side of the cylinder one with tight and the other with leaking packing. The changes in the area of reverse expansion can be clearly identified.

5 Piston and rider rings

To determine wear of the piston and rider rings, what is called the “rod drop analysis” is widely used. This involves the piston rod drop being continuously measured by a proximity sensor. In the course of its life wear of the piston ring leads to measurable piston rod drop.

For this purpose an induction proximity sensor is fitted to the packing. For signal analysis either individual interval values at specific points on the piston head are measured or segmented analyses are conducted over the whole signal flow. These are particularly advantageous as possible interruptions caused, for instance by particles or lubricant residues on the piston rod, can be better identified and therefore can prevent erroneous interpretations. In addition, when there is damage to the piston rod connections to the piston or to the crosshead, information can be gained from the total rod drop signal as the piston rises. Figure 6 contrasts the course of a rod-drop-signal in Good Condition to the signal where the piston rod-crosshead connection is broken. Due to the loose connection the position signal shows strong vibrations in certain areas.

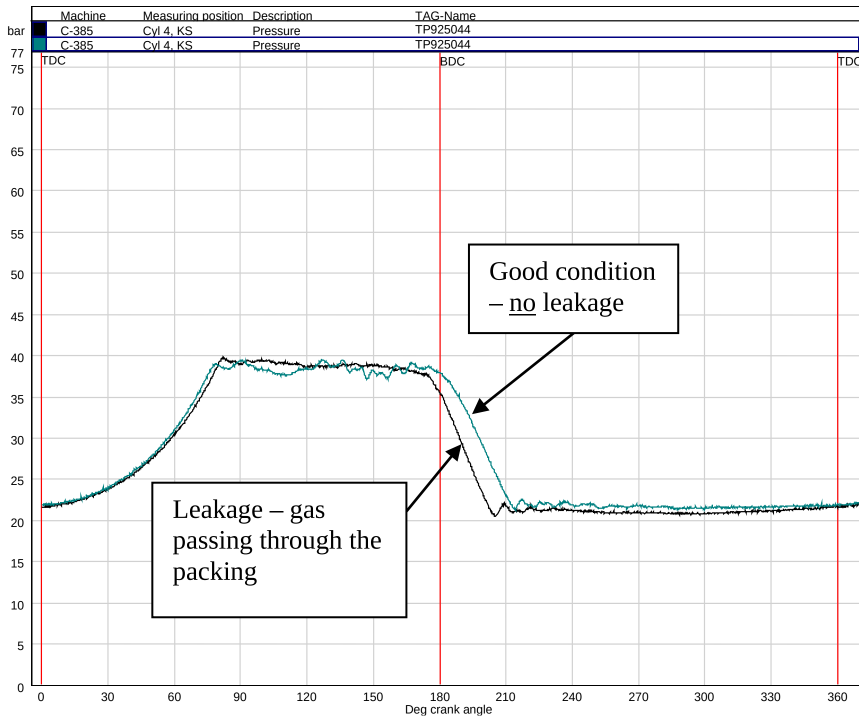

Fig. 5: Effect of leaking packing on the indicated pressure course

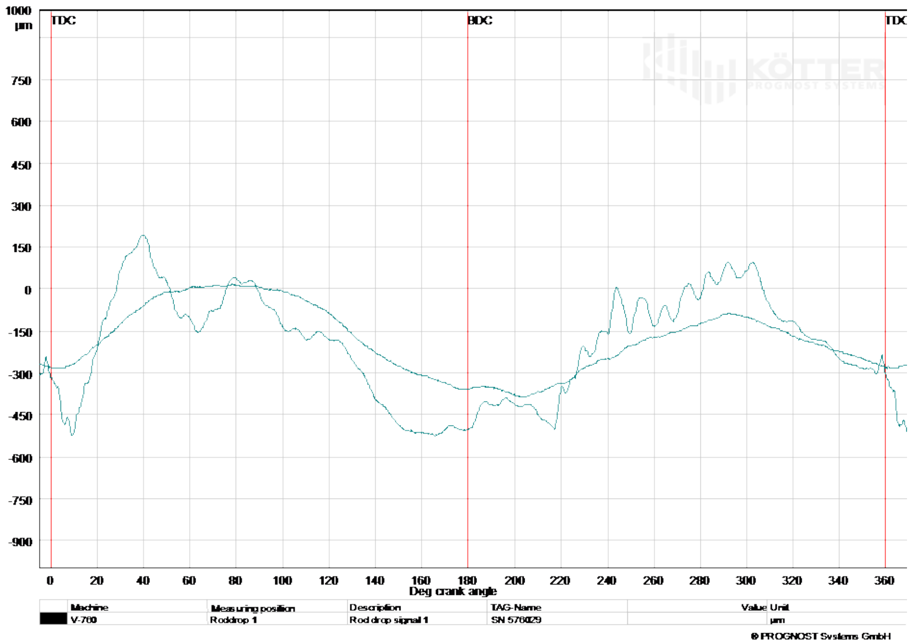

Fig. 6: Comparison of normal rod drop signal with distorted curve due to loose connection between crosshead and piston rod

Determining rider ring wear requires monitoring of the measured values over a long period of time in the form of a trend. Figure 7 (see next page) represents the trend values of four segments of a piston rod analysis over a period of about ten weeks. The values shown give the distance measured from the piston rod to the sensor in the crank angle areas 0-10 degrees, 80-90 degrees, 170-180 degrees and 350-360 degrees. The exponential increase in wear is clearly identifiable.

On inspection the piston rings were found to be badly worn. A more detailed study of the cylinder lubricant (see Figure 1 with 5.1% of the causes) found a partially blocked lube oil inlet pipe. The reason for the particularly rapid rod drop lay therefore in the inadequate lubrication of the cylinder.

6 Valve unloaders

Regulation of the capacity of reciprocating compressors in recent years has attained ever increasing priority through changing production requirements and increased efforts to save energy. The reverse flow principle of controlling capacity that has been known for decades became available for a large number of compressors due to improved lifting devices and suction valves.

Effective function monitoring of these control devices is possible with p-V diagram analysis. With measured pressure course the installed control stages of each individual cylinder can be checked. If p-V diagrams are analysed in the framework of operation point-dependent threshold monitoring, possible errors in control can be automatically identified and pin-pointed to a cylinder. Figure 8 shows the four p-V diagrams of two structurally identical double-acting cylinders with valve unloaders. The lifting device on one of the cylinders is obviously not working properly.

In many instances more information on the proper adjustment of a lifting device can be gained from vibration monitoring at the cylinder.

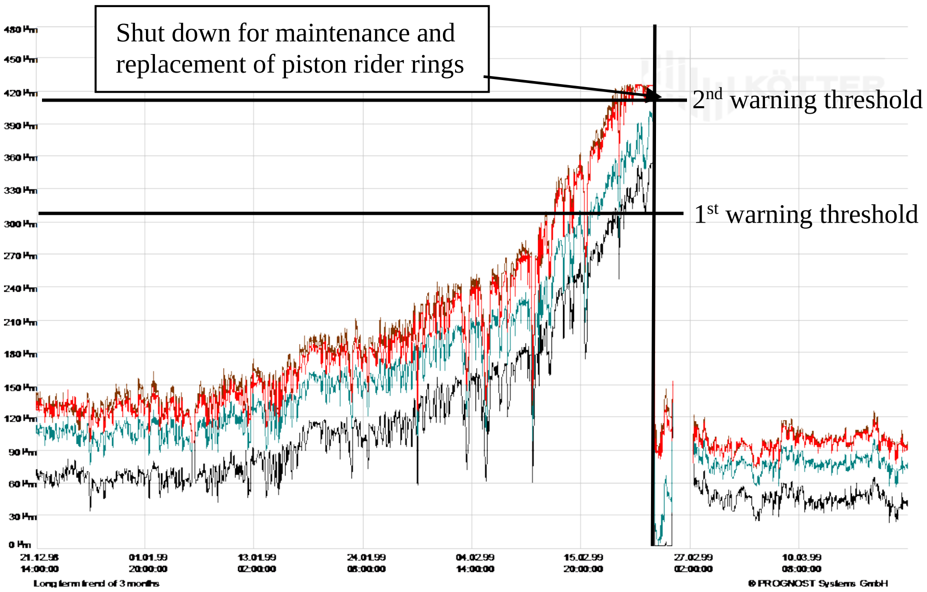

Fig. 7: Rod drop values of segments 1, 9, 18 and 27 (1 hour average) within the last 8 weeks prior to the shutdown for replacement of the worn rider rings

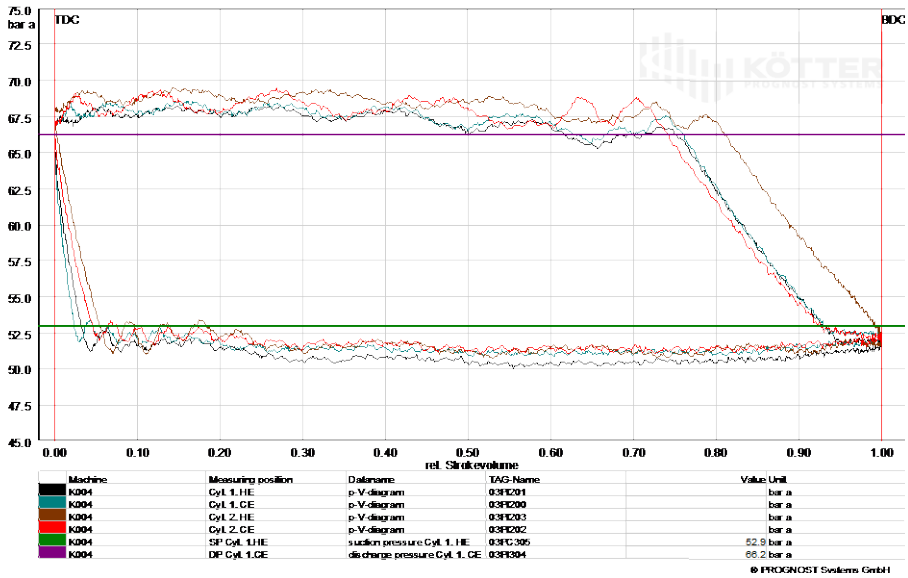

Fig. 8: P-V diagrams of a one stage two-cylinder compressor with one defective valve unloader

7 Running gear

When monitoring the condition of a compressor running gear and its components various methods are employed. For wear monitoring of bearings (crankshaft, connecting rod) temperature measurement points are used as a rule which are advantageous because of their simple assembly and low acquisition costs. These are frequently supplemented by flow measurements and temperature measurements of the lubricating oil.

As regards general operational moni-toring of the machine and to avoid mechanical cones-quential damage vibration monitoring is used. Here, depending on the type of compressor in the area of the crank or the crosshead slide, vibration sensors are mounted. Essentially the same technical requirements exist as regards frequency resolution for the sensor and measurement chain as for vibration measure-ment on the cylinder.

In analysing the vibration processes segmented analyses have proven their worth in terms of rapid diagnosis. This means that vibration peaks, e.g. at characteristic load change points, can be effectively moni-tored. Figure 9 shows the measured vi-bration peaks in the area of the crosshead slide that occur at the time of rod load reversal typically when there is increased bearing play in the crosshead bolt.

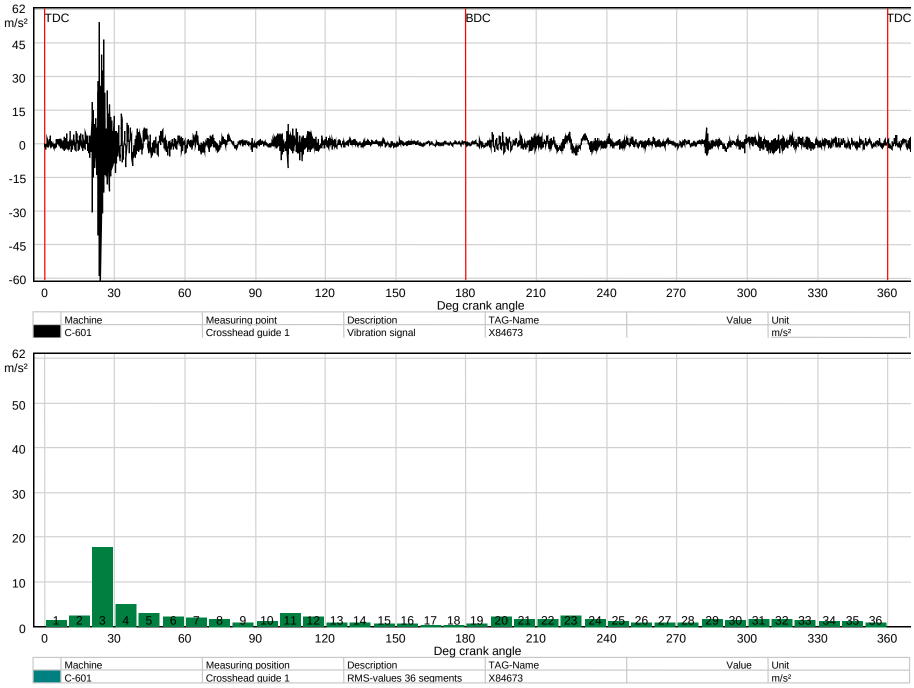

Fig. 9: Measured vibration and related segmented RMS-values showing a peak at the rod load reversal

Figure 10 shows the vibration signals of a four-crank horizontal opposed compressor measured on the crosshead slide which triggered an alarm. Inspection of the machine revealed a defect in the crosshead bolt which had already destroyed the safety plate. With the rapid alarm it was possible to avoid complete removal of the crosshead bolt and prevent substantial consequential damage to the crank mechanism.

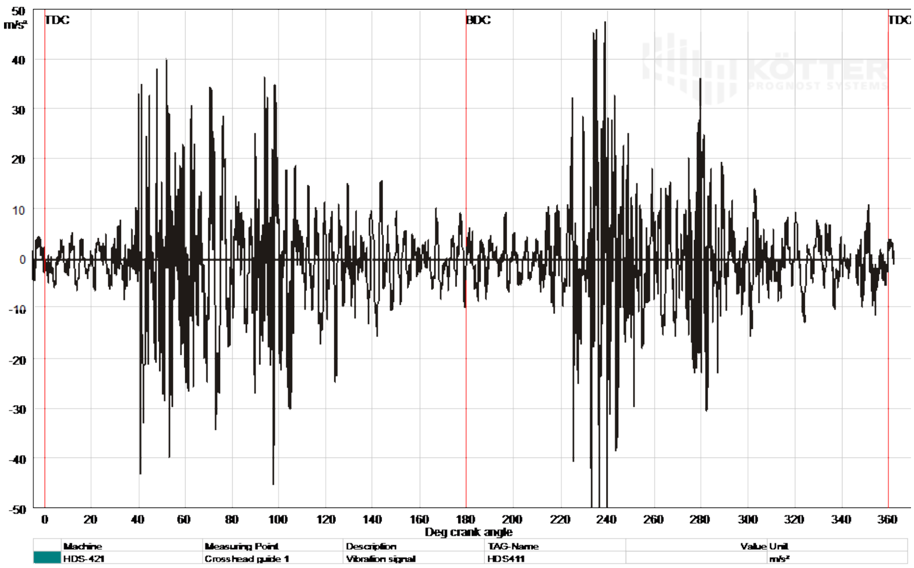

Fig. 10: Vibration signal measured at a crosshead slide with damage at the crosshead safety-plate

To avoid consequential damage it is important to conduct continuous contemporaneous analysis and quickly alert operating staff or shut down the machine. These requirements are met by special continuous monitoring which subjects the vibration signals of each crank revolution - individually and segmented - to a threshold check (safety monitoring). In addition, frequency analyses (FFT) of specific signal sections, e.g. of the load exchange areas, can provide information on structural and mechanical changes.

8. Conclusion

Experience in operating reciprocating com-pressors has highlighted the focal point of maintenance work. Above all the main areas for using condition monitoring are in the area of seal components (valves, packings, piston rings) and the machine. Methods and systems for condition analysis for the areas in question have been known and tested for many years.

Certain analytical methods, such as p-V diagram analysis or vibration analysis, make it possible to assess several components with one recorded signal. But in order to get a better assessment of the condition of one group of components, different measurements are applied. Here, rapidly-changing condition values, such as vibrations, and longer fluctuating values, such as temperatures, supplement each other and these are monitored according to changing operating conditions. Both these kinds of condition values have to be integrated in a modern monitoring system thereby offering the whole maintenance team via a computer network a single analysis and diagnostic interface.

The systems available on the market specially for reciprocating compressors, e.g. the PROGNOST-NT which has been in use for over 10 years, are by now being successfully employed by a large number of maintenance teams in the refining, chemical and natural gas industry and have achieved a respectable state of the art. In years to come, mathematical models will increasingly be used to display the mechanical and physical processes of the reciprocating compressor. In this way, in the future an even more detailed error recognition and diagnosis system will be possible, enabling all technical possibilities to be used in an economically-run plant.

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SULZER BURCKHARDT

Virtual Product Development - A Method for Increasing the Reliability and Efficiency even in the Reciprocating Compressor Industry

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th / 18th, 2001, The Hague

Abstract:

The virtual, digital product is a substantial prerequisite for the concurrent or simultaneous engineering. On the basis of a central (CAD-)data model the virtual product development is able to consider all aspects from the concept to the implementation phase.
First experiences with the virtual product development of a piston compressor show how static and dynamic Finite-Element-Analysis, component optimizations as well as casting and manufacturing improvements can be incorporated into an integrated development process. Individual components were optimized so as to allow for considerably higher specific loads at comparable total costs. The dynamic behavior of the entire compressor has been also substantially improved.
It is shown how the models can be employed in a range of further activities, extending from the manufacturing, pulsation and vibration studies up to the documentation production.

1 Introduction

Product and process technology in the surrounding field of the reciprocating compressor industry is rapidly evolving. Customers are placing an increasing emphasis on quality and reliability, looking at the same time for a cost-effective compressor [1].

There is no doubt that the product quality is of greatest importance for a reciprocating compressor manufacturer to survive in the extremely competitive global market. To respond to the increasingly dynamic and challenging market demands, the manufacturers have to modernize their products and product lines. Therefore the aging but proven products, strongly grown casting pattern and casting pattern variations consequently are requiring a complete redesign of the products.

The conventional way of redesigning a product leads to rework, iterations with non-value-added changes and recalls, which can easily use up the profit margins of a new product. If the work is done using an integrated development environment, a reduction of the design cycle time and a product value improvement at the same time is possible[2].

2 Comparison of the Traditional and Concurrent Engineering Approaches

2.1 Traditional Engineering Approach

The traditional engineering approach [Fig. 1] is also known as "Serial Engineering". This conventional product development process has been largely sequential in nature and segmented with some very hard breaks between the process phases i.e. each discipline performs its own individual function and passes the results to the next discipline in the serial chain. Typically, there is very little or no interaction at all among various disciplines, which in many cases causes a rapid growth of designs without regard to the impact on the rest of the organization. This thus leads to problems later in the product development cycle. The most obvious barrier is the separation of various functions. The most critical breaks are the gaps between design and engineering and again engineering and manufacturing. Once a new product design is verified, by either simulation or hardware prototyping or both, it is tossed "over the wall" to the manufacturing, test, quality and service engineers for review. Continual changes usually must be made as problems are usually discovered later in the process. This invariably adds to overall development cost and time to market.

In terms of product quality, the emphasis is more on correction than prevention. Design errors are detected only during manufacturing rather than eliminated during the design stage in order to prevent them from occurring in the production stage. This process inevitably leads to reworks. Generally manufacturing becomes more expensive due to the additional number of tests and checks required.

Information flow is mostly one way and is severely restricted as the information is stored on many media in many locations as a result of recreating an items geometry or other product design information in such areas as marketing proposals, conceptual design, detail design, engineering analysis, detail drafting, fabrication or assembly sketches, NC programming etc. As a result, additional effort is required to manage and maintain all the redundant design information sources, to transfer data from one system to another and to coordinate the activities spread among the company.

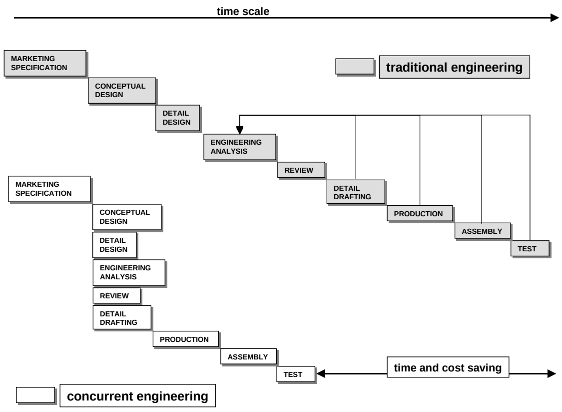


Fig. 1: Traditional engineering approach vs. Concurrent Engineering approach

2.2 Concurrent Engineering Approach

The Concurrent Engineering (CE) approach [Fig. 1] encourages teamwork and forces to focus on the expertise from all the disciplines that are involved to work closely together in parallel right from the early stage of the product design and development stage. In order to realize the effective teamwork, sharing of ideas and goals has to go beyond immediate assignments and departmental loyalties. Tradeoffs regarding ease of production, testing and servicing are made along with product performance, size, weight, parts and cost tradeoffs. When a design is verified, it is already manufacturable, testable, serviceable and of high quality. Hence, multiple iterations on the product changes commonly found in sequential processes are eliminated. This will minimize the time from the concept to the product release.

In addition to CE, working in parallel or concurrent activities allows multi-disciplinary teams to remove any problems at the earliest stages. This will prevent any design problems to surface later in the process and help to reduce the number of checks and tests required later in the process.

Information sharing is extremely important and necessary for effective CE implementation. The organization is unified with mechanism for storage, control and retrieval of all information and data relevant to the product. All functions have access to these resources. Information exchanges among all teams are strongly encouraged.

3 Prerequisites

Concurrent engineering is a way to promote the co-operative working of all disciplines involved in a product's development cycle. It increases the capabilities and assists the workings of multi-disciplinary teams to manage complexity. The development process becomes shorter because constraints and requirements that often cause delays and costs are managed in a timely manner. It is an investment whose advantages are many, such as: reduced life-cycle cost, improved quality and reduced time from concept to the production of parts and systems. If CE is consistently applied, a reduction of 50% in both time and costs for the product development process is possible.

But there are organizational and technical prerequisites necessary for the realization of CE:

1. Organizational structure

Creation of an organizational structure for working in teams and a strict project management are the basis for CE. A hierarchical organization structure becomes incapable of coordinating the many steps needed to provide effective early involvement and parallel design. Cross-functional product development teams are a way to breakdown this organizational complexity and put together the necessary skills and resources to support more effective product and process development.

2. Technical management

The realization of CE also requires evaluation, use of appropriate new technologies and integration, co-ordination and management of computer-based tools. Synchronization of data and information is also essential for optimal results and can be solved using an engineering data management system. These technical prerequisites are leading to a system as shown in Fig. 2 with the digital product database as the central point for all other CAx technologies involved in the process.

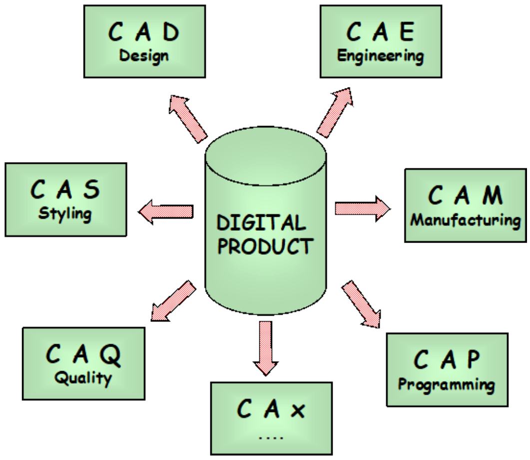


Fig. 2: The digital product in the surrounding field of CAx-techniques

Furthermore the digital product can be seen as the central point for all directly and indirectly related processes. Fig. 3 shows how sales and service departments among others are connected to the digital product database.

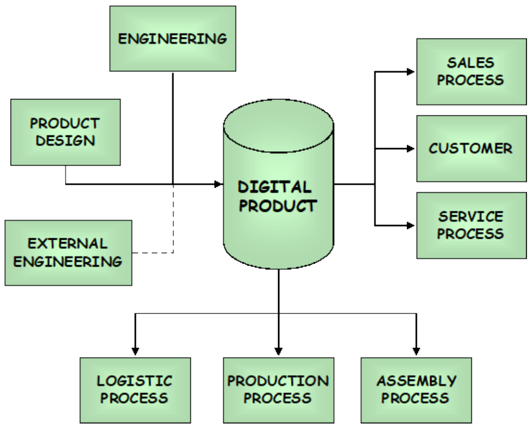


Fig. 3: The digital product in the surrounding field of the business processes(according to [3])

3. Soft factors

There are a number of soft factors which should not be underestimated. The greatest challenge is not the implementation of new techniques, business practices or technology, but overcoming the organizational barriers, accepting the considerations of internal and external customers and suppliers, as well as overcoming the resistance to changing the way things are done.

4 Practical Experiences

In the following, only CAE results in the field of static and dynamic optimization analysis for crankshaft, compressor frame, distance piece and cylinder are presented. The optimizations always had the goal of making improvements with respect to manufacturing economics, reliability and ease of maintenance. The optimizations combined with an intensive cooperation with the foundry and the manufacturing department ensure that:

- the castings are suited to modern casting technology (such as reducing foundry molding cores).
- the number of casting pattern and casting pattern variations are minimized.
- the design considers the API618-guideline.
- the optimized individual parts allow a 45 % higher specific load at comparable total costs.

4.1 General Remarks

As mentioned above the CAD-model is the central part for all disciplines being involved in the process. But it is not sufficient to build a model and hope that the model could be used for further investigations especially for CAE.

Experiences show that the criteria to define the model quality can be grouped in geometric features, non-geometric attributes, and form and format of the model. For the form and format of the model, like the description of the structural layout of the model, the modeling strategy, naming conventions within the model etc., standardized methods how to build up the CAD model are a necessity to guarantee the further use of the models for various CAx techniques. Non-geometric attributes, like numerical data, material attributes, parameter settings could also be solved successfully through a defined standard. For the geometric features, like excessive warping of faces, wrong normal directions of faces, discontinuities and gaps between adjacent faces etc. software tools that automatically prove and correct the design are the recent developments in this field. So the quality of the CAD-model could be improved in an early stage of the process[4].

Another important factor for success are the CAD engineers themselves. The CAD user should have a common understanding of CAE. So its possible that simple parts or in general linear calculations could be treated without the help of the CAE specialist, thus minimizing the up-front CAE calculations for the specialists. And for example for the nonlinear FEA the CAD engineer should be able to prepare the model in a way that it could be used for further investigations. Preparing the model means neglecting for example threads in the model or very small radii for an optimization run to reduce the time for the analysis.

4.2 The CAD Model

The following figures are showing a small extract of the work carried out. The model in Fig. 4 is ready for the CAE. Details are removed and the model is reduced to the characteristics necessary for a FEA.

Fig. 4: CAD-model ready for CAE

Finally the various optimized parts (Fig. 5-6) are digitally arranged. With such a digital mock up as shown in Fig. 7 it is possible to run a motion analysis proving that each part could be assembled or disassembled and is working without colliding with other parts.

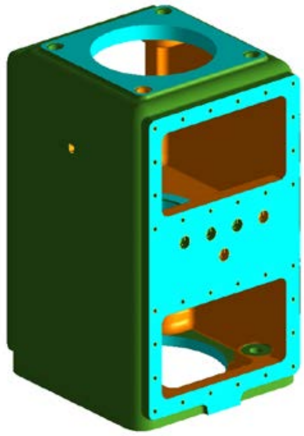
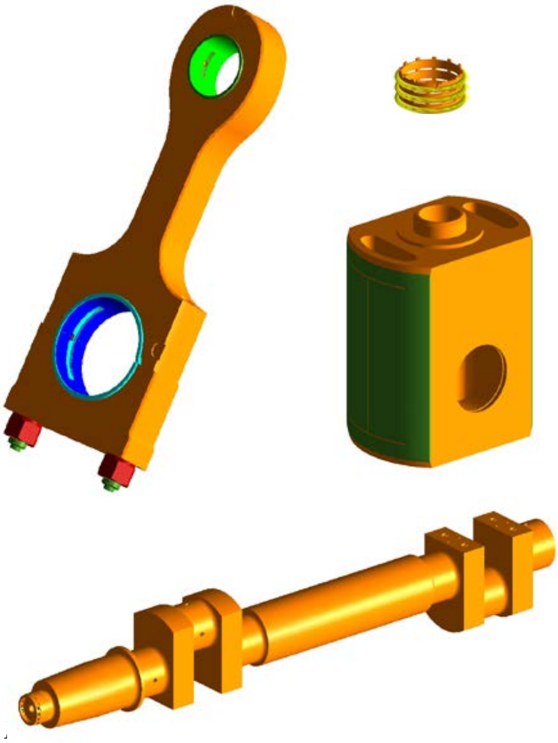


Fig. 5: Optimized distance piece



(crankshaft, crosshead, connecting rod with bearings, oilscrapers)

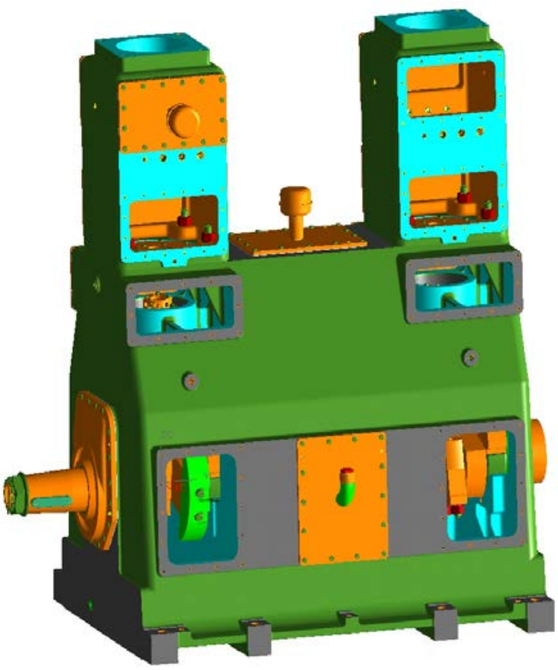


Fig. 7: Digital mock up

4.2 Optimization of the Crankshaft

As a standard procedure the crankshaft calculation has been done so far using a simplified model. Each crankweb is calculated individually on the assumption of a bending beam under known load forces. Calculation is done in several load steps for a complete revolution and the stresses in the fillet radii are calculated using the appropriate form factors. There is no consideration of dynamic effects. The influence of a flywheel wobbling effect is also neglected.

To consider these restrictions and therefore increase or optimize the specific load capacity, a sophisticated simulation technique was used. Special algorithms make the program used[5] be a unique tool for most accurate nonlinear analysis of the cranktrain dynamic and the fatigue behavior of the crankshaft. Herein the actual dynamic behavior of rotating engine components - such as the flywheel wobbling effects, the elastohydrodynamic conditions in the bearings and the non-linearities in the cranktrain are considered.

Using the CAD-Model and transforming it to a 3D-solid element model for a full dynamic analysis is not a practicable way. The transformation would lead to FE-models with more than 100k elements, which could not be dealt with for optimization. Economically the dynamic analysis should be performed using a realistic model that consists of beam and mass elements. Therefore only the specific crankshaft parts as shown in Fig. 8 are modeled. The bending and torsion stiffness and the center of gravity are calculated and transformed to the simplified model, Fig. 9. To verify the quality of the beam/mass model, a modal analysis for both the models could be performed additionally.

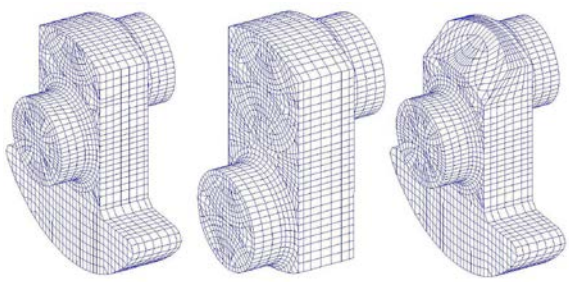


Fig.. 8: Different crank web types

Considering elastohydrodynamic conditions in the bearings and the stiffness of the crankcase the beam/mass model (including the flywheel) is used to calculate the dynamic behavior of the system due to maximum gas load, mass forces and maximum torque loading cases. Optimization is done with respect to the crankweb geometry such as width, fillet radii and mass distribution.

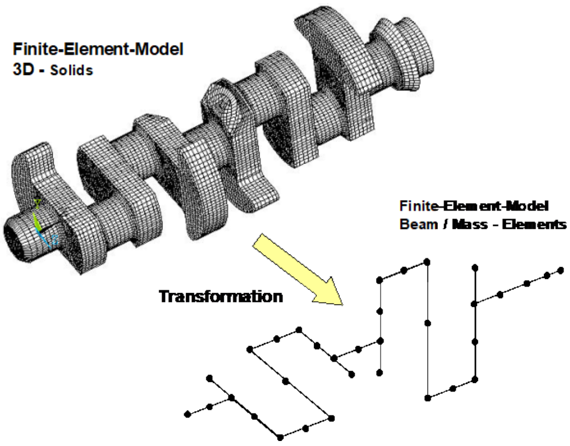


Fig. 9: Transformation from a solid model to a beam/mass model

The calculated reaction forces and moments for the optimized solution were used as input for the complete 3D-Model and the safety factors for fatigue behavior were determined. Fig. 10 shows the variation of the safety factor for fatigue fracture over an engine speed range from 250 to 400 rpm. For the highest strained point of each of the four crankwebs a minimum safety factor was found at 330 rpm. The finally optimized solution for the complete crankshaft can be seen in Fig. 11. Static and cyclic load capacities for the critical point for the crankweb fillets are shown. A load capacity of 100 % corresponds to the maximal admissible load if appropriate safety factors are built in. The maximum load is considered in the static load carrying capacity, while in the cyclic load carrying capacity the mean stress and the corresponding stress amplitude were taken into account.

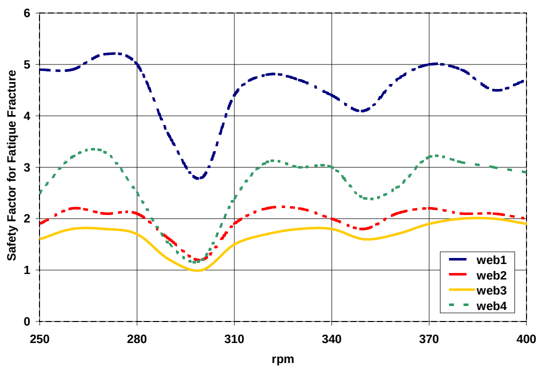


Fig. 10: Variation of the safety factors for fatigue fracture for different compressor speeds

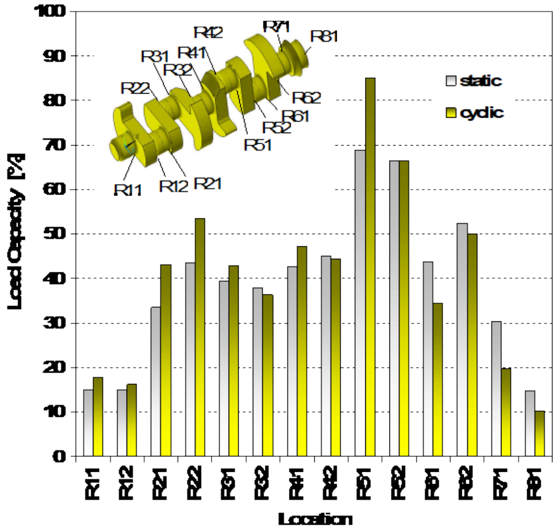


Fig. 11: Static and dynamic load carrying capacities for the crankweb fillets

4.3 Modal Analysis

One of the most challenging problems was the optimization of the dynamic behavior of the whole compressor including crankcase, distance pieces and cylinder. Especially for the vertical type, one of the stringent boundary conditions was the optimization of the stiffness in such a way that the first natural frequency of the compressor should be higher than 20 Hz (second order harmonic frequency, assuming an compressor speed of 600 rpm) for the largest cylinder. The initial, non-optimized configuration shows a first natural frequency of 12 Hz for a single throw compressor with the largest cylinder.

After some preliminary work two important questions arose. On the one hand, the outside shape of the compressor is of interest, i.e. is the silhouette conical (shape A), cylindrical (shape B) or bottle-shaped (shape C) as shown in Fig. 12. On the other hand, the question has to be answered as to how the thickness of individual walls should be optimized, thus achieving both a high stiffness and a low total weight. High stiffness of the compressor-frame leads to higher natural frequencies and for cost effectiveness a lower weight is reasonable.

Deviating from the idea of the CE using a central CAD-Model, an additional optimization to determine the fundamental construction principles, based on a shell element model, was performed. Optimization with the shell element models could be done within a reasonable time and is therefore more economical.

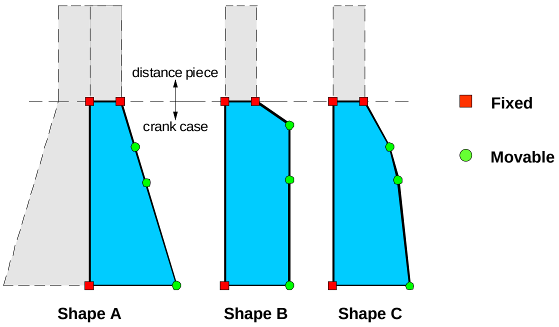


Fig. 12: Different sectional shapes of a compressor frame

The starting point for the optimization process is the “as designed” model. The first step is to specify an objective or “goal” function, which is minimized during the optimization process. Other parameters to be specified are functional constraints or limits on the state variables and the design variables. During the optimization, a series of simulations is launched to determine the optimal design variables. The final set of variables yields a result that optimizes the design based upon the user-specified objective, performance constraints and variables.

An extract of the optimizations is shown in Fig. 13 – 14. Fig. 13 shows the first natural frequency depending on one shape variation, namely the width of the crankcase. Among other optimization results, the optimization finally leads to the shape C (“bottle shape”). The variation of the wall thickness for different parts and the effect on the first natural frequency and the total weight is shown in Fig. 14. In this case the objective function was the total mass of the structure and the functional constraint was specified as a natural frequency of 21 Hz. The design variables are the wall thicknesses and, as shown in the Fig.14, the optimum was achieved with an increased wall thickness of the distance piece.

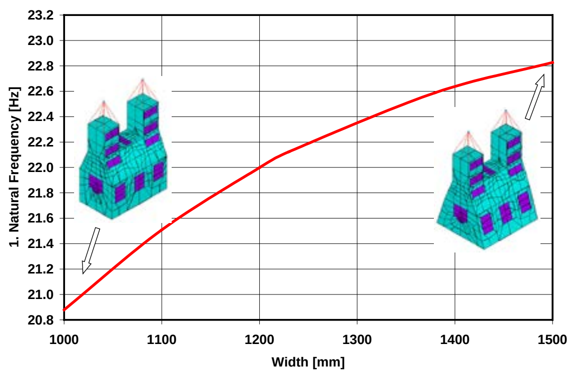
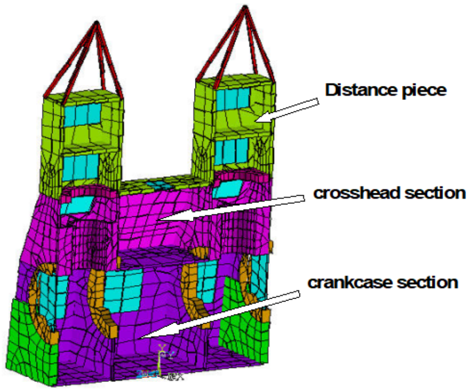


Fig. 13: Change of the first natural frequency with respect to the width of the frame

These fundamental construction principles were taken into account for the realization of the final parts. The optimized wall thickness was discussed with the internal and external foundry specialists and the final optimization took place using the central CAD-model and running the analysis using the full 3D-model.



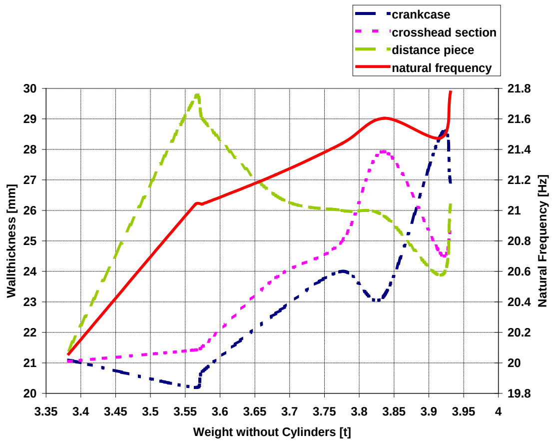


Fig. 14: Optimization of the different wall thicknesses for compressor frame and distance piece

4.4 Static Analysis

Complex calculations are necessary in order to answer the question of the life cycle and fatigue behavior of different components. An in-house developed program based on the FKM guideline[6] is used to evaluate and optimize the fatigue behavior.

First of all, the equivalent loadcases for the minimum and maximum loads are calculated to determine the mean stress and the stress amplitude for the components. Fig. 15 shows one quarter of a single throw, vertical compressor crankcase with the distance piece and the two corresponding loadcases for the upper and lower dead center.

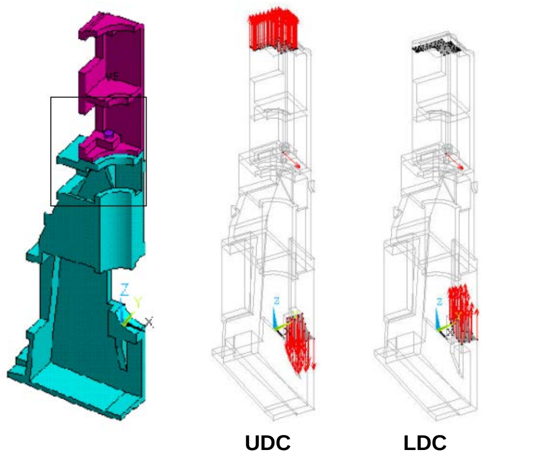


Fig. 15: Quarter model of a single throw compressor and the corresponding loadcases

Safety factors or the load carrying capacity factors respectively, are therefore determined using a reduced Haigh diagram (Fig. 16). Reduction is made on the "amplitude axis" with respect to surface roughness, size of raw material, loaded volume etc. Then the calculated service stress is checked for the position in the "safe area" and the equivalent safety factors are calculated.

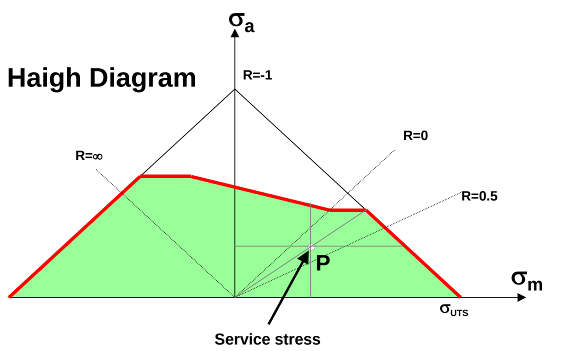


Fig. 16: Haigh diagram

Especially the load transition within the screwed joint areas of the components (frame to distance piece and again distance piece to cylinder) were optimized, improving thus the fatigue behavior. Due to the contact elements between the different parts, all FEA are of the non-linear type.

Fig. 17 shows the static and cyclic load capacity for the screwed joint areas of the frame and the distance piece. Once again a load capacity of 100 % corresponds to the maximal admissible load if appropriate safety factors are built in. The maximum load is considered in the static load carrying capacity, while in the cyclic load carrying capacity the mean stress and the corresponding stress amplitude were taken into account.

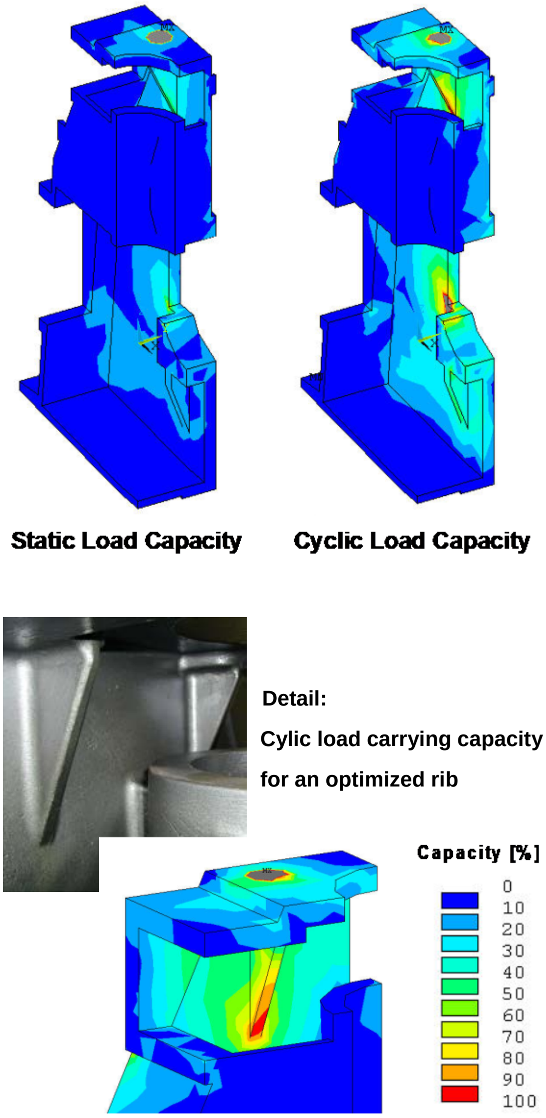


Fig. 17: Static and cyclic load carrying capacity

5 Next Steps

The next important step in the virtual product development cycle will be the CAM linkage. Therefore customer specific machining and machining processes should be integrated in order to capture manufacturing knowledge and to preserve it for repeatable and consistent use. First efforts were made to build the model of one of the specific CNC-machines and the current activities deal with the integration of the appropriate machining tools.

Another benefit of the virtual product development is the fact that the complete information about the structural stiffness of the compressor is available. Determination of the stiffness is done during the optimization procedure or subsequently could be done in a very short time. The structural stiffness could be used in a API compressor manifold study thus increasing the accuracy of the pulsation /

vibration study. Fig. 18 shows a possible combination of the compressor and the surrounding dampers, coolers and piping. Current activities are dealing with the experimental verification of the calculated result.

Finally the pictures and drawings could be used in the sales and customer supports fields either for creating close-to-reality sales documentation or for descriptive customer manuals.

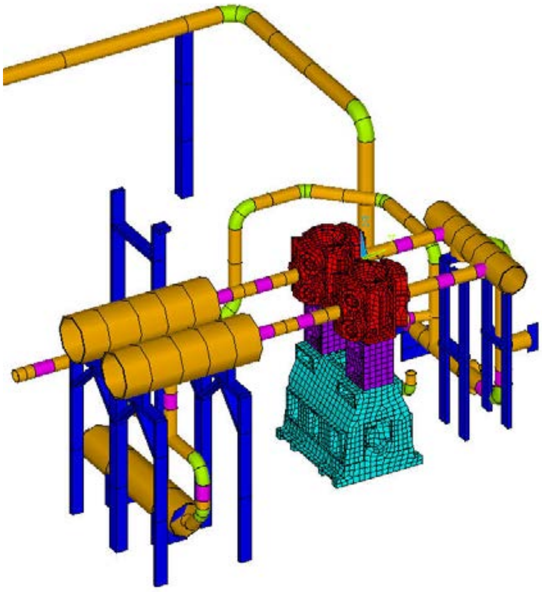


Fig. 18: Configuration for a compressor manifold study

6 Conclusions

The virtual, digital product is a substantial prerequisite for the concurrent or simultaneous engineering. On the basis of a central (CAD-) data model the virtual product development is able to consider all aspects from the concept to the implementation phase.

First experiences with the virtual product development of a piston compressor show how static and dynamic Finite-Element-Analysis, component optimizations as well as casting and manufacturing improvements can be incorporated into an integrated development process. Individual components were optimized so as to allow for a 45% higher specific load at comparable total costs. The dynamic behavior of the entire compressor has been also substantially improved. This was realized by means of optimized, rigid distance pieces and crankcases thus shifting the first natural frequency from 12 Hz to over 20 Hz. Next steps will be the use of the models in a range of further activities, extending from the manufacturing, pulsation and vibration studies up to the documentation production.

The experiences also showed that the virtual product development gives the opportunities and encourages to

- define and keep standards
- preserve specific process know-how
- automate the CAD process
- define and standardize optimization procedures
- increase the product quality
- use rapid prototyping tools
- optimize the product with respect to manufacturing, casting, assembly etc.

Finally, low cost cannot be managed into a product, it must be engineered into a product[7]. For this, the virtual product development supports the necessary design for cost, which seeks to minimize life cycle cost to the customer by designing high quality into the product.

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Minimizing Compressor Life Cycle Costs Through Product Line Design

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Life Cycle Costs – Reciprocating Compressors in the Focus of Function, Economics and Reliability
The Hague, May 17th to 18th, 2001

Abstract:

The requirement to become increasingly competitive has led companies to a better understanding of total life cycle costs of facilities and the equipment used in them. This has resulted in different criteria for selecting equipment, which cause changes in the design of the compressors and the way the manufacturers support the equipment with spare parts, service and technical information after the sale. The application of moderate speed, short stroke compressors can result in lower life cycle costs for many process applications. These units offer reliability equivalent to traditional process compression equipment while providing total life cycle cost reductions through savings in the equipment and spares costs, elimination of the requirement to purchase insurance spare parts for the facility and reductions in the time of unscheduled outages through short spares delivery.

1 Introduction

The requirement to become increasingly competitive has led companies to a better understanding of total life cycle costs of facilities and the equipment used in them. This has resulted in different criteria for selecting equipment, which cause changes in the design of the compressors and the way the manufacturers support the equipment with spare parts, service and technical information after the sale. These changes have been accelerated as company structures move from large organizations with functionally separate departments to smaller organizations with more team-based approaches to business. Whereas the older organization focused on strong functional goals, the newer organization structures support broader project goals based on a wider range of needs and trade-off decisions between conflicting project requirements. For example, in the past a rotating equipment group made equipment selections primarily based on reliability, power and initial capital cost. Maintenance and parts costs came out of a different budget. When custom features were added that made a unit as reliable as possible, capital costs increased and delivery was extended. There were intense purchase negotiations to obtain the best price. Many manufacturers responded by reducing the initial equipment cost but increased the spares pricing to remain profitable.

This example can be contrasted against equipment design and selection process for leased oil and gas production compression in North and South America. Approximately half of all new equipment going into this market is now leased rather than purchased. A contract for a 1 MW class gas engine driven compressor is for an agreed fixed price per month and is typically based on a 97% mechanical availability. If the unit does not meet the availability the price is reduced. If the unit continually or significantly falls below the required availability, it is removed. The leasing company must have a reliable unit and a clear understanding of total life cycle cost to be successful. Parts cost, their delivery and the ability to respond quickly and effectively to operational difficulties are as important as the original capital cost. These companies have an in-depth understanding of the total life cycle cost of the equipment and use it to specify the equipment design and select the manufacturer.

2 Components of life cycle cost

A wide variety of items determine the total life cycle cost of a reciprocating compressor. Monetary expenditures such as the initial capital power costs will have a direct and usually easily quantifiable effect on the total life cycle cost. The time required for start-up, spare part delivery periods for unforeseen maintenance events, the time required to obtain specialized technical information or the delivery period of the compressor package can have both direct and indirect financial impacts.

Direct effects can include loss of revenue for a facility operating at full capacity and additional shipping or additional personnel costs. Indirect costs are associated with the delay in project revenues on the project's overall financial performance. Now most projects are reviewed on a project cash flow basis. Investment decisions for stand-alone facilities are commonly based on a calculation of the internal rate of return (IRR) on the cash investment. This analysis takes into account the amount and timing of investment and revenues to determine the return on the investment. The more quickly a project starts earning a revenue stream the higher the return. Delays or interruptions in revenue lower the return. The ability to respond quickly to a part requirement can have a significant effect on the project's financial return especially if it happens at the start of a project where the funds' impact on the IRR is not discounted by time.

The following lists represent some of the items that must be considered in understanding the life cycle cost of a reciprocating compressor installation:

Direct components

- Project feasibility study and initial engineering
- Compressor procurement
- Engineering cost to interface the compressor package into the facility
- Compressor package capital cost
- Installation
- Power cost
- Utility costs
- Consumable parts longevity and cost
- Maintenance parts usage and cost
- Insurance spare parts purchase and inventory costs
- Operating manpower
- Maintenance manpower
- Unscheduled shutdowns (reliability)
- Salvage value

Indirect components

- On time delivery of technical information required for facility engineering
- Compressor delivery period

- On-time package delivery
- Installation time requirements
- Installation difficulties
- Start-up time requirements
- Start-up problems
- Unscheduled shutdowns due to incorrect equipment application
- Ability to operate over expected operating range
- Availability of technical information to support operations and maintenance
- Operations and maintenance training
- Extended downtime due to parts availability

3 Life cycle component cost comparison

The amount of life cycle cost attributable to individual components varies greatly depending on the project. The largest is almost always due to the direct and indirect effects of unscheduled outages. Compressor valves, pressure packings and process upsets are the primary cause of these outages. The economic effect is largely determined by the time required to develop a technical solution and the time needed to obtain any required parts. The actual value attributable to unscheduled outages varies greatly but must always be minimized through correct compressor design, rapid delivery of components and the ability to correct the problem quickly.

The second highest cost factor is normally the power costs for the installation. For an electric motor driven reciprocating compressor this will vary greatly because of the difference in energy cost for the project. Typical costs for a 950 kW unit over a 20 year project can total from about $US 5 to 12 MM based on energy cost of $US 0.03 to 0.07 per kW-hr.

The costs in Figure 1 are representative of the total cost of a 950 kW API-618 reciprocating compressor over a 20 year period. The actual cost will vary substantially with the project requirements and the method of accounting for manpower and overhead charges.

The power cost over the life of a project is nearly identical for all reciprocating compressors designed specifically for the application and configured with similar capacity control devices. Product design decisions can play an important role in decreasing other project costs through reduction in the compressor package and parts cost, decreases in compressor and parts delivery and making technical information on the unit available quickly.

Indirect project costs vary greatly in the value assigned to them. They can be small if the facilities can makeup the lost revenue quickly, or so large they can cost more than the equipment in a few days. Manufacturers can minimize indirect costs by providing a reliable design properly configured for the application with the shortest possible delivery times for parts.

Total System Costs Over 20 Year Project Life 950 kW API-618 Reciprocating Compressor Electric Motor Driver (Values in $US)	
Power cost ($US 0.03 to 0.07 / kW-hr)	$5,000,000 to $12,000,000
Compressor package price	$950,000
Operating and Maintenance	$800,000
Installation	$750,000
Insurance spare parts and inventory costs (@15%/yr)	$600,000
Procurement and engineering contractor services	150,000

Figure 1

4 Process compressor product line design decisions

4.1 Product line design objectives

Objectives were developed to guide Ariel's efforts in developing the design and support requirements for the process market. These objectives were directed toward minimizing the total life cycle costs of a compressor installation. One of the first decisions required was whether to follow the commonly accepted machine design standards in the industry or proceed with different designs that would decrease the total life cycle costs but that the user could be hesitant to accept. Interestingly the same decision was faced a number of years ago in the oil and gas production and gas storage markets. The major design changes were the use of short-stroke compressors, elimination of the cylinder water jacket and elimination of cylinder liners. The following objectives can be used to guide product design decisions:

- Reliable, conservatively applied basic compressor designs
- Develop specialized component options required for selected commonly seen applications
- Pursue applications where pre-design components and application data can be used repetitively
- Develop the ability to engineer and package the compressors in accordance with the needs of the customer and requirements of the installation location
- Integrate the design, application, supply and service of the valve, pressure packing and non-metallic manufactures into the product line.
- Provide the customer a compressor package at a lower price than available with traditional designs
- Provide spare parts at prices equal to or lower than available through any other source
- Ship the majority of spare parts within 48 hours and new cylinders within 1 week as can be achieved for production, storage and transmission markets
- Make technical information easily available
- Train the users in the operation and maintenance of the compressor
- Develop long-term relationship with the users

4.2 Compressor Design

Process compressors typically operate at piston speeds of 3.8 to 4.8 m/s for lubricated and no more than 3.8 m/s for non-lubricated applications. Short-stroke compressor designs can be operated at these speeds and offer savings due to their inherently smaller size. These frames and the pre-designed components on which they are based have been in use for many years thereby verifying design and component reliability. Valve, pressure packing and non-metallic material manufacturer experience on process applications is directly applicable at these speeds. The use of components used for other applications results in lower costs and raw material available from stock. Production manufacturing techniques and pre-designed components, in conjunction with material availability provides the ability to meet the spare parts delivery goal. Design features that support life cycle cost reduction include:

- Forged steel crankshafts and connecting rods
- Well-defined quality control procedures that have a history of high first pass yields
- Driver speeds allowing the use of induction motors that are substantially less costly than a slower speed synchronous alternative
- Pistons long enough to achieve 0.035 N/mm2 wear band loading on non-lubricated designs
- Cylinders with standard provisions for measurement of compressor performance or installation of on-line performance measurement and predictive maintenance systems
- Existing designs for the majority of options required for process applications such as long two compartment distance pieces, purged water-cooled packings and NACE design
- Detailed published technical information on the various pre-designed components

4.3 Component options

The majority of design options required for the process market exist for short stroke designs. Additional component options for process valve designs, pressure packing capable of supporting a variety of non-metallic materials and alternative non-metallic material for wear bands and piston rings are easily created.

The key to a good design remains communication of the application requirements, component design, trade-off decisions and the project objectives. The detailed design knowledge of the component manufacturer must be integrated into overall compressor design and component application criteria. The objective of using common components and material to decrease costs and shorten delivery is no different than for the other compressor components. Where high component volume is not available the stocking of the raw materials used to make the items is needed to achieve a quick parts delivery.

5 Design and application guidelines

The capability to properly configure a compressor for a specific application is critical to reliable operation and minimizing costs. Proper application also minimizes the indirect financial effects of extended start-up periods or reductions in the plant throughput during the initial operating period. Without this knowledge base the availability of the correct component options is worthless. Availability of this information is even more important as companies downsize or outsource their engineering function. The decrease in employee numbers in these departments makes training new people more difficult. Additionally company mergers and closures and the retirement of many experienced people has resulted in difficulty finding some written standards and the loss of much undocumented experience.

A product line based on a group of pre-designed frames, cylinders and options makes capturing and documenting the technical aspects of the design relatively straight-forward though still a time consuming task. The objective is to maintain detailed design and application information in a verified and controlled document similar to how component designs are verified and controlled. Making information available electronically would aid both the manufacturer and customer knowing the product and how to correctly apply it.

6 Compressor packaging

The design and components used in packaging a compressor for process applications are unique to each project. Vessels and piping are designed specifically for the flow, pulsation control and instrumentation requirements, and they must be certified in accordance with the local code requirements. The package structure, equipment layout, auxiliary equipment and instrumentation vary with each project. In addition to the components supplied, the engineering standards to be met and certification requirements are also unique.

Goals for a successfully operating project are the same as the purchaser's: on-time delivery, successful start-up and reliable operation of the package. The process by which packagers are selected, approved and audited includes:

- Define the requirements for a company that engineers and fabricates packages for the process industry
- Identifying various packagers experienced in regional requirements
- Audit the firm's capabilities to meet these requirements
- Investigate the company's past projects and reputation in the industry
- Audit initial package designs to be in accordance with Ariel's packaging standards
- Review the fabrication of the initial units
- Review the performance of the units after they are placed in operation in the field
- Continually audit package engineering and fabrication through visits to the company, visits to field installations and feedback from the customer

The use of specialty packagers provides a unique ability to supply timely engineering and package components appropriate for the customer and project destination. Additionally, since the packagers are usually local to the customer communication is improved and total shipping charges are lower.

7 Spare parts

The use of pre-design components with high usage rates in multiple applications yields a number of cost savings for the user of the equipment. Larger production volumes allow raw materials to be ordered based on historical usage rates. Additionally, committed foundry space can be reallocated to obtain raw materials quickly. Both of these capabilities support the rapid delivery of spare parts.

Short deliveries provide the customer the option to order items other than typical consumable parts. In our experience few customers order and stock insurance spare parts when they can be obtained within acceptable periods. This provides a significant reduction in parts and inventory costs.

8 Service

Designing components that are easy to replace in the field can reduce Service costs. One way to do this is to design a single component rather than an assembly that is composed of a number of smaller parts. This makes the component installation less time consuming and reduces the risk of the assembly being built incorrectly.

An example of this concept is providing a single crosshead assembly rather than as a built-up assembly. A crosshead for a long stroke unit needs to be adjustable at site because the large units could not be machined and reassembled to tolerances that insured the crosshead and distance piece was in the proper position. The assemblies were designed to allow the crosshead shoes to be shimmed to the correct crosshead position. Crosshead shoe fasteners were then lock-wired to prevent damage if the fastener became loose. Crosshead shoe replacement was desirable because the reliability of the lubrication was not as good as it is today and it was less expensive to replace the shoes than a complete assembly. Manufacturing tolerances on short-stroke compressors are capable of maintaining the proper position of the crosshead. Additionally lubrication systems and lubricants are much better than those available when many of the long-stroke design concepts were developed. Today few compressors lose crosshead assemblies in operation. It is nearly as cost effective to provide a single piece crosshead as to replace shoes.

Another important design requirement is to insure that once various components are assembled onto the compressor frame there is maintenance access. The availability of computer aided design has made this relatively easy to check but accessibility must be part of the basic design because there may not be methods to provide access once the designs are finished.

Quickly accessible technical information is important in maintaining a unit. The use of a limited number of pre-designed compressor components simplifies the development and maintenance of technical information for the complete compressor. Making the information available electronically facilitates easy access and availability of the data. Ariel created a call center to answer questions on existing units. The call center has access to all unit records as well as spare parts availability, drawings that can be transmitted by fax or email and a wide variety of other capabilities such as checking valve springing for alternate operating conditions. The availability of this type of information can significantly reduce service costs and compressor outages.

Providing cylinders with provisions for the field installation of a complete on-line performance monitoring package can limit downtime by enabling the use of a portable monitoring systems to quickly diagnosing performance and pulsation problems. These systems can also perform on-line torsional analysis of the system.

9 Conclusion

The application of moderate speed, short stroke compressors can result in lower life cycle costs for many process applications. These units offer reliability equivalent to traditional equipment while making available reductions in initial package and parts cost. Additionally, short parts delivery reduces the cost of unscheduled shutdowns by getting the unit back in service quickly.

The amount of savings varies depending on the project. Savings can generally be expected due to the following:

Compressor package price

- Short stroke compressors are normally less costly because they are inherently smaller for a given requirement and are built with pre-designed components
- The ability to utilize an induction motor can reduce driver cost

Installation and start-up

- Smaller package size reduces installation costs
- Compressor packages as large as 4 MW can be delivered and installed as a single unit reducing on-site assembly
- Short stroke compressors are normally mechanically run at the manufacturer thereby reducing the risk of mechanical delays during start-up

Operations and maintenance

- Pre-designed components used in both oil and gas production and process applications offer cost savings due to their volume of manufacture
- Short part deliveries minimize the length of unexpected shutdowns
- Smaller component size and built-in assembly tolerances reduce the time and complexity of maintenance

Insurance spare inventory

- Short delivery of insurance spares eliminates the need to purchase these components and the inventory cost to hold them
- Obtaining insurance spares at the time of need insures the parts are in new condition

Short stroke compressor alternatives are available for the majority, but not all, applications. Component options must also be available to configure the compressor correctly for the intended use. Applications where short stroke compressors are available include, but are not limited to, both lubricated and non-lubricated offerings for:

- Hydrogen and hydrogen mixtures
- Inert gas
- Methane
- Ethylene
- Hydrocarbon mixtures
- Natural gas
- Acid gas services
- Refrigeration
- CO2 / Syn-gas

Where short stroke machinery is available reductions in total life cycle cost can be achieved through savings in the equipment and spares costs, elimination of the requirement to purchase insurance spare parts for the facility and reductions in the time of unscheduled outages due to short spares delivery.

Evolving Technology to Monitor Dynamic Bending Strain on a Crankshaft

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability

May 17th / 18th, 2001, The Hague

Abstract:

This paper presents a technology to record dynamic strain using a strain gage and data acquisition system mounted on the crankshaft. An RF signal triggers data acquisition at a fixed crank angle. With on-board power and memory to run remotely for several days, data is periodically downloaded for assessment.

The technology has potential for condition monitoring and for machine development testing.

Under PRC *I*nternational (PRC*I*) and Gas Machinery Research Council (GMRC) support, the technology has been tested on two compressors. The paper presents the concept, development experience, results, and relates the measured data to model predictions.

1 Introduction

In the United States, the gas transmission industry relies heavily on integral reciprocating engine compressors – many installed 30 to 50 years ago or more. Crankshaft integrity concerns all operating companies and a common cause of crankshaft bending failure is thought to be misalignment, aggravated by deterioration of the foundation. To help protect against such failures, most operating companies take annual crank web deflection measurements, and schedule regrouting when web deflections become excessive.

Unfortunately, this web deflection measurement is less than satisfactory. It is difficult and uncomfortable to make; it does not reflect operating conditions of load or temperature; its accuracy is questionable because data cannot be acquired over a full revolution; and it normally requires judgement during the measurement and recording process - particularly as to the angle at which each data point is taken. It is an infrequent, periodic measurement, and clearly does not provide continuous condition assessment or have any chance of catching a sudden deterioration. Moreover, the measurement of static web deflection, while the unit is barred over, does not directly measure the stress state of the shaft. Static deflection measurements reflect the orientation of the shaft sitting <u>on the bearing</u> and do not account for frame deflections that are both load and speed dependent. The condition induced by the static misalignment does not reflect the gas loads from the engine and compressor, nor any possible resonance effects which can amplify all sources of excitation.

The need exists for a tool with which to better assess dynamically varying strain on crankshafts, and for knowledge of how misalignment-induced stresses combine with load-induced stresses to produce the combined operating crank stress field. This paper, first, presents a device labeled the "Strain Data Capture Module (SDCM)," which should help meet these needs. The next several sections of this paper discuss the features of the SDCM, the preliminary data acquired using this unit under GMRC and PRC*I* funding, and the results of a preliminary analysis to be combined with the measured data which provides insight into the stress field of the running crankshaft.

In recognition of its widespread use, the GMRC has also supported research to improve the practice of web deflection measurement, and to enhance the interpretation of this periodic conditon monitoring technique. The paper summarizes this complementary work, which has produced a widely distributed software package called "WEBMAP™," and is developing an alternative to the widely used Caldwell criterion for assessing web deflection severity.

2 Strain Data Capture Module Description

The Strain Data Capture Module (SDCM) is a microprocessor controlled device designed to acquire and store dynamic strain data in high G loading operating conditions. Currently, the software is configured to acquire the data synchronously with the shaft rotation by subdividing the crank rotation into 32 equal angular increments and acquiring data at each increment. The SDCM acquires the shaft position dependent data and has the potential to average each point over 16 crank revolutions, or to acquire data for individual cycles. The data is stored in nonvolatile memory for subsequent download. Each time wave record is time stamped and combined with the unit rpm. For multiple crank web measurements, the devices are triggered simultaneously using a RF link from transmitting antennae mounted in the crankcase. In this fashion, each unit acquires the data simultaneously, phased to a known crank position. Currently, the device is software configured to acquire data at five-minute intervals allowing for up to five days of testing. Since foundation thermal misalignment has a time constant of several days and reflects a distinct day/night cycle, the module has been configured to investigate and document the web strain over an extended period of time. Note that the device is easily reconfigurable to allow more frequent acquisitions and different data acquisition strategies. The unit is battery driven and utilizes a power management system for long term testing, which incorporates a sleep mode of programmable duration. The device shown in Figure 1 is completely enclosed in an aluminum housing with no external active components, except for its connections to the strain gage.

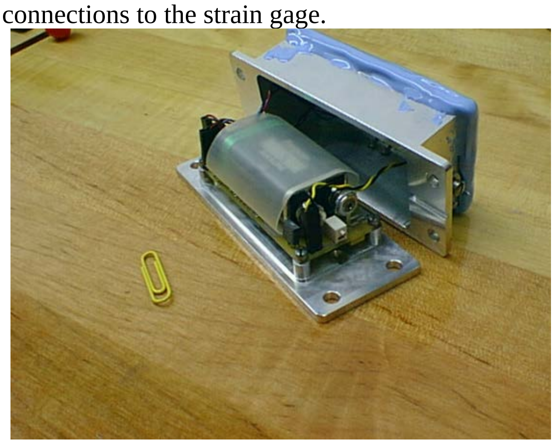

Figure 1: Strain Data Capture Module

3 Bench Testing of the Strain Data Capture Module

Several iterations of the SDCM were developed, and its current form has evolved through incremental improvements in reliability, signal integrity, and mounting/G loading capability. Bench testing on an inertial test rig was completed to document improvements and final design capabilities. The bench test rig used a cantilever mounted variable speed motor with variable imbalance load. Figure 2 presents the peak-to-peak strain data extracted from waveforms acquired and stored during a speed sweep test. The data has been plotted *versus* the square of the speed, and indicates a linear relationship as expected. The waveform data was normalized with the square of the speed, and is plotted as a function of crank angle in Figure 3. As expected, a sine wave (inertial excitation response) is seen. The scatter reflects the noise floor in the device arising from both digital bit and electrical noise. The noise floor for the SDCM is estimated to be 10 microstrain (peak-to-peak).

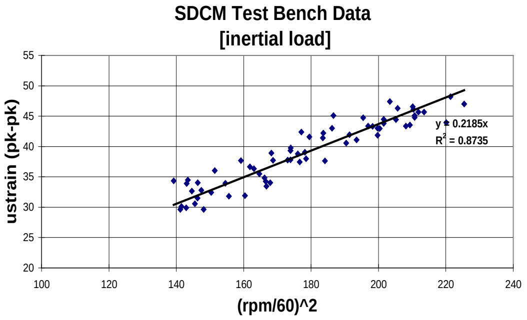

Figure 2: Bench Test Inertial Load Data

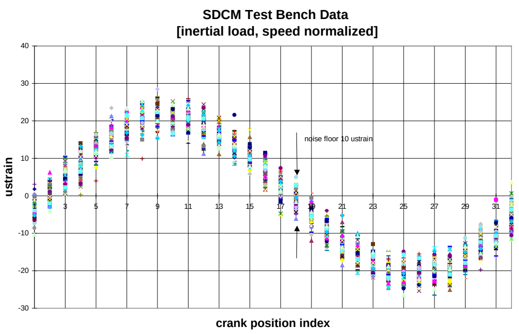

Figure 3: Speed^2 Normalized Waveform Data

4 Installation and Operation on a Small High Speed Reciprocating Compressor

Figure 4 shows the SDCM installed on one web of the single throw Ariel compressor in the GMRC Reciprocating Compressor Test Facility. The SDCM is mounted with four No. 10 machine screws. Note that the mounting location is a low stress area on the end of a web. During initial testing, the compressor was run up to 1200 rpm, giving rise to peak G loads significantly higher than those to be expected in a 300 rpm compressor (even accounting for the larger physical size). The compressor was configured with the crank end valve removed, effectively providing a single acting unit. During a series of constant speed tests (600 rpm), the unit was progressively loaded by increasing system ratio. Figure 5 shows the variation of strain with crank angle at these different loads. Strain variation, not unlike the variation typical of cylinder pressure, is seen in this data. The peak-to-peak amplitude increases with load, as emphasized more strongly in Figure 6, where peak-to-peak strain is plotted as a function of load number. These self-consistent results provide confidence in the SDCM's sensitivity to crankshaft strain variation, and the effectiveness of its response to change in load on the rotating crankshaft. All the data shown in Figures 5 and 6 was acquired with the SDCM operating remotely mounted on the crankshaft web. Sixteen revolutions of data were averaged to produce the strain data with each set acquired at the end of a five-minute sleep cycle. The data acquisition was triggered (timed) by a single RF transmitter with the antenna in the top crankcase, firing at top dead center of the compressor cylinder.

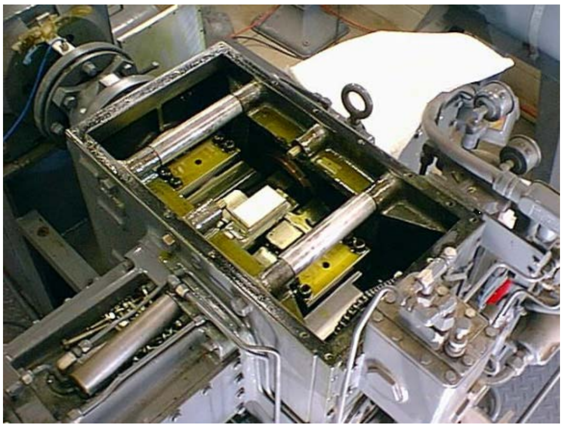

Figure 4: SDCM Installed on RCTF Ariel

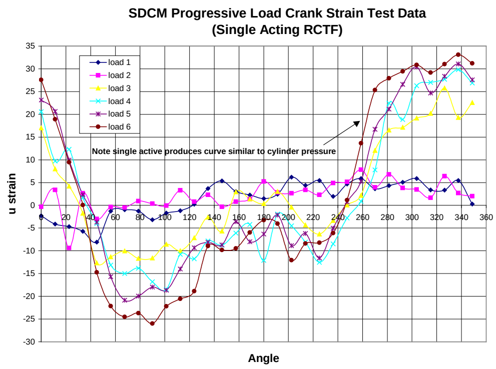

Figure 5: Waveform Data from RCTF Increasing Load Tests

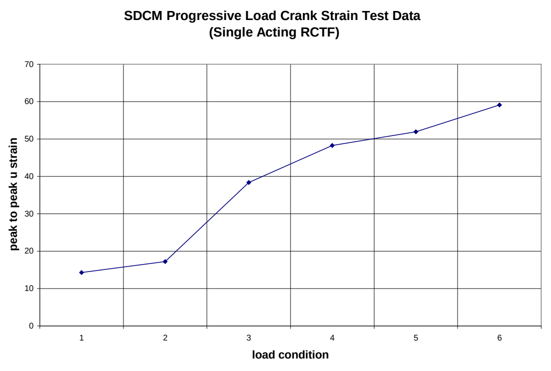

Figure 6: Peak-to-Peak Strain Data from RCTF Tests

5 First Field Test of Strain Data Capture Module

With encouraging results from facility tests, six modules were built and on August 16 through 18, 2000, a test was performed on an HBA6 at Tennessee Gas Pipeline's Kinder Compressor Station. One module was mounted per crank throw, some on the web nearest the oil pump, and some on the web nearest the flywheel end. Strain gages were located as close to the crank pin as possible, but because of physical constraints, were typically separated from it by one-half to one-inch. Each strain gage location was measured from the outer edge of the web, so that they could be referred back to a finite element model of the crank throw. Figure 7 shows one of the SDCMs installed on the crankshaft. To expedite the installation, a drill template was prepared and glued to each web allowing drilling and tapping without the chance of damaging the SDCM. Figure 8 shows the backside of the HBA6 with the covers closed after installation of the modules. Close inspection of this photograph shows leads to the RF antennae - positioned at four locations in the crankcase.

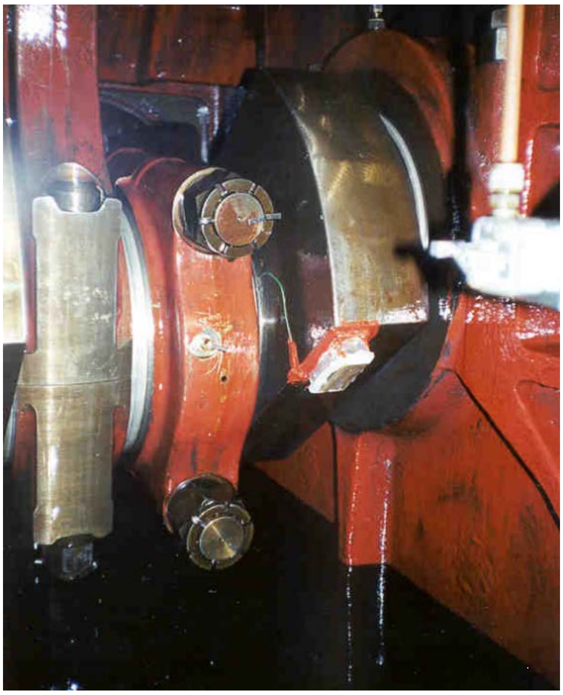

Figure 7: SDCM Installed on HBA6 Web

Figure 8: HBA6

Prior to closing the covers, a set of web deflection readings was taken on each crank throw as the unit was barred over. This was immediately followed by acquiring a set of static strain measurements on throw Nos. 1 and 3 at 90-degree intervals as the unit was again barred over. Both web deflection and strain data are key to strain data calibration and estimate of strain gage location effects, as discussed in the next section. Table 1 presents the summary of measured web deflections and static strain values.

The static web deflection data can be interpreted through the WEBMAP™ software to provide a deflected shape of the crankshaft in the vertical and horizontal directions. As shown in Figure 9, based on this analysis, the crankshaft is relatively well aligned. The highest web deflection value was two-thousandths of an inch on throw No. 3, compared to a Caldwell web deflection criterion of 3.1 thousandths of an inch for this compressor. The web map deflected shape shows a maximum shaft deflection of about five-thousandths of an inch.

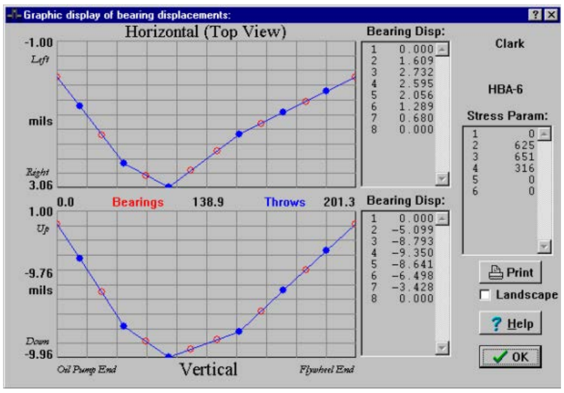

Figure 9: WEBMAP ™ Crank Deflection Shape

After closing the covers, the HBA6 unit was started and ran for three hours as an initial functional test period. The modules were inspected and data was downloaded from several of the modules. Data suggested triggering problems on some of the modules, and as an attempted short-term remedy, one antenna was moved down closer to the crank throw of interest. The covers were closed and the unit ran for about 15 hours overnight from 4 p.m. to 7 a.m. on August 17-18, 2000. After opening the unit, peak temperatures on the surface of the web were measured at 150º F.

The module on throw No. 1 performed perfectly, as intended. It acquired 12 records per hour for 15 hours. Each of those records represents the average of 16 revolutions. Figure 10 presents the variation of strain with crank angle obtained by averaging multiple records from throw No. 1. Note that throw No. 1 does not have a compressor cylinder attached. The zero crank angle position represents top dead center for power cylinder No. 1. As expected for the crank pin vertically upwards, the downward load from the power cylinder produces a positive or tensile strain on the inside surface of the crank web. As the crank throw rotates, the bending strain seen by the strain gage reduces, turns negative, and near bottom dead center is close to zero. It starts to increase again as the power cylinder compresses a mixture of fuel gas and air prior to top dead center. The strain varies from about 17 microstrains maximum to -10 microstrains minimum – a total of 27 microstrains peak-to-peak. Note that these numbers are at the gage location, not the peak numbers in the web. Estimation of peak number from gage location numbers is established in the next section.

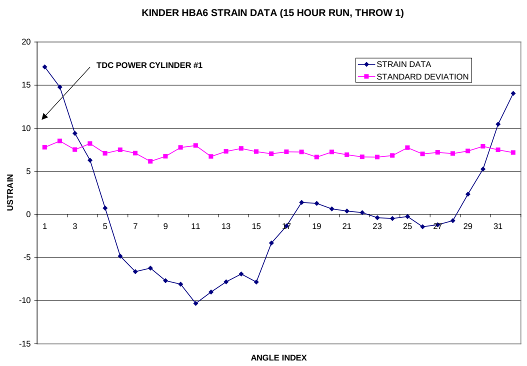

Figure 10: HBA6 Measured Strain (Throw 1)

Four other strain data capture modules produced a number of good records interspersed with records which appear to have been mistriggered in some way. By editing these records, it was possible to extract useful data, and Figure 11 shows the variation in strain obtained at throw No. 3. This throw has both a power cylinder and a compression cylinder. It reflects a different variation of strain with crank angle and the peak-to-peak strain variation is higher. Superimposed on this graph is the standard deviation for every point in the record. Note that the sharpest variation in strain occurs between Top Dead Center (TDC) for the power cylinder and TDC for the attached compressor cylinder.

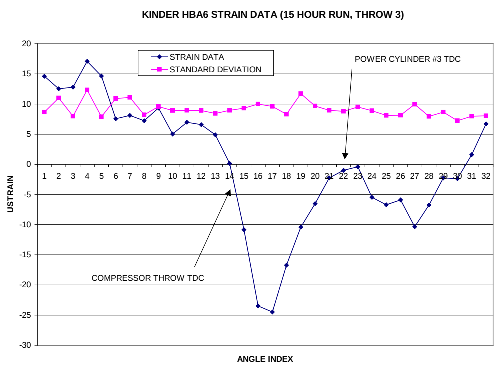

Figure 11: HBA6 Measured Strain (Throw 3)

For the six strain data capture modules installed, one provided 100 percent successful data, four provided partial data, which could still be used, and one produced no data. Triggering remains the primary problem, still to be addressed. The wavelength of the transmitted TDC pulse is less than half a meter, which makes it susceptible to reflections from the metal structure. Reflections can produce null points and regions about these null points where the signal is low. Thus, the most obvious solution to the problem is to increase the power of the RF signal. To increase RF power from the current level of 100 mW is straightforward. Additional tests with different power levels, frequencies, protocols, and data post-processing have helped to identify a more consistent and robust approach to the acquisition of triggered data, which should increase the success rate of the method in the future.

6 Influence of Strain Gage Position and Estimation of True Peak Stresses

The strain on the inside surface of the crank web near to the crank pin varies rapidly in the region of the crank pin reaching a maximum value right at the radius between crank web and crank pin. Using the measured web deflection numbers of Table 1, combined with the static strain values, gives the ratio of strain per mil of web deflections and a mechanism to calibrate peak stress to the measured strain values. Figure 12 shows a finite element model of one of the HBA6 crank throws comprising a crank pin, two crank webs, and part of the adjacent main journals. The faces of the crank webs have been pushed apart by a total of 0.5 thousandths of an inch at the centerline of the main journals. This corresponds to the measured web deflection of 2 mils at throw No. 3. Using a nominal Young's modulus of 30 x 10^6 psi and the static values recorded in Table 1, the stress at the strain gage location for throw No. 3 is predicted to be 690 psi. The finite element model estimates the stress to be 585 psi. The gage averages over a finite area and the model is reporting the stress slightly inside the surface. Given these two considerations, the agreement is quite good. A sub-model was used to refine the finite element mesh and establish the peak stress levels in the root/filet of the web. Figure 13 presents these results. For the web geometry of the HBA6, the ratio of peak stress in the filet to stress measured at the strain gage location is 8.75. This is the multiplier that has to be applied to the strain gage measured stress. The peak-to-peak strain measured from throw No. 3 (Figure 12) is 42 microstrain. Using a Young's modulus of 30 x 10^6 psi and the stress multiplier of 8.75, results in a maximum peak-to-peak stress level estimate of 11,025 psi. Note that this particular unit has no history of crank failures and this stress estimate then represents our first reference point in an acceptable unit.

Table 1: Web Deflection and Static Strain Data

Recorded Web Deflections (mils)

		OPE	Throw				
Reading	**Estimated Angle**	1	2	3	4	5	6
1	140	0	0	0	0	0	0
2	180	0	-0.00025	0.00025	0.00025	-0.00025	0
3	22	0.00025	-0.00025	0.00025	0	-0.00025	0
4	270	0.00025	-0.00025	0.0015	0	0	0
5	315	0.0005	0	0.002	0	0	0
6	345	0.0005	0	0.002	0	0	0.00025
	peak to peak	**0.0005**	**0.00025**	**0.002**	**0.00025**	**0.00025**	**0.00025**

Recorded Static Strain (ustrain)

Estimated Angle*	OPE	**Throw**
	1	3
0	19	7
90	5	-16.5
180	5.5	-13
270	9	0
peak to peak	**18**	**23.5**

* 0 at power cylinder #1 TDC

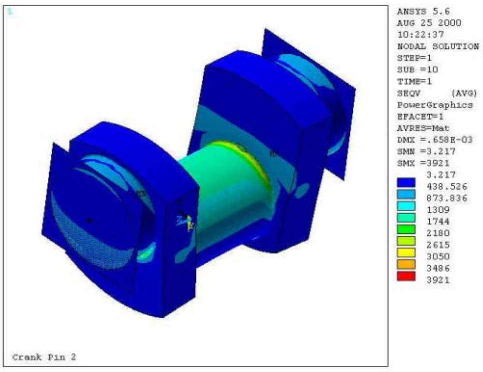

Figure 12: Course Grid Static Stress Field with Applied Web Deflections

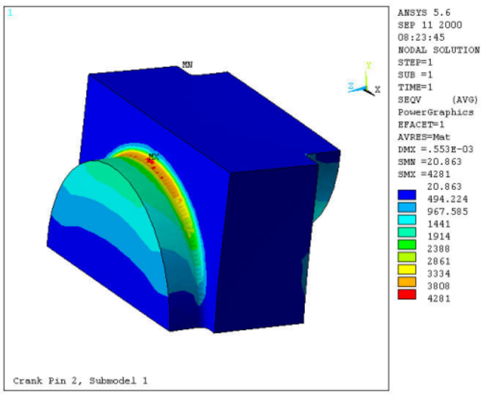

Figure 13: Refined Sub-Model Static Stress Field with Applied Web Deflections

7 Gas Load Stress Estimation

Dynamic cylinder pressure was acquired on the unit during the SDCM test program. Figure 14 presents the power cylinder and compressor head end and crank end pressure data. This data can be used to estimate the pin loads both for webs with attached compressor cylinders and for those without. Figures 15 and 16 present the estimated gas driven pin loads. Using the finite element model of the web, we can estimate the stress at the strain gage location for varying crank angles. Figure 17 presents the finite element strain predicted at the gage location at five angular positions, assuming the power cylinder gas loads of Figure 14. The peak-to-peak values predicted at the gage location are approximately 12.5 microstrain. As expected, this is less than the measured value of 27 microstrain peak-to-peak. Note that the sum of the peak-to-peak misalignment strain (Table 1) of 18 plus the 12.5 microstrain predicted by the gas loads is close to the measured value of 27 microstrain peak-to-peak.

Any operator of slow speed two-stroke engines is aware that considerable variation occurs from cycle-to-cycle with periodic extremes of misfire and detonation. Thus, a planned next step in the data acquisition and analysis process is to assess the influence on stress of cycle-to-cycle variation in the power cylinder loads, superimposed on the average values addressed here.

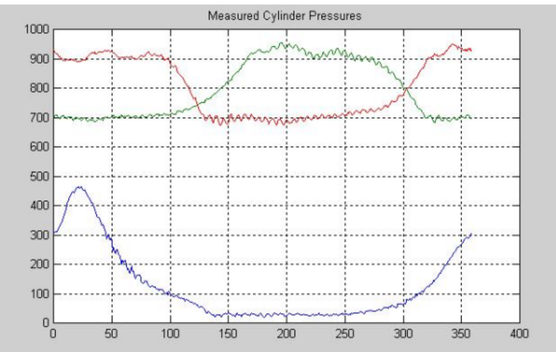

Figure 14: Measured HBA6 Cylinder Pressures

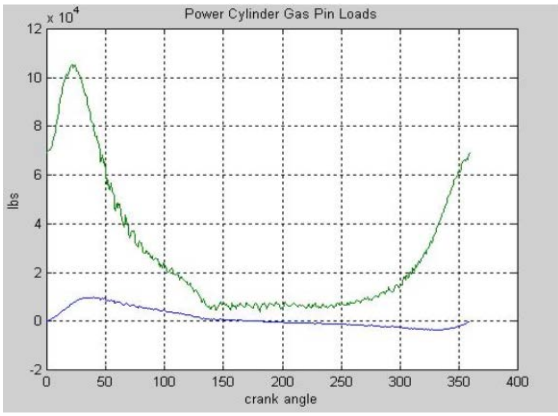

Figure 15: Power Cylinder Pin Loads

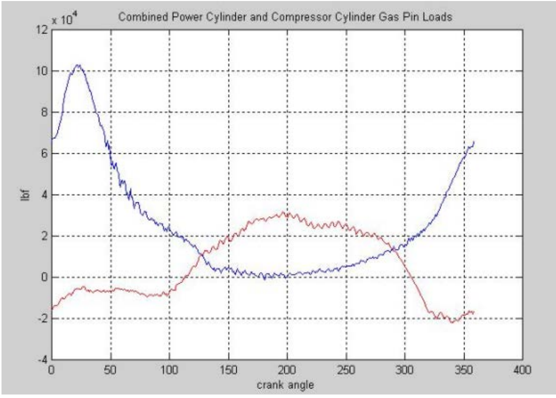

Figure 16: Combined Power and Compressor Cylinder Gas Pin Loads

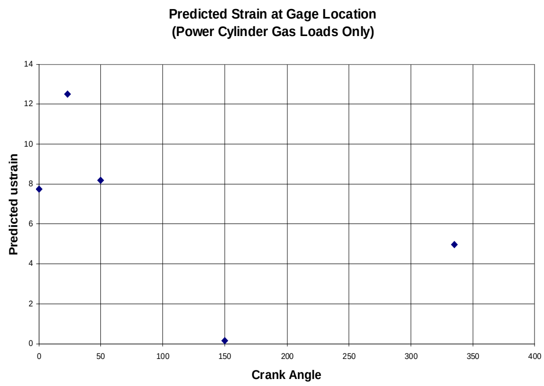

Figure 17: Predicted Strain using Power Cylinder Gas Loads at Pin 1

8 Enhancing the Use of Web Deflection Measurement

The paper, so far, has focused on a method to measure crank strain during operation. In recognition that the majority of operators of large integral compressors will continue to use traditional methods for some time to come, the GMRC has also developed technologies which will enhance the acquisition and interpretation of traditional web deflection data. The most widely deployed of these technologies is a software package entitled WEBMAP™[1]. This software combines the web deflection measurements of all throws on the shaft, and integrates the implied shaft curvature over the shaft length. This yields the deflected shape of the shaft in its bearings and the misalignment of the bearings relative to each other, as Figure 9 has already illustrated. The added insight, given by observing the deflected shape, enables operators to better assess the state of health of a compressor installation, but still leaves open a criterion which improves on the Caldwell rule of thumb (0.182 mils per inch of stroke). Most recently, Harrell[2] has shown for the majority of engine compressors in use in the U.S. pipeline system that a single relationship can relate stress to web deflection and a few geometric factors:

$$S_{mas} = C * \frac{Pr^m}{Th^n} * Wd^p \qquad (1)$$

where:

C	=	a constant.
Pr	=	the pin radius,
Th	=	the throw distance.
Wd	=	the web deflection.

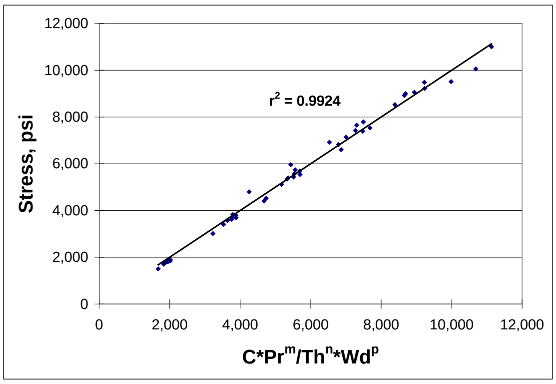

Figure 18: Crankshaft Stress Prediction Relationship

Figure 18 confirms the consistency of this relationship for about 11 different compressor models. Harrell goes on to build on this relationship a method for assessing the ability of a shaft with misalignment to survive for an acceptable life, by comparing the stress inferred from the web deflection against a design stress based on the S-N curve for the shaft, accounting for stress concentration in the fillet radii of the main journal, and a dynamic stress multiplication factor guided by SDCM results.

Figure 19 shows a device developed by Indikon – an indicator with electronic readout for both angle and deflection. Installed across the webs, this device will provide a stream of web deflection values tied directly to the crank angle while the unit is barred over. WEBMAP™ can now read this stream of values directly, which improves data quality and consistency while making the process of web deflection measurement less unpleasant.

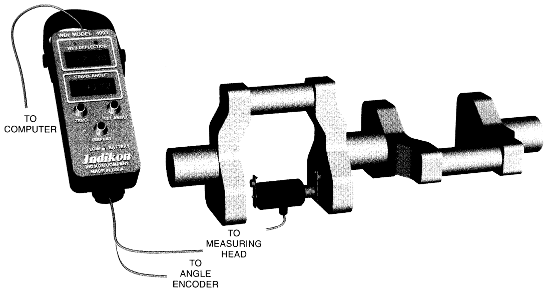

Figure 19: Electronic Dial Indicator

9 Summary

The SDCM has successfully acquired data from both a small high-speed unit and an industry representative slow-speed compressor. The data acquired from the HBA6 is consistent with the levels predicted from the combination of applied static misalignment low and gas loads. Peak-to-peak strain levels of 42 microstrain were recorded. Using a finite element model calibrated to measure slow roll data, the peak-to-peak stresses in the root of the web are estimated to be 11,000 psi. This particular unit has not experienced any crank related problems and these levels represent our first effort at setting acceptable levels for screening. Future efforts will focus on refining the SDCM and improving the triggering capability, as well as using the data acquired by the SDCM to refine predictive models. The role the SDCM will play in condition monitoring remains to be fully defined; advanced versions with two-way digital RF transmission have been bench-tested and have considerable promise. The developments in web deflection measurement and interpretation, summarized in the paper, also provide some near-term benefits to users of this widely deployed condition monitoring process.

10 References

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[2] Harrell, J. P. and Wattis, G. N., "An Improved Criteria for Assessing Crankshaft Misalignment," *Proceedings*, GMRC Gas Machinery Conference, Colorado Springs, Colorado, October 2-4, 2000.

[3] Harrell, J. P., "WEBMAP™ User's Manual, www.gmrc.org.

Evaluation of Low Frequency Pulsation Damping Devices

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Flow and Structural Dynamics (PULSIM)

TNO TPD Delft, The Netherlands

Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability May 17th / 18th, 2001, The Hague

Abstract:

Reciprocating compressors are strong sources of pulsating flow in process installations. Since the typical frequencies related to these pulsation sources are related to compressor speed and thus in the order of 3 to 10 Hz and higher harmonics thereof, the pressure pulsations are only slightly damped by viscothermal effects. Therefore they can be a cause of strong pipe vibrations even at large distances from the compressor. In order to dampen the pressure pulsations, various pulsation damping devices can be applied in the installation. First of all, pulsation bottles are installed at the compressor suction and discharge side, to dampen the high frequency components of the pulsating flow generated by the compressor. Furthermore orifice plates are often installed at the compressor nozzles and line connections of the pulsation bottle. In order to dampen the resulting pressure pulsations in the installation, additional pulsation dampers can be installed such as orifice plates, perforated plates, side branch resonators, damper internals, secondary pulsation bottles or acoustical filters.

In a joint research project of the EFRC a number of damping devices has been evaluated, both experimentally and analytically. Both the effective damping and pressure drop have been determined as a function of frequency and flow rate. A single bore orifice plate, a perforated plate and a perforate plate with venturi-shaped holes with an equivalent effective area were compared. The experiments were carried out in the air test facility of TNO TPD in Delft at atmospheric pressure under well-defined resonance conditions. 4” sized damper plates with effective open area of 40% and 50% were evaluated in a frequency range between 10 and 100 Hz, and at a mean flow velocity of 2.5 - 25 m/s in 4 inch piping. The experimental results are be presented and the effectiveness of the various damper designs has been compared with a theoretical model for the frequency range and flow rates specified.

1 Introduction

A pulsating flow in a compressor installation can result in strong dynamic forces, which act on bends, reducers, valves and closed branches. In particular reciprocating compressors are known sources of low frequency pulsating flows. The pulsation bottles at the suction and discharge of the compressor can only effectively eliminate the pulsations above a typical frequency, such that the low frequency pulsations, typically a number of higher harmonics of the compressor speed, remain in the pipe system. Typical for the low frequency pulsations is that the damping by viscous or thermal effects is very small, and the pulsations travel over larger distances than higher frequencies. As a result, pulsations caused by the compressor can induce vibrations far away from the compressor. When the frequency of the dynamic forces caused by the pulsating flow coincides with mechanical resonance frequencies of the pipe system, unacceptable piping vibrations and dynamic stress levels may occur, which finally can lead to fatigue failure of the installation. This vibration problem is usually solved by

- Adding clamp supports to increase the stiffness of the existing supporting.
- Reducing the pulsation forces by pulsation dampening devices.

The first solution is not always feasible, since a certain flexibility of the piping is required to allow for thermal expansion. Improving the supporting can lead to lower vibration levels, but also to higher stress levels in the piping. Also, contrary to the second approach, this approach does not reduce the source of the vibrations.

By damping the pulsating flow to an acceptable level according to API618, the excitation forces are minimised. Already in 1916[1], a constriction in the flow in the form of a concentric orifice plate was known to be an effective pulsation dampening device, when located near pulsation bottles or heat-exchanger vessels. In 1980 Bechert[2] showed that a perforated plate is also an effective dampening device for pulsating flows. Bechert showed, that in particular the damping of high frequency pulsations was improved compared to an orifice plate. Leistrit[3] already developed a pulsation damper, based on this principle in 1954. Recently, a pulsation damper design based on this principle, but with improved perforation geometry was patented[4]. Also compressor manufacturers have developed their own devices for pulsation dampening[5]

Figure 1.1. Various low frequency pulsation dampers, orifice plates and perforate plates.

It is known that orifice plates are less effective in compressor applications where screw compressors are the main source of pulsating flow. This is caused by the fact that the frequencies involved are much higher than in reciprocating compressor installations. However, also for reciprocating compressors, the relevant pulsation frequencies tend to increase for example for machines running at high speed and because of the increased interest in higher harmonics causing high frequency vibrations. Also, the range of frequencies involved increases in particular for installations with capacity control by means of variable speed drivers.

So far it is not clear up to which frequency orifice plates are as effective as perforated plates, as predicted by the quasi-steady, low frequency theory[2]. This has been the reason for the members of the European Forum for Reciprocating Compressors (EFRC) to initiate an experimental evaluation project of dissipative pulsation dampers used in the reciprocating compressor installations. In this joint research project, carried out by the TNO Institute of Applied Physics, the effectiveness of a number of dissipative pulsation dampers is measured as a function of mean flow rate, effective flow area and frequency.

In the next section, a short literature review is presented and the quasi-steady theory for the dampening effect is presented. In section 3, the experiments are described and the results are discussed in section 4. The conclusions are summarised in section 5.

2 Literature review

2.1 Damping devices

Dissipative pulsation dampers can be divided into two main classes. The first class is consists of a volume filled with porous material, which reduced the high frequency sound, by visco-thermal dissipation. Sometimes resonance is created in the damper in order to enhance the dissipation. For low frequencies the dampers based on this principle would become too large for practical applications. The second class of dampers relies on the generation of regions of strong vorticity in which acoustic energy is dissipated effectively. The vorticity creation can be done in several ways, which results in various types of damper plates as discussed below. In general, the flow area is reduced, and a vortex layer with strong vorticity is generated. This results in a pressure drop over the damper plate. As a result, these dampers are only effective in presence of a mean flow.

2.1.1 Orifice plate

The oldest and simplest damping device is the orifice plate[1]. This device consists of a plate with single concentric hole and is placed in a pipe section carrying a mean flow, preferably near a large vessel such as a pulsation bottle, scrubber or a heat exchanger. Several designs are known. An orifice plate with a hole with a straight boring is easiest to produce, but can be susceptible to the production of high frequency noise and pure tones (whistles)[6]. A special design developed for the application as a flow measurement device has a diverging section downstream[7]. In order to reduce the pressure drop, a design with a converging and diverging section has also been evaluated. For specific applications, also other designs are being used. For example, for hyper compressor applications, orifice plates with an eccentric hole are being used, which results in an open connection at the bottom of the plate. In the current evaluation five plates with single holes are evaluated. Two with straight boring, with open area ratios 0.40 and 0.51, one with a converging channel, one with an venturi-like channel and one with a eccentrically located straight boring.

2.1.2 Multiple Holes

Damper plates with multiple holes are being used in practice as so-called flow straightners, p.e. Laws design, which also effectively reduces the pulsation level. In the current evaluation a plate with 5 holes with straight boring is evaluated with open area ratio 0.40 and 0.51.

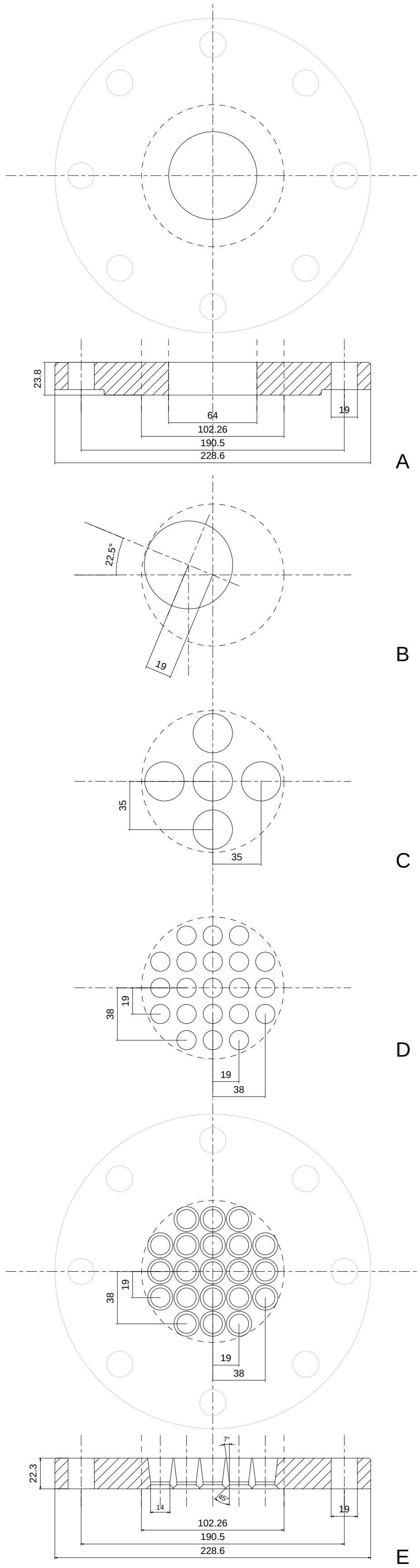


Figure 1.2. Various low frequency pulsation dampers (a,b) orifice plate (c,d) perforate plate (e) perforated plate with venturi-type holes

2.1.3 Perforated plates

Perforated linings are being used in mufflers to damp acoustic noise, for example in car engine exhausts. The effectiveness of perforated plates and regular arrays of slits in damping low frequency sound was shown by Dowling and Hughes[8]. The critical parameters, which determine the effectiveness, are the open area ratio and the Strouhal number based upon the hole diameter and the velocity in the holes.
Recently, the low frequency behaviour of orifices, slits and perforated plates was also studied experimentally by Durrieu et al.[9]. In the current evaluation a plate with 21 holes with straight boring is evaluated with open area ratio 0.40.

2.1.4 Kotter Design

The damper design patented by Kotter is based on the same principle of the perforated plate, however with a particular design of the perforation. Each hole consists of a converging and diverging section, thereby creating a venturi-like hole geometry. Apart from the advantages of an ordinary perforated plate, a lower pressure drop is claimed at an equal damping effectiveness. In the current evaluation a perforated plate with 21 holes with venturi-type geometry is evaluated.

2.1.5 NEA Umlenkkorper

Neuman und Esser has designed a pulsation damper based on the principle of generating pressure loss over a perforated plate however, the perforations are created in a small volume, through which the flow is diverted, see figure 2.1

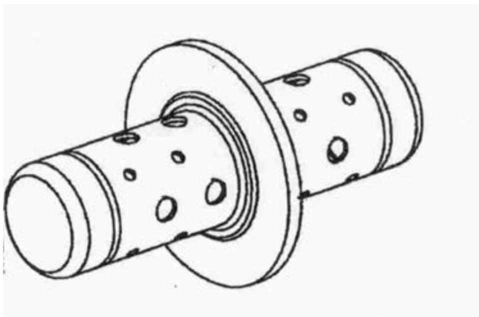


Figure 2.1 NEA Umlenkkorper

2.2 Quasi-steady theory of damping

Theoretical evidence of the dampening effect of orifice and perforated plates for low Mach number flow and low frequencies was developed by among others Howe[10], Bechert[2], Cummings and Eversman[11] and recently by Wendeloski[12]. Initially, the damping and reflection of acoustic waves at a pipe exit with a nozzle contraction was studied.
It is assumed that the mean flow generates a pressure drop over a nozzle, an orifice plate or a perforated plate, which has a significant impact on the reflection coefficient. The basic model for a pipe end is shown in figure 2.2

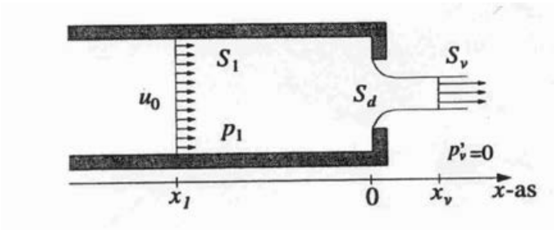


Figure 2.2 Quasi-steady model of a damper plate at the end of a pipe.

Important characteristic number are

- $M = U/c$ Mach number
- $Sr = fD/U$ Strouhal number
- $He = fD/c$ Helmholtz number
- $Re = uD/\nu$ Reynolds number
- $\alpha = Sd/S_1$ Open area ratio
- $\Gamma = Sv/Sd$ Vena-Contracta ratio

Where U is the flow velocity, c is the speed of sound, f the pulsation frequency, D the pipe diameter, ν the kinematic viscosity, Sd, S_1 and Sj the area of the orifice, pipe and jet respectively.
In the present evaluation, we consider a low Mach number $M << 1$, high Reynolds number $Re >> 10^4$ flow, and we consider low frequency pulsations for which $He << 1$. Therefore, in the area of the contraction compressibility effects can be neglected. The Strouhal number, area contraction ratio and vena contraction ratio are varied in the evaluation around a value below 1.

Considering the conservation of mass and Bernoulli's theorem, we find

$$S_1 u_1 = S_v u_v \qquad (1)$$

$$p_1 + \tfrac{1}{2} \rho u_1^2 = p_v + \tfrac{1}{2} \rho u_v^2$$

Eliminating p_v and using $p_1 = p_{10} + p_1' e^{i\omega t}$, with $p'/p_o < 1$ and similar equations for u_1 we find for the pressure drop due to the steady flow

$$\begin{aligned} p_{10} - p_{v0} &= -\tfrac{1}{2} \rho u_{10}^2 (1 - S_1^2/S_v^2) \qquad (2) \\ &= -K^* \tfrac{1}{2} \rho u_{10}^2 \end{aligned}$$

Where K is the loss coefficient for a steady flow.

For the first order unsteady flow

$$p_1' - p_v' = -\rho u_{10} u_1' (1 - S_1^2/S_v^2) \qquad (3)$$

For low frequency pulsations, the unsteady flow can be described by two plane waves travelling in up- and downstream direction

$$p_1' = p_1^+ + p_1^- \qquad (4)$$

$$u_1' = (p_1^+ - p_1^-)/\rho c$$

The damping of pulsations is caused by a change in pressure reflection coefficient R_p and impedance Z of the pipe end. These are defined as

$$R_p = p_0^-/p_o^+ \qquad (5)$$

$$Z = p_0/\rho c u_0$$

For an open pipe end, or a pipe entering a large vessel, the reflection coefficient R_p = -1, i.e. all acoustic energy is reflected with a phase shift of 180^0, i.e perfect reflection. The impedance Z = 0.
From equation (3-5) and using that p_v' = 0 for an open end, we find

$$R_p = (M_1 - M_b)/(M_1 + M_b) \qquad (6)$$

$$M_b = S_v^2/(S_1^2 - S_v^2)$$

As a result, the reflection of pressure waves can become zero, when the Mach number is equal to M_b, which is dependent on the open area ratio α.
The quasi-steady theory was extended to different end geometries such as orifices, perforated plates and slits. Also non-linear effects were taken into account by Ingard and Ising[13] and Cummings[14]. Experimental work was carried out by Salikuddin and Plumbee[15]. They observed no large differences between a perforated plate and an orifice plate with respect to damping of low frequency sound.

In practical applications, the orifice plate usually cannot be located at the pipe end, but in a pipe section very close to a large vessel. In this case, the pressure fluctuation p_v' downstream the orifice is not zero, and pressure recovery takes place in the downstream section of the orifice, see figure 2.3.

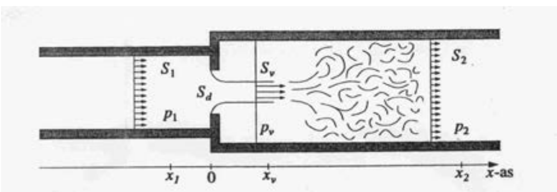


Figure 2.3 Quasi steady model of a damper plate in a pipe near an open end (Here, $S_1 = S_2$).

In this case the set of equations given by (1) have to be extended with a conservation of momentum across the turbulent region downstream the nozzle, where kinetic energy is converted to potential energy, i.e. pressure,

$$S_1 u_1 = S_v u_v \qquad (7)$$

$$S_2 u_2 = S_v u_v$$

$$p_1 + ½ \rho u_1^2 = p_v + ½ \rho u_v^2$$

$$S_2 p_v + S_v \rho u_v^2 = S_2 p_2 + S_2 \rho u_2^2$$

This results in a total pressure loss of

$$p_{10} - p_{20} = ½ \rho u_{10}^2 (1- S_1/S_v)^2 \qquad (8)$$

A similar exercise leads to a reflection coefficient equal to

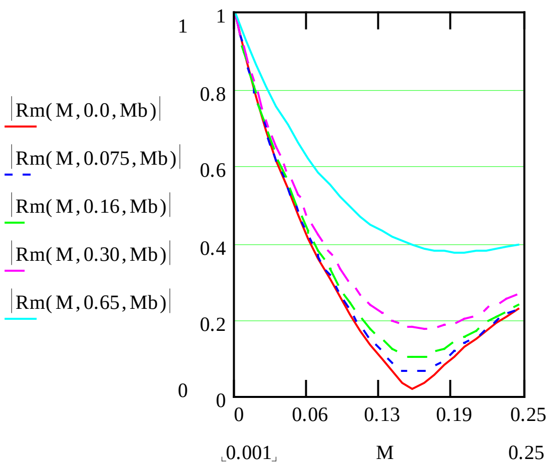
$$R_p = \frac{M_1 e^{-ikL/(1+M)} - R_u (M_1 - 2M_b) e^{ikL/(1-M)}}{(M_1 + 2M_b) e^{-ikL/(1+M)} - R_u M_1 e^{ikL/(1-M)}} \qquad (9)$$

with, taking into account that $S_1 = S_2$,

$$M_b = (S_v/(S_1 - S_v))^2 \qquad (10)$$

in which k is the wave number ω/c and L is the distance of the orifice to the pipe end, at which the reflection coefficient is R_u. An ideal reflection coefficient R_u = -1 of an open pipe end has been assumed. For L tends to zero, equation (8) is equal to (6), but with a different value of Mb. The quasi steady prediction of the reflection coefficient as given by equation (8) as a function of the mean flow Mach number is shown graphically in figure 2.4 for a number of Helmholtz numbers kL.

(a)



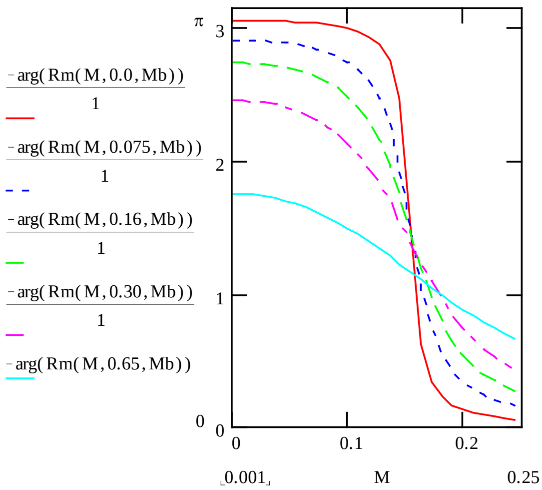


Figure 2.4 Quasi-steady prediction of the pressure reflection coefficient R_p. as a function of Mach number M and Helmholtz number kL for an open area ratio α = 0.4 and a vena contracta of 0.7, i.e. Mb = 0.14 (a) absolute value of R_p (b) phase of R_p.

When the orifice is not located at the end of a pipe, a non-reflective termination, with Rp = 0, cannot be obtained. The minimum value of Rp remains around M = M_b. For the minimum value of Rp, the strongest damping of acoustic resonance is obtained. A good comparison between the various damper plates therefore is the reflection coefficient, which can be obtained for a given open area ratio.
The dependence on the open area ratio S_v/S_1 is shown in figure 2.5. The open area ratio determines the value of M_b according to eq. (9) and is not only changed by a change of hole diameter or number of holes via S_d/S_1, but also by a change in hole geometry which changes the vena-contacta factor S_v/S_1.

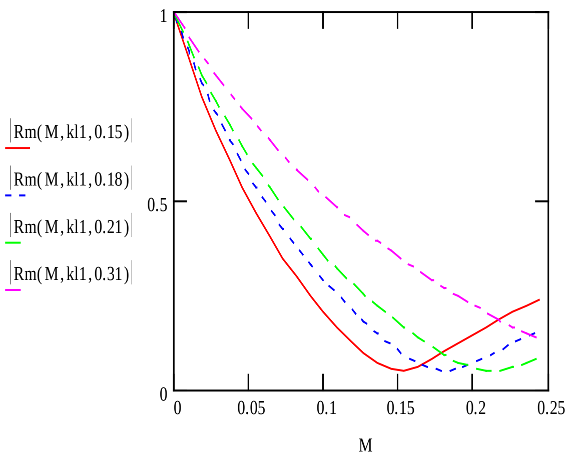


Figure 2.5 Quasi-steady prediction of the pressure reflection coefficient R_p.as a function of Mach number M and open area ratio, i.e. for Mb = 0.15, 0.18, 0.21 and 0.31.

The effectiveness of a damper plate can be evaluated from the standing wave ratio SWR, which is the ratio between the amplitude of the pressure antinode and the pressure node in the pipe section. The SWR is related to the reflection coefficient as

$$SWR = (1 + |Rp|)/(1 - |Rp|) \qquad (10)$$

For a perfect standing wave, which exists in a pipe section when |Rp| = 1, and the pressure nodes are equal to zero, the amplitude of the standing wave ratio is infinite. For a perfectly damped resonance, the reflection coefficient Rp = 0, i.e. a non-reflecting pipe termination and SWR = 1.

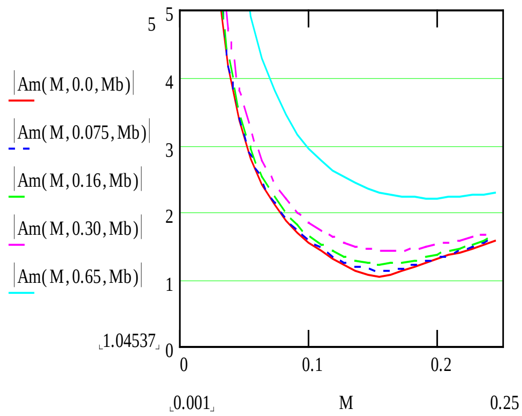


Figure 2.6 Quasi-steady prediction of the stadning wave ratio .as a function of Mach number M and Helmholtz number kL(corresponding to 10, 20, 40 and 86 Hz) for an open area ratio of α = 0.4 and a vena contracta of 0.7, i.e. Mb = 0.14.

Alternative means to evaluate the effectiveness of damper plates was suggested by Bechert[2]. He used the sound power reflection coefficient as defined by

$$A = W_-/W_+ = (1-M_1)^2/(1+M_1^2)\,|R_p|^2 \quad (11)$$

to evaluate the effectiveness of orifice plates compared to perforated plates. This coefficient describes the acoustic power reflected at a restriction in the flow. Also the relative impedance of the restriction can be measured, which is defined as

$$Z = p'/\rho c u' \qquad (12)$$

$$Z = p^+ + p^- / p^+ - p^- = (1 - R_p) / (1 + R_p)$$

The impedance is the ratio between the pressure and velocity fluctuation at the restriction and can be related to the pressure reflection coefficient according to eq. (11). The impedance Z is zero for an ideal open end with R_p = -1, and infinite for a closed end with R_p = 1.

Both energy reflection coefficient and impedance are related to the pressure reflection coefficient. We will consider only the standing wave ratio SWR in the further evaluation.
The quasi-steady theory of Bechert[2] has recently been extended for compressibility effects by Hofmans[16]. Corrections of equation (6) for compressibility effects can be found in and Durrieu et al.[9]. These effects appear to be relevant at a Mach number larger than 0.08.

2.2 Pressure drop

The quasi-steady theory uses a pressure drop obtained from the steady flow through an orifice or perforated plate to determine the low frequency reflection coefficient for plane acoustic waves. It is assumed, that the pressure drop is not strongly affected by the unsteady flow, and that the vena contracta of the jet is constant. This assumption has been validated by numerous experimentals[2,9,16].

2.2.1 Static

For the calculation of the reflection coefficient via equation (8), the vena-contracta has to be known. This value can be obtained from the static pressure drop across the damper plate. In figure 2.4 the pressure drop is shown for the four damper plates studied.

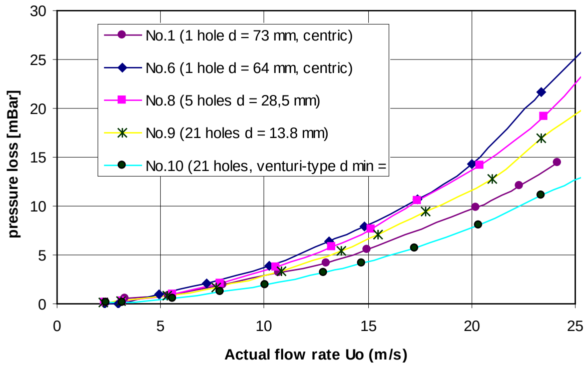


Figure 2.4 Static pressure drop for the four damper plates evaluated with area contraction ratio 0.4.

The corresponding vena contracta factor can be determined from eq. (7 and 8). The lowest pressure drop corresponds to the highest value of Γ.

2.2.2 Dynamic

In a pulsating flow, the pressure drop over the orifice plate increases. The increase depends strongly on the flow pulsation amplitude u_1'/Uo and the pulsation frequency. In figure 2.5, the (time averaged) pressure drop is shown for orifice plate nr. 6, the flow pulsation amplitudes varied from 100% for 10Hz down to 20% for 125 Hz.

Plate nr.	***Type and # holes***	***D_{eff} [mm]***	***Open area ratio***	***K = 1/M_b = (1-$S_1/S_v)^2$***
1	Orifice, 1	73	51%	2.2
6	Orifice, 1	64	39%	4.5
8	Perf., 5	64	39%	5.6
9	Perf., 21	64	39%	4.8
10	Perf.,21, venturi-type	64	39%	3.2

Table 2.1 Vena contracta factors for five damper plates evaluated from the static pressure drop in figure 2.4.

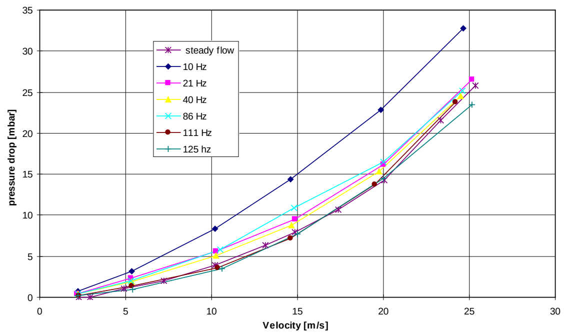


Figure 2.5 Dynamic(time averaged) pressure drop for the orifce plate nr. 6 for various pulsation frequencies and amplitudes.

In particular at low frequencies, a strong pulsating flow can lead to a significant increase of total pressure drop over the orifice plate. In flow measurement, this effect leads to flow measurement errors known as the *square-root-error*.

3 Experimental evaluation

3.1 Experimental setup

3.1.1 flow rig

The experimental evaluation of the various damper plates was carried out at the low pressure air flow loop at the Institute of Applied Physics of TNO in Delft. A schematic drawing of the installation is shown in figure 3.1. Air is compressed by a centrifugal compressor to a pressure of 8 bara, the maximum flow rate is 1300 Nm^3/h. The volume flow rate is measured by a turbine flow meter, whereby corrections for pressure, temperature and compressibility of the air are taken into account. Two reference lines, 3" and 4" are available to cover the full range of the installation with sufficient accuracy (<0.3%).

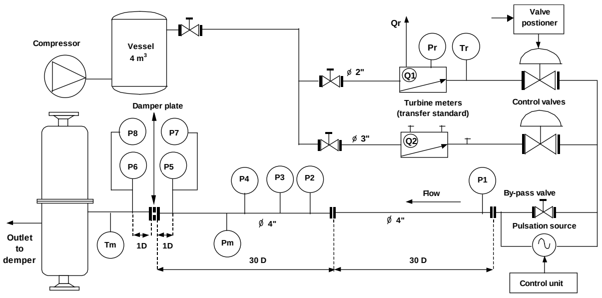


Figure 3.1 The experimental flow facility of TNO where the evaluation of pulsation damper plates was carried out.

A pressure regulator reduces the pressure from 8 bara to atmospheric pressure, thereby preventing pressure pulsations generated downstream the regulators to travel upstream towards the turbine flow meters.

3.1.2 Pulsation source

The pulsation source is a rotating cylinder, which blocks the flow periodically, thereby generating strong pressure pulsations up to 10%, depending on the frequency and downstream conditions, see figure 3.2. By creating a by-pass flow the pulsation levels can be reduced. Changing the rotating speed of the cylinder can vary the frequency of the pulsating flow.

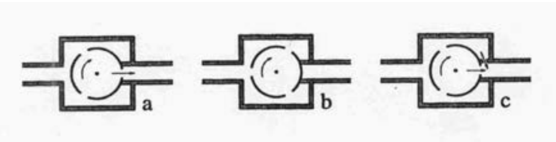


Figure 3.2 Pulsation source

Downstream the pulsation source is the measurement section, which consists of a 6 meter long 4" pipe ending in a larger vessel of volume $0.5m^3$. The damper plate is located between the pipe section and the vessel at a distance of 0.4 m from the vessel, which is the minimum distance possible. The static pressure is measured 1-D up- and downstream of the damper plate, and at 6 locations along the pipe dynamic pressure measurements are performed. To correct for density changes near the damper plate, also the temperature is measured. The dynamic sensors are located at 0.1m downstream and 0.1, 0.94, 1.525, 2.115 and 5.90m upstream of the damper plate.
An example of the dynamic pressure measurement is shown in figure 3.3 for a mean flow velocity of 10 m/s and a pulsation frequency of 20 Hz.
From figure 3.3 it can be shown, that a clear standing wave pattern exists in the measurement section caused by a resonance between the pulsation source and the large vessel.

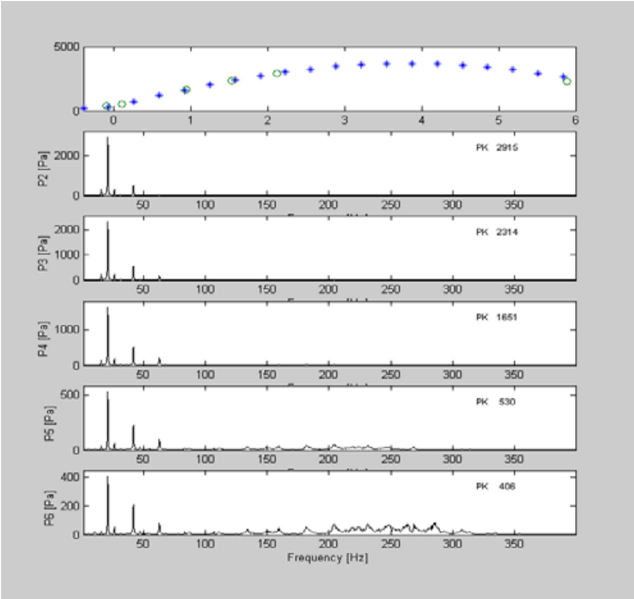


Figure 3.3 Standing wave pattern and frequency spectrum of the dynamic pressure measurement in the measurement section.

It can also be seen from figure 3.3 that close to the vessel inlet, high frequency noise is generated, probably caused by the turbulent jet entering the vessel. These frequencies however are damped very effectively.

3.1.3 Multi-microphone method

A two-microphone method (see for example Peters at al.[17,18]) is being used to determine the pressure reflection coefficient. From two dynamic pressure measurements, and taking into account the phase relationship, the amplitude of the up- and downstream travelling pressure waves (see eq. (4)) can be determined. The difference in up- and downstream convection velocity of the pressure waves, c - Uo and c + Uo, respectively and the viscothermal damping has to be taken into account. Depending on the pulsation frequency and the associated wave length, a certain combination of pressure sensors is used to determine the reflection coefficient at the damper plate. Also with this method, the standing wave pattern in the pipe can be reproduced as shown in figure 3.3, and the amplitude of the resonance can be obtained.

3.1.4 Test conditions

Measurements have been carried out at 6 different velocities, 2.5, 5, 10, 15, 20 and 25 m/s. Ambient conditions for pressure and temperature were measured at each condition, but are close to the atmospheric pressure (1 bara) and room temperature (20ºC). Measurements were carried out at resonance conditions in this section, i.e. at 10, 20, 40, 80, 112 and 125 Hz.

3.2 Results

3.2.1 Reflection coefficient

The reflection coefficient has been determined as a function of Mach number U/c and for various frequencies. In figure 3.4 the reflection coefficient measured for a single orifice plate nr. 6 (see table 2.1) is shown. The results can be compared with the quasi-steady model shown in figure 2.4. Both the amplitude and phase of the reflection coefficient compare accurately with

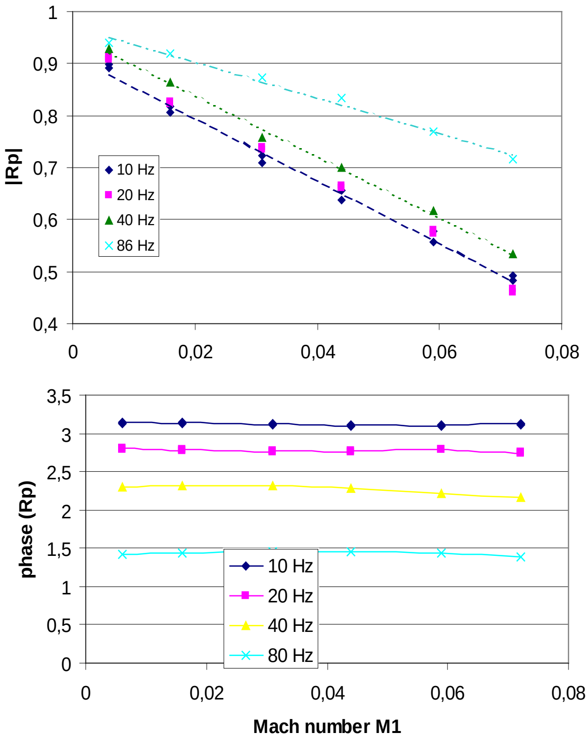


Figure 3.4 Pressure reflection coefficient R_p. as a function of Mach number M and Helmholtz number kL for an open area ratio of α = 0.4 and a vena contracta of 0.7, i.e. Mb = 0.14 (a) absolute value of R_p (b) phase of R_p. The results can be compared with the theoretical prediction shown in figure 2.4.

3.2.2 Standing wave ratio

The effect of a change in reflection coefficient of the damper plate on the standing wave pattern is shown in figure 3.5 for an area ratio a = 0.4 and a vena contract of 0.72 for three frequencies, 20, 40 and 80 Hz.

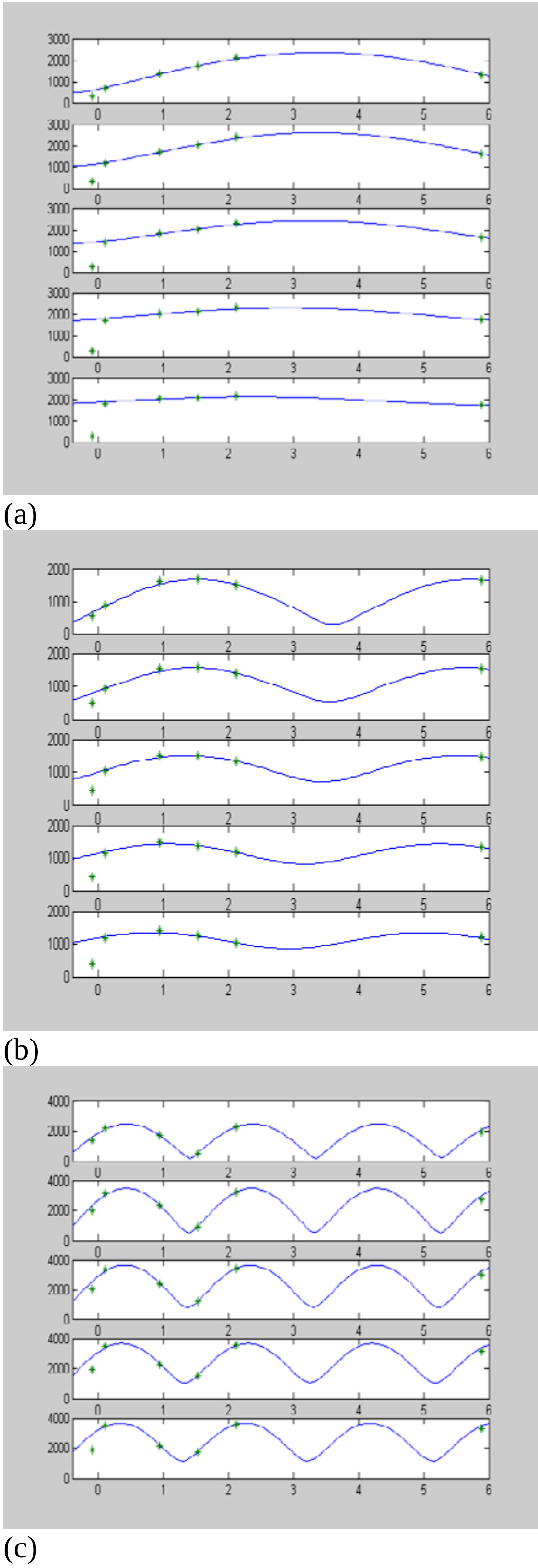


Figure 3.5 Standing wave pattern as a function of flow rate (for each graph 5, 10, 15, 20 and 25 m/s from top to bottom) for three frequencies, (a) 20 Hz (b) 40 Hz (c) 86 Hz. The standing wave ratio is the maximum divided by the minimum value. Plate #6 with α = 0.4 and a vena contracta of 0.7.

3.2.3 Effect of open area ratio

In figure 3.6 the effect of the open area on the standing wave ratio is shown. The damping caused by plate #1 with α = 0.5 is compared with plate #6 with α = 0.4, both with a single straight hole.

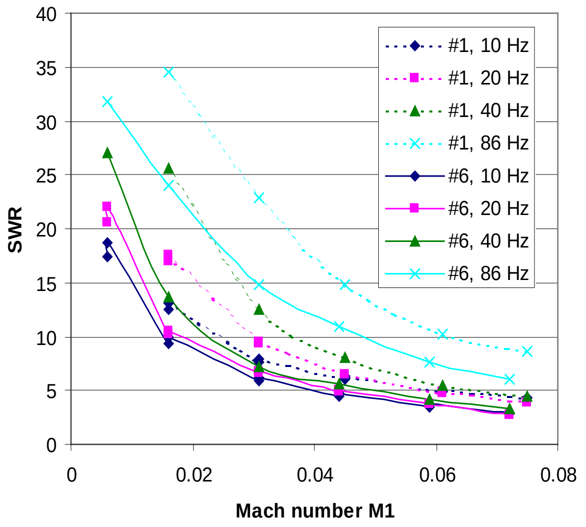


Figure 3.6 Standing wave ratio as a function of flow rate for four frequencies. A comparison is made between plate #1, with α = 0.5 and plate #6, with α = 0.4.

3.2.4 Effect of number of perforations

In figure 3.7 the effect of the number of perforations on the standing wave ratio is shown. The damping caused by plate #6 with one hole, #8 with 5 holes and #9 with 21 holes, all with open area ratio α = 0.4 is shown.

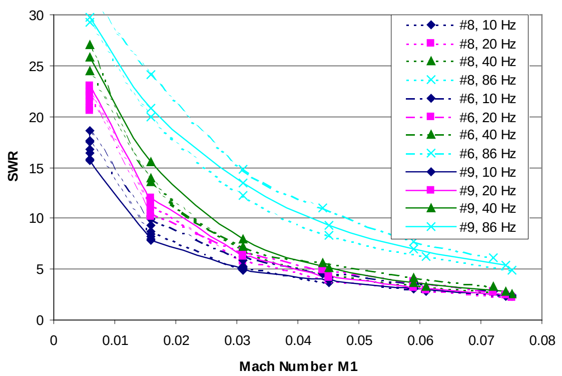


Figure 3.7 Standing wave ratio as a function of flow rate for four frequencies. A comparison is made between plate #6, #8 and #9 with 1, 5 and 21 holes, respectively. All plates with α = 0.4.

3.2.5 Effect of perforation geometry

In figure 3.8 the effect of perforation geometry on the standing wave ratio is shown. The damping caused by plate 9 with α = 0.4 and 21 holes is compared with plate 10 with α = 0.4, 21 venturi-shaped holes.

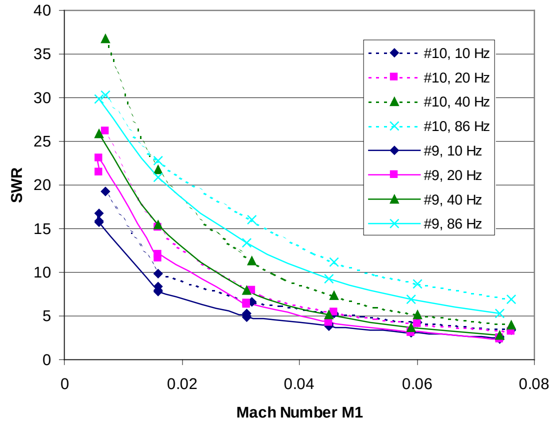


Figure 3.8 Standing wave ratio as a function of flow rate for four frequencies. A comparison is made between plate #9, with straight holes and plate #10, with venturi-shaped holes, both with α = 0.4.

3.2.6 Effect of pressure drop

In figure 3.9 the effect of the pressure drop over the plate, expressed in a K-value (see eq. (2)) on the standing wave ratio is shown. The damping caused by the various plates fall on a single line which shows that at least for low frequencies, the damping only depends on the overall pressure drop.

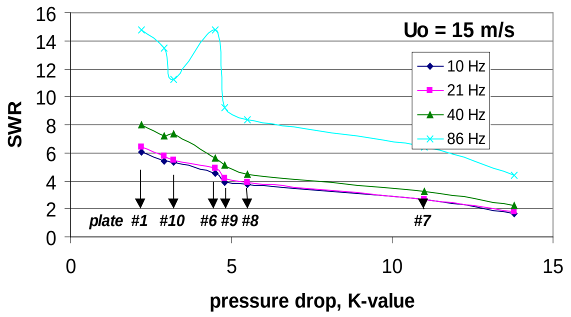


Figure 3.9 Standing wave ratio as a function of pressure drop for four frequencies. A comparison is made between various plates with 1, 5 and 21 holes, respectively and various hole sizes and geometries. The gas velocity is 15 m/s, M = 0.045.

4 Discussion

It appears that many parameters influence the damping effectiveness of the plates. In particular the open area ratio, the Mach number or velocity, the Helmholtz number or frequency, the number of holes and the shape of the holes. However, it appears that at low frequencies, all data collapse to a single curve which only depends on the pressure drop independent of other parameters related to the damper plate geometry (figure 3.9). At high frequency, the perforated plates are a more effective damper compared to a single orifice, albeit independent on the shape of the perforations.

Nonlinear effects: In the theoretical model non-linear effects have not been taken into account. Also the equations have been linearised, which is not allowed for $u'/U \sim 1$. The low frequency experiments were carried out with flow pulsation amplitudes in the order of one, which may explain the differences between the predicted and measured reflection coefficient. Also for $M > 0.08$, equation (9) is not valid anymore because of compressibility effects.

Effect of location of the orifice: The effectiveness of the orifice plate reduces at higher frequencies because the orifice is not located exactly at the end of the pipe, i.e. at the pressure node, but at a distance 0.4m upstream in the pipe section. As a result, at high frequencies, for which $kL > 1$, the orifice is located at a distance $\lambda/4$ from the pressure node, i.e. in a pressure anti-node and not effective.

Other damper designs: The results of the evaluation of the 'NEA Umlenkkorper' and other damper plate design have not been presented since the fall outside the scope of this paper. However, the results of all tests are summarised in a report which can be obtained via the EFRC.

5 Conclusion

A quasi-steady model has been developed to describe the damping of low frequency resonant pressure waves. The effect of Mach number U/c, orifice size S_d/S_1 and Helmholtz number kL on the reflection coefficient and the standing wave ratio is predicted by this model. At low Helmholtz numbers, i.e. $kL \sim 0$, A perfect damping can be obtained for a given area ratio α at a specific mach number Mb, wich is related to α according to eq. (6) and (10). For finite values of kL, i.e. when the orifice is not located at the pipe end, the dampng is less effective. According to the theoretical model, the damping is only determined by the total pressure drop over the plate, and not by number or the local geometry of the holes. Experimentally these effects have been evaluated for various plate designs. The following effects were evaluated:

Effect number of holes: increasing the number of holes but with an equal open area ratio α improves the effectiveness at high frequencies. For 10, 20 and 40 Hz no significant difference has been observed.

Effect of hole geometry: changing the shape of the holes does strongly effect the pressure drop. As a result, the damping is changed. For example, a perforated plate with 21 holes with venturi-shaped holes is less effective than a similar plate with straight holes (figure 3.4). However the total pressure drop is also less.

Effect of pressure drop: According to the quasi-steady theory, the pressure drop is the main parameter which determined the damping of the resonance. This has been validated for low frequencies for a number of plates with different pressure drops. A orifice plate and a venturi-shaped perforated plate with an equal pressure drop are equally effective for 10, 20 and 40 Hz (figure 3.5).

Effect of eccentricity hole: The pressure drop over an orifice plate with an eccentric hole is significantly larger than that of a standard orifice. As a result, the damping effectiveness is larger.

Summarising it can be concluded that for applications with low frequency pulsations, such as reciprocating compressor installations, orifice plates are equally effective as perforated plates. The pressure drop is the only parameter that determines the damping for low frequencies. A model to predict the damping of single and multi-hole plates has been validated with the experiments.

The open area that is required to achieve maximum damping (reflection free termination), will give in most cases a pressure drop that is unacceptably high and is therefore not feasible. As a result it will always be necessary to make a trade-off between pressure drop and dampening effect.

For installations with high frequency pulsations, such as installations with screw compressors, roots blowers or centrifugal compressors, pulsation dampers with multiple holes are advised because of the higher effectiveness at high frequencies.

The hole geometry is not important for the damping. However, it may be required to round the upstream edges in order to achieve the required open area, which is smaller for rounded holes.

Another reason can be that whistle tones are avoided. Also here a trade-off is to be made with cost of manufacturing.

6 Acknowledgements

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The Limits of Linear Calculations of Gas Pulsations in Reciprocating Compressor Plants

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability
May 17th / 18th, 2001, The Hague

Abstract:

The guided fluidic waves which are the reason for gas-pulsations in reciprocating compressor units are submitted to deformation during their spreading. The latter results from the different spreading speed of the wave element within the wave and leads to the well-known rising behaviour of wave fronts. In the mostly linear compution systems for simulation of gas-pulsation reciprocating compressor units , such deformation cannot be considered. By the damping effect of the friction course between fluid and pipe wall a further deformation of waves occurs. This deformation counteracts the pitching effect. The entire deformation of waves is thus defined by pitching behaviour and damping. Because of the difficulty of deformation analysis mostly the examination of the deformation behaviour of the waves in the lines that is to be expected is omitted when simulating gas-pulsation in reciprocating compressor units, or reference is made to the already known limits of gas condition for linear compution. This contribution gives a characterization of deformation behaviour of waves guided in lines and presents a procedure which permits an approximative estimate of the errors to be expected in linear compution of pulsating gas-flow in pipes at reasonable expense.

1. Einleitung

Leitungsgeführte fluidische Wellen unterliegen bei ihrer Ausbreitung einer Deformation, die sich aus den unterschiedlichen Ausbreitungsgeschwindigkeiten der Wellenelemente innerhalb einer Welle ergibt und zu dem bekannten Aufsteilverhalten der Wellenfronten führt (Abb. 1).

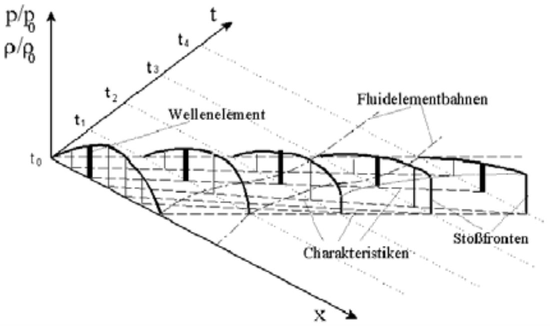


Abb.1: Deformation einer hinlaufenden Welle

Die bei pulsierenden Gas-Strömungen in Rohrleitungen auftretenden Wellendeformationen bestimmen im Wesentlichen die Wahl des möglichen Berechnungsverfahrens. Unterliegen die leitungsgeführten Wellen einer erheblichen Deformation, die zum starken Aufsteilen der Wellenfronten führt, so können diese nur mit nichtlinearen Berechnungsverfahren hinreichend genau beschrieben werden. Nichtlineare Berechnungsverfahren für pulsierende Gas-Strömungen sind sehr kompliziert und erfordern einen erheblichen Rechenaufwand.

Kolbenverdichteranlagen besitzen in der Regel umfangreiche und komplizierte Anlagenstrukturen. Die Simulation von Gas-Pulsationen in ihnen erfolgt aus diesem Grund zumeist mittels überschaubarer linearer Berechnungsverfahren. Die Grenzen der Anwendung dieser Verfahren werden in der Regel nur durch eine Begrenzung der maximal auftretenden relativen Dichteänderung in den Wellen beschrieben. Damit werden weitere wesentliche Einflüsse des Gaszustandes und der geometrischen Abmessungen der Rohrleitung auf das Deformationsverhalten leitungsgeführter Wellen nicht berücksichtigt.

Numerische Untersuchungen von Wellenvorgängen in unterschiedlichen Anlagen haben ergeben, dass obwohl die üblichen Grenzwerte der relativen Dichteänderung im Bereich für eine Zulässigkeit linearer Berechnungsverfahren lagen, dennoch unzulässig hohe Wellendeformationen auftraten. Andererseits ergaben sich bei Überschreitung der Grenzwerte der Dichteänderung dennoch relativ geringe Wellendeformationen.

Dieser Beitrag beinhaltet eine Charakterisierung des Deformationsverhaltens von leitungsgeführten Wellen und stellt ein Verfahren vor, das mit erträglichem Aufwand eine näherungsweise Überprüfung der zu erwartenden Fehler bei einer linearen Berechnung der pulsierenden Gas-Strömungen in Rohrleitungen ermöglicht. Er soll auch zu weiteren Arbeiten auf diesem Gebiet anregen.

2. Voraussetzungen

Das im Folgenden beschriebene basiert auf der Analyse der strömungsmechanischen Vorgänge an der Front einer Welle. Es werden hier beginnend die harmonische Wellenform und der Zustand in einer reflexionsfreien Leitung mit kreisförmigem Innenquerschnitt untersucht.

Für die kontinuierlich nur hinlaufenden Wellen werden als Wellenfronten die Knotenpunkte des Geschwindigkeitsverlaufes bezeichnet, an denen die Fluidgeschwindigkeit in der Welle von der negativen zur positiven Richtung wechselt. Als Wellenende werden die Knotenpunkte bezeichnet, an denen die Fluidgeschwindigkeit von der positiven zur negativen Richtung wechselt.
Zuerst wird der Wellenverlauf in einem ruhenden Fluid untersucht. In einer weiteren Betrachtung laufen die Wellen über eine zeitkonstante Strömung. Für beginnend nichtharmonische Wellenverläufe wird ein Näherungsverfahren vorgeschlagen [Ste00].

Den hier durchgeführten Betrachtungen zu leitungsgeführten Wellen liegt eine grundlegende Arbeit von E. Becker [Bec70] zugrunde, die sich mit der Entstehung von Verdichtungsstößen in ebenen Wellenfeldern befasst.

3. Änderung der Beschleunigung eines Fluidelementes an der Wellenfront

Die Ursachen der Deformation einer Welle im kompressiblen Fluid sind:
Aufsteilung, d. h. unterschiedliche Ausbreitungsgeschwindigkeit der Wellenelemente innerhalb einer Welle mit der Wirkung der Aufsteilung im vorderen und der Abflachung im hinteren Bereich einer positiven Halbwelle.
Dämpfung infolge Reibung, d. h. Fluidreibung ausgehend von der Scherströmung an der Rohrwand mit Wirkung der Dämpfung der gesamten Welle.

Betrachtet man den Bewegungsverlauf an der Wellenfront aus der Sicht eines Beobachters, der sich mit der Wellenfront bewegt, so strömen aus dessen Sicht die Fluidelemente mit der Ausbreitungsgeschwindigkeit der Welle in die Wellenfront hinein und werden dann auf die Fluidgeschwindigkeit in der Welle beschleunigt. Führt man für die in die Welle eintretenden Fluidelemente eine Koordinate *s* ein, beginnend an der Wellenfront in die Welle hineinlaufend, so gilt für die zeitliche Ortsveränderung eines Fluidelementes unmittelbar nach der Wellenfront

$$\frac{ds}{dt}(s,t) = {}^{0}a + v(s,t) \quad , \qquad (01)$$

mit a^0 als Ausbreitungsgeschwindigkeit der Wellenfront und *v(s, t)* der Geschwindigkeit eines Fluidelementes in der Welle

Man kann nun nach den Gesetzen der substantiven Ableitung aus (1) die Beschleunigung des Fluidelementes bilden und weiterhin deren örtliche und zeitliche Änderung. Dabei ist zu beachten, dass die Ursache der Beschleunigungsänderung ein konvektiver Bewegungsablauf ist, somit gilt bei der zu bildenden substantiellen Ableitung der Beschleunigung nur der sich ergebende konvektive Anteil.

Für die Wellenfront mit *v(s=0, t=0)* ergibt sich daraus unter Vernachlässigung der Krümmung der zeitlichen Abstandsänderung der Fluidelemente die zeitliche Änderung der Beschleunigung eines Fluidelementes an der Wellenfront zu

$$\left.\frac{db}{dt}\right|_{s=0} = a_0 \frac{\partial \dot{v}}{\partial s} + a_0 \frac{\partial v}{\partial s}\frac{\partial \dot{s}}{\partial s} \quad . \qquad (02)$$

Der erste Term in (02) berücksichtigt die Geschwindigkeitsänderung der Fluidelemente infolge Dämpfung. Der zweite Term in (02) berücksichtigt die Geschwindigkeitsänderung der Fluidelemente infolge der zeitlichen Änderung der Koordinatenabstände der den Wellenelementen zugeordneten Fluidgeschwindigkeiten, er beinhaltet das Aufsteilverhalten der Welle an der Wellenfront.

4. Zur Aufsteilung der Wellenfront

Man kann sich eine Welle in eine sehr große Zahl sehr kleiner Wellenelemente unterteilt vorstellen, in sogenannte Machsche Elementarwellen. Diese besitzen jeweils unterschiedliche Zustandsgrößen. Die Ausbreitungsgeschwindigkeit eines Wellenelementes hängt von seinen Zustandsgrößen Druck bzw. Dichte ab, größere Werte der Zustandsgrößen führen zu höheren Ausbreitungsgeschwindigkeiten und umgekehrt. Die Wellenelemente besitzen somit unterschiedliche *x*-*t*-Ortskurven, also unterschiedliche Charakteristiken. Dies führt im Verlauf der Wellenbewegung zu Abstandsänderungen zwischen den Wellenelementen und zu ihrem gegenseitigen Überlaufen. So wird der vordere Teil einer Welle durch immer neue Wellenelemente mit höheren Ausbreitungsgeschwindigkeiten gebildet, die sich auf die Wellenfront zu bewegen. Die Welle wird deformiert, sie steilt auf. Hat das zustandsgrößte Wellenelement die Wellenfront erreicht, so hat sich eine Stoßfront gebildet und die Wellenfront breitet sich mit deren Ausbreitungsgeschwindigkeit aus.
Für die zur Aufsteilung führenden Vorgänge werden isentrope bzw. adiabate Zustandsänderungen vorausgesetzt.

Die aus der Sicht des vorstehend angeführten Beobachters in die Welle einströmenden Fluidelemente nehmen die Zustandsgrößen der nachfolgenden Wellenelemente an. Ihre zeitliche Wegänderung nach dem unmittelbaren Eintritt in die Wellenfront ist

$$\dot{s} = a_0 + \left.\frac{\partial a}{\partial s}\right|_{s=0} ds + \left.\frac{\partial v}{\partial s}\right|_{s=0} \quad . \qquad (03)$$

Mit den Isentropenbeziehungen für die Zustandsgrößen, der Impuls- und Kontinuitätsgleichung und mit (03) wird die örtliche Änderung der zeitlichen Wegänderung, also die Änderung zum nächsten
Wellenelement

$$\frac{\partial \dot{s}}{\partial s} = \frac{\kappa+1}{\kappa+1}\frac{\partial a}{\partial s} \quad . \qquad (04)$$

Mit (04) und einigen geeigneten Umwandlungen sowie der Definition einer Anfangsbeschleunigung eines Fluidelementes zu Zeit *t=0* an der Wellenfront

$$b_0 = \left.\frac{\partial v}{\partial t}\right|_{s=0,t=0} \qquad (05)$$

wird der die Aufsteilung betreffende Term in (02) zu

$${}^{0}a \frac{\partial v}{\partial s}\frac{\partial \dot{s}}{\partial s} = \frac{\kappa+1}{2}\frac{1}{a_0} b_0 \quad . \qquad (06)$$

5. Zur Dämpfung infolge Reibung

Für die hier durchgeführten Untersuchungen wird der Bewegungsablauf an der Wellenfront einer beginnend harmonisch verlaufenden Welle betrachtet. Für die Bestimmung der Dämpfung infolge Reibung kann man deshalb von der differentiellen Charakterisierung der Zustandsverläufe absehen und von einer hinlaufenden Welle in geschlossener Darstellungsform ausgehen.

$$v(x,t) = e^{j(\omega t + \varphi)} \, \hat{v} \, e^{-j(P - jD)\frac{\omega x}{a_0}} \Big|^{\Re} \quad . \qquad (07)$$

In (07) bedeutet *x* die Koordinate in Wellenlaufrichtung. Das Phasen- und Dämpfungsmaß ergeben sich zu

$$P = \sqrt{1 + {}^{\Im}R} \quad \text{und} \quad D = \frac{{}^{\Re}R}{2\sqrt{1 + {}^{\Im}R}} \qquad (08)$$

mit R als eine komplexe Reibungsfunktion nach [Ste87] und [Ste98]. $\Re$ und $\Im$ deuten auf Real- und Imaginärteil hin. Aus (07) wird die Beschleunigung und weiterhin die Änderung der Beschleunigung der Fluidteilchen an der Wellenfront gebildet. Transformiert man diese Beziehungen mit

$$x = a_0 t - s$$

auf die Koordinate *s*, so ergibt sich der Realteil der Beschleunigung zu

$$\left.\frac{\partial v}{\partial t}\right|_{s=0,t=0} = \omega\, \hat{v} = b_0 \qquad (09)$$

und die Änderung der Beschleunigung zu

$$\left.\frac{\partial \dot{v}}{\partial s}\right|_{s=0,t=0} = -D\frac{\omega^2}{a_0}\hat{v} \quad . \qquad (10)$$

Mit der Definition (05) und (09), (10) ergibt sich der erste Term von (02) zu

$$a_0 \left.\frac{\partial \dot{v}}{\partial s}\right|_{s=0,t=0} = -D\omega\, b_0 \quad . \qquad (11)$$

6. Kriterien zum Aufsteilverhalten

Die zeitliche Gesamtänderung der Beschleunigung der Fluidelemente an der Wellenfront nach (2) ergibt sich aus der Summe der Änderung infolge des Aufsteilens (06) und infolge der Dämpfung (11). Mit den in [Bec70] verwendeten und die Aufsteilung und Dämpfung charakterisierenden Parametern

$$\Gamma_0 = \frac{\kappa+1}{2}\frac{1}{a_0} \quad , \quad \Omega_0 = D\omega \qquad (12)$$

wird dann die Änderung der gesamten Beschleunigung

$$\left.\frac{d\,b}{d\,t}\right|_{s=0,t=0} = \Gamma_0\, b_0^{\;2} - \Omega_0\, b_0 \quad . \qquad (13)$$

Aus (13) ist ersichtlich, dass bei Gleichheit der beiden Glieder der rechten Seite die Änderung der Gesamtbeschleunigung zu Null wird. Daraus lässt sich eine kritische Beschleunigung ableiten

$$b_{krit} = \frac{\omega_0}{\Gamma_0} \quad , \qquad (14)$$

für die gilt: Ist die Beschleunigung zur Zeit $t = 0$ der Fluidelemente b_0 an der Wellenfront gleich der kritischen Beschleunigung b_{krit}, so ist die Änderung der Beschleunigung der Fluidelemente Null, d. h. die Welle wird nicht deformiert.

Nun wird eine **Wellendeformationskennzahl σ** definiert, die gleich dem Verhältnis der kritischen Beschleunigung zur Anfangsbeschleunigung der Fluidteilchen an der Wellenfront ist

$$\sigma = \frac{b_{krit}}{b_0} = \frac{\Omega_0}{\Gamma_0\, b_0} \quad . \qquad (15)$$

Mit der Anfangsbeschleunigung nach (09) und der Machzahl für die Amplitude der beginnend harmonischen Welle

$$\hat{Ma} = \frac{\hat{v}}{a_0} \qquad (16)$$

ergibt sich die Wellendeformationskennzahl zu

$$\sigma = \frac{2}{\kappa+1}\frac{D}{\hat{Ma}} \quad . \qquad (17)$$

Die Wellendeformationskennzahl charakterisiert das Deformationsverhalten einer beginnend harmonischen Welle im ruhenden Fluid und es gilt:

$\sigma = 0$: deformationsfreie Welle,
$\sigma > 0$: abflachen der Welle infolge Dämpfung
$\sigma < 0$: aufsteilen der Welle

7. Vorgänge im bewegten Fluid

Die Wellen laufen nun über ein mit zeitkonstanter Geschwindigkeit in Laufrichtung der Wellen. strömendes Fluids hinweg. Hier ergibt sich analog zu (01) für die zeitliche Örtsveränderung unmittelbar nach der Wellenfront

$$\frac{ds}{dt}(s,t) = {}^0a\left(1 - Ma_0\right) + v(s,t) \qquad (18)$$

und für die zeitliche Änderung der Beschleunigung an der Wellenfront analog (02) gilt

$$\left.\frac{db}{dt}\right|_{s=0} = a_0\left(1-Ma_0\right)\frac{\partial \dot{v}}{\partial s} + a_0\left(1-Ma_0\right)\frac{\partial v}{\partial s}\frac{\partial \dot{s}}{\partial s} \qquad (19)$$

Geht man hier ebenfalls wie in den Abschnitten 4, 5 und 6 vor, so ergeben sich die charakteristischen Parameter zu

$$\Gamma_0 = \left(1-Ma_0\right)\frac{\kappa+1}{2}\frac{1}{a_0}$$
$$\Omega_0 = \left(1-Ma_0\right)D\,\omega \qquad (20)$$

und damit eine Gesamtänderung der Beschleunigung

$$\left.\frac{d\,b}{d\,t}\right|_{s=0,t=0} = \left(1+Ma_0\right)\Gamma_0\, b_0^{\;2} - \Omega_0\, b_0 \quad . \qquad (21)$$

Für Wellen, die sich über ein mit konstanter Geschwindigkeit in Wellenlaufrichtung strömendes Fluid ausbreiten wird die Wellendeformationskennzahl

$$\sigma = \frac{1}{1+Ma_0}\frac{2}{\kappa+1}\frac{D}{\hat{M}a} \quad . \tag{22}$$

Sie besitzen gegenüber Wellen im ruhenden Fluid nach (17) ein etwas erhöhtes Aufsteilverhalten.

8. Deformationsverlauf einer Welle

Wellenverläufe mit einer Wellendeformationskennzahl σ >1 können bedenkenlos mit linearen Berechnungsverfahren untersucht werden. Für Wellenverläufe mit σ <1 erfolgt eine aufsteilende Wellendeformation. Numerische Untersuchungen zeigen, dass der Aufsteilvorgang an der Wellenfront nicht linear zur Wellenlauflänge erfolgt.
Nimmt man geringe Ungenauigkeiten infolge schwacher Deformation der Wellen bei einer Simulation der Pulsation in Kauf, so können unter bestimmten Voraussetzungen für Wellenverläufe mit σ <1 auch linearen Berechnungsverfahren angewendet werden. Die Grenzen der Anwendung lassen sich aus dem Deformationsverlauf einer Welle ableiten.

Geht man von einer der Beziehung (13) entsprechenden Differentialgleichung

$$\dot{b} + \Omega_0\, b - \Gamma_0\, b^2 = 0 \tag{23}$$

aus, bei der nun gilt $b = b(s{=}0,\, t)$, und löst diese mit der Anfangsbedingung für $t = 0$ ist $b = b_0$, so ergibt sich nach [Bec70] mit (14) und (15) hierfür eine Beziehung für die zeitlich sich ändernde Beschleunigung der Fluidelemente an der Wellenfront

$$b = \frac{b_{krit}}{1-(1-\sigma)e^{\Omega_0 t}} \quad . \tag{24}$$

Die Aufsteilung erreicht den Stoß, wenn die Beschleunigung an der Wellenfont $b \to \infty$ geht, das ist der Fall wenn der Nenner in (24) zu Null wird. Die hierzu erforderliche Zeit wird damit

$$t_{st} = -\frac{1}{\Omega_0}\ln(1-\sigma) \tag{25}$$

und mit der Ausbreitungsgeschwindigkeit der Wellenfront a_0 bzw. $a_0\ (1+Ma_0\ )$ wird die Lauflänge der Welle bis zu Stoß

$$x_{st} = -\frac{a_a(1+Ma_0)}{\Omega_0}\ln(1-\sigma) \quad . \tag{26}$$

Hier und künftig ist zu beachten, dass für $Ma_0 > 0$ σ nach (22), andernfalls nach (18)) anzuwenden ist.

Für eine Beurteilung des Deformationsverlaufes einer Welle, beginnend am Leitungsanfang und längs der Rohrleitung bis zum Erreichen des Stoßes, eignet sich der Verlauf der Änderung der Beschleunigung an der Wellenfront, in der Form

$$\frac{b}{b_0} = \frac{\sigma}{1-(1-\sigma)e^{\frac{\Omega_0}{a_0(1+Ma_0)}x}} \quad . \tag{27}$$

(27) erfüllt die Grenzwerte für $\sigma = 1$ und für $x = x_{st}$.
Eine Deutung von (27) erfolgt anschaulich über die Steigung der Wellenfronten. Bildet man die Steigung der Wellenfront mittels einer normierte Steigung wie folgt

$$\left.\frac{dv}{ds}\right|_{s=0} = \frac{b_0}{a_0}\frac{2}{\pi}\frac{d\left(\frac{v}{\hat{v}}\right)}{d\left(\frac{s}{\lambda/4}\right)} \sim b_0 \tan\alpha = b \tag{28}$$

und berücksichtig, dass für die Wellenfront einer harmonischen Welle bezogen auf die Koordinatenrichtung *s* die normierte Steigung $\tan\alpha = 1$ ist, so ist mit (28)

$$\left.\frac{dv}{ds}\right|_{s=0} \Big/ \left.\frac{dv}{ds}\right|_{s=0,t=0} = \frac{b}{b_0} = \tan\alpha \quad . \tag{29}$$

(29) lässt sich auch direkt aus (28) ableiten. Man kann nun (27), also b/b_0 , als die normierte Steigung der Wellenfront der aufsteilenden Welle zur normierten Steigung der Wellenfront der beginnend harmonischen Welle deuten. Instruktiv ist vor allem, wenn man die für die Steigungen sich ergebenden Winkel α betrachtet.

9. Zu beliebigen Wellenstrukturen

Alle vorstehenden Betrachtungen erfolgten an einer hinlaufenden reflexionsfreien harmonischen Welle. In Kolbenverdichteranlagen existieren jedoch periodische eingeschwungene beliebig geformte Wellenstrukturen. In welcher Weise lassen sich die vorstehenden Untersuchungen nun auf die Gas-Pulsation in Kolbenverdichteranlagen übertragen?

Die eingeschwungene Wellenstruktur ist das Ergebnis der Überlagerung hin- und rücklaufender Wellen. Diese hin- und rücklaufenden Wellen unterliegen separat der Deformation. Die Schwierigkeit besteht nun, aus einer eingeschwungenen berechneten Wellenstruktur die zu untersuchenden hin- bzw. rücklaufenden Wellen zu separieren. Das gelingt für eine eingeschwungene Wellenstruktur näherungsweise, wenn der Verlauf der Druck- Δp und Geschwindigkeitsschwankungen Δv bekannt sind und der Wellenwiderstand näherungsweise 1 gesetzt wird. Dann gilt für jede Stelle *x* einer Rohrleitung

$$\frac{\Delta p}{\rho_0\, a_0} = \hat{v}_h\, e^{j\varphi_h} + \hat{v}_r\, e^{j\varphi_r}$$

(30a)

und
$$\Delta v = \hat{v}_h \, e^{j\varphi_h} - \hat{v}_r \, e^{j\varphi_r} \, . \tag{30b}$$

Mit (30a), (30b) ergibt sich für eine hinlaufende Welle

$$\hat{v}_h = \frac{1}{2} e^{-j\varphi_h} \left(\frac{\Delta p}{\rho_0 \, a_0} - \Delta v \right) \quad . \tag{31}$$

Wendet man (30) auf eine Stelle maximaler Druckschwankung an, so kann man damit folgern, dass hier $\varphi_h \to 0$ geht. Die Amplitude der hinlaufenden Welle lässt sich dann aus (31) bestimmen. Man kann ebenso auch eine Stelle maximaler Geschwindigkeitsschwankung für dieses Verfahren benutzen. Pic-to-pic Darstellungen eignen sich gut für eine Stellenauswahl.

Die in Abschnitt 4 durchgeführten Untersuchungen zum Aufsteilverhalten sind unabhängig von der Wellenform, für sie ist die entscheidende Ausgangsgröße die Anfangsbeschleunigung der Fluidteilchen an der Wellenfront. Diese Anfangsbeschleunigung lässt sich bei beliebigen Wellenstrukturen nicht mehr wie in Abschnitt 5 aus der Wellenamplitude bestimmen.
Der in Abschnitt 5 untersuchte Einfluss der Dämpfung bezieht sich auf eine beginnend harmonische Welle. Man kann davon ausgehen, dass die hier gefundenen Zusammenhänge zwischen den Zustandsgrößen Dämpfungsmaß, Kreisfrequenz und Anfangsbeschleunigung an der Wellenfront auch für nichtharmonische Wellenformen gelten. Diese Zustandsgrößen jedoch für einen nichtharmonischen Wellenverlauf zu bestimmen ist problematisch.

Im Folgenden werden Vorschläge für eine Betrachtung des Deformationsverhaltens von nichtharmonischen Wellenverläufen vorgestellt.

In vielen Fällen existiert bei periodischen Wellenverläufen in Kolbenverdichteranlagen eine Harmonische, die den überwiegenden Anteil in der Wellenstruktur bildet, also eine sogenannte Vorzugsharmonische. In einem solchen Fall genügt es, das Deformationsverhalten dieser harmonischen Teil-Welle zu untersuchen.

Sollte in einer beliebigen Wellenstruktur keine Vorzugsharmonische existieren, so wird ein Approximationsverfahren auf der Grundlage der Betrachtung harmonischer Wellen vorgeschlagen.
Bei harmonischen Wellen erfolgt die größte Geschwindigkeitsänderung der Fluidteilchen an den Knotenpunkten, die entsprechend vorstehenden Kriterien die Wellenfronten darstellen. Für Wellen mit beliebigen Wellenformen müssen auch derartige Knotenpunkte und somit Wellenfronten festgelegt werden. Von einer solchen Wellenfront ausgehend wird eine **harmonische Ersatzwelle** definiert, die näherungsweise das gleiche Verhalten an der Wellenfront, also gleiches Deformationsverhalten besitzen soll wie die zu betrachtende beliebige Welle.

Die Dämpfung einer Welle wird durch ihre Form und Frequenz beeinflusst. Ausgehend von der Form der beliebigen Welle wird ein Wellenscheitel definiert, der einer Ersatzamplitude entspricht. Es wird nun vorgeschlagen, eine Ersatzkreisfrequenz zu bilden, mit der sich der Abstand von der definierten Wellenfront zum definierten Wellenscheitel als Viertelwelle, also $\lambda/4$, einer harmonischen Ersatzwelle ergibt. Es soll gelten

$$\omega_E = 2\pi \frac{a_0}{\lambda_E} \quad . \tag{32}$$

Die Steigung der Wellenfront der nichtharmonischen Welle gegenüber einer harmonischen Welle wird durch den Korrekturfaktor χ beschrieben. Geht man davon aus, dass eine harmonische Welle an der Wellenfront die Steigung $\chi = 1$ besitzt, so gilt für die interessierenden nichtharmonischen Wellen mit größerer Steigung $\chi > 1$. Die Beschleunigung der Fluidelemente zur Zeit $t = 0$ ist dann

$$b_{0E} = \chi \, \omega_E \; \hat{v} = \omega_E \; \hat{v}_E \quad . \tag{33}$$

Mit der Ersatzfrequenz und der Ersatzamplitude in (33) ist die harmonische Ersatzwelle bestimmt, für die anstelle der nichtharmonischen Welle das Deformationsverhalten ermittelt werden kann. In diesem Zusammenhang soll nochmals darauf hingewiesen werden, dass hier nur die Änderung der Steigung der Wellenfront als ein Maß für das Deformationsverhalten einer Welle untersucht wird, die Form der Welle und deren Änderung jedoch nicht.
Entsprechend der Möglichkeit der Approximation der nichtharmonischen Welle zu einer harmonische Welle ist das vorstehende Verfahren eine mehr oder weniger grobe Näherung.

10. Zu den Grenzen der linearen Berechnung

Die Festlegung von Grenzen für eine lineare Berechnung der Wellen lässt sich nicht so einfach beantworten. Für Wellenstrukturen, bei denen die Deformationskennzahl > 1 ist, kann eine lineare Berechnung unbedenklich durchgeführt werden.

Für Wellenstrukturen, bei denen die Wellendeformationskennzahlen < 1 sind, können dennoch lineare Berechnungsverfahren angewendet werden. Jedoch müssen für derartige Wellenstrukturen das Deformationsverhalten- bzw. Aufsteilverhalten wie vorstehend angeführt ermittelt werden. Dann muss entschieden werden, welchen Fehler man für eine Simulation der Pulsation noch zulassen kann.

11. Numerische Untersuchungen

Im Folgenden werden die Ergebnisse einiger numerischer Untersuchungen mittels „ Mathcad“ dargestellt.

Nach ersten Untersuchungen ist der Einfluss eines bewegten Fluids mit einer zeitgemittelten Mach-Zahl bis zu Ma_0 = 0,2 auf das Deformationsverhalten sehr gering.

Es wird somit eine **beginnend harmonische hinlaufende Welle im ruhenden Fluid** mit folgenden Parametern betrachtet:

Fluid Luft	
kinem. Viskosität	ν = 1,5∗ 10-6 m/s2
Isentropenexponent	κ = 1,4
Ruheschallgeschwindigkeit	a0 = 342 m/s
Rohrinnen-Radius	R = 0,05m

Als Variable für den Frequenzeinfluss werden die modifizierte Reynoldsche Zahl

$$\mathrm{Re}_\omega = \frac{R^2\,\omega}{\upsilon}$$

und als Variable für die Geschwindigkeitsamplitude der Welle die mit ihr gebildete Machsche Zahl verwendet

$$\hat{M}a = \frac{\hat{v}}{a_0} \quad .$$

Ihr Parameterbereich ist:

modifiz. Reynoldsche Zahl	$\mathrm{Re}\omega = 10^3$ bis 10^5
Machsche Zahl	Ma = 0,03 bis 0,2

Für das hier berechnete Dämpfungsmaß wurde entgegen [Ste78], [Ste99] die Kompressibilität vernachlässigt. Mit Mathcad lassen sich nur diskrete Werte für das Dämpfungsmaß ermitteln, der nachfolgend im Diagramm 1 dargestellte Verlauf des Dämpfungsmaßes über der modifizierten Reynoldschen Zahl ist das Ergebnis einer Spline-Interpolation separat berechneter Werte.

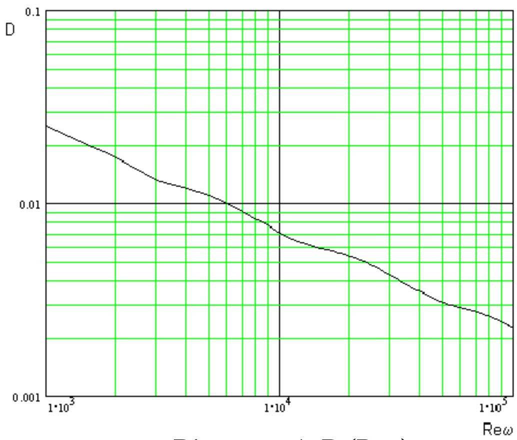


Diagramm 1: D (Re_ω)

Für konstante Re_ω Zahlen zeigt das nachfolgende Diagramm 2 die Deformationskennzahlen σ über den Mach-Zahlen Ma der Amplituden.

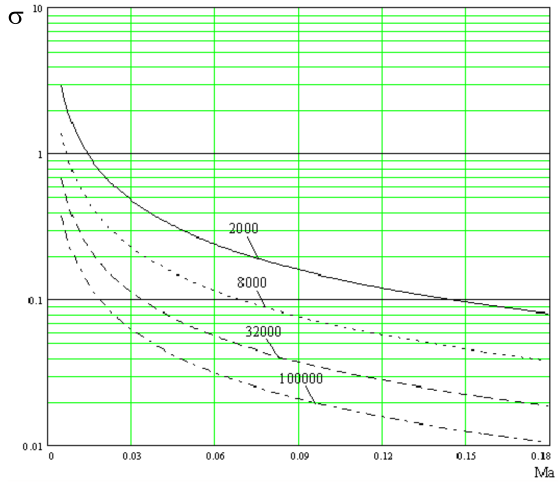


Diagramm 2: σ (Ma) für Re_ω=konst.

Diagramm 2 zeigt, dass fast für den gesamten betrachteten Parameterbereich $\sigma < 0$ gilt.

Ebenfalls für konstante Re_ω zeigen die nachfolgenden Diagramme 3a und 3b für unterschiedliche Parameterbereiche die Lauflänge x_{st} in *m* über die Deformationskennzahlen σ

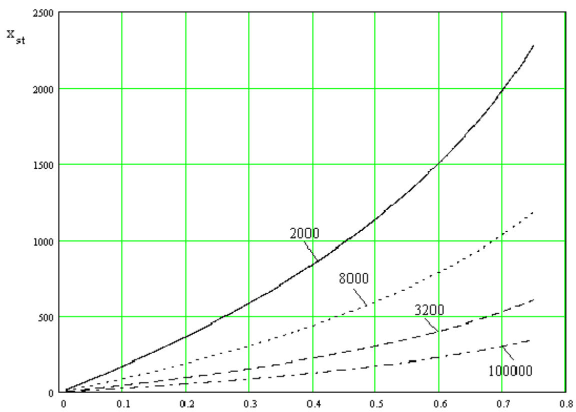


Diagramm 3a: x_{st} (σ) in *m* für Re_ω=konst.

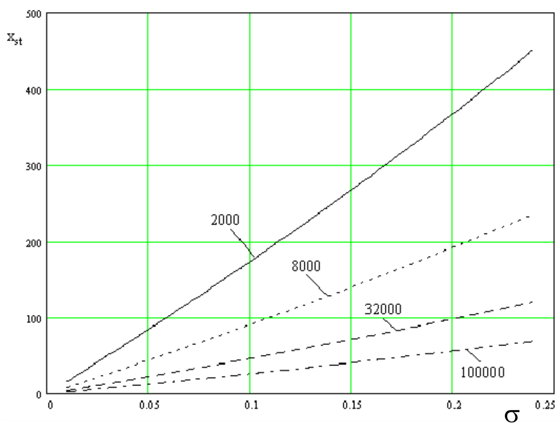


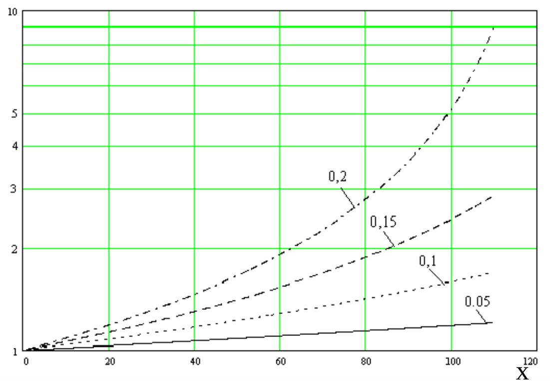
Diagramm 3b: x_{st} (σ) in *m* für Re_ω=konst.

Die Lauflängen bis zum Stoß x_{st} haben keinen direkten Bezug zu den Grenzen der linearen Berechnung, sie sind jedoch bezüglich des Deformationsverhaltens sehr instruktiv.

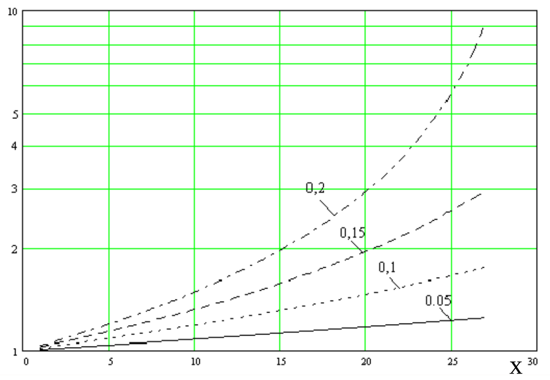
Im Folgenden zeigen die Diagramme 4a, 4b, 4c und 4d für konstante Ma-Zahlen der Amplitude von *Ma = 0,05; 0,1; 0,15; 0,2* die Änderung der normierten Steigung *tan α* an der Wellenfront über der Lauflänge der Welle.

Die Verläufe der *tan α* -Werte, welche die Veränderung der Steilheit der Wellenfront mit der Lauflänge der Welle repräsentieren, ermöglichen eine Aussage über den Deformationsverlauf einer Welle infolge ihrer Ausbreitung.

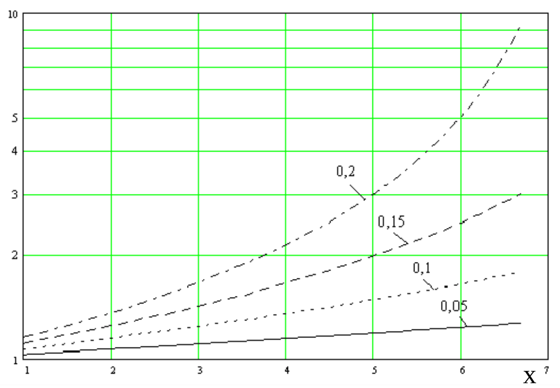
tan α



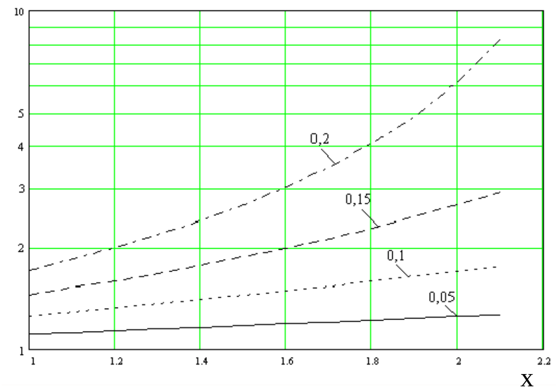
Diagr.4a: tan α (x_{st}), Ma = konst. für Re_ω=2000



Diagr.4b: tan α (x_{st}), Ma = konst. für Re_ω=8000



Diagr.4c: tan α (x_{st}), Ma = konst. für Re_ω=32000



Diagr.4d: tan α (x_{st}), Ma = konst. für Re_ω=100000

Um eine Beurteilung der normierten Steigung der Wellenfront zu erleichtern, ist im anschließenden Diagramm 5 die normierte Steigung *tan α* über dem zugehörigen normierten Winkel *α* der Tangente an der Wellenfront aufgetragen.

tan α

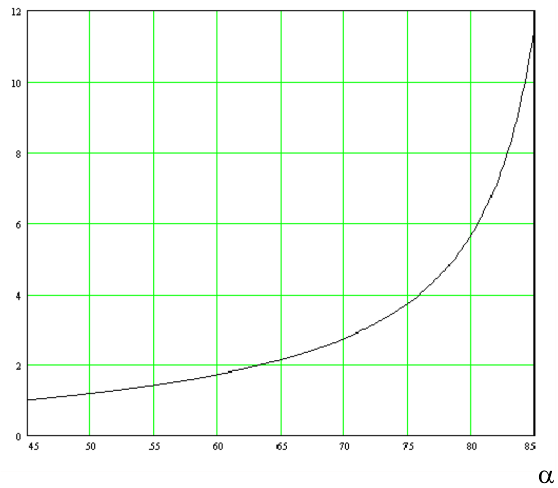


Diagramm 5: tan α über den Winkel α in grd

Aus den Diagrammen 4a bis 4d ist ersichtlich, dass die Mach-Zahl der Amplitude, die gleich der relativen Dichteänderung ist, einen wesentlichen Einfluss auf das Deformationsverhalten einer Welle besitzt.

Eine relative Dichteänderung $\Delta\rho / \rho_0 < 0{,}1$ wird in der Praxis zumeist als Kriterium für eine Anwendung linearer Berechnungsverfahren angewendet.

Die Diagramme 4a bis 4d zeigen nun auch die Einflüsse von Frequenz, Viskosität und Rohrquerschnitt, durch Re_ω dokumentiert, und den wesentlichen Einfluss der Wellenlauflänge, in der Anlage die Rohrlänge, auf das Deformationsverhalten.
Es zeigt sich hier, dass eine Begrenzung der Anwendung linearer Berechnungsverfahren nur durch die Angabe der zulässigen relativen Dichteände-

rung nicht ausreicht. Sie besitzt jedoch, hier dargestellt durch die Machschen Zahlen mit der Wellenamplitude (Diagramme 4), einen erheblichen Einfluss auf das Deformationsverhalten.

Man sollte für eine Simulation der Pulsation in Verdichteranlagen von einem vorgegebenen noch zulässigen Fehler bezüglich der Deformation der Wellenfronten ausgehen, also einen zulässigen Grenzwert für die veränderte Steigung der Wellenfront festlegen.

Aus den Verdichter- und Anlagenparametern lassen sich die für eine Deformationsanalyse erforderlichen Parameter überschlägig bestimmen. Mit dem vorstehend angeführten Verfahren kann man entsprechend den Gegebenheiten zulässige Grenzparameter festlegen, diese können sein die Kreisfrequenz der Erregung, die Mach-Zahl der Amplitude bzw. die zulässige maximale relative Dichteänderung und die Lauflänge der Welle.

Für beginnend harmonische Wellen bzw. harmonische Ersatzwellen in einem ruhenden Fluid, näherungsweise auch in einem bewegten Fluid, können die Grenzparameter mit den Diagrammen 4 bestimmt werden.

12. Zusammenfassung

Die Simulation von Gas-Pulsationen in Kolbenverdichteranlagen erfolgt zumeist mit Hilfe linearer Berechnungsverfahren, so bspw. bei PULSIM [Ega00] und bei PURO2000 [Ste99].

Die bei fluidischen Wellen auftretenden Deformationen infolge ihres Aufsteilverhaltens setzen den linearen Berechnungsverfahren bezüglich einer erzielbaren Genauigkeit Grenzen.

In dieser Arbeit wird für ausgehend harmonische Wellenformen in ruhenden und bewegten kompressiblen Fluiden das Deformationsverhalten von leitungsgeführten Wellen beschrieben.

Für beliebige Wellenformen, von denen man in der Praxis der Simulationsberechnung stets ausgehen kann, wird ein mögliches Näherungsverfahren angeführt.

Das vorstehend dargestellte Verfahren zur Untersuchung des Deformationsverhaltens von Wellen bei Gas-Pulsationen in Kolbenverdichteranlagen gibt näherungsweise Aufschluss über ein mögliches Aufsteilen von Wellen und über die zu erwartende Größe der Deformation in den Rohrleitungen einer zu berechnenden Anlage.

Für Vorgänge mit einer Wellendeformationskennzahl $\sigma \geq 1$ tritt keine Aufsteilung der Wellen auf. Hier können bedenkenlos lineare Berechnungsverfahren angewendet werden.

Für $\sigma < 1$ steilen die Wellen auf. Auch für diese Vorgänge lassen sich zumeist lineare Berechnungsverfahren anwenden, jedoch sollte man hierfür das Aufsteilverhalten der Wellen nach dem vorstehend angeführten Verfahren überprüfen, um Aufschluss über mögliche wesentliche Ungenauigkeiten zu erhalten.

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Effective Combination of On-Site Measurements and Simulations for a Reciprocating Compressor System

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Life Cycle Costs - Reciprocating Compressors in the Focus of Function, Economics and Reliability

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Abstract:

The capacity of two parallel operating BORSIG reciprocating compressors, installed at the ÖMV refinery in Vienna, had to be increased. TNO TPD was ordered to investigate the dynamic effects on compressors and pipe work due to the increased flow to avoid vibration and fatigue problems afterwards. For this purpose a pulsation and mechanical response analysis according the API standard 618[1], design approach 3, has been carried out. On-site vibration measurements have been carried out to check the dynamic behavior of the existing supports and modifications have been recommended. Moreover, the results are used in the mechanical analysis to achieve maximum accuracy. As a result of the pulsation and mechanical response study additional modifications were recommended to improve the dynamic behavior of the system.

All the recommendations have been implemented and finally on-site vibration measurements of the modified system showed that the vibrations were allowable throughout the system. With this approach costly corrective additional measures afterwards were not necessary.

1 Introduction

Two BORSIG reciprocating compressors, which are installed at the ÖMV refinery in Vienna, are operating since 1982. Due to the increasing demand of the process, the flow of the compressors had to be increased by 10%, which is achieved by means of an increase of the stroke. For this purpose a new crankshaft and electric motor is installed. It is expected that the pulsations and consequently the vibration and cyclic stress levels will increase due to the increased capacity. This can lead to vibration and fatigue problems because local vibration problems have occurred already since commissioning. Therefore, TNO TPD was ordered to investigate the dynamic behavior of the modified system to prevent vibration and fatigue problems.

For this purpose the performance of the original pulsation dampers was investigated as a first step in the project.
The second step of the analysis was the calculation of the pulsation levels and pulsation induced forces for the original piping layout. These levels have been compared with allowable levels of the API 618 Standard, 4th edition of June 1995. It appeared that the pulsation levels exceed the allowable levels. One of the solutions to decrease the pulsation levels is to install orifice plates.
However, the cost-effectiveness of an orifice needs careful consideration, particularly when operational (fuel) and/or capital investment costs (driver) are significant contributors to life cycle costs. A cost benefit analysis factored into a life cycle cost analysis could motivate the serious consideration of other solution options.

To keep all pulsation levels within the allowable API 618 limits for this system would imply too much unacceptable pressure loss. This would mean that the power of the new driver is too small, which means that the guaranteed flow and/or pressure ratio cannot be met. Therefore, it has been investigated with a mechanical response analysis if possible vibration and/or fatigue problems, due to the exceedings of the allowable API limits, will occur.

In the mechanical analysis the vibration and cyclic stress levels have been calculated and in case of exceedings additional pipe supports, and/or modification of existing pipe supports have been advised. This approach is also stated in the 3rd edition of the API 618 (note in chapter 3.9.2.7) and becomes more and more an accepted design approach[2].
The advantage of a mechanical response analysis during the design stage is that the support layout can be optimized for all possible operation conditions. This means that fatigue failure can be avoided because the dynamic stresses are also calculated. The experience of TNO TPD is that 90-95% of the vibration and fatigue problems can be solved during the design stage with this approach.

Pipe systems, which are subjected to dynamic loads, must be adequately supported by means of clamp type supports. However, many so-called rest type supports have been used in this system, which are not always suitable to restrain the dynamic loads. Therefore, field measurements have been carried out to check the boundary conditions as used in the mechanical response study. From the measurements it appeared that the measured dynamic behavior of several pipe supports were different from the assumed dynamic behavior. Therefore, it was advised to modify the layout of these supports.

After all recommended modifications of the pulsation and mechanical response study had been carried out, the system was start up at the end of 1999. Finally, field measurements were carried out on the modified system to verify the results of the simulations and it appeared that the system was operating without any vibration problems.

2 Brief description of the system

The investigated system consists of the suction 1st stage, interstage and discharge 2nd stage pipe system of two 3 Mw Borsig reciprocating compressors. The gas is a mixture of hydrogen and hydrocarbons. The compressors run at a fixed speed of 298 RPM and have two cylinders for the 1st stage and two cylinders for the 2nd stage. Each stage has a common damper for the two cylinders. The capacity is controlled by means of valve lifters from 100%, 75%, 50% to 25% load. The two compressors are connected to a common interstage pipe system.

	Load condition (%)	
	Compressor A	Compressor B
Parallel operation	100	0
	75	25
	50	50
	25	75
	0	100
Single operation	100	100
	75	75
	50	50
	25	25

Table 1: Load conditions

3. Damper check[2,3]

3.1 Introduction

The first step in the project was to check the pulsation levels and pulsation-induced forces of the existing pulsation dampers for the increased flow condition. A large amount of money will be saved when the performance of the existing pulsation dampers meets the requirements for the increased flow condition. During this check an "endless" line has replaced the pipe system or in other words a reflection free termination point.
During the damper check the exceedings of the pulsations near compressor valves can be effectively reduced by the installation of orifice plates or other damping devices at the cylinder flange. The increased power consumption of these damping devices has been calculated, optimized and discussed with the compressor vendor.

3.2 Allowable pulsation levels at line nozzle

The calculated pulsation levels at the line connection, for the endless line condition, should be lower than the allowable levels for the pipe system. This is because pulsation levels at the line connection, or anywhere else in the pipe system, will be higher in case there are resonances in the system. The presence of other compressors or cylinders should also be accounted for. An empirical guideline is to allow a maximum pulsation level of 80% of the allowable level in case of simple pipe systems. For more complex systems, e.g. parallel running machines and interstages, a reduction to 70% of the allowable level has shown to be a good criterion.

3.3 Results

From the calculations it appeared that for the existing layout the dampers sufficiently reduce pulsations. Also the pulsation-induced vibration forces are moderate (maximum 7.3 kN peak-to-peak). However, it was noted that there was a significant contribution (although allowable) of higher harmonics. Therefore, it was recommended to install orifice plates at the compressor flanges of each stage to reduce the pulsations and pulsation-induced forces with a frequency of 10-15 times the compressor speed.

4 Pulsation study

4.1 Introduction

The objective of the pulsation study is to predict the pulsation levels in the pipe system for all operating conditions. The study also covers the investigation of modifications required to reduce pulsation levels to acceptable levels. Pulsations may cause large vibration forces, malfunction or damage of the compressor valves and errors in flow metering stations. The pulsation study addresses these three aspects.
For the pulsation study a special computer program, called PULSIM[3,4] has been applied. Both compressor and pipe system have been modeled using simulation elements like compressor cylinders, pipe elements, T-branches, elbows, volumes, valves, etc. During the pulsation study special attention has been given to resonance conditions which can occur within the range of compressor speeds (and its harmonics) and the range of speed of sound in which the system can operate. The pulsation levels at these worst-case conditions have been compared with the allowable API 618 levels. Measures to eliminate or to dampen these resonances have been investigated in case the allowable limits have been exceeded.

4.2 Results

The pulsation study has been carried out for all operating conditions as indicated in table 1. It appeared that highest pulsation levels occurred at parallel operation of the compressors both at 50% load condition. The results have been summarized in table 2. Therefore, modifications have been investigated for this condition.

Stage	Compared to the allowable API level	Number of recommended orifice plates (1)
Suction 1st	2.0	1
Discharge 1st	1.8	2
Suction 2nd	2.3	3 (2)
Discharge 2nd	5.3	3

Table 2: Summary of the results of the pulsation study for the original layout

(1) This number is in additional to the recommended orifice plates in the cylinder nozzles

(2) It is also recommended to increase a part of the diameter of two relief lines

From the results of the system with the recommended orifice plates at the cylinder flanges and at the line connection, it appeared that the pulsation levels still exceed the allowable levels for some parts. As discussed earlier, additional pressure loss could be installed. However, this could mean that the power of the new driver is too small, which means that the guaranteed flow and/or pressure ratio cannot be met. Also an increase of the volume of the dampers, the installation of additional volumes, or larger pipe diameters would lead to high costs. Therefore, it is investigated with a mechanical response analysis if vibration and/or fatigue problems, due to the exceedings of the allowable API limits, will occur. This approach is also stated in the 3rd edition of the API 618 (note in chapter 3.9.2.7) and becomes a more accepted design approach.

5 Mechanical response study

5.1 Introduction

One of the objectives of the pulsation study is to reduce the pulsation induced vibration forces to a minimum. However, unallowable vibrations and cyclic stresses can occur in case a mechanical natural frequency is close to or coincides with a frequency component of the pulsation induced vibration forces, even in case the pulsation levels itself are below the allowable level. The objective of the mechanical response study is therefore to check the design of the pipe system, including the pipe supports and the construction on which the supports are mounted, to make sure that the vibration levels and the cyclic stresses are lower than the allowable levels.
The advantage of a mechanical response analysis during the design stage is that the support layout can be optimized for all possible operation conditions. This means that fatigue failure can be avoided because the dynamic stresses are also calculated. The experience of TNO TPD is that 90-95% of the vibration and fatigue problems can be solved during the design stage with this approach.

Special attention is paid to local flexible parts[2,7] such nozzle/shell intersections, compressor flange and local flexible steel structures. The mechanical response study has been carried out according to the API Standard 618, design approach 3 after the customer has decided which of the recommended modifications of the pulsation study will be implemented. In the mechanical response analysis the following calculations have been carried out:

- Mechanical natural frequencies and mode shapes;
- Forced response due to pulsation induced pulsation forces.

For the above mentioned calculations the general-purpose finite element program ANSYS[5] is used.
The pipe system is divided into basic parts (called finite elements) such as straight pipe sections, elbows, T-joints, flanges, reducers, constructions on which pipe supports are mounted, etc.

The pulsation forces have been calculated with the PULSIM[3,4] program for the pipe system with the finally accepted modifications. When the vibration and cyclic stress levels exceed the allowable levels, modifications have been investigated to decrease the levels to allowable levels. This can be achieved by shifting the natural frequencies far enough from the excitation frequency. Shifting the natural frequency was achieved by the installation of extra pipe supports and by increasing the stiffness of the structures on which the supports are mounted.

5.2 The general approach for this project

Pipe systems, which are subjected to vibrations, must be adequately supported by means of so-called clamp type supports of which an example is shown in figure 1.

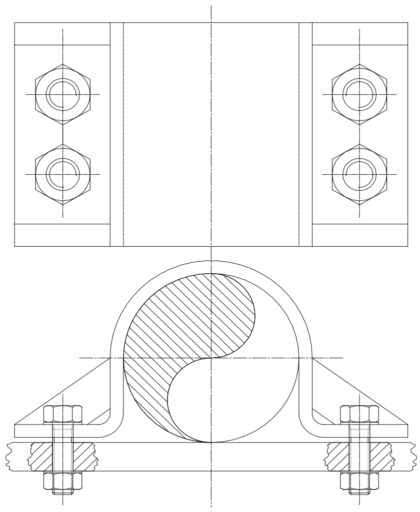

Figure 1: Example of a clamp type support

However, many pipe systems are optimized from the thermal and not from the dynamic point of view. This means that so called guide and rest type supports are frequently used. A photograph of a typical rest type support, as used in this system, is shown in figure 2.

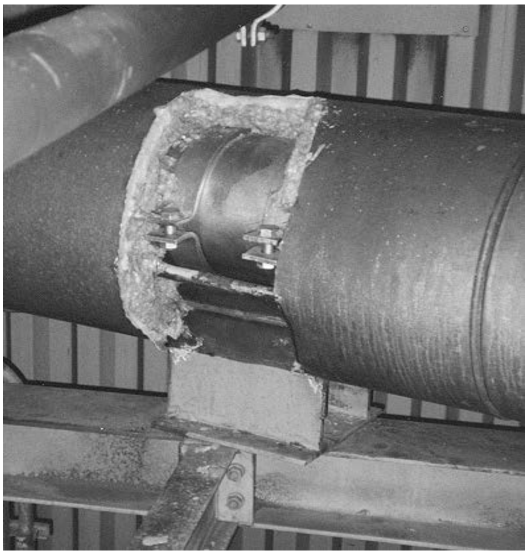

Figure 2: Photograph of a rest type support

However, for these supports it is difficult to determine the dynamic restraint directions. This is also due to the fact that these supports are non-linear, which is an additional complicated factor because the harmonic response analysis is carried out in the frequency domain, which is only applicable for linear systems. To judge if a guide and/or rest type support restrains a dynamic motion, the definition "dynamically fixed" has been introduced as follows:

A pipe is "dynamically" fixed for one or more translation and/or rotation directions when the construction, which holds the pipe to the structure, is able to withstand the dynamic loads.

A dynamically fixed support can be achieved by means of stiff enough pipe support construction, which does not allow a dynamic movement and/or rotation of the pipe in a certain direction. This means that no clearance between the pipe and the support construction is allowed. This can be achieved by the installation of a clamp type support, of which an example is shown in figure 1.

Another possibility to achieve a dynamic restraint is to ensure that the friction force (caused by the weight) between the pipe and the pipe support structure is higher than the pulsation-induced reaction force.
Most of the times the static loads are much higher than the pulsation-induced forces. Therefore, with this approach it can be achieved that the support is both dynamically fixed and statically loose.
However, in most of the times the static loads are not known during the mechanical (dynamical) response study and it is also possible that the calculated static loads differ from the static loads in the field. This is also one of the reasons why rigid pipe clamps are recommended for systems which are due to dynamic loads.

Pipe supports drawings and the results of the static (thermal) stress calculation were not available for the pipe system of this project. Therefore, it has been assumed in the mechanical analysis that the rest type and guide type supports are dynamically fixed in the three translation directions. These assumptions have been verified later on during field measurements (see also chapter 5.5) because the accuracy of the calculated mechanical natural frequencies, and consequently the vibrations and cyclic stresses, strongly depends on the boundary conditions[2].
Structures, on which pipe supports are mounted, such as dummy legs, single beams, A-type frames and small and large pipe racks have been used to support the weight of the pipe system and pulsation dampers. Most of these structures have been designed only for static loads and have a relatively small dynamic stiffness. Neglecting the mass and flexibility of such structures can lead to an overestimate of the natural frequency, by as much as 50 percent[2]. Therefore, these local flexible structures have been included directly in the mechanical model.
Larger structures are normally only included in the model when the stiffness (in comparison with the pipe stiffness) is too small. Drawings of these large structures were not available and therefore they have not been included in the mechanical model for this project.
However, field measurements have been carried to determine if the stiffness of these structures was sufficient. To determine which supports are dynamically fixed and which large structures are too flexible the following approach has been applied:

- In case the vibration level in a certain translation direction is lower than 5 mm/s RMS (Root Mean Square) it has been assumed that the support is dynamically fixed;

- In case the vibration level of the piping exceeds 5 mm/s RMS, additional measurements have been carried out on the structure on which the pipe supports have been mounted. There are the following possibilities:

    1. The vibration level is below 5 mm/s RMS. This means that the friction force between the pipe and the underlying construction is not sufficient to restrain the dynamic motion. For this case it is recommended to modify the layout of the pipe support in such a way that it is able to withstand the dynamic motions. This can be achieved by the installation of clamp (see figure 1) or so

called hold down type supports (see figure 3 and 4);

2. The vibration level is above 5 mm/s RMS:
   a) In case both vibration levels of the structure and piping differ less than 5 mm/s RMS it is assumed that the support is dynamically fixed on the structure. In case both measured vibration levels are within the allowable level of 15 mm/s RMS no additional measures are necessary. In case the allowable level is exceeded it is recommended to modify the layout of the pipe support in the same way as described in 1;

   b) In case both vibration levels differ more than 5 mm/s RMS it is assumed that the friction force is not sufficient to restrain the dynamic motion (dynamically loose). For this case it is recommended to increase the friction force between the pipe and the structure on which the support is mounted. This can be achieved by the installation of so-called hold down type supports (see 5.2.1). In case the vibration level of the structures on which the support is mounted exceeds the allowable level of 15 mm/s RMS it is also recommended to increase the stiffness of the structures.

However, for this system it appeared that clamp type support could not be installed because of the too high thermal stresses. Therefore, hold down type supports have been applied.

5.2.1 Hold down type supports

A hold down type support is able to restrain the dynamic motions by means of a friction force and besides it allows thermal motions if designed well. The dynamic restraint is achieved by applying a friction force between the support and the structure on which the support is mounted. The friction force F_w must be at minimum the pulsation induced reaction force on the support. The minimum required preload F_n in the bolts is calculated according equation (1). The thermal motion is achieved by means of applying a clearance between the bolds and the plate of the support (e.g pipe shoe). Examples of hold down type supports are shown in figures 3 and 4. The spring hold down type of figure 4 has to be applied when a thermal motion in the vertical direction must be possible. A disadvantage of these supports is that it requires more maintenance because the preload must be maintained.

$$Fn \geq \frac{Fw}{f} \quad [N] \qquad (1) \qquad (1)$$

In which:
F_n = minimum required preload in the bold [N]
F_w = pulsation-induced force reaction force on the support [N]
f = friction coefficient, which depends on the applied materials (for steel-steel contact normally a value of 0.3 is used)

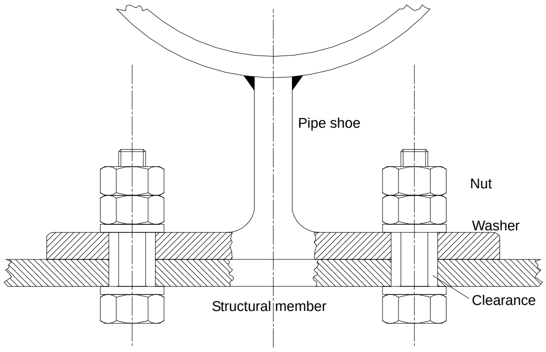

Figure 3: Example of a hold down type support which does not allow thermal motion in the vertical direction

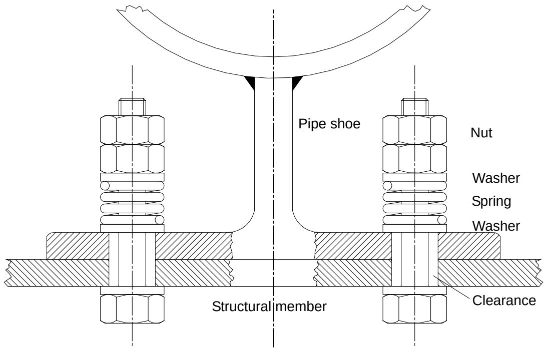

Figure 4: Example of a spring hold down type support which allows thermal motion in the vertical direction

5.3 Allowable levels for the mechanical response study

5.3.1 Allowable cyclic stress

An allowable cyclic stress according the API[1] Standard 618 of 179 N/mm^2 peak-to-peak have been applied, which is based on the endurance limit of the material. This value is only valid for carbon steel pipe with an operating temperature below 371ºC. All other stresses must be within the applicable code limits.

5.3.2 Allowable vibration levels

To make sure that equipment such as temperature transmitters, pressure gauges, valves, flanges etc. will not fail due to too high vibrations, the vibration levels have also been calculated. During the analysis the vibration levels have been calculated for the worst-case conditions, which means that the acoustical and mechanical natural frequencies coincide. This is a conservative approach and in the calculations an allowable level of 30 mm/s RMS has been applied therefore, which is higher than the applied allowable level of 15 mm/s RMS during field measurements. This is a safe approach because the maximum cyclic stress, which is the most critical parameter, is always calculated during the mechanical response analysis.

5.4 Results of the mechanical response study

From the results of the pulsation study it appeared that the highest pulsation levels occurred at parallel operation of the compressors both at 50% load condition. Therefore, the mechanical response analysis has been carried out for this condition. In table 2 the calculated maximum vibration and cyclic stress levels have been summarized. It can be concluded that the maximum calculated vibration levels for the assumed (and later on checked) restraints of the supports still exceed the allowable level of 80 mm/s peak-to-peak for each stage. Fatigue failure can occur in the discharge 2nd stage because the calculated maximum cyclic stress exceeds the allowable level of 179 N/mm^2 peak-to-peak. To decrease the vibration and cyclic stress levels a combination of additional pipe supports and modifications of the layout of the existing pipe supports have been recommended, which have been summarized in table 3. For the suction 2nd stage it appeared that an additional pipe support was necessary at a location where installation was not possible. Therefore, it was finally advised to install additional pressure loss at the line connection of this damper to reduce the pulsation-induced forces throughout this part of the system.

	Vibrations	Cyclic stress
Stage	Compared to the allowable level	Compared to the allowable level
Suction 1st	1.20	0.47
Discharge 1st	2.40	0.23
Suction 2nd	1.93	0.47
Discharge 2nd	16.53	2.73

Table 2: Summary of the calculated maximum vibration and cyclic stress levels

Stage	Number of additional supports	Number of modified supports
Suction 1st	1	-
Discharge 1st	2	2
Suction 2nd	-	3
Discharge 2nd	3	2

Table 3: Summary of the recommended modifications

5.4.1 Example of one of the investigated mechanical models

For the calculations of the suction 2nd stage it appeared that the calculated maximum vibration level occurred at the bypass section at a frequency of 6.1 Hz. A photograph of this part is shown in figure 6. The mechanical model of this part is shown in figure 5 and the accompanying mode shape at 6.1 Hz is shown in figure 6. From the mode shape it can be seen that the maximum deflection occurs in the Z direction. From the measurements it also appeared that the maximum levels occurred at this location. However, the measured vibration level was above the allowable level but approximately a factor 3 smaller than the calculated level. This is due to the following facts:

- In the mechanical response analysis the worst-case situations are calculated, which do not necessary occur during the field measurements;
- The measurements have been carried out on the original system with a lower flow and consequently lower pulsation and vibration levels.

From the measurements and the calculations it was recommended to increase the stiffness of the supporting steel structures, as shown in figure 6. From the calculations and measurements it appeared that the maximum vibration level at this section was decreased to acceptable levels by this measure.

Figure 6: Photograph of the bypass section

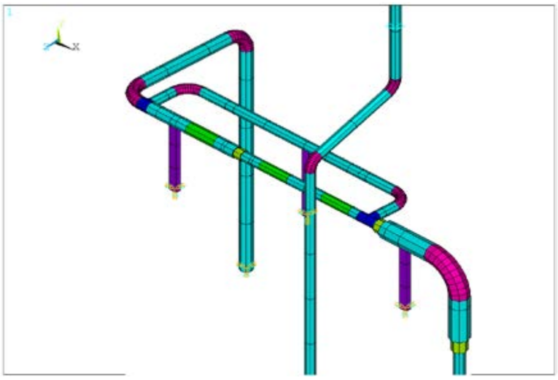

Figure 6: Plot of the model of the bypass section

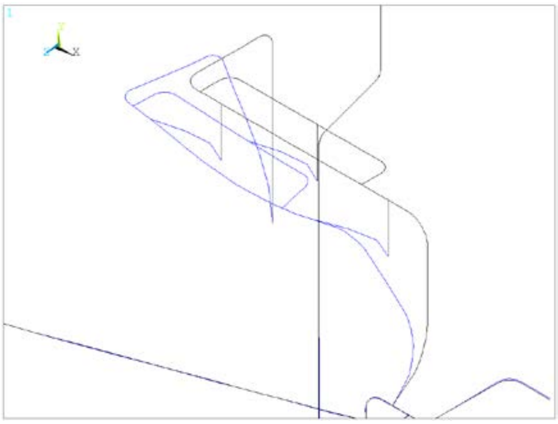

Figure 7: Mode shape of 6.1 Hz

5.5 Results of first measurements

As indicated in 5.2 the first measurements have been carried out to investigate the actual boundary conditions, and to investigate if corrective measures were necessary. Moreover, the measurement results have been used in the mechanical analysis. From the results of the pulsation and mechanical response study it appeared that the highest pulsation levels occurred at parallel operation of the compressors both at 50% load condition. Therefore, the measurements have been carried out at this condition. table 4 gives a summary with the advised modifications after the first measurements were carried out to achieve dynamically fixed points.

Stage	Number of increased structure stiffness	Number of supports with increased friction
Suction 1st	1	1
Discharge 1st	5	8
Suction 2nd	2	3
Discharge 2nd	7	7

Table 4: Advised modifications after the first measurements

6 Results of second measurements

After the recommended acoustical and mechanical modifications were carried out (results of the pulsation and mechanical response study and first measurements), a second field survey has been carried in December 1999 for the system with the increased capacity. From visual inspection it appeared that all recommendations of the pulsation study, mechanical response study and first vibration measurements were carried out properly. From the vibration measurements it appeared that the vibration levels were within the allowable level.

7 Summary and Conclusions

The damper check has shown that the volume was sufficient and that the pulsation–induced vibration forces were acceptable. However, it appeared that some high frequency components were found in the pulsations. Therefore, it was recommended to install orifice plates at the cylinder flanges to avoid that pulsations of these frequencies would lead to high pulsations and vibrations in the pipe system.

In the pulsation study of the original layout, exceedings of the API levels occurred. Therefore, orifice plates were advised to decrease the levels. To keep all pulsation levels within the allowable API 618 limits would imply too much unacceptable pressure loss for this system. This would mean that the power of the new driver is too small, which means that the guaranteed flow and/or pressure ratio cannot be met. The installation of more volume in the system would also lead to high costs. Therefore, it has been investigated with a mechanical response analysis if possible vibration and/or fatigue problems, due to the exceedings of the allowable API limits, will occur.

The advantage of a mechanical response analysis during the design stage is that the support layout can be optimized for all possible operation conditions. This means that fatigue failure can be avoided because the dynamic stresses are also calculated in the mechanical response analysis. The experience of TNO TPD is that 90-95% of the vibration and fatigue problems can be avoided during the design stage with a pulsation and mechanical response analysis. With additional field measurements the success rate can even be increased.

Pipe systems, which are subjected to dynamic loads, must be adequately supported by means of clamp type supports. However, many so-called rest type supports have been used in this system, which are not always suitable to restrain the dynamic loads. Therefore, field measurements have been carried out to check the boundary conditions as used in the mechanical response study. From the measurements it appeared that the measured dynamic behavior of several pipe supports were different from the assumed dynamic behavior. Therefore, it was advised to modify the layout of these supports.

From the mechanical response study it appeared that the vibrations are decreased to acceptable levels with some additional pipe supports and an increase of the stiffness of some steel structures. However, for the suction 2nd stage it appeared that an additional pipe support was necessary at a location where installation was not possible. Therefore, it was finally advised to install additional pressure loss at the line connection of the damper.

After all recommended modifications of the pulsation and mechanical response study have been carried out, the system was start up at the end of 1999. Finally, field measurements were carried out on the modified system to verify the results of the simulations and it appeared that the system was operating without any vibration problems.

This project has shown that for existing systems a combination of simulations and additional field measurements, of which the results are used in the mechanical analysis, are effective to optimize the system from the dynamic point of view to prevent vibration and fatigue problems. With this approach costly corrective additional measures afterwards were not necessary.

8 Acknowledgements

The authors would like to thank ÖMV and BORSIG for their permission to use the results of this project.

9 References

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