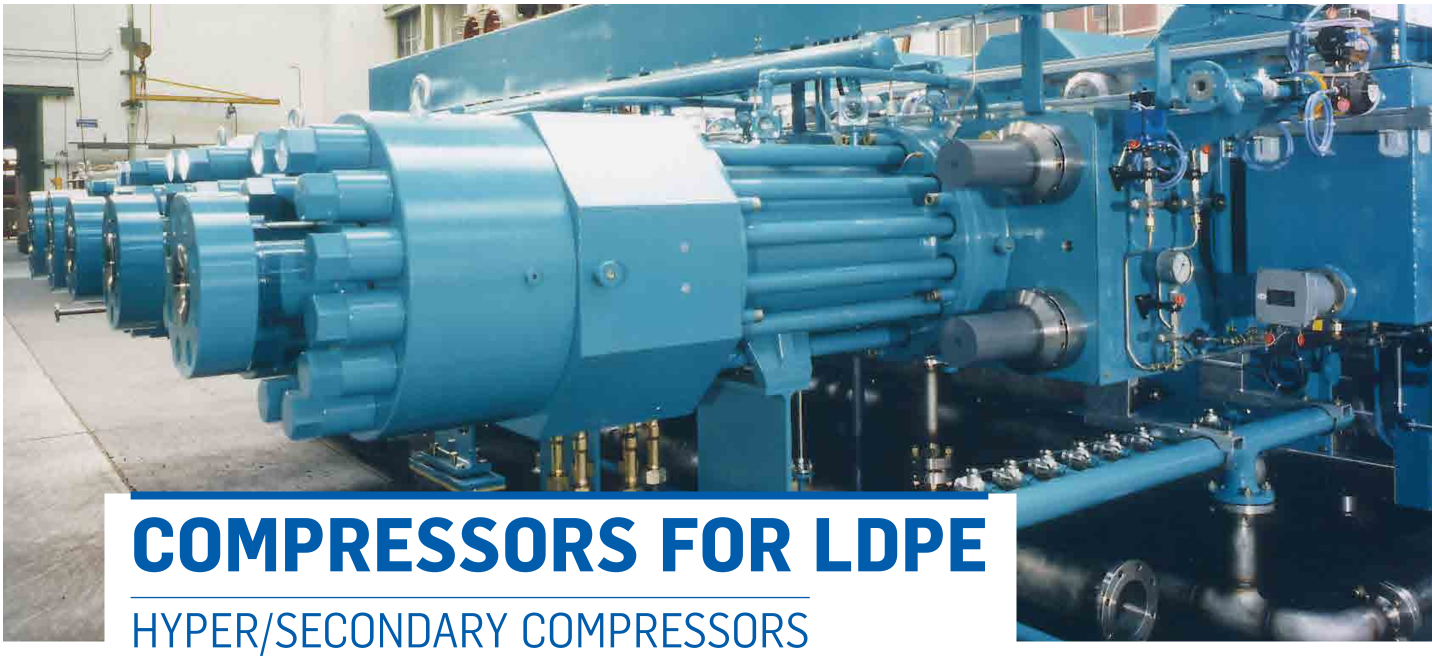
Burckhardt Compression



COMPRESSORS FOR LDPE

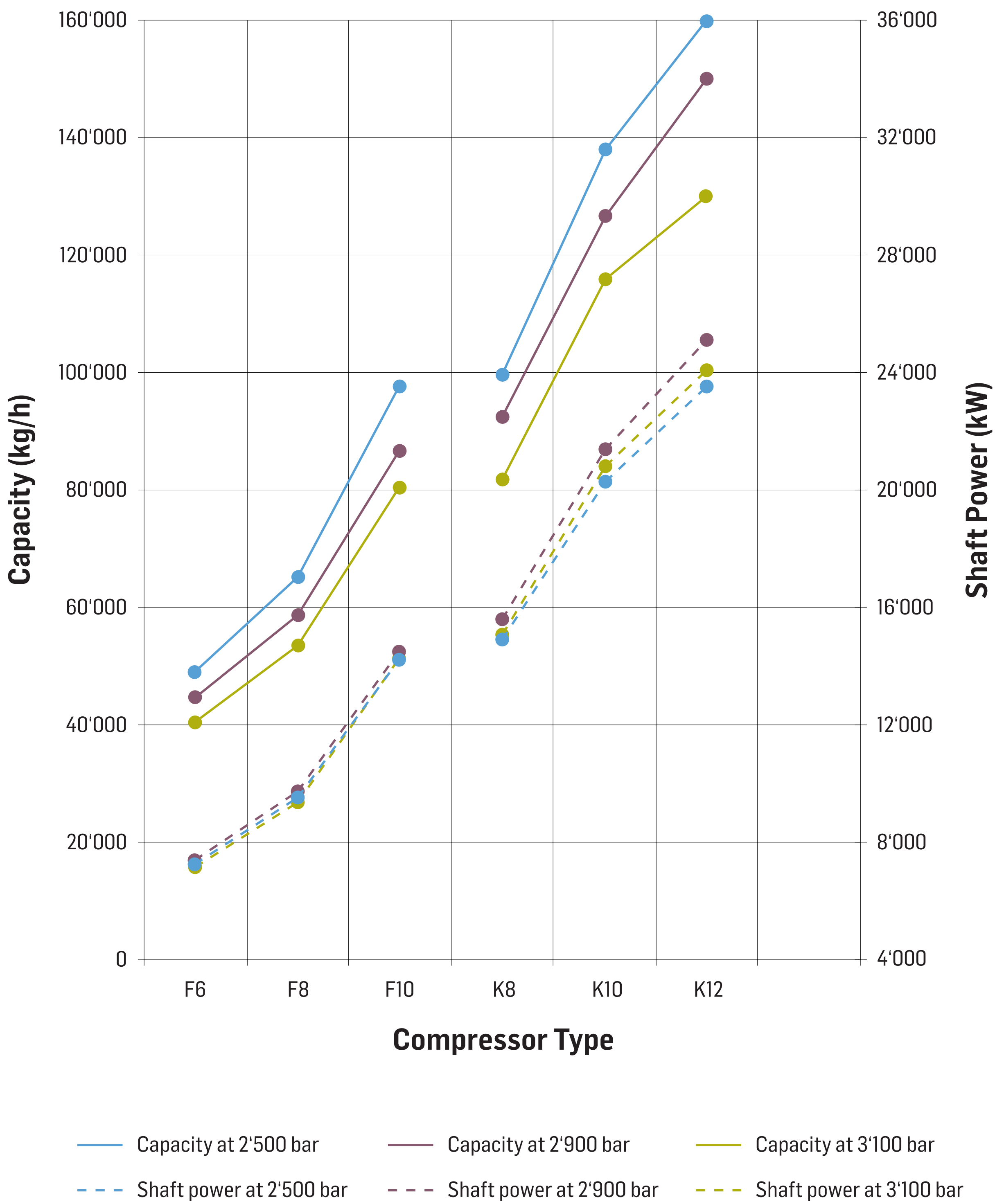
HYPER/SECONDARY COMPRESSORS

Compressors for a Lifetime™

THE WORLD'S LARGEST SINGLE FRAME RECIPROCATING COMPRESSOR

Burckhardt Compression Hyper/Secondary Compressor Type K12
Reliance Industries (LDPE), Jamnagar, India
Motor power: 30'000 kW

CAPACITY AND SHAFT POWER
HYPER/SECONDARY COMPRESSORS



TECHNOLOGY AT ITS BEST
SAFETY AND RELIABILITY

KEY COMPRESSOR COMPONENTS – FOR BEST PERFORMANCE AND LONGEST LIFETIME

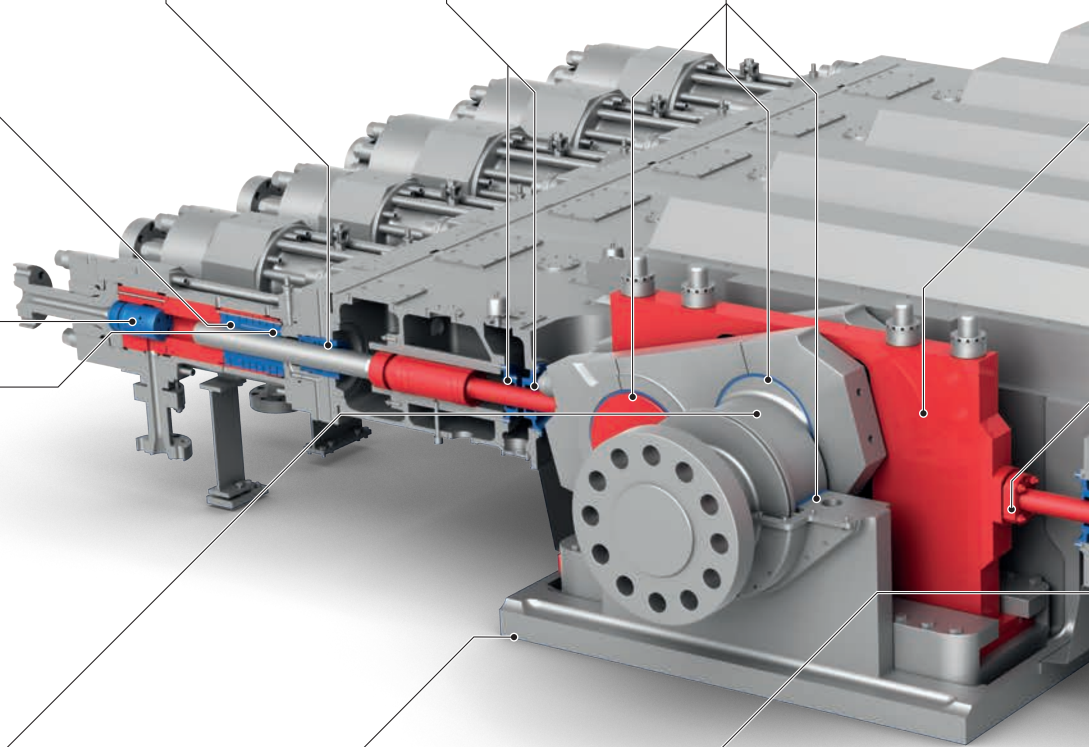
CENTRAL VALVES
– Combined suction/discharge multi-poppet valve for compact design
– In full metal execution to withstand wear for long durability
– Autofrettage treated to withstand high pressures
– Optimized flow for best performance and long lifetime

HIGH-PRESSURE PACKINGS
– Applied technology based on decades of experience
– Extremely high sealing efficiency and lifetime
– Different materials and designs for various copolymer applications

LOW-PRESSURE PACKINGS
– Provides extra safety
– Avoids flow of ethylene into the distance piece

OIL SEALS
– Separate seals on crankgear and distance piece side
– Self-centering design

BEARINGS
– Large diameters for low specific load
– Expected lifetime > 15 years
– Tri-metal galvanic layers for increased wear protection



HIGH-PRESSURE PACKING CUPS
– Special design to withstand fretting
– Surface treated to avoid fissures

RUGGED DESIGN – FOR DURABILITY

INSTRUMENTATION (NOT ILLUSTRATED)
– Latest technology implemented
– For safe-guarding (protective circuits) and condition monitoring

CRANKSHAFT
– With camshaft design
– Without webs
– Without bores
– Rigid against torsion and bending

CRANKGEAR
– Three types to minimize cylinders in operation
– Robust design with low specific loads
– Welded construction for K type
– Cast construction for H and F types
– Separate oil tank, no oil sump

LARGE CROSS SECTIONS OF COOLING/FLUSHING PIPING
– Extra safety in case of a plunger failure

IN-HOUSE DESIGNED AND MANUFACTURED MAIN PARTS – FOR RELIABILITY

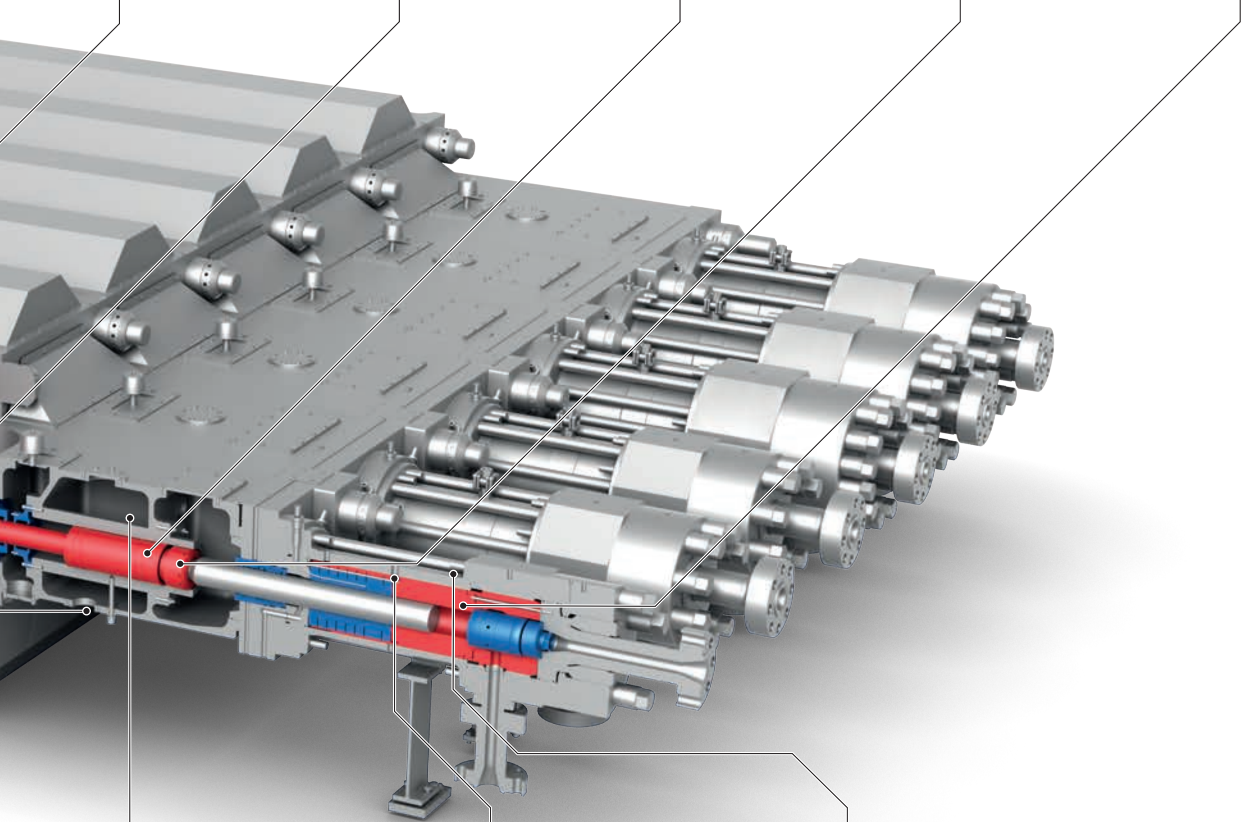
CROSSHEAD FRAMES
- Bottom guidance for easy service access
- Massive construction for low vibration level

ELASTIC ROD COUPLINGS
- Low tolerance connection for excellent guidance of the auxiliary guide and plunger

AUXILIARY GUIDES
- With elastic rod
- To eliminate undesired plunger movement
- For exact plunger alignment

PLUNGER COUPLINGS
- Simple taper design to avoid tractive forces
- For less adjustment work and easy access

BURCKHARDT HYPROPACK™ – CARTRIDGE SYSTEM
- Pre-assembled module including base cup, high pressure cups, cups for the guide ring and the throttle ring, shrunk liner (cylinder) and head core (valve seat)
- For fast exchange and service



DISTANCE PIECES
- To prevent ethylene from entering the crankgear
- For increased safety
- Double-compartment design
- No explosion relief valve required
- Vented interspace for extra safety

SAFETY DEVICES
- Rupture disk provides extra safety in case of leakage of high pressure cups

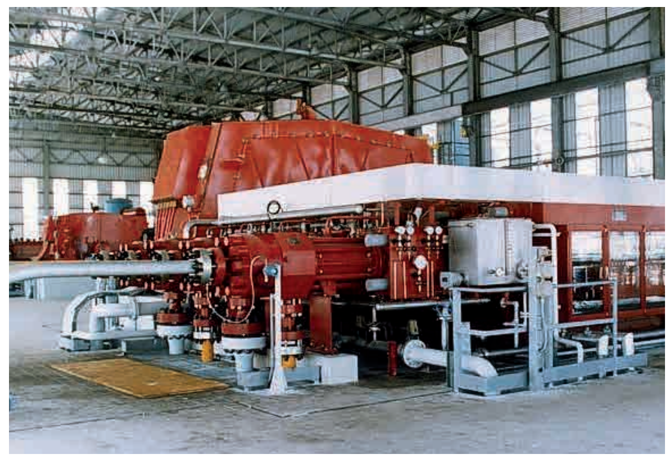
CYLINDER TIE BOLTS
- Extra long and elastic design
- Allow safe gas decomposition

COMPRESSOR SPECIFICATIONS
FOR INCREASED RELIABILITY AND AVAILABILITY

THREE DIFFERENT FRAME SIZES TO MINIMIZE CYLINDERS IN OPERATION

To cover the increasing demand for higher capacities in today's LDPE plants, three different frame sizes with progressively larger cylinder diameters, longer strokes and higher admissible frame power are available. This minimizes the number of cylinders in operation for maximum availability of the client's plant.

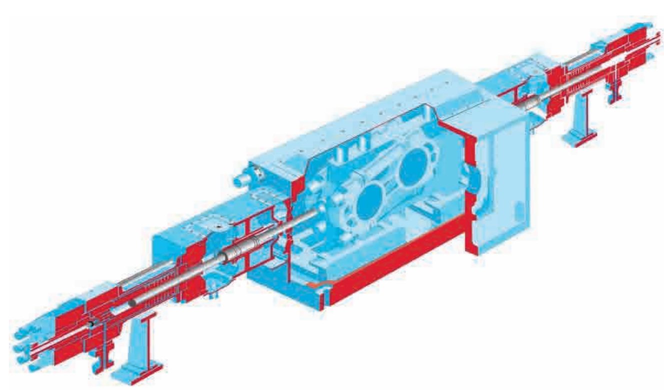
FRAME TYPE	H	F	K
Maximum number of cylinders	8	10	14
Maximum gas load (kN)	1'100	2'000	3'500
Maximum rod load (kN)	900	1'750	2'600
Maximum stroke (mm)	360	410	450
Maximum speed (rpm)	257	231	215
Maximum frame power (kW)	8'000	20'000	38'000



SIGNIFICANTLY LOWER SPEED

Hyper Compressors from Burckhardt Compression are normally operating at a low nominal speed. The low nominal speed ensures minimum wear and tear of moving parts and therefore maximizes reliability and availability.

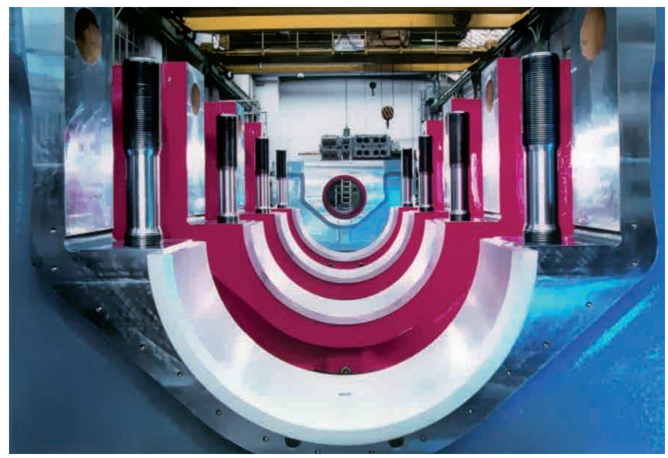
FRAME TYPE	H	F	K
Maximum number of cylinders	8	10	14
Maximum gas load (kN)	1'100	2'000	3'500
Maximum rod load (kN)	900	1'750	2'600
Maximum stroke (mm)	360	410	450
Nominal speed (rpm)	200	200	200
Maximum speed (rpm)	257	231	215
Maximum frame power (kW)	8'000	20'000	38'000



COMPRESSOR FEATURES
OPTIMIZED CRANKCASE AND CRANKSHAFT

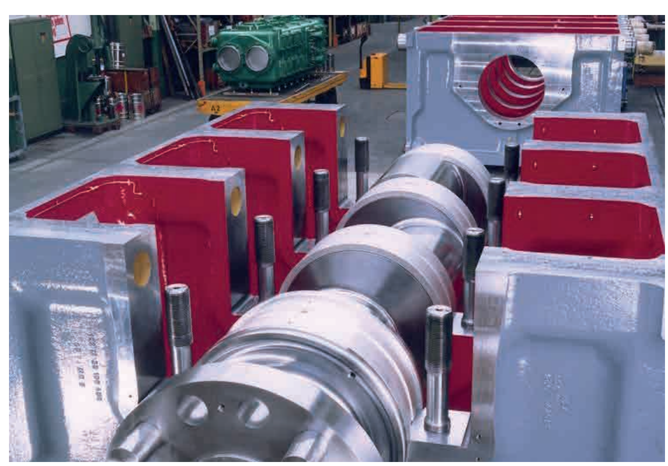
ROBUST CRANKCASE DESIGN FOR RELIABLE PERFORMANCE

The crankcase is either made from cast steel (type F, H) or welded steel (type K). Our state-of-the-art laser measurement system guarantees highest quality standards and confirms a very robust design of the crankcase.



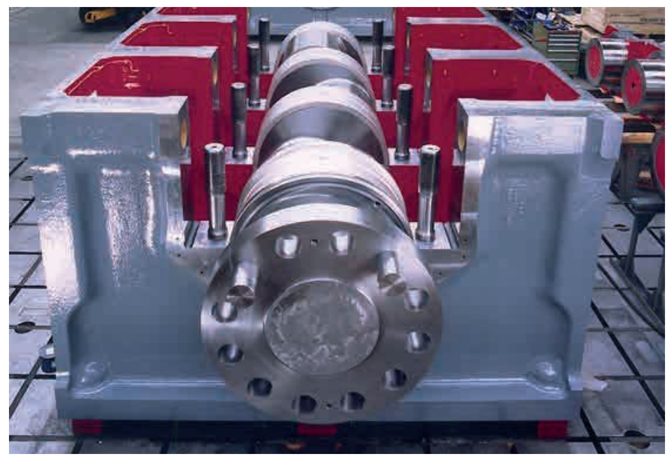
NO OIL SUMP IN THE CRANKCASE

Because the oil is stored in a separate tank in secure distance of the compressor, there is practically no risk of oil mist explosion or fire in the crankcase caused by overheated bearings. Therefore, an explosion relief valve for the crankcase is not required.



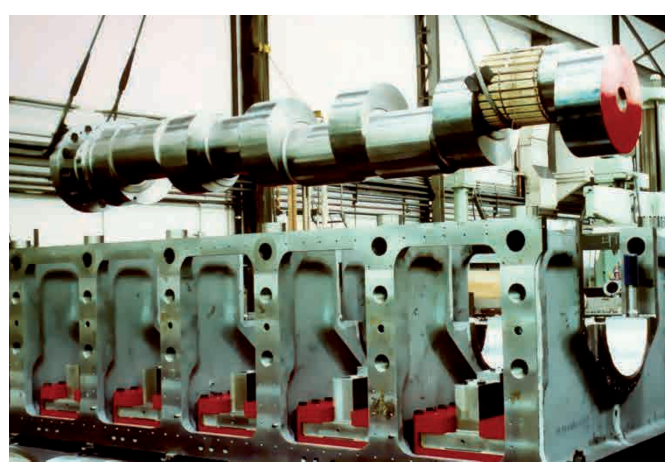
LARGE BEARING DIAMETERS FOR LOW SPECIFIC LOADING

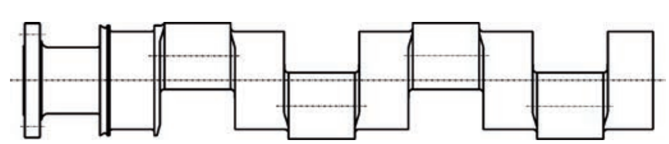
The large bearing diameters create very low specific loadings of the bearings. In particular, the most critical crosshead pin bearing – moving only in a very small angle – is generously sized. The bearing with a tri-metal galvanic flash layer (between 1.0 and 2.5 mm) are insensitive to oil impurities. According to our experience, the mean time between overhaul of these bearings is longer than 15 years (100'000 hours).



CRANKSHAFT WITHOUT WEBS AND INNER HOLES

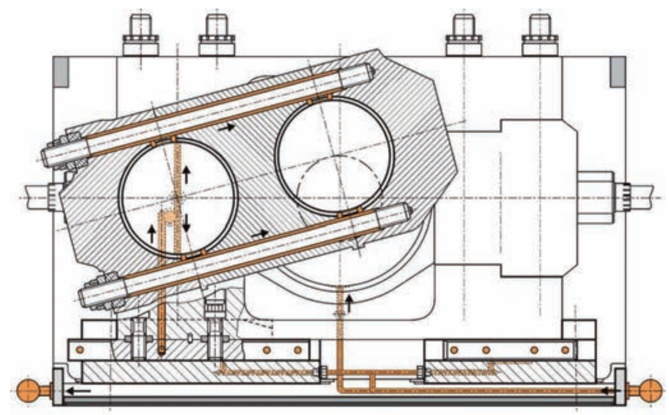
The crankshaft is looking like a camshaft, there are no webs between the bearing areas. The lubrication is effected from outside the crankshaft, there are no lubricating holes inside the crankshaft. Due to this construction the shaft is very rigid against torsion and bending.





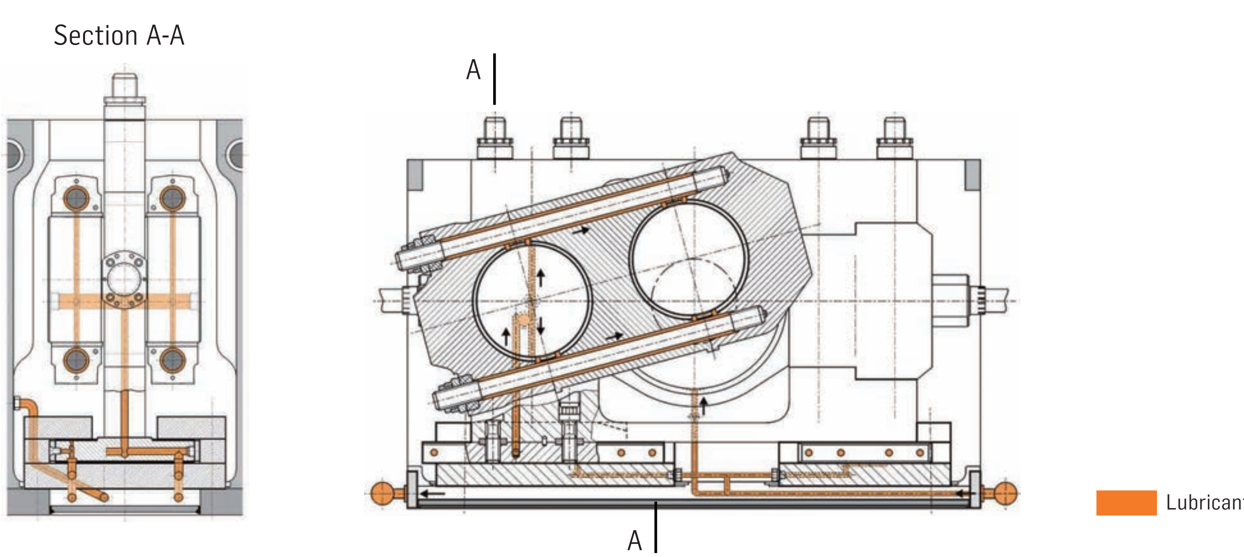
RIGID CONNECTING ROD FOR OPTIMUM PRESSURE DISTRIBUTION

The connecting rod has a massive construction for smooth power transmission on a low vibration level. It is made from cast steel and offers a 3-piece design for easy bearing replacement. The hydraulic tensioning connection provides optimal tensioning to the crankshaft and the crosshead frame bearing avoids bearing fretting. Lubrication bore connections for bearing lubrication oil feed are also integrated.



CRANKCASE LUBRICATION SYSTEM

Lubrication of main, crank and crosshead pin bearing.



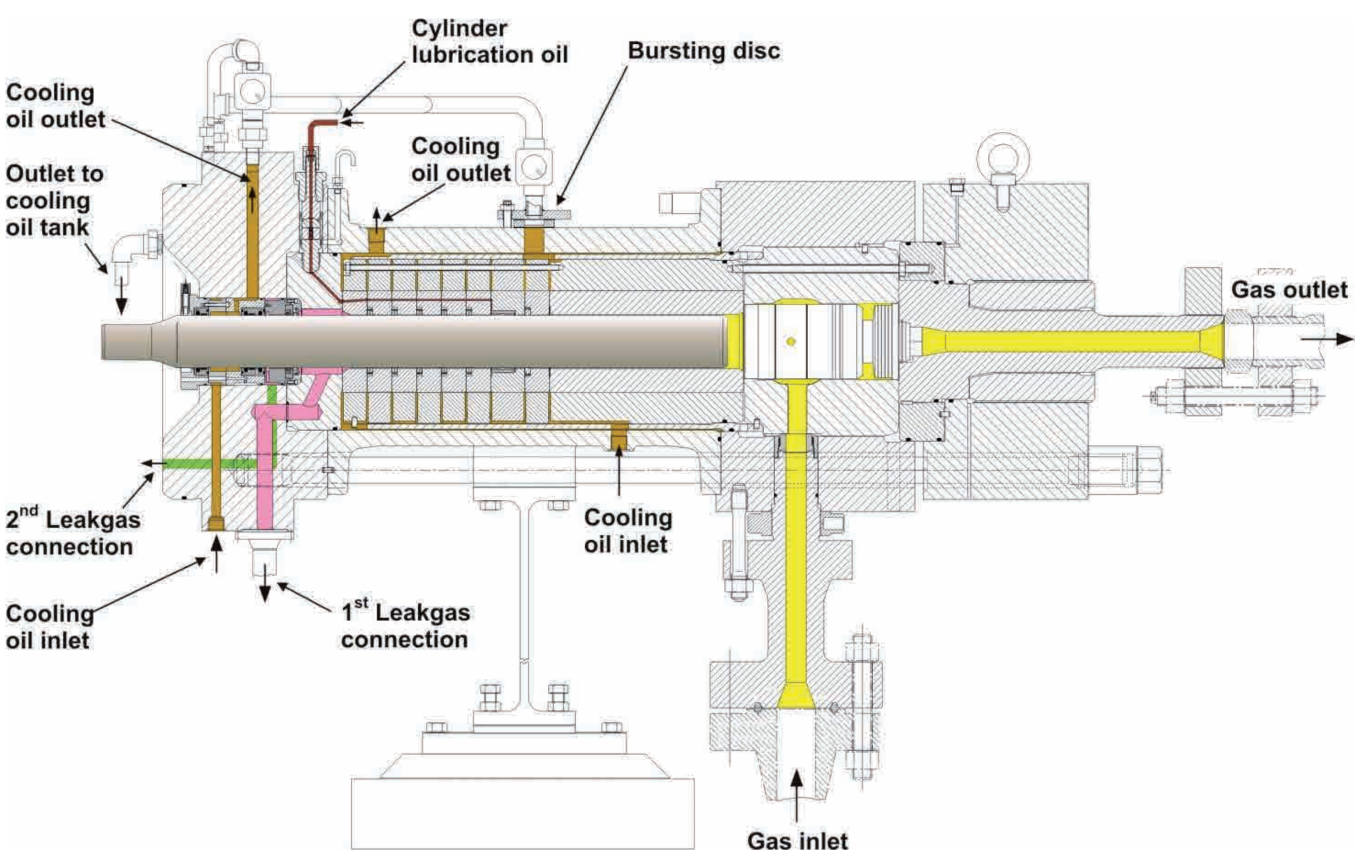
COMPRESSOR FEATURES
CYLINDER LAYOUT FOR HIGHEST DEMANDS

IMPLEMENTED FEATURES FOR EASY AND FAST MAINTENANCE

Its hydraulic tightening ensures even distribution of forces and it has extra long cylinder tie bolts as well as an elastic design as a safety feature in case of excessive pressure inside the cylinder.

BURSTING DISC ON CYLINDER BODY WITH LEAK GAS EMERGENCY PIPING

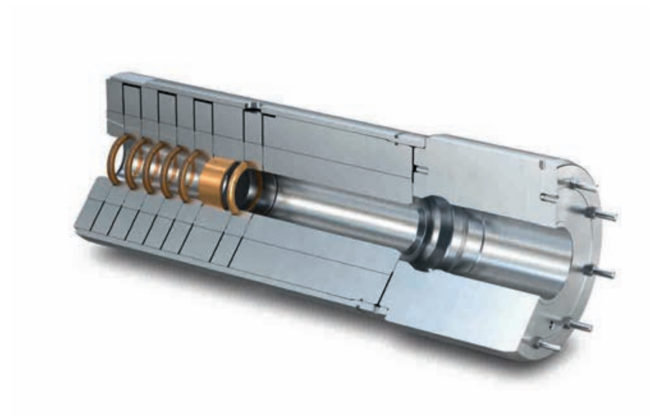
In case of a high pressure packing cup failure, leak gas enters the cooling oil system. The bursting disc, which is set to 16 bar g, allows the leak gas in the cylinder body to escape through the emergency leak gas piping into the distance piece.



COMPRESSOR FEATURES
HIGH-PRESSURE PACKING

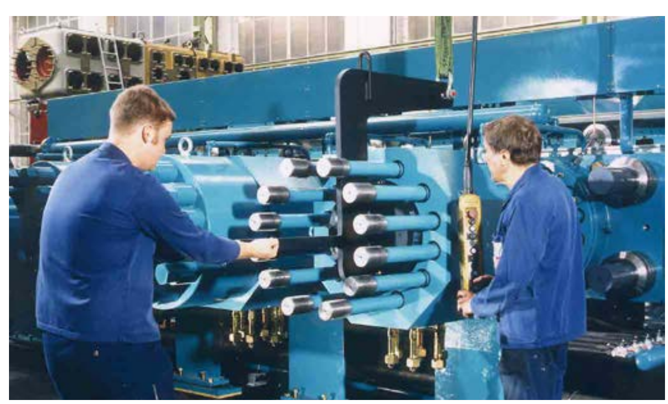
SPECIAL SHAPE OF HIGH PRESSURE INNER PACKING CUPS

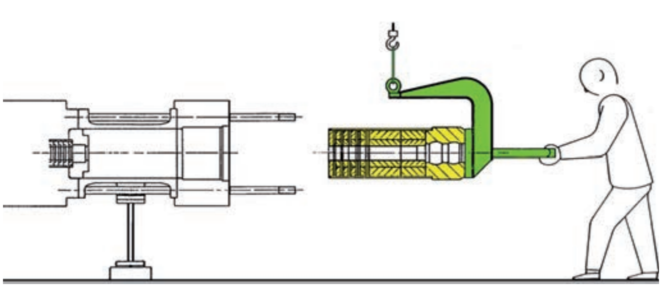
The high pressure packing chambers are manufactured on state-of-the-art CNC machines. A special shape of the inner cups avoids fretting of the surfaces. In addition, a special material surface strengthening procedure is used to avoid fissures. Highly exposed areas are also treated using autofrettage.

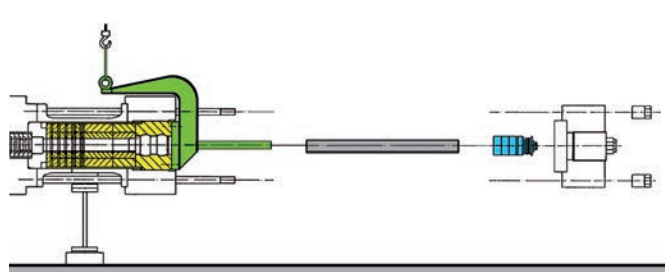


BURCKHARDT HYPROPACK™ – FAST CYLINDER EXCHANGE AND SERVICE

The pre-assembled unit containing the inner parts of the high pressure cylinder, called cartridge, can be exchanged and replaced in a short time by a spare cartridge. This results in a shorter down-time of the plant as the client can change the ring equipment of the high pressure packing, while the compressor is running with the spare cartridge.

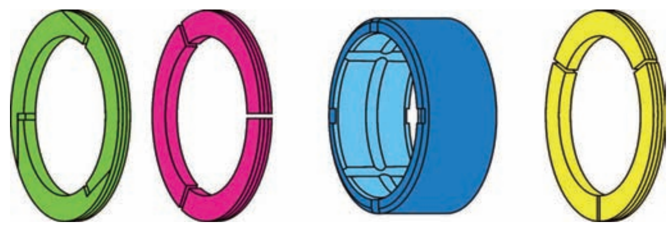


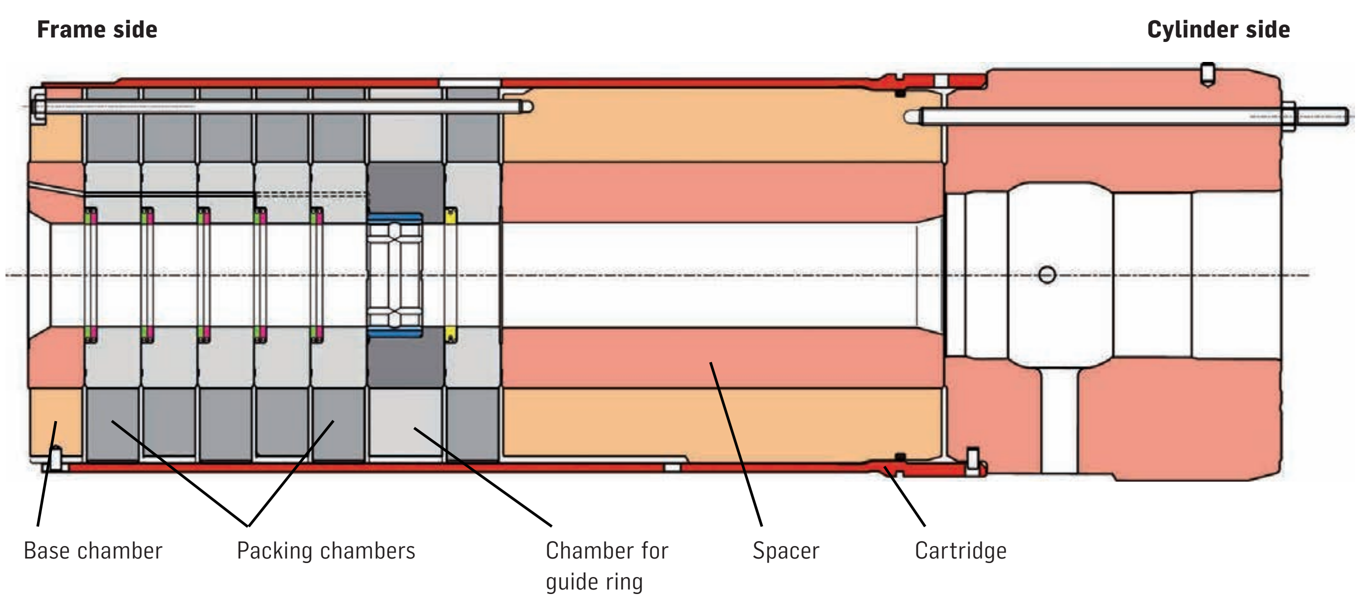




BURCKHARDT HYPROPACK™ – PACKING AND RING DESIGN FOR INCREASED SAFETY

In case of a sudden complete failure of all sealing elements of one cylinder, a very high ethylene gas flow, in the worst case the complete compressor capacity, is able to release through the ring gap between plunger and packing chambers into the leak gas system. On the assumption that the gas will expand under adiabatic conditions with a pressure of max. 40 bar.





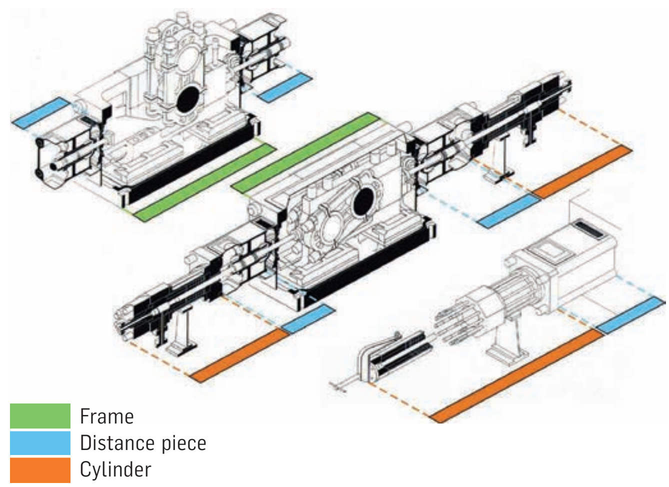
COMPRESSOR FEATURES

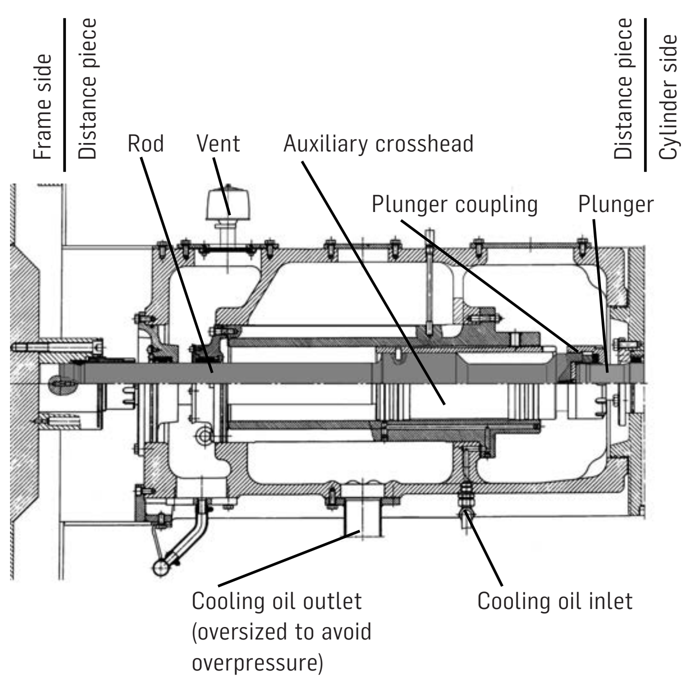
ADDITIONAL DISTANCE PIECE FOR HIGHER SAFETY

DISTANCE PIECE ISOLATES THE CRANKCASE FROM THE CYLINDER

The additional distance piece prevents ethylene gas from entering the crankcase, during normal operation as well as in case of an accident. In case of a complete plunger failure the inner cylinder diameter works as a leak gas piping and up to five times of the compressor mass flow occurs in the distance piece. The cooling oil return piping is oversized to allow gas pressure release during an incident. Therefore, an explosion relief valve is not required for our secondary crankcase.

In case the plunger coupling would be arranged inside the crank case there is a great risk of fire when the ethylene gas gets in contact with the hot oil. It is very uncertain if the crank case walls would withstand an ethylene overpressure of nearly 10 bars.

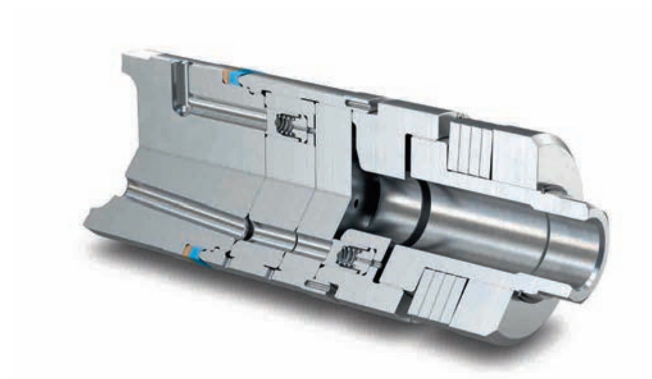




COMPRESSOR FEATURES
HIGH-PERFORMANCE VALVES

VALVES WITH OPTIMIZED SHAPE

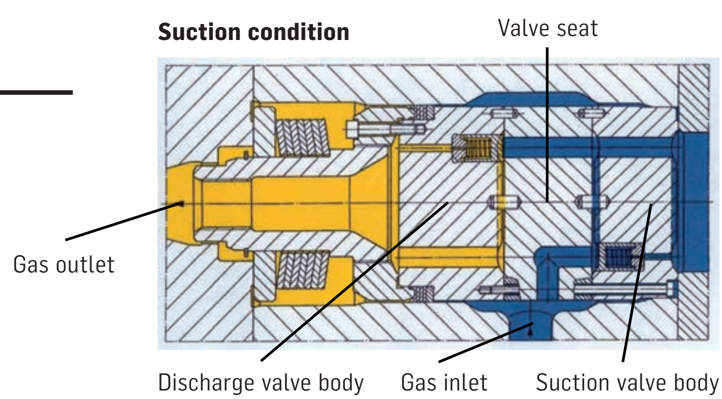
– Optimized by finite element analysis
– Avoids fretting on the cylinder cup
– Prevents external leakage
– Special strengthening procedure to avoid fissures
– Highly exposed areas treated using autofrettage

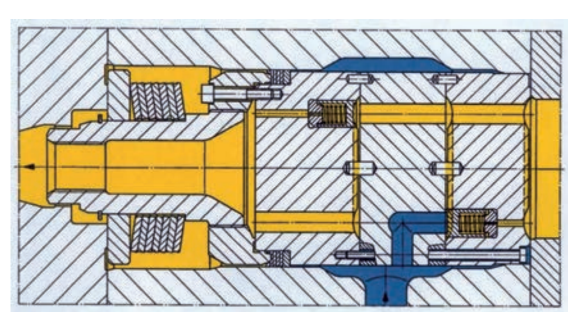


CENTRAL VALVE – COMBINED SUCTION AND DISCHARGE

– Compact design
– Full metal execution
– Different surface treatments
– Optimized flow
– Easy maintenance

Suction condition





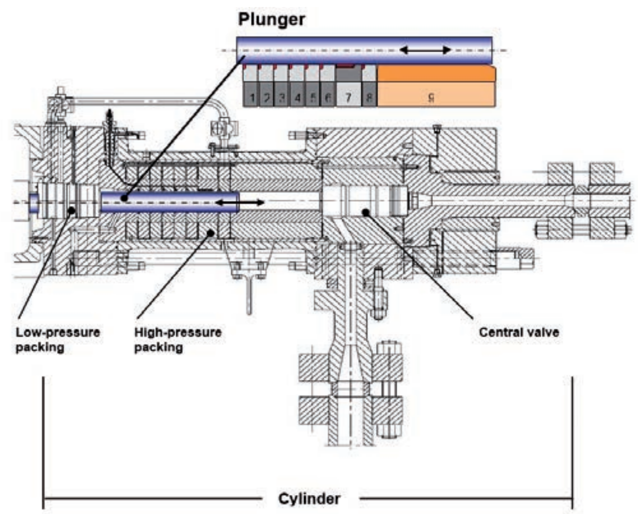
Discharge condition

COMPRESSOR FEATURES

ADVANCED PLUNGER DESIGN

PLUNGER QUALITY ASSURED BY BURCKHARDT COMPRESSION

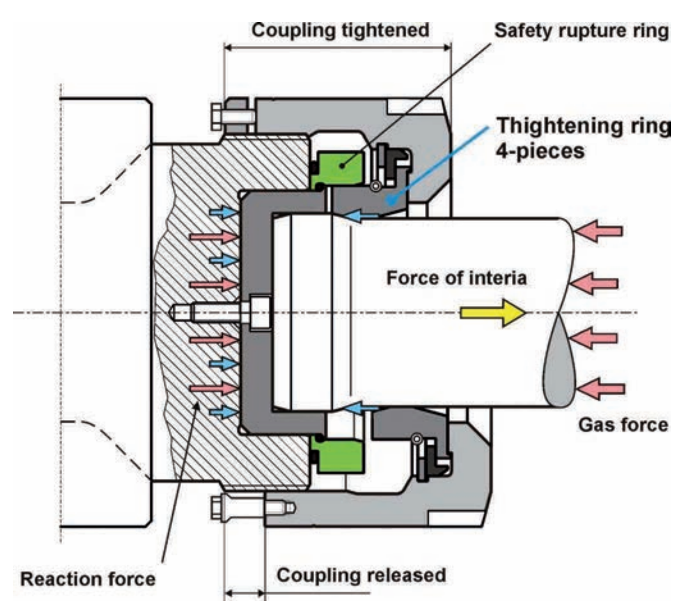
The plunger's fist stage is made from tungsten carbide plated and the second stage from tungsten carbide solid. It offers one touch connection without any alignments being necessary and meets highest demands on surface finish and tolerances. Specifications and quality is assured by Burckhardt Compression.



CONUS PLUNGER COUPLING FOR FLEXIBLE CONNECTION OF PLUNGER AND CROSSHEAD

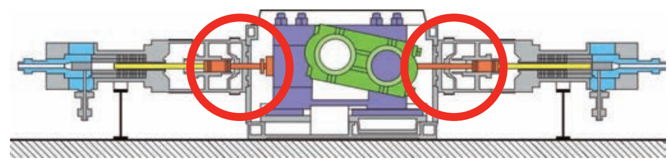
The conus plunger coupling never needs alignment during erection or assembly. Easy access due to good accessibility from outside the compressor and simple assembly are the main advantages of this construction. An additional safety ring inside the coupling protects the compressor against damage due to tractive forces.

The coupling compensates wear of plunger guide bush or misalignment of plunger to avoid bending forces on the brittle tungsten carbide plunger.



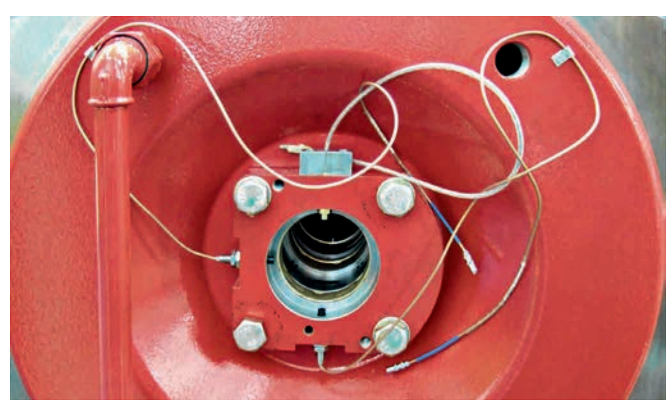
AUXILIARY CROSSHEAD TO ELIMINATE UNDESIRED PLUNGER MOVEMENT

The auxiliary crosshead compensates all movements and forces caused by the crank case parts. A failure of motion work parts will not cause any damage to the plunger. This, and the excellent guidance due to low tolerances, result in a long life time of the high pressure packing.



PLUNGER TEMPERATURE AND VIBRATION MONITORING

The constant monitoring of the plunger vibration, with two eddy-current probes (contactless) located next to each plunger, as well as of the plunger temperature, by a temperature element inside of a bronze sensor (in contact with the plunger), allows the high pressure to run for its entire lifetime.



LUBRICATION SYSTEMS
SUPPORTING HIGHEST EFFICIENCY

LUBRICATION CRANK GEAR

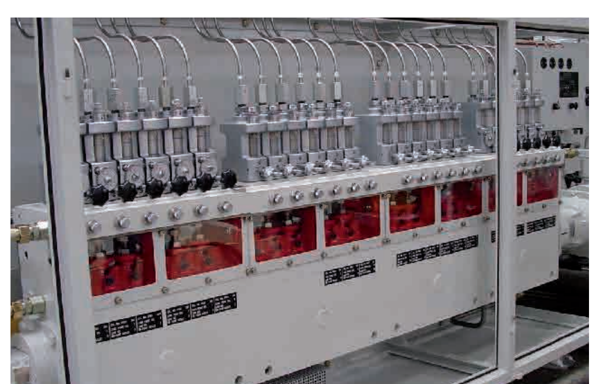
- Separate skid
- Reliable, proven components
- Full redundancy
- Low maintenance
- High availability
- Low lubricant consumption
- Low life cycle costs

COOLING AND FLUSHING

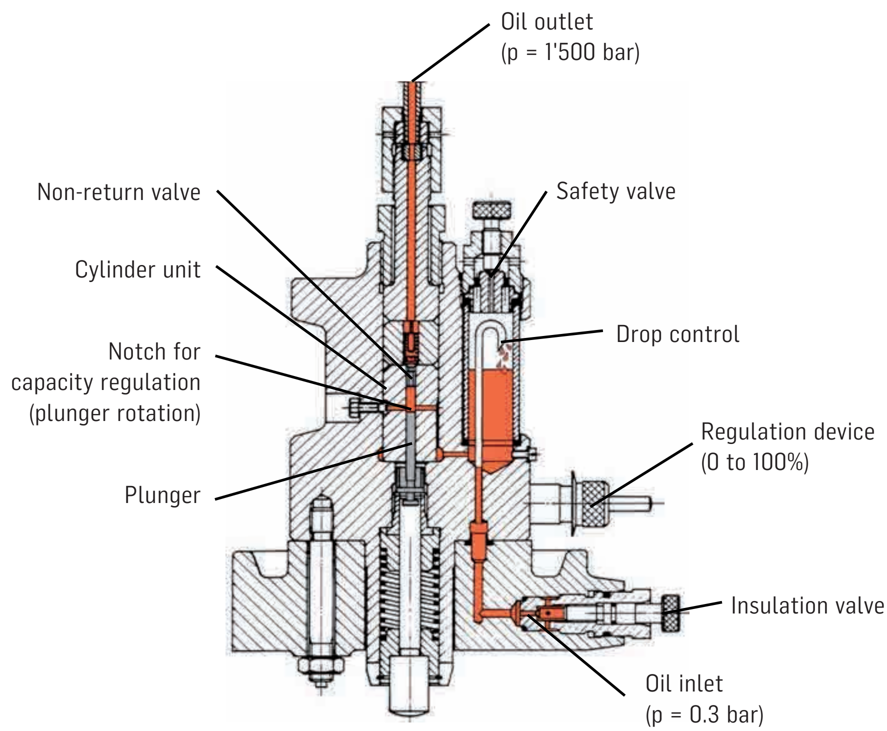
- Separate oil reservoir, skid-mounted
- Two motor driven oil pumps (main and standby)
- Double oil cooler
- Double oil filter of special design
- Full redundancy
- Low down-times
- Low production losses
- High reliability
- Low maintenance

CYLINDER LUBRICATION GROUP

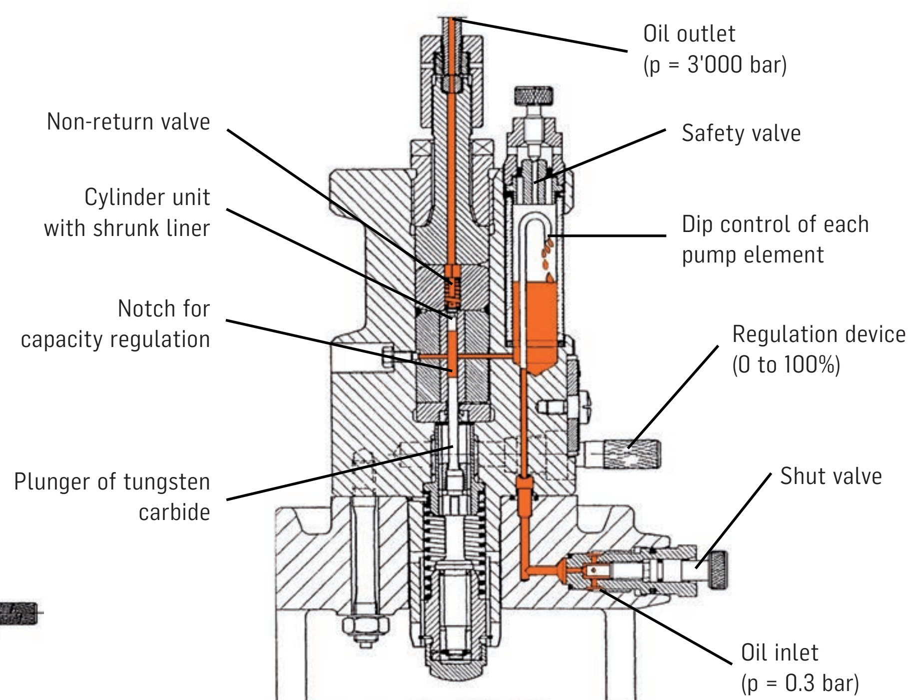
- Single pump per lubrication point for high reliability and highly effective lubrication distribution
- In-house engineered, customer-proven design
- Multiple adjustment possibilities
- Low maintenance
- High availability
- Low lubricant consumption
- Low life cycle costs

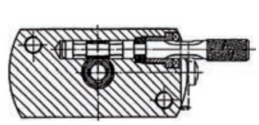


TYPE C



TYPE CH



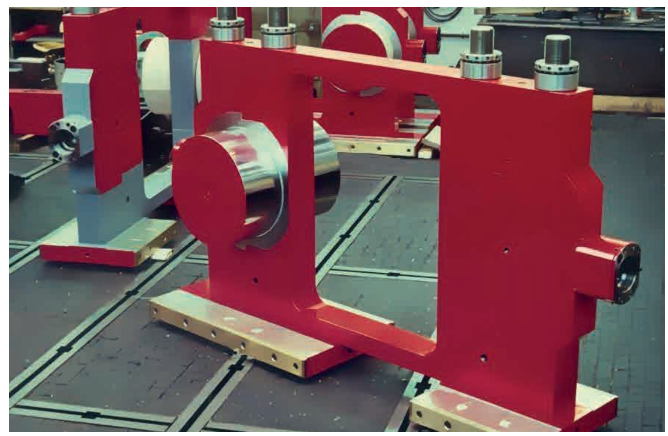


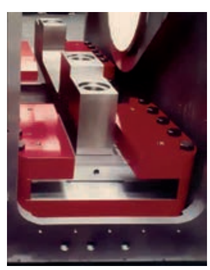
Fine pump adjustment

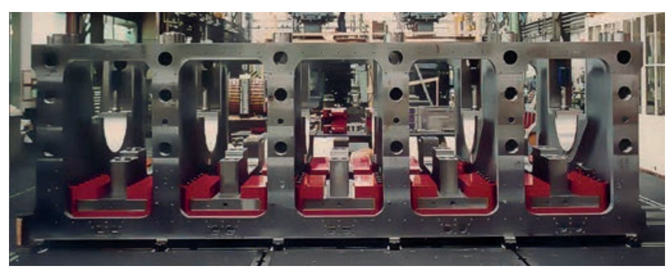
COMPRESSOR FEATURES
BOTTOM GUIDANCE

BOTTOM GUIDANCE FOR EASY ACCESS TO THE FRAME WORK

All motion work parts, like bearings and connecting rod, are easily accessible. There are no internals in the bearing axis to ensure an easy and safe maintenance.







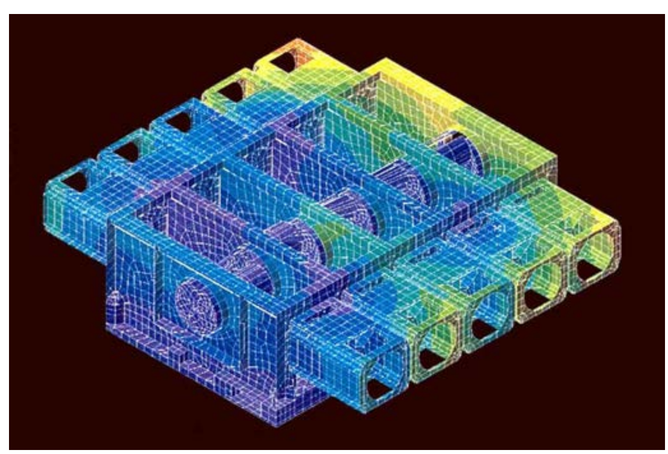
IN-HOUSE EXPERTISE
DESIGN & ANALYSIS

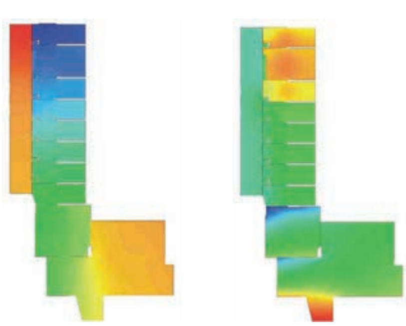
DESIGN FOR LOWEST MAINTENANCE

- Design with lowest number of cylinders in the market
- Resulting in minimized number of wear parts
- Low speed to reduce wear
- Advanced lubrication system for all main bearings and the entire cylinder assembly
- Easy and quick access to all wear parts

IN-HOUSE FINITE ELEMENT ANALYSIS

- Tribology
- Mechatronics
- Finite Element Analysis (FEA)
- Computational Fluid Dynamics (CFD)
- Steady and dynamic simulations of operating conditions
- Verification of results on test beds

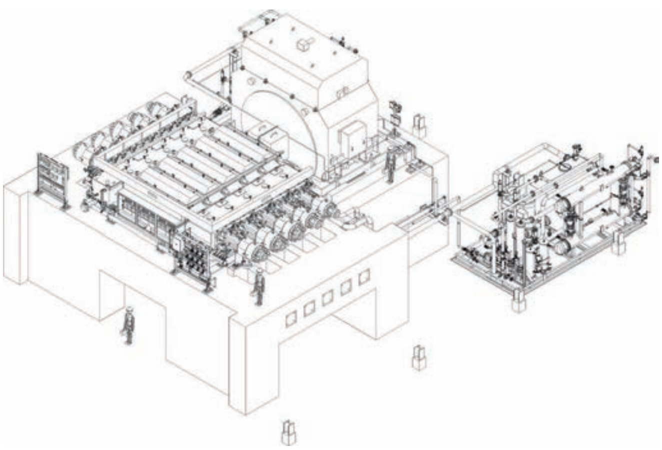


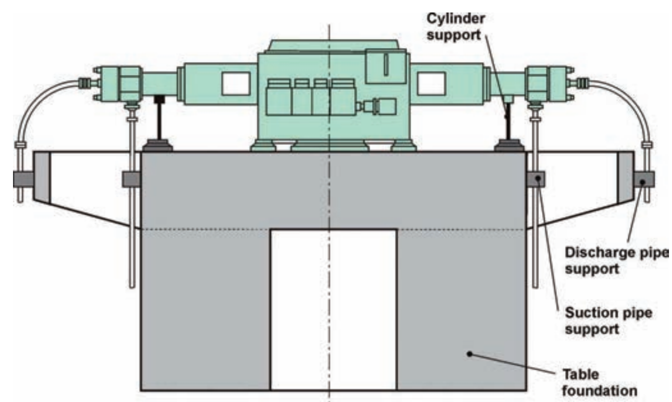


High-pressure packing clearance optimization

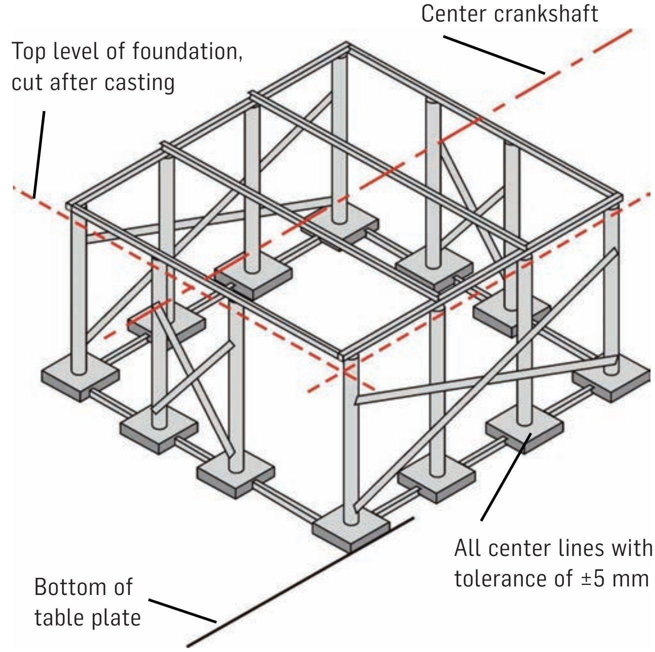
FOUNDATIONS
FOR BEST STABILITY

CONCRETE FOUNDATION

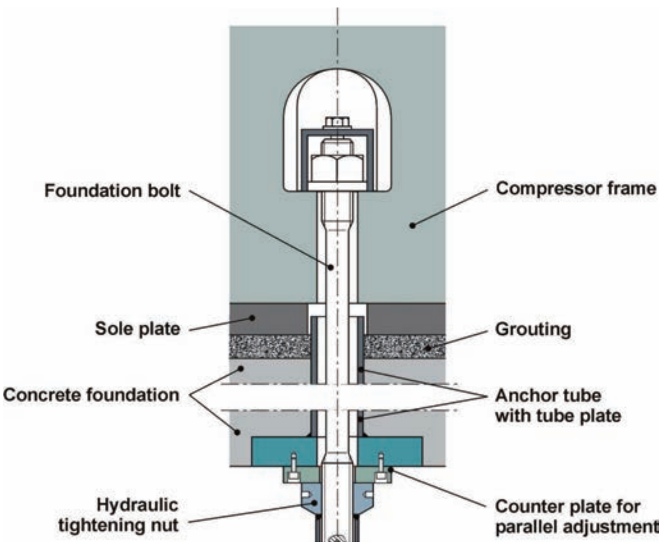




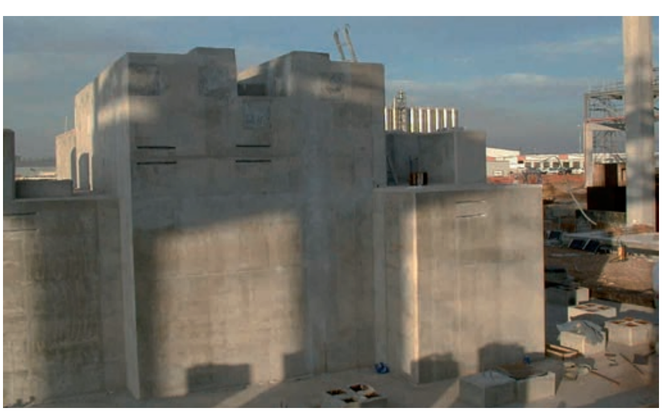
FOUNDATION TUBES



CRANKCASE ANCHORAGE



SIDE FOUNDATION



TRANSPORTATION & ASSEMBLY
OF THE COMPLETE COMPRESSOR SYSTEM

TRANSPORTATION

The entire transport can be arranged by Burckhardt Compression:
– Reliable scheduling
– No deviations
– Reliable installation time
– No time losses

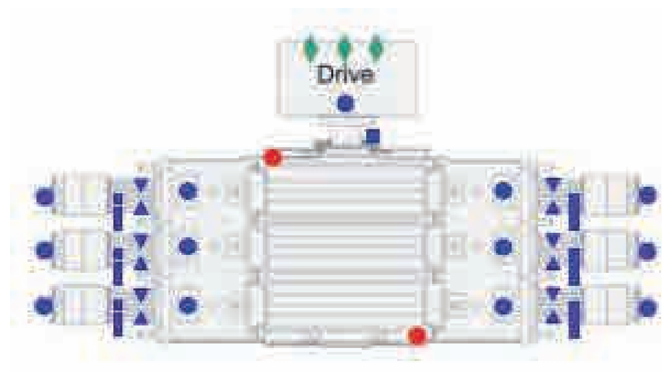
MOTOR ASSEMBLY PLANT SIDE

Careful transportation of the motor with the transfer crane to the compressor building. After positioning of the motor onto the foundation, provisional alignment of the compressor-motor-train.

CONDITION MONITORING
MEASURING AND FAILURE DETECTION

PROGNOST®-NT

Asset performance management system for reciprocating compressors:
– 20% of all LDPE compressor systems are equipped with Prognost®-NT
– Early failure detection: Shut-down the machine before a part is severely damaged. Unscheduled shut-downs can be avoided.

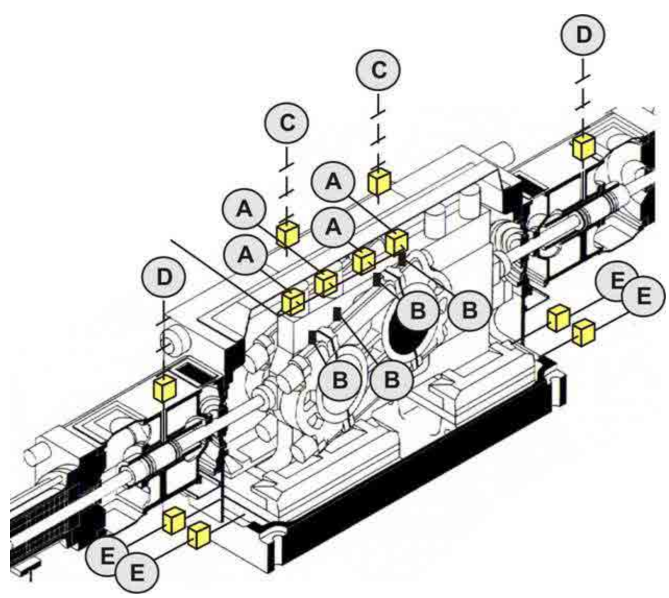


Analog Input Signals

- 1 trigger sensor
- 12 acceleration sensors on CHS and cylinder
- 1 acceleration sensor on motor
- 2 velocity sensors on frame
- 12 strain gauge rings
- 12 plunger run out sensors
- 3 temperature sensors (existing)

SENTRY SYSTEM

Wireless bearing temperature measurement system:
– System of real-time monitoring of bearing temperature on non-stationary moving parts
– One temperature control signal per crankshaft revolution
– Temperature trend indication



UNEQUALED CAPABILITIES
FOR YOUR HYPER EQUIPMENT

REQUIREMENTS & SITUATION ANALYSIS

- Process Compressor condition change
- Improved design to replacement parts with excessive wear
- Damaged parts/failure
- Worn out parts
- Precarious or unsafe operating conditions
- Troublesome parts
- Maintenance and spare parts logistic scheduling

EVALUATION

- Root cause analysis/failure analysis
- Risk/Condition and Material analysis
- Feasibility studies
- Finite Element Analysis (FEA) Studies
- Non-destructive testing (NDT)
- Stock recommendations
- 3D Modelling
- Computer Fluid Dynamics (CFD) Analysis

ENGINEERING

- On-site, and/or in-house measurements of components
- Recalculation and dimensioning
- Engineering for repair procedures
- Reverse engineering/Retrofit
- Material selection and sizing
- Load optimization
- Thermodynamics

MANUFACTURING

- Highest quality assurance
- Reproduction (1:1 replacement incl. integration of latest quality standards)
- In-house precision manufacturing of parts
- Fast track production upon request
- OEM guarantee
- Specific in-house manufacturing (e.g. Autofrettage)

FIELD ACTIVITIES

- Extensive service network with our specialized field service engineers/troubleshooters
- Spare parts framework agreements
- Worldwide distribution and service center network
- Dismantling/Reassembly of parts
- Large parts stock
- Burckhardt e-Shop™ spare parts identification and ordering system

INCORPORATING FULL DOCUMENTATION OF REQUIRED GUIDELINES AND STANDARDS INCL. CERTIFICATION

INTERACTION WITH CUSTOMER REGARDING SPECIFICATIONS AND SCOPE OF WORK

REVAMP, RETROFIT & TURN-KEY
ADJUSTED TO CHANGING REQUIREMENTS

REVAMP – REJUVENATE OR TUNE YOUR COMPRESSOR SYSTEM

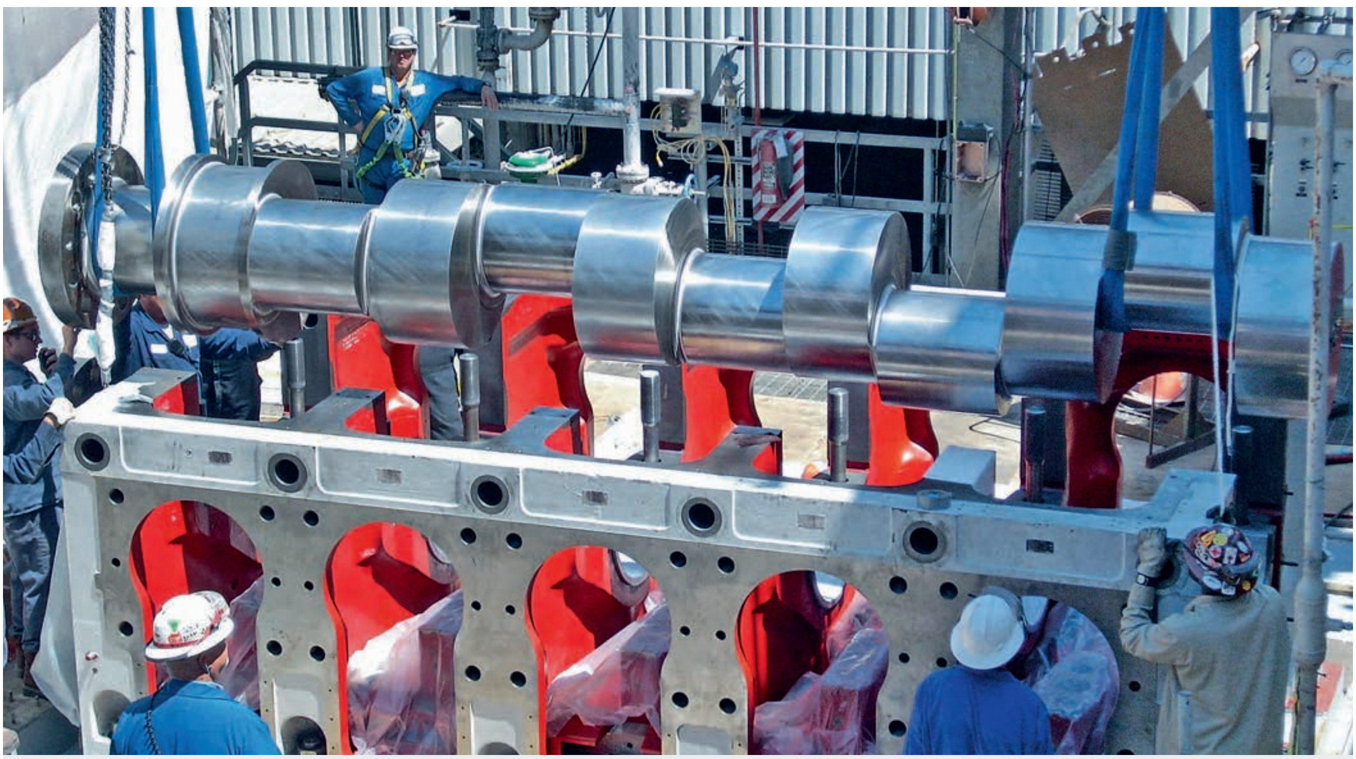
Changes to a compressor and/or compressor system for operational, technological, economical or environmental reasons, which can combine activities such as:
- Debottlenecking
- Capacity change
- Reapplications
- Modernizations

RETROFIT/REVERSE ENGINEERING – PERFECTLY ADAPTED COMPONENTS & IMPROVED QUALITY

Replication and substitution of existing components with components of original design and latest quality standards.

TURN-KEY PROJECT – ONE SOURCE FOR YOUR COMPLETE SOLUTION

Solutions provided with sole responsibility for the complete execution of the contractually specified scope of works from the planning to the commissioning activities.



REVAMP CASE

Compressor type: Hyper Compressor (10 Cylinder)

Application: Low-density polyethylene production

Year of installation: 1978

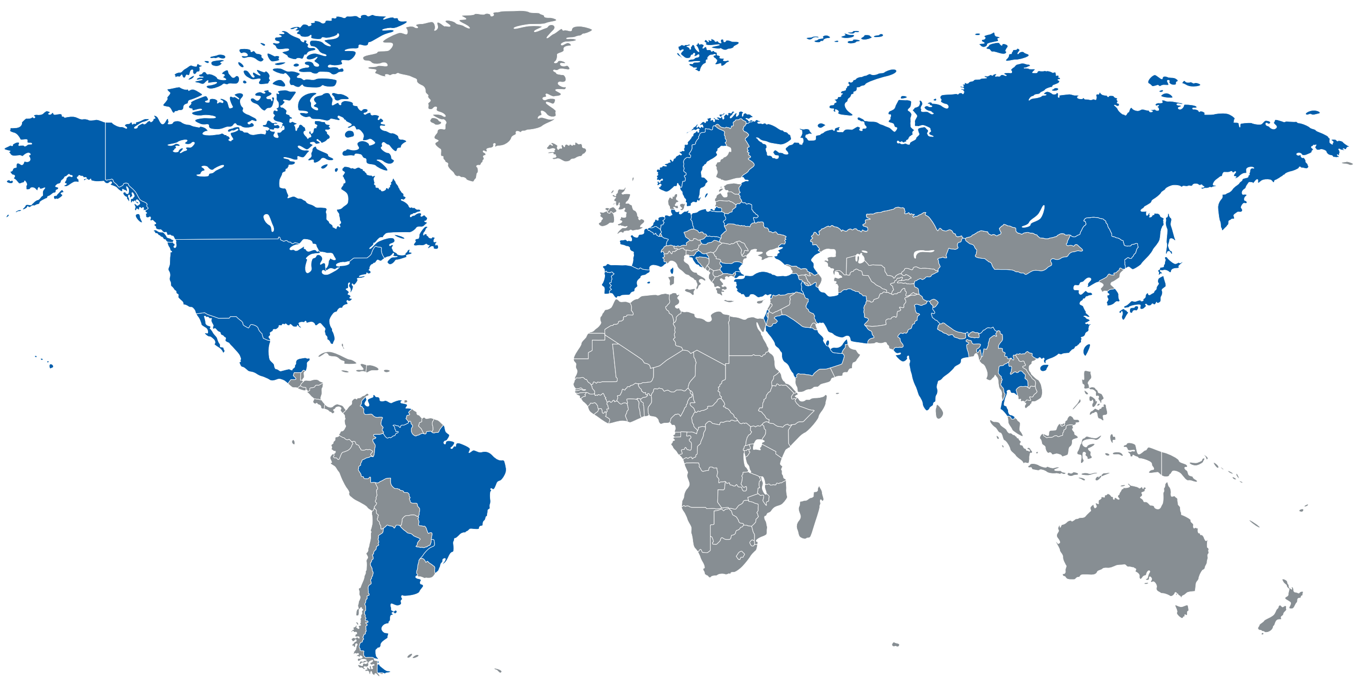
Situation/Customer requirements
- Original foundation (provided by 3rd party) without oil resistant protection. Movement of the system led to damage of foundation
- Replacement of the damaged foundation and a complete revamp of the compressor system was required

Scope of supply and service
- Complete disassembling of compressor and connections
- Removal of old foundation and replacement with new, state of the art foundation
- Local refurbishment of entire compressor, motor and auxiliaries
- Concurrent coordination of all necessary suppliers and services

Technical highlights
- Heavy duty equipment combined with high precision execution
- Highly accurate machining of large parts (e.g. crankcase)

WORLDWIDE REFERENCES

OVER 150 YEARS OF EXPERIENCE



ARGENTINA

Zulia	K10

BELARUS

Novopolotsk	F8

BELGIUM

Zwijndrecht	F10

BRAZIL

Camacari	2x F8
Triunfo	F6, F10

BULGARIA

Burgas	F6

CANADA

Edmonton	H6

CHINA

Beijing	F6
Daqing	F8, K8
Lanzhou	K8
Lianyungang	K8, K10
Maoming	K10
Nanjing	2x K8
Ningbo	K6
Qilu	F8
Shanghai	F8, 2x F6
Tengzhou	F8
Urumqi	K10
Yanshan	3x F8
Yulin	K10

CROATIA

Zagreb	F6

FRANCE

Balan	F8
Berre	K10
Carling	2x F8
Dunkerque	K10, F6(8)
Fos sur Mer	2x H6
Gonfreville	F8
Mont	F4

GERMANY

Worringen	H6

INDIA

Baroda	2x F6
Jamnagar	K12
Nagothane	F8

IRAN

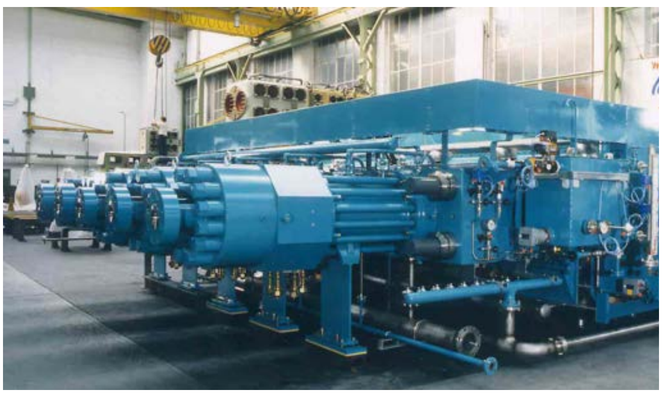
Assaluyeh	K10
Bandar Imam	K10
Marun	K10
Sanandaj	K10

ISRAEL

Haifa	F8

JAPAN

Chiba	F6
Kashima	F10
Kawasaki	2x F6
Oita City	F8



TYPE K10

Motor power	24'500 kW
Industry	LDPE
Customer	Compagnie Pétrochemique de Berre
Location	France

TYPE K8

Motor power	n/a
Industry	LDPE
Customer	Qatar Petrochemical Co.
Location	Umm Said, Qatar

Yokkaishi	F10

MEXICO

Veracruz	3x F6

NETHERLANDS

Geleen	F8, 2x F10

NORWAY

Stathelle	2x F8

POLAND

Mzrip-Plock	2x F6

PORTUGAL

Sines	2x F6

QATAR

Mesaieed	K10
Umm Said	K8, K10

RUSSIA

Angarsk	4x H6
Kazan	2x F8
Novy Urengoy	3x F8

SAUDI ARABIA

Al Jubail	K8, K12, 2x K10
IPC	K8
Sadara	K10

SINGAPORE

Singapore	F8

SLOVAKIA

Bratislava	F8, K8

SOUTH KOREA

Daesan	F10, K10, 2x F8
Ulsan	F4
Yosu	F4, F6, F8

SPAIN

Tarragona	F8

SWEDEN

Stenungsund	K12

TAIWAN

Mailiao	K6, K8

THAILAND

Map Tha Phut	K10
Rayong	2x F8

TURKEY

Aliaga	F8

UAE

Ruwais	K12

USA

Beaumont	F6
Clinton	F8
Deer Park	F10
Plaquemine	K10
P. Comfort	K8+8

VENEZUELA

Zulia	K10

SCIENTIFIC PAPERS

Dynamic Pressure Distribution in the Cylinder Packing of Compressors for Very High Pressures

K. Scheuber, Burckhardt Compression

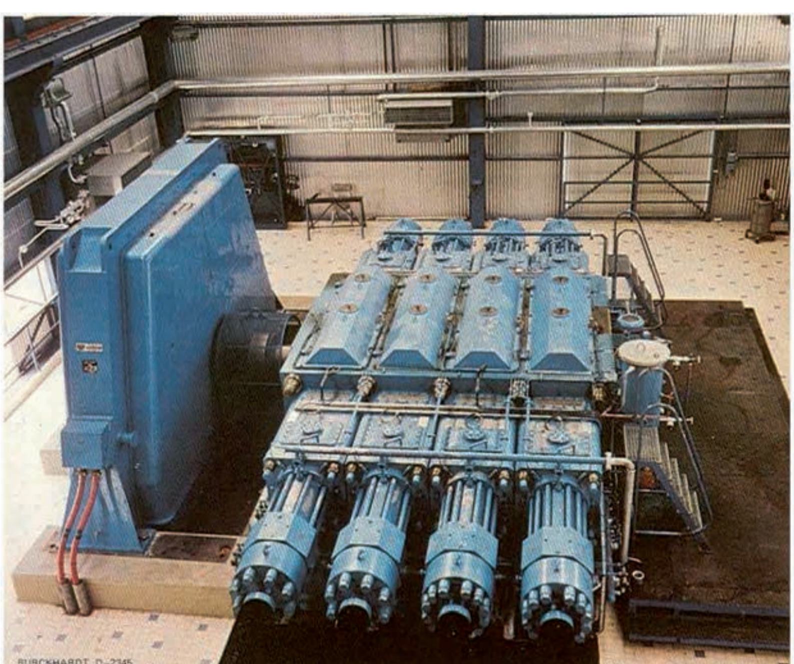
The production of polyethylene by the high-pressure process requires pressures up to 3'500 bar, which are delivered by reciprocating compressors (fig. 1). On these machines, each plunger (smooth piston rod) is sealed by several self-adjusting bronze rings. Because the pressure in the cylinder alternates rapidly between suction and end pressure, non-controlled, static pressure distribution is able to establish itself between the sealing rings. Calculation is difficult, and the results questionable.

For optimal dimensioning and designing, exact knowledge of the phenomena occurring is very important. Accordingly measurements were made of the pressure vs. time behavior at pressures up to 2'500 bar. A report on the results is given below.

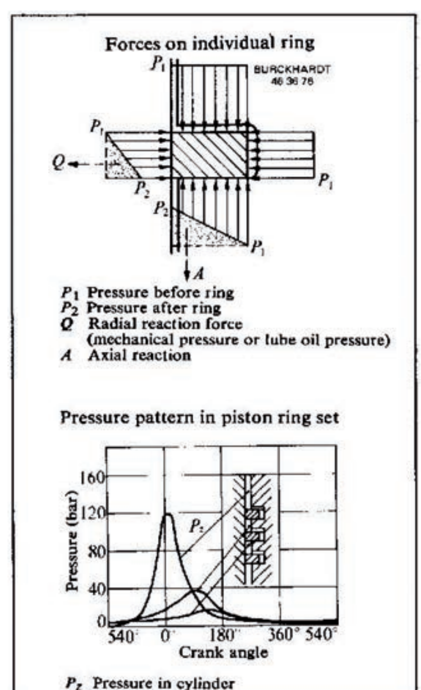
Fundamentally speaking there are two possibilities for generating and sealing very high pressures in reciprocating compressors:

- Moving piston with piston rings acting against the stationary cylinder liner
- Packing with stationary sealing elements acting against the reciprocating plunger

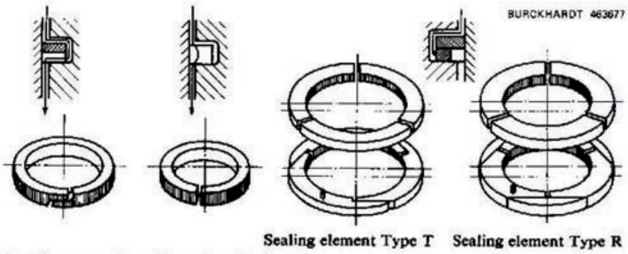
In recent years, packings have found general favour for sealing very high pressures. One of the main reasons for this is the ability to feed the lubricating oil directly to the friction point. Very exacting demands are placed on the sealing elements by the sealing at high pressures between 1'500 and 3'500 bar required in the polymerization of high-pressure polyethylene. These elements must withstand exceptionally severe loads.



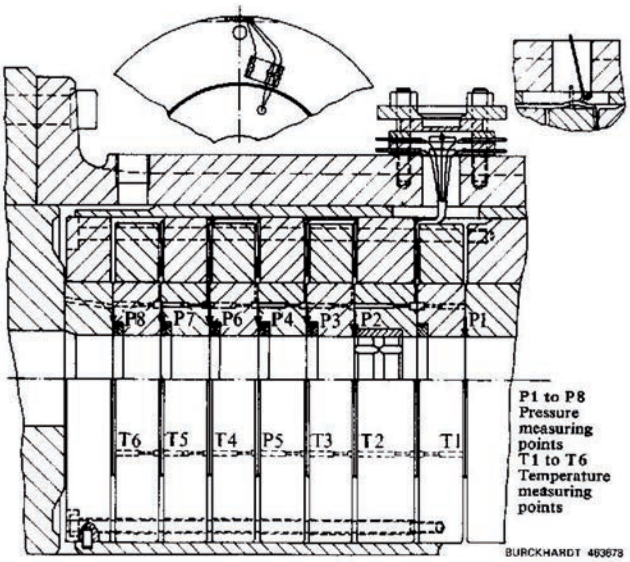
1 Secondary compressor Type F8: suction pressure 250 bar, delivery pressure up to 3400 bar, output up to 80 t/h, power up to 11 MW.



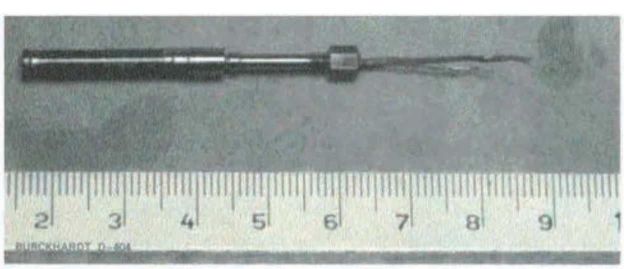
2 Mode of operation of piston rings.



3 Piston and sealing ring designs.



4 Arrangement of the measuring points.



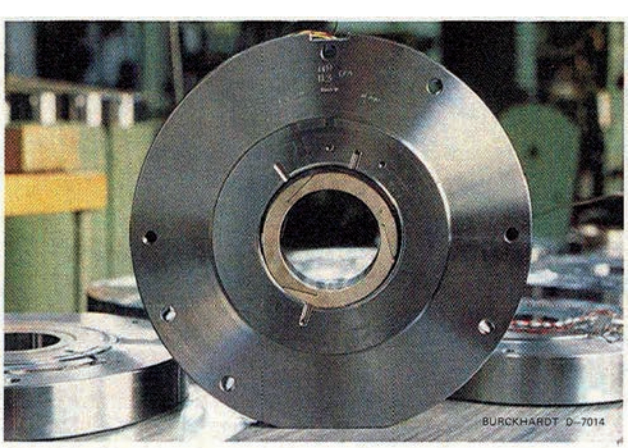
5 Pressure transducer with Manganin wire.

With increasing plant size, long-life sealing elements are assuming major importance, for every interruption means considerable loss of production. But before seal life can be improved, it is necessary to know how the seal functions. It is characterized by the pressure drop across the packing. We have therefore measured the pressure distribution in high-pressure packings over longer periods of productive operation.

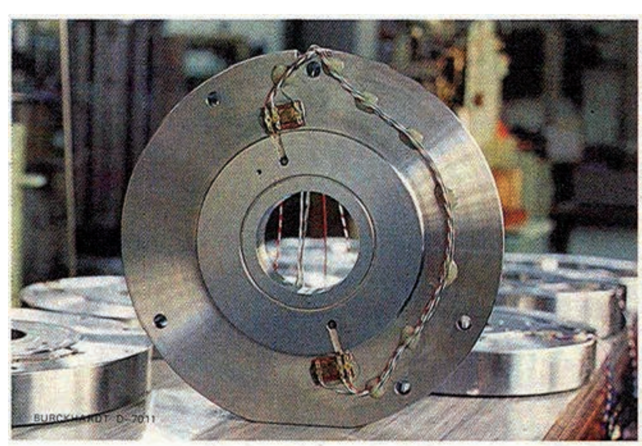
MODE OF OPERATION OF PACKINGS

Sealing elements function in principle like piston rings, except that whereas the latter slide against the mating part with their outer surface, sealing elements slide with their inner (bore) surface against the other part. Under certain assumptions, the "effective" gas pressure which governs the loading of the running surface can be calculated for the sealing conditions on piston rings [1, 2, 3] (fig. 2). The pressure drop, to which all elements of a seal contribute, presupposes a certain permeability (leakage cross-section).

Here there is a basic difference between piston rings and high-pressure packing, however, for the gap clearance of the tangential sealing ring is concealed by a three-piece front ring (fig. 3). In this way, no defined passage cross-section is left free, which with extreme pressure is also desirable to minimize leakage losses. However this gives rise to another sealing mechanism, and the pressure drop can no longer be calculated because the geometry of the leakage cross-section cannot be determined now. It may happen for example that a single element provides the entire sealing. Consequently only measurement can reveal the actual pressure distribution in the packing.



6 Cup ring with sealing element inserted.



7 Reverse side of cup ring with compensation circuit.

CONCEPTION OF THE MEASUREMENTS

The pressure distribution was to be measured under real service conditions over a long period. Important above all was to establish the long-term behavior taking wear into account, and not just register an instantaneous state.

Arrangement of measuring points

Figure 4 shows a section through a packing to be measured, with the pressure measuring points P1 to P8. Symmetrical to these are the temperature measuring points T1 to T6. The purpose of the temperature measurements was to ascertain the measuring errors due to heat. Pressure and temperature transducers were fitted inside the actual cup rings. To make this possible, two problems had to be solved first: where to find a suitable extreme pressure transducer (fig. 5) and how to fit it with sufficient strength. Figure 6 shows a cup ring (gland) with sealing element inserted, while Figure 7 shows the reverse side with compensation circuit glued on.

The strength problem was solved by leading the pressurized gas from the division between two chamber rings through a groove to the pressure transducer. In this way, a radial transverse hole is dispensed with, and the pressure transducer can be arranged longitudinally. This gives strength conditions exactly the same as those with the lube oil feed, which is proven.

Since no suitable pressure transducer was available commercially, Burckhardt Compression developed its own Manganin wire transducer, the resistance of which changes under pressure (fig. 5). Temperature influence is small.

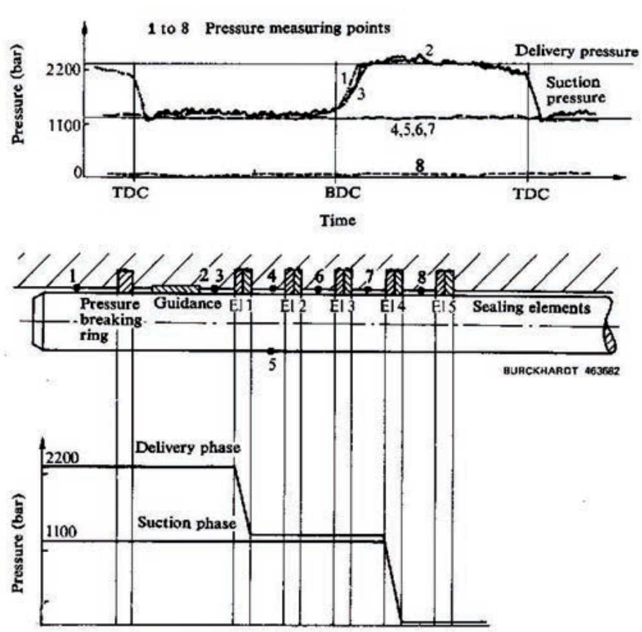
Prior to fitting it in the measuring seals, the transducer was subjected to the following tests:
- Static loading up to 4'000 bar
- Dynamic loading with 1.5 x 10^7 load cycles (0 to 2'000 bar)

Compared with the familiar measurement in the lube oil lines the arrangement of the transducers in the gland itself offers the following advantages:
- High accuracy (no transmission column between sealing element and measuring point)
- Pressure can be measured on each individual sealing element

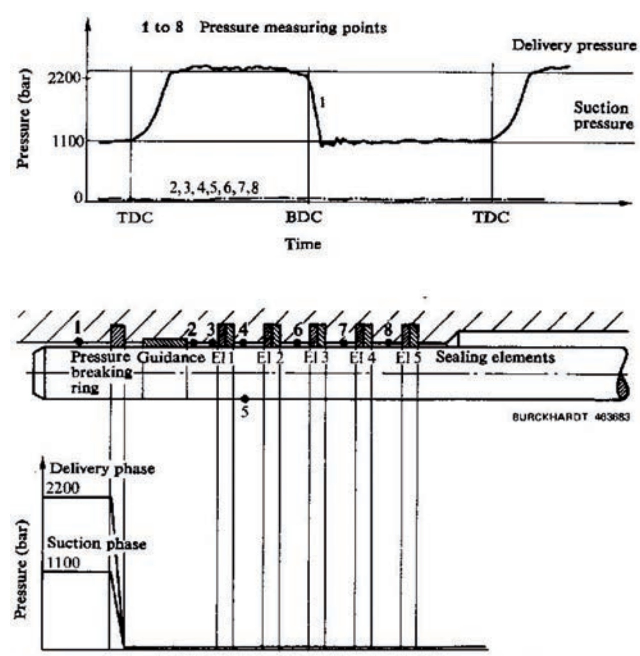
Extent of the measurements

The measurements were performed on various compressors of the F type. For each compressor, three different measuring packings were prepared, designated variants A, B and C. Within one packing, all measuring points were scanned simultaneously.

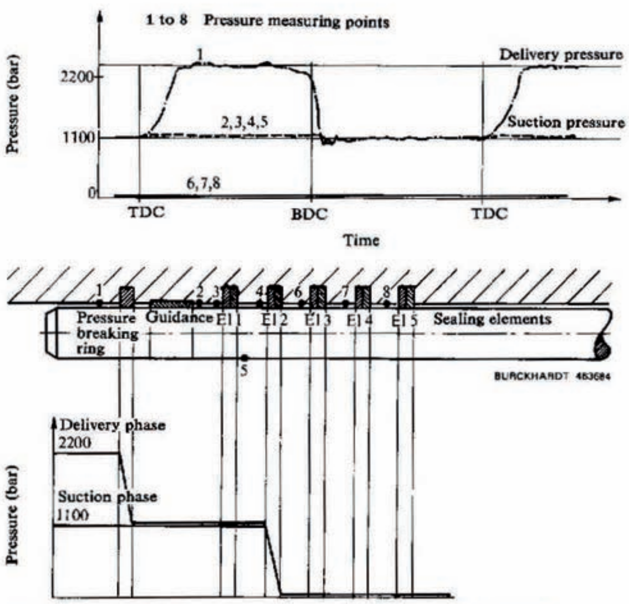
At the same time as the pressure measurements in the packing, the pressure in the cylinder lube oil lines was measured with Bourdon gauges. These measurements served as a check on the absolute pressure level (zero point).



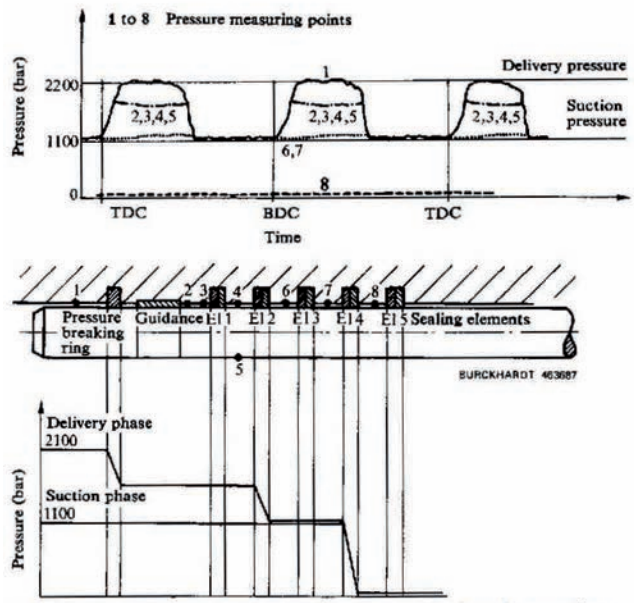
8 Typical pressure behaviour in measuring packing A (three-pice, radially split pressure breaking ring, five sealing elements Type T).



9 Pressure behaviour in measuring packing B (one-piece pressure braking ring, three-piece radially split front ring with one-piece ring, four sealing elements Type T) from commissioning to 300 h operation.



10 Pressure pattern in measuring packing B after 500 h operation.



11 Pressure pattern in measuring packing B after 3000 h operation.

RESULTS OF THE MEASUREMENTS

All measured curves graphed below relate to second-stage cylinders of the same secondary compressor with the following data:

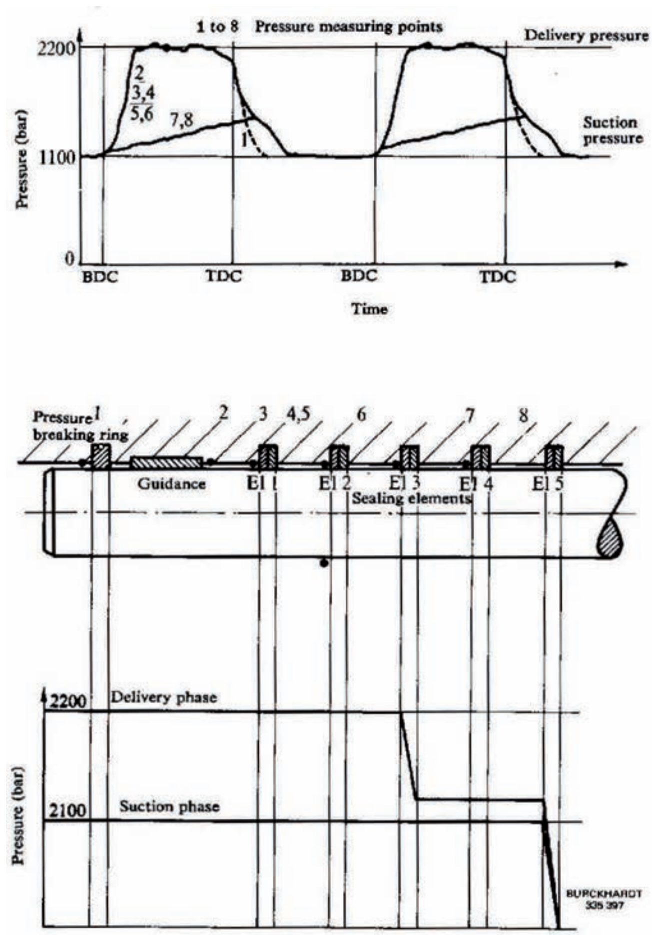
Plunger diameter	89 mm
Plunger stroke	370 mm
Compressor speed	200 rev/min

Figure 1 shows a compressor of this type series. Out of the large number of measured results we shall confine ourselves here to a comparison of the two seal variants A and B, with the emphasis on B. These two packings are fitted with the following sealing elements:

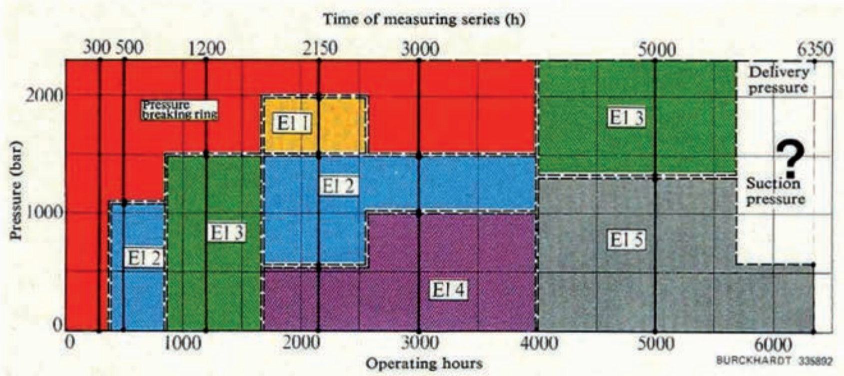
Measuring packing Type A
– One three-piece pressure breaking ring
– Five sealing elements Type T (fig. 3) consisting of a three-piece front ring and a tangentially split sealing ring

Measuring packing Type B
– One solid pressure breaking ring
– One combined pressure breaking ring (consisting of a three-piece radially split front ring and a one-piece ring)
– Four sealing elements Type T

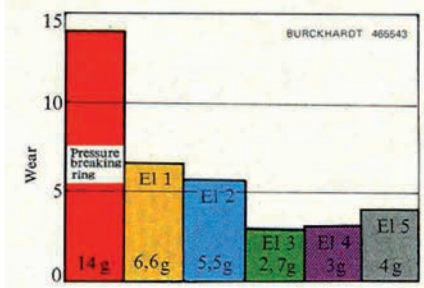
Measured results with packing A
Figure 8 shows a typical pressure plot. The three-piece pressure breaking ring has only a slight throttling action. During the compression phase the pressure rise to the first sealing element following is slightly delayed, but it



12 Pressure pattern in measuring packing B after 5000 h operation.



13 Behaviour of the individual sealing elements in measuring packing B.



14 Wear on the individual sealing elements of packing B after 6350 h operation.

is not reduced. During the delivery phase, the full cylinder pressure is applied behind the pressure breaking ring, and in the expansion phase the pressure behind the pressure breaking ring follows the cylinder pressure without delay. During the suction phase the pressure behind the pressure breaking ring is equal to the cylinder pressure.

Accordingly the pressure distribution in the packing is as follows: The first sealing element (El 1) takes up the entire alternating pressure component. Behind element 1 the pressure is constant, i.e. equal to the suction pressure of 1'100 bar. Elements 2 and 3 do not have any pressure-reducing action. The last element but one (El 4) takes up the entire constant pressure component and seals against the atmosphere. The last element (El 5) likewise serves no purpose as far as pressure is concerned. At the bottom of figure 8 the pressure is plotted over the width of the packing. Since only two elements are effective, they are stressed with commensurate severity. This sealing curve might appear different at another point in time, however, especially if he two effective elements have sustained a lot of wear and the sealing has been taken over by other elements.

This packing has been tested over a long period, and changes have been observed due mainly to the pressure breaking ring. After a few hours running-in, pressures established themselves which remained unchanged for some 300 hours. The entire sealing (alternating and constant pressure components) is provided solely by the one-piece pressure breaking ring (fig. 9).

After 500 hours of operation (fig. 10) the pressure breaking ring took over the entire alternating pressure; behind it the pressure is constant and equal to the suction pressure. Element 2 seals the entire constant pressure. The first and last elements are ineffective.

After about 3'000 hours of operation the pressures have become stabilized (fig. 11). The one-piece pressure breaking ring still takes up about half of the alternating pressure component, element 2 takes up the rest of the alternating pressure, while element 4 seals the entire constant pressure. Elements 1, 3 and 5 are ineffective.

After 5'000 hours of Operation (Fig. 12) the following situation has established itself. The pressure breaking ring and the first two elements are ineffective. The third element seals about three-quarters of the alternating pressure, element 4 is ineffective, and element 5 seals the entire remaining pressure. The operational behavior of this packing was measured during 6'350 hours of operation at certain intervals. The interpolation of the pressure distribution between the individual measuring series as shown in figure 13 has been assumed arbitrarily. Nevertheless it may be said with confidence that during the first 4'000 hours the pressure breaking ring played a major part. Although the seal was still serviceable after 6'350 hours, it was dismantled in order to examine the condition of the sealing elements at this time. Shortly before the seal was changed, its leakage loss amounted only to 0.3% of the total compressor discharge.

Figure 14 shows the wear, ascertained from the difference in weight between the new and used elements.

CONCLUSIONS

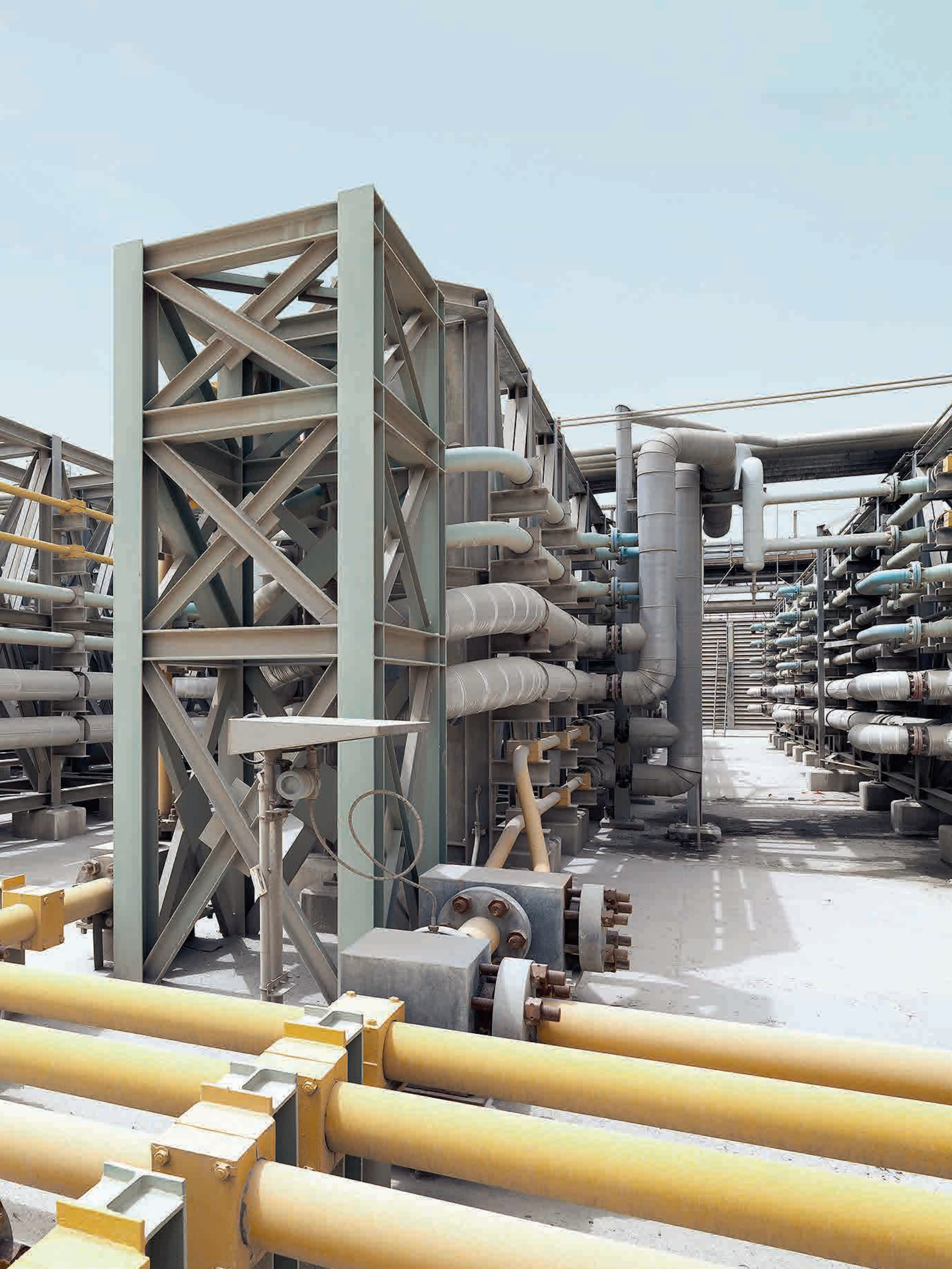
With the development of special Manganin wire pressure transducers it has become possible to carry out extended measurements of the pressure distribution within the packings of compressors for very high pressures. This enables the behavior of the scaling elements to be studied over long periods, ascertaining the optimal design for the individual elements and the best arrangement for them. Investigations of this kind take years and are very costly.

Through the measurements performed to date it has been established that the pattern of the pressure drop in the seal remains stable over a long period. The measurements have revealed also that no uniform division of the pressure drop between all elements took place in any of the packings examined here during 6'350 hours of operation. On the contrary, two or three elements always provide the sealing. This fact demonstrates the importance of measuring the pressure at each sealing element, and not omitting odd elements as happens when the pressure is merely measured at the lube oil feed points.

These measurements are being continued at the time of writing. Further disclosures are expected from the results.

Bibliography

[1] EWEIS, M.: Reibungs- und Undichtigkeitsverluste an Kolbenringen. VDI-Verlag, Berlin (1935), Forschungsheft 371.
[2] ENGLISCH, C.: Kolbenringe für grosse und mittelgrosse Schiffsdieselmotoren. Motortechnische Zeitschrift (MTZ), Vol. 28 (1967), No.3.
[3] EBERLE, F.: Zur Mechanik der Kolbenringdichtung. Motortechnische Zeitschrift (MTZ), Vol. 28 (1967), No. 3.



Gas Pulsations and Vibrations in LDPE Plants

J.R. Olivier, K. Scheuber, Burckhardt Compression

Today, plants for the production of low density polyethylene (LDPE) are built with a capacity of 150'000 ton/yr. in one line and one high-pressure compressor. Production lines with a capacity of 200'000 ton/yr. are already in the planning stage.

The high-pressure compressors grow bigger and bigger, and their typical problems get increasingly important. Some of these problems concern all sons of vibrations which can influence the safety of a plant, particularly:

1. Oscillation of the foundation caused by inertia forces and moments of the reciprocating compressor.
2. Torsional and bending vibrations of the shaft line of motor-compressor, caused by the piston forces from gas pressure and inertia.
3. Vibrations of different parts of the compressor, such as bearing covers, valve heads, etc., mainly caused by internal forces from gas pressure and inertia.
4. Gas pulsations in the pipes, caused by the pulsating flow from the cylinders.
5. Vibrations of the pipes, caused by the gas pulsations and the movement of the valve heads.
6. Noise, created by hammering of metallic parts and the gas flow in ports and pipes.

It is very important to have all these vibrations under control to avoid harmful effects or even accidents.

1. OSCILLATION OF THE FOUNDATIONS

Compressor and motor are generally placed on a common heavy block. The movement of this block should not exceed ± 0.1 mm in each direction. The proper frequency of the system is generally between the first and the fifth order of the compressor speed. It is a good policy to place the foundation block on a layer of gravel, thus creating the best conditions for an exact calculation.

2. TORSIONAL AND BENDING VIBRATIONS OF THE SHAFT

Generally, reciprocating high-pressure compressors are directly coupled with slow-speed electric motors, having a short and rigid shaft with only one outboard bearing. In this case, torsional vibrations present no problem, since the proper frequency is normally higher than the 12th order, the corresponding harmonic components of such high orders are generally small.

Some uncertainties enter into the calculation of the proper frequency of bending, since it is difficult to estimate the influence of oil film thickness and elastic deformation of the bearings. Therefore, a reasonable safety margin is necessary between the compressor speed and the calculated proper frequency. The magnetic forces acting on the rotor must be taken into consideration. Generally, the first correct frequency is found to be between 800 and 2'000 rev./min. which has to be compared with a compressor speed of about 200 rev./min.

3. VIBRATIONS OF DIFFERENT PARTS OF THE COMPRESSOR

All the parts of a compressor are caused to vibrate by the alternating inner forces. Many disturbances, e.g. beginning cracks, ruptures, loose fixations, worn out bearings, etc., can vary the stiffness of an element, the causing force or the damping, and thereby the form of the vibration. Measuring and analyzing these vibrations is therefore an important help in controlling a machine, and possibly an important part of a remote control system, which should replace the trained ear of the operator. To develop such a remote control system, measurements in the shop and in the plant were necessary to find an answer to the following questions:

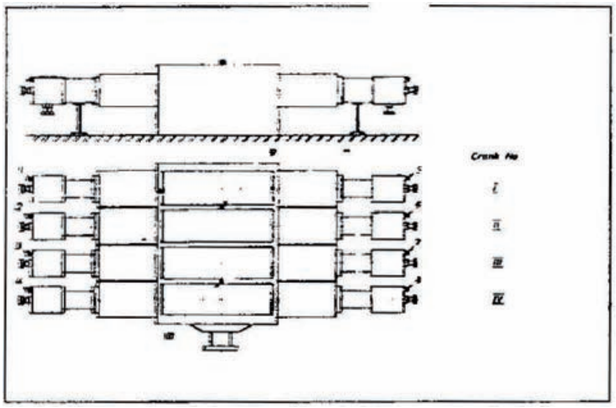


Figure 1. Positions of the sensors.

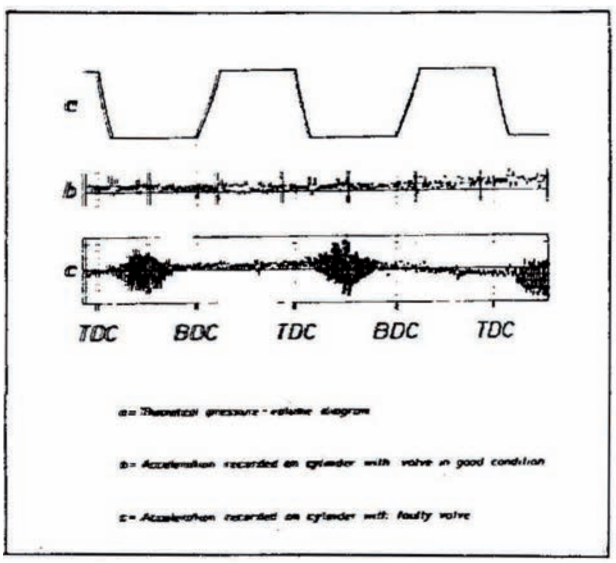


Figure 2. Acceleration of vibration.

1. Can abnormal vibrations be distinguished from the normal ones?
2. What is the range of frequencies and amplitudes of such vibrations?
3. Which is the most characteristic value (amplitudes, speed or acceleration)?
4. How many sensors are necessary and where should they be placed?
5. Which value gives the best information (amplitude, effective value, etc.)?

Today it is possible to answer all these questions and to obtain such a remote control system. Figure 1 shows, for instance, where sensors are placed on a reciprocating high-pressure compressor. Each of the sensors, 1 to 8, regulates one cylinder and sensors 9 and 10 control the whole of the motion work. The sensors record seismic accelerations in a frequency range from 10 Hz to over 20 kHz.

Their sensitivity is very high and they can indeed record irregularities, as shown in the following example: Figure 2 shows vibrations actually recorded on two identical cylinders. The upper record, at b, shows a cylinder where the combined suction and delivery valve is in good condition. In the lower one, at c, a defective valve allows a backflow into the cylinder during the suction stroke, which was confirmed by a higher gas temperature in this cylinder.

All sensors measure acceleration only. We found that monitoring accelerations is sufficient for accident prevention.

The lifetime of the sensors, according to the supplier's information, should be up to 100'000 hours. The explosion proof installation is protected by break-down diodes.

4. GAS PULSATION IN PIPES

Calculation of gas pulsations is an important part of the layout of the pipe-work of a reciprocating high-pressure compressor. Several methods exist for this calculation. Mostly, a simulator is used, based on the analogy of electrical and acoustical phenomena. Since the theory uses linear equations the method is not absolutely exact, but measurements have proved the error to be so very small that the results of the simulator can safely be used as base for the calculation of pipe vibrations.

Figure 3 shows a comparison of calculated and measured gas pulsations in the delivery line of a first-stage cylinder. This example concerns elbow B1 in figure 4. Gas pulsations in the pipework should be as small as possible. Generally, admissible amplitudes of the complex wave are in the range of 10% p.t.p. of the mean pressure; a value of 10 to 15% is admitted for the suction line of the first stage and the interstage pipes, and 10% or less for the delivery line second stage. It is important to have large sections in the pipe near the cylinder, where gas velocity calculated on the mean piston speed should not be higher than 10 meter/sec.

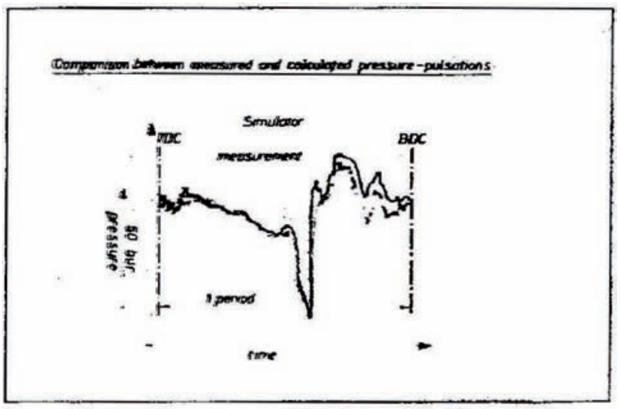


Figure 3. Comparison between measured and calculated pressure pulsations.

5. VIBRATIONS OF THE PIPES

The calculation of the vibrations of the pipes is very complex and requires much work. But it is necessary to determine vibrations already during the layout and to avoid later costly modifications. Figure 5 shows the way to solve this problem. Several general structural programs exist today, based on the method of finite elements and developed for static and dynamic analysis. Frequently, such programs are used for the layout of atomic power stations. They can also be used for the pipework of LDPE plants, but the program should permit the direct solution of a periodical phenomenon.

Calculation as a transient (aperiodical) phenomenon, using an approach step by step would consume too much computer time. The different steps of such an examination will be explained in more details by the following example: Figure 4 shows the delivery pipe first-stage from the cylinder to the intercooler with the chosen fixations.

On this pipe, calculations and measurements for comparison have been made.

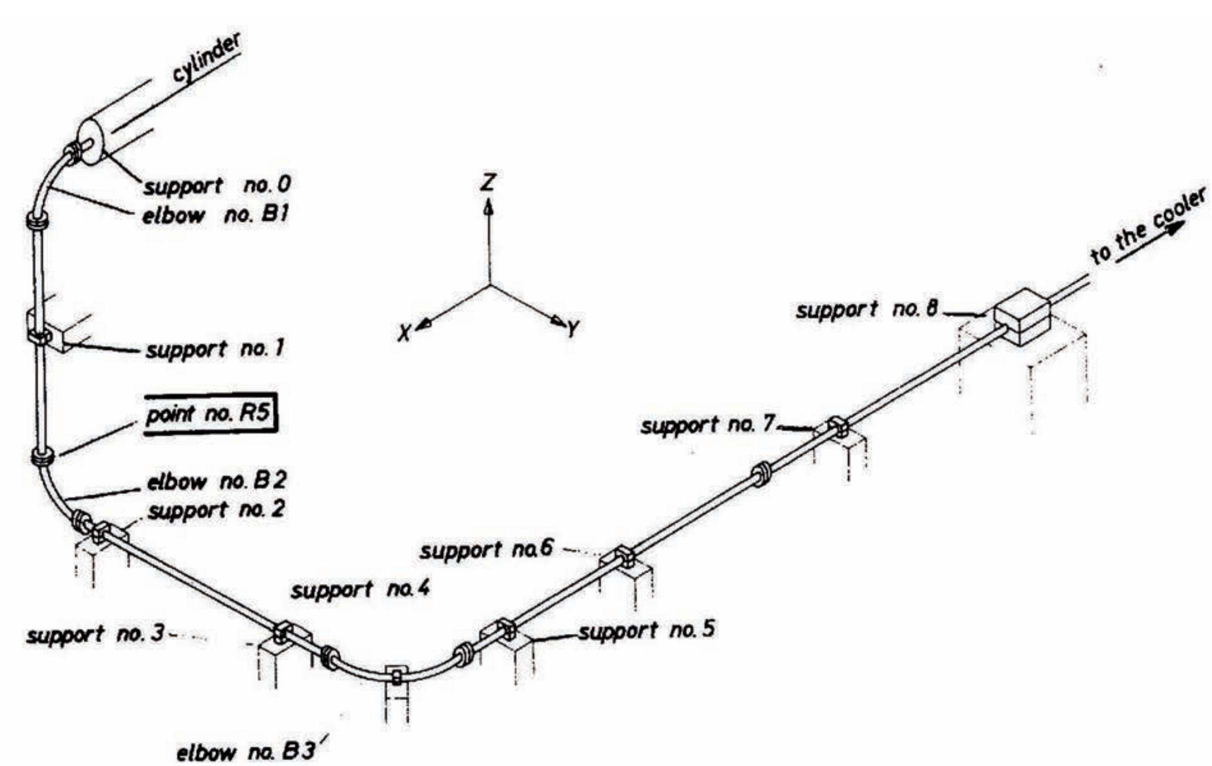


Figure 4. Piping layout.

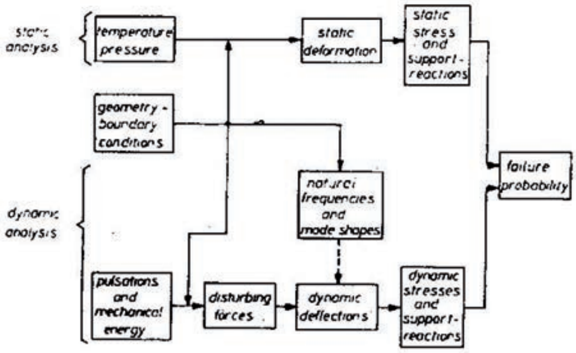


Figure 5. Schematic flowchart of calculations.

First step: Static analysis. The static translations, forces and supports and efforts on the pipe resulting from nominal pressure and temperature are each calculated and then composed. This calculation is known and therefore not explained in detail.

Second step: Analysis of proper frequencies and forms. Already this analysis is very instructive, since it will show resonance cases. The pipe can vibrate at a very great number of proper frequencies, each one having its proper form. It is good policy to keep the lowest proper frequency as high as possible to avoid resonance with the first important harmonic of the gas pulsations. Figure 6 shows the proper frequencies of the pipe treated in this example and the harmonic analysis of the gas pulsations in elbow B1. It is remarkable how close together proper frequencies with corresponding forms can appear. It follows that tuning of the pipe to low frequencies cannot be recommended. Supports for the pipe should be as rigid as possible.

Third step: Dynamic analysis of amplitudes and stresses. Gas pulsations, being the main cause of pipe vibrations, are calculated first. Even if they have relatively small amplitudes (10% p.t.p.), they can still create great stresses, as will be shown here. Figure 7 shows the gas pulsations in two neighboring elbows B1 and B2, as found with the help of the Simulator. Inner pressure in each elbow acts with a certain force on the pipe.

Pressure vibrations on ethylene at high pressure are very rapid, like in a liquid. Therefore, within a short time (about 75 bar), a pressure difference P1 - P2 exists between elbows B1 and B2 due to the phase difference. From this, a periodical force of about 2 ton results at each piston stroke; this has to be taken up mainly by the support, to avoid movements of the pipe.

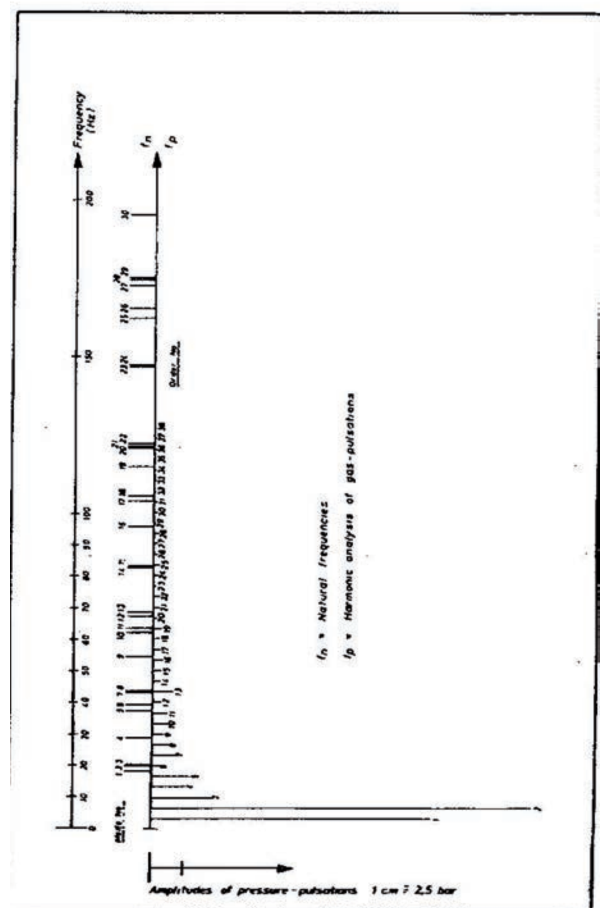


Figure 6. Frequency analysis.

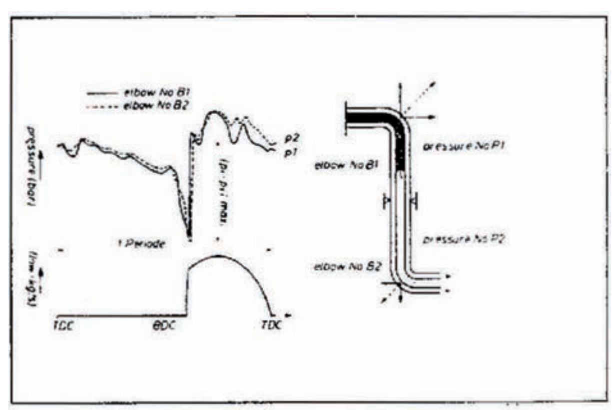


Figure 7. Gas pulsations in the two elbows.

Apart from these vibrations, created by rapid variations of pressure, resonances will occur, which can be dangerous if their frequency is lower than 30 times the compressor speed.

Another source of pipe vibration is the movement of the cylinders, resulting from the varying forces during compression and the elasticity of the materials of construction which has to be taken into account. The momentum created by variations in the direction of the gas flow can be neglected, because it is very small and amounts to max. 20 kp.

The structural program used for this job permits not only to calculate the proper frequencies with the corresponding forces of vibration, but the amplitudes and the resulting stresses as well. Different assumptions must be made at the start, and they can have a great influence on the accuracy of the result. This concerns the stiffness of the pipe supports and their resistance against sliding, the damping, and the point of attack of the existing forces.

Measurement in the laboratory and in the plant were necessary to find the right value for these factors. Figure 8 shows the comparison between measured and calculated values for the above-mentioned example.

Finally, the calculated resulting stresses are compared with the admissible values of fatigue resistance of the pipe material. The aim is not to exceed alternating stresses of ± I kp./sq. mm.

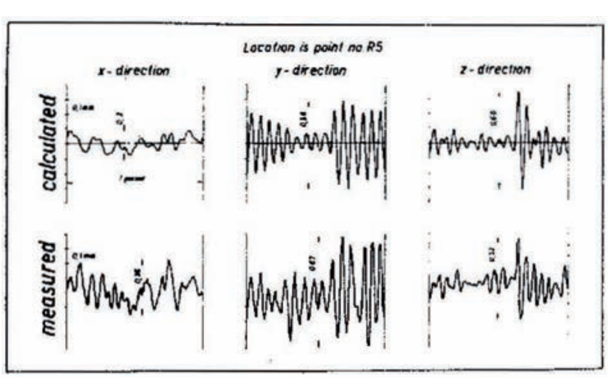


Figure 8. Comparison between calculated and measured deflections.

The expenditure for a complete examination with the above-mentioned three steps ~an be considerable. Depending on the complexity of the system, up to I'500 man-hr. for preparation and evaluation and 12 hr. computer time may be necessary. This expenditure can be reduced if the first two steps are considered sufficient. In the project state, valuable hints will result from these.

6. NOISE

The slow-speed high-pressure compressors do not create a noise problem. The maximum noise level of such compressors with their driving motors does not exceed 85 dBA, and the octave band sound pressure levels are below the values of curve ISO N 80. In the plant it is difficult to reproduce exact noise measurements because the background noise from the neighboring machines such as primary booster compressors and the gas flow in by-pass lines is mostly higher.

Burckhardt Compression

RECIPROCATING COMPRESSORS
LEADING TECHNOLOGY FOR
LOWEST LIFE CYCLE COSTS

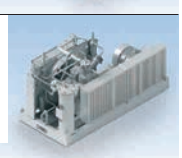
Laby® Compressors
Contactless
and oil-free

Laby®-GI Compressors
Fully balanced

Process Gas Compressors
API 618 – lowest life
cycle costs

Hyper Compressors
Safe and reliable
up to 3'500 bara /
51'000 psia

Standard High Pressure Compressors
Compact package for
demanding applications



COMPRESSOR COMPONENTS
BEST PERFORMANCE
AND LONGEST LIFETIME

- Compressor valves
- Redura® rings & packings
- Capacity control systems
- Capital parts
- Labyrinth piston compressor components
- Hyper/secondary compressor components

SERVICES
THE FULL RANGE

- Burckhardt Valve Service
- Spare parts logistics
- Field service
- Technical support
- Revamps & upgrades
- Component repair
- Condition monitoring & diagnostics
- Training

FOR ALL YOUR RECIPROCATING COMPRESSORS

Burckhardt Compression AG
CH–8404 Winterthur
Switzerland
Tel.: +41 52 262 55 00
Fax: +41 52 262 00 51
24 hours emergency phone: +41 52 262 53 53
info@burckhardtcompression.com
www.burckhardtcompression.com

Your local contact

Compressors for a Lifetime™

Compressors for LDPE – 11/2018