The Institution of Mechanical Engineers

Proceedings 1969–70 · Volume 184 · Part 3R

ꞮECIPROCATING AND ROTARY
S: DESIGN AND OPERATIONAL PROBLEMS

A Conference arranged by the
uid Plant and Machinery Group
13th–16th October 1970

The Institution of Mechanical Engineers

Proceedings 1969–70 · Volume 184 · Part 3R

INDUSTRIAL RECIPROCATING AND ROTARY COMPRESSORS: DESIGN AND OPERATIONAL PROBLEMS

A Conference arranged by the
Fluid Plant and Machinery Group
13th–16th October 1970

1 BIRDCAGE WALK · WESTMINSTER · LONDON · S.W.1

INDUSTRIAL RECIPROCATING AND ROTARY COMPRESSORS: DESIGN AND OPERATIONAL PROBLEMS

Industrial Reciprocating and Rotary Compressors: Design and Operational Problems

A CONFERENCE was held at the Institution of Mechanical Engineers, London, from the 13th to 16th October 1970. It was sponsored by the Fluid Plant and Machinery Group of the Institution; the Ministry of Technology; the British Cryogenics Council; the British Compressed Air Society and the British Mechanical Engineering Confederation Ltd. The conference was formally opened by Mr I. Maddox, C.B., O.B.E., F.R.S., and 298 delegates registered to attend.

The papers were divided into eight sessions for presentation and discussion.

Tuesday, 13th October

Session 1; Chairman: Mr G. F. Arkless, B.Sc., C.Eng., F.I.Mech.E.
Papers 20 and 21

Session 2; Chairman: Mr M. H. Vogel, C.Eng.
Papers 8, 7 and 10

Wednesday, 14th October

Session 3; Chairman: Mr G. F. Arkless, B.Sc., C.Eng., F.I.Mech.E.
Papers 13, 2, 3, 23 and 9

Session 4; Chairman: Mr R. P. Clark, B.Sc., C.Eng., M.I.Mech.E.
Papers 1, 11 and 17

Thursday, 15th October

Session 5; Chairman: Mr R. P. Clark, B.Sc., C.Eng., M.I.Mech.E.
Papers 5, 18 and 22

Session 6; Chairman: Mr M. H. Vogel, C.Eng.
Papers 6 and 15

Friday, 16th October

Session 7; Chairman: Mr I. E. Graham, C.Eng., F.I.Mech.E.
Papers 14, 16 and 19

Session 8; Chairman: Mr E. Lacy-Hulbert
Papers 12 and 4

A Conference Dinner was held in the Westminster Suite of the Europa Hotel, London, on the evening of Wednesday, 14th October.

The members of the Planning Panel responsible for organizing the conference were: Mr G. F. Arkless (Chairman), Mr R. P. Clark, Mr I. E. Graham, Mr A. Johnson, M.A., Mr E. Lacy-Hulbert and Mr M. H. Vogel.

CONTENTS

Paper 1

INDUSTRIAL RECIPROCATING COMPRESSORS FOR VERY HIGH PRESSURES

C. Matile*

The industrial use of very high pressures (above 1000 atm) is limited today to the production of low-density polyethylene. After a survey of the general conditions of thermodynamics, different partial aspects are dealt with from the point of view of development design, and operation.

INTRODUCTION

INTIMATELY BOUND to the chemical industry and based on a long evolution, the main stages of which were the liquefaction of air and the synthesis of ammonia, the technique of using very high pressures was eventually perfected from developments in the manufacture of low-density polyethylene. This is now the only industry requiring large reciprocating compressors for very high pressures, as the pressures necessary for other main chemical processes have been steadily reduced since 1945. For this reason, the present paper will be restricted to the ethylene compressors.

Considering that the classical designs of reciprocating high-pressure compressors cover an uninterrupted range up to about 1000 atm, 'very high pressures' will imply those above 1000 atm.

A characteristic feature of the high-pressure ethylene polymerization process is that a very large difference in pressure is necessary between the inlet gas entering the reactor and the outlet of the recycle gas. The recirculators, generally called secondary compressors, work between two limits, i.e. 100–300 atm on the suction side and 1500–3500 atm on the delivery side, for most of the existing processes. As the coefficient of reaction lies between 16 and 30 per cent, the secondary compressors have to handle three to six times the fresh gas quantity, being thus by far the most powerful machines in the production stream. Their unit capacities which, when the industrial expansion first began, were of 4–5 ton/h, now lie between 15 and 50 ton/h and their power requirement per unit has been increased from 600 hp up to some 10 000 hp.

The MS. of this paper was received at the Institution on 27th February 1970 and accepted for publication on 7th April 1970. 22

* *Manager, The Burckhardt Engineering Works Ltd, Dornacherstrasse 192/210, 4002–Basle, Switzerland.*

As the whole operating range of these secondary compressors takes place well above the critical point of ethylene, the thermodynamic behaviour of the fluid lies somewhere between that of a gas and that of a liquid. This peculiar condition has two main effects. The first is a very small reduction of the specific volume with increasing pressure; for instance, at a temperature of 25°C the specific volume is 3 dm^3/kg at 100 atm, 2 dm^3/kg at 700 atm, and 1·5 dm^3/kg at 4500 atm. The second effect is a very moderate rise of adiabatic temperature with increasing pressure; for instance, with suction conditions of 200 atm and 20°C the delivery temperature will reach only 100°C at 2000 atm.

These particular thermodynamic conditions greatly influence the design of high-pressure ethylene compressors. Compared with the classical reciprocating compressor, the compression ratio is of little practical significance, the important factor being the final compression temperature which should not exceed 80–120°C, depending on process, gas purity, catalyst, etc., in order to avoid premature polymerization. The influence of the cylinder dead clearance on the volumetric efficiency is slight because of the small reduction of specific volume, and very high compression ratios are therefore possible with quite admissible efficiency. In addition, the stability of intermediate pressures depends chiefly on the accuracy of temperatures; for instance, in the case of a two-stage compression from a suction pressure of 200 atm to a delivery pressure of 2500 atm, a drop in the first-stage suction temperature from 40 to 20°C will cause the intermediate pressure to rise from 1000 to almost 1600 atm.

For these reasons, a secondary compressor is required which has only one or two stages, despite the very large pressure differences involved. However, this again compels the designer to face extremely high mechanical strains due to the high amplitude of pressure fluctuation in the cylinders.

Finally, an additional and sometimes disturbing feature of ethylene must be mentioned. If the gas reaches a very high pressure and a high temperature simultaneously (which can easily occur in a blocked delivery port owing to the very low compressibility), it will decompose into carbon black and hydrogen in an exothermic reaction of explosive character. This danger must be taken into account by the designer.

CYLINDERS AND PISTON SEALS

Sealing of the high-pressure compression chamber is a major problem, and this could be solved by avoiding friction between moving and stationary parts. This has been realized for laboratory equipment and small-scale pilot plants by the use of either metallic diaphragms or mercury lutes in U-tubes, and such arrangements are still in use for research purposes. In addition, they have the advantage of avoiding any contamination of the compressed gas by any lubricant. Unfortunately, chiefly for economic reasons, they proved to be impracticable for industrial compressors, at least in the present state of techniques. Thus, as labyrinth seals are out of the question for very high pressures, friction seals have to be accepted; in fact, two solutions are currently used—moving and stationary seals.

Metallic piston rings are the only sort of moving seals used in the large reciprocating type of compressor. They are generally made in three pieces: two sealing rings, each covering the slots of the other, and an expander ring behind both of them which also seals the gaps in the radial direction. The materials used are special grade cast iron, bronze, or a combination of both, with cast iron or steel for the expander. The piston, of built-up design, comprises a series of supporting and intermediate rings with a guide ring on top of them and a through-going bolt (two different designs are shown in Fig. 1.1). All parts of the piston are made of high tensile steel and particular care must be given to the design and to the stress calculation of the central bolt, which is subjected to severe strain fluctuations.

The use of piston rings allows for a simple cylinder design, the main part of which is a liner that has been thermally shrunk to withstand the high variations of the internal pressure (see Figs 1.2 and 1.3). The inner sleeve, which was previously made of nitrided steel, is now generally of massive sintered material like tungsten carbide. The use of this expensive material is justified by two beneficial qualities: it possesses an extremely hard surface and has a high modulus of elasticity. The first considerably improves the conditions of friction and greatly reduces the danger of seizure. Owing to the high modulus of elasticity, the amplitude of the 'breathing' movement under the internal pressure fluctuation is much smaller than with steel, and thus the stress variations in the expanded outer sleeves is appreciably reduced. However, as these sintered materials have a very poor tensile strength, care must be taken to ensure that the inner sleeve is always under compression, even if the temperature increases. This is the main purpose of the external cooling of the liner.

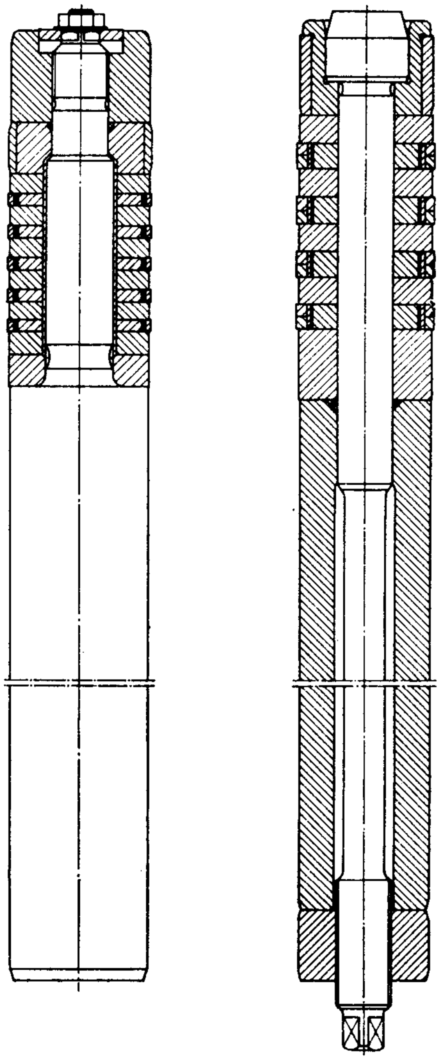


Fig. 1.1. High-pressure pistons with piston rings

Packed plungers are the other answer to piston sealing. Although some manufacturers still use packings of hard plastic materials (nylon or similar), the most widely used packings are the metallic self-adjusting type. They are usually assembled in pairs, the actual sealing ring tangentially split into three or six pieces being covered by a three-piece radially cut section. Both are usually made of bronze, kept closed by surrounding garter springs, and held in place by locating and supporting steel plates. These plates must also be thermally shrunk to resist the high variations in internal pressure. Unfortunately the use of sintered hard materials is restricted by the fact that the supporting plates are subjected, in the axial direction, to

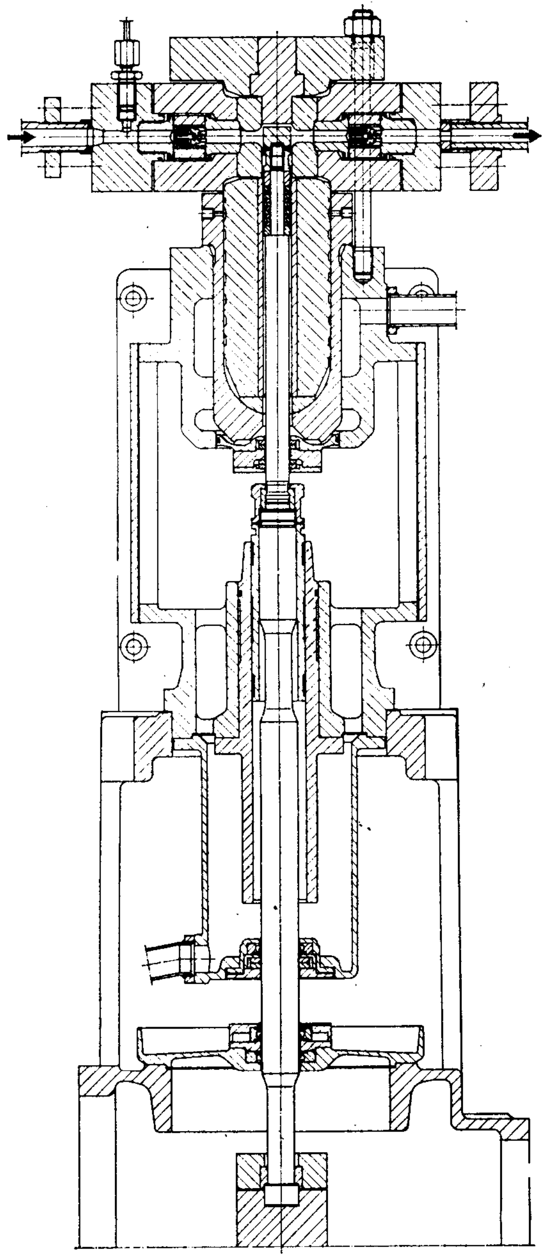


Fig. 1.2. High-pressure cylinder for moderate end pressures

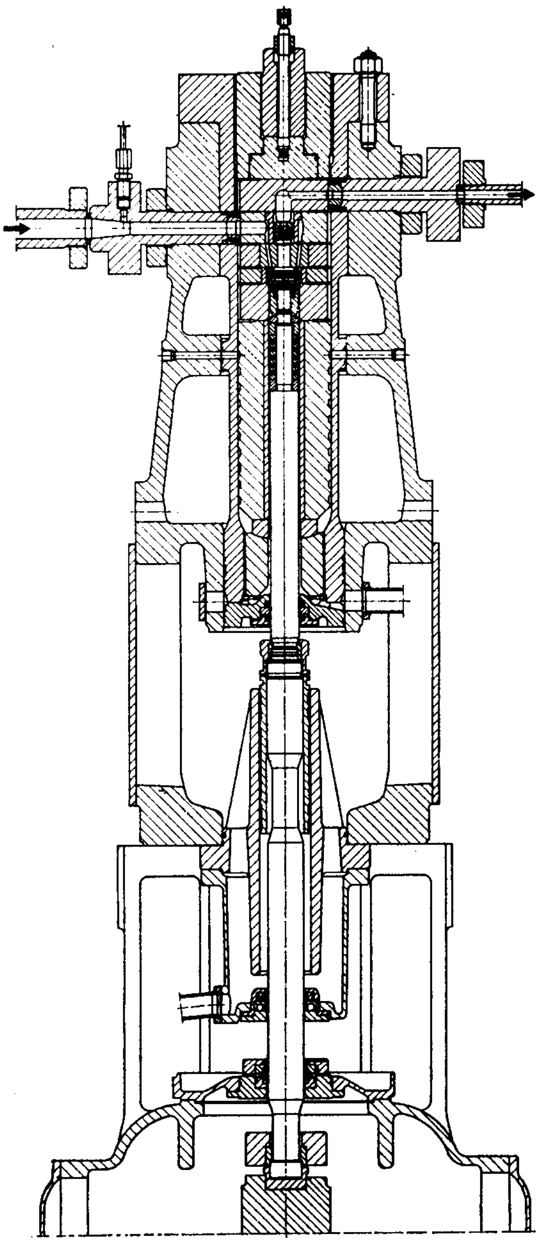


Fig. 1.3. High-pressure cylinder for medium end pressures

heavy bending and shearing forces which these materials generally cannot stand. To improve the friction conditions of the packing rings, the high tensile steel supporting discs are frequently surface hardened or plated with carbide. The plungers are made of nitrided steel for use in moderate pressures, and for higher pressures are of steel, plated with hard materials. For very high pressures the use of solid bars of hard metal is the best wear-resistant solution for both plungers and packings. The disadvantage of the packed plunger design lies in the much larger joint diameters of the static cylinder parts, which require two to three times higher closing forces than the piston ring design. Large cylinders, such as the one shown in Fig. 1.4, need a pretensioning of the cylinder bolts to about 10 times the maximum plunger load. This ratio is higher for smaller cylinders.

For piston rings and packed plungers the optimum number of sealing elements appears to be four or five. In

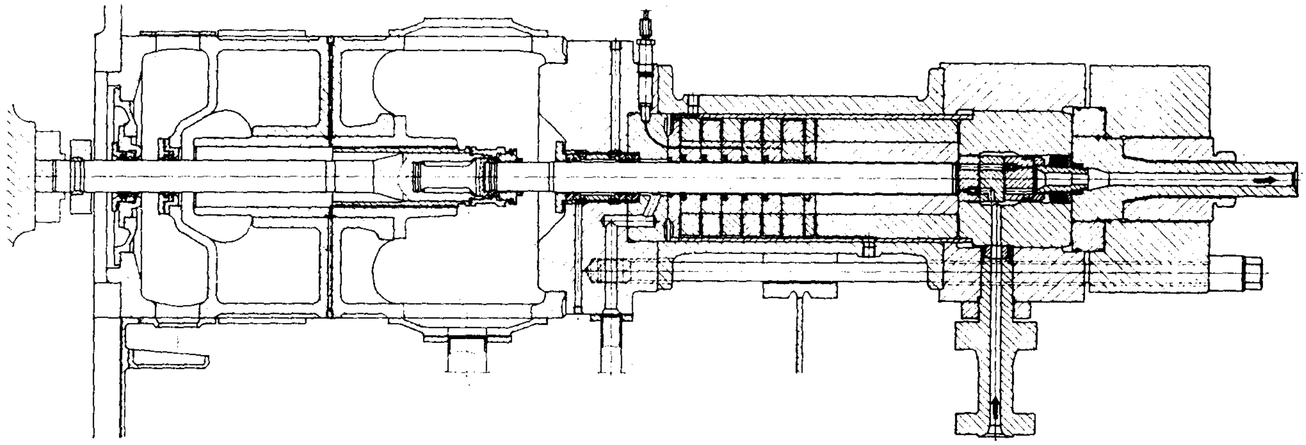


Fig. 1.4. Gas cylinder for very high end pressures

both solutions, it is essential that the piston be accurately centred if the seals are to be effective; this is the reason for the guiding ring within the cylinder and for the additional guide at the connection between the piston and driving rod. At the base of the cylinders an additional low-pressure gland allows gas leaks to be collected and the plunger to be flushed and cooled. Other separate glands positioned on the rod connecting the piston to the drive (see Figs 1.2 and 1.3) prevent the cylinder lubricant from mixing with the crankcase oil, and as the intermediate space is open to atmosphere, it is impossible for gas to enter the working parts.

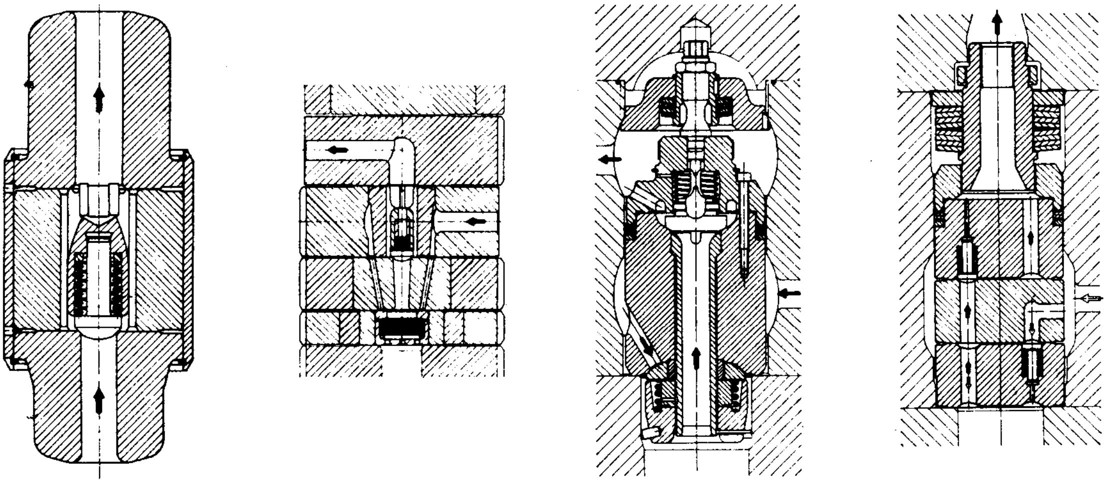
From the point of view of design and maintenance, piston rings would appear to be the most adequate solution, and they are currently used for pressures up to 2000 atm, or in some circumstances up to 3000 atm. The choice between them and the packed plungers depends largely on the process and type of lubricant used. One difficulty is that normal mineral oils are dissolved by ethylene under high pressure to such an extent that they have no longer any lubricating power. The glycerine used in earlier machines has been widely replaced by paraffin oil, either pure or with wax additives, which is much less diluted by the gas than other mineral oils. However, it is a rather poor lubricant, and is inferior to the various types of new synthetic lubricants, which are generally based on hydrocarbons. The basic difference between piston rings and plunger packings is that the latter may be lubricated by direct injection, while piston rings are lubricated indirectly. This may be an advantage since the low polymers carried by the return gas back from the reactor are reasonably good lubricants. However, too large an amount of low polymers causes the rings to stick in their grooves, and some kinds of catalyst carriers also brought back by the gas are excellent solvents for lubricants. Thus, the most convenient solution has to be selected for each specific case. In general, for higher delivery pressures (above 2000–2500 atm) better results are obtained with the use of packed plungers.

CYLINDER HEADS AND VALVES

It is relatively simple to construct a vessel that will resist 2000 atm, but the problem becomes more intricate when the vessel must withstand, for years, a pressure which fluctuates between 300 and 2000 atm at a frequency of 3–4 Hz. The leading idea of the designer must be to divide a complicated problem into a series of simple ones, each of which is then accessible to accurate methods of investigation. If this is done properly, it is possible to divide a large piece at the very places where inadmissible changes of stresses would occur, and to keep the combined strains in each item within tolerable limits. The examples of cylinders shown in Figs 1.2, 1.3, and 1.4 illustrate the result of this method. The striking feature is the very simple shape of all pieces subjected to high pressure.

A first obvious result is that suction and delivery valves have to be located in a separate cylinder head. Fig. 1.2 shows one type of cylinder head that can be used for moderate pressure fluctuations (up to amplitudes of about 1200 atm) and moderate cylinder dimensions. The intersection of the gas passages with the main bore is located in a small forged core, shrunk in a heavy outer flange, and pressed by the upper cover in the axial direction. This piece has a quite symmetrical shape with carefully rounded internal edges. By dismantling only the upper cover, it is possible with this design to pull out the complete piston with its rings through the central hole without disconnecting the gas pipes and without removing the valves. A typical valve for this kind of cylinder head is shown in Fig. 1.5*a*. The same valve is used on the suction and delivery side, the two endpieces being differently shaped to avoid incorrect assembly. The valve is held against the central head piece by the connection flange of the gas piping, as a kind of composite lens.

For higher amplitudes of pressure and larger cylinders, cross bores and duct derivations must be taken away from the area of large pressure fluctuations. This is effected by the use of central valves, combining suction and delivery valves into one concentric set. For cylinders of moderate



a b c d

Fig. 1.5. Different designs of suction and delivery valves

size this can be done as shown in Figs 1.3 and 1.5*b*: the different valve elements are located in a succession of simply shaped discs, with the same diameter as the cylinder liner and piled up on top of it. The lower two discs, which have been produced by the shrinking technique, receive the pressure fluctuation in their central hole, while the upper two, which contain the radial bores for the gas connections, are subjected only to static pressure.

For still larger cylinders, the combined valve is assembled as a separate unit, in order to keep compact dimensions and weights, and is inserted into the central hole of the cylinder head core, as shown in Fig. 1.4. The two valves illustrated in Fig. 1.5*c* and *d* are designed on the same basic principle, the last one, used in very large cylinders, being fitted with multiple suction and delivery poppets in order to reduce the moving masses. The entire valve body is subjected to suction pressure on the outside and only to the pressure fluctuations in the longitudinal hole. The suction pipe is connected to the radial bore of the cylinder head core as shown in Fig. 1.4. Separation of suction and delivery pressures is assured by the circumferential self-sealing ring of hard plastic material, as shown in Fig. 1.5*c* and *d*. The entire valve is pressed on the end of the cylinder liner by the difference of pressures: the set of plate springs visible in the figure have only to maintain the valve against pressure drop during periods of operation on by-pass. The gas delivery pipe is connected radially to the core piece (like the suction one) for moderate delivery pressures, and axially for higher pressures (as shown in Fig. 1.4).

All components subjected to high stresses, particularly the internal cylinder elements under high tri-dimensional fatigue strains, are generally investigated at the design stage by three different methods. The first is a conventional calculation of combined stresses, based on the classical hypotheses, using computer programs as far as convenient. The second approach is that of the frozen stress technique of photoelasticity applied on resin models cast either on full scale or on slightly reduced scale: it supplies accurate information about the course of the two main stresses in every plane section within the material. The third method is a direct measurement of the superficial stresses by means of strain gauges on the actual component subjected to the full prestressing and internal pressure. A variation of this last method consists in stress-measuring on an enlarged model made of a low-modulus material like aluminium: it provides better information through strain gauges on small rounded edges, and allows progressive modification of such places in an attempt to reach an optimum. Comparison of results of these different methods gives a very useful reciprocal check on their exactness and accuracy.

DRIVE

Different types of driving mechanism, actually in use or having once been in use on industrial reciprocating compressors for very high pressures, are diagrammatically illustrated in Fig. 1.6. The first two (Fig. 1.6*a* and *b*) have been extensively used during the initial period of development and are still applied to smaller units. They are characterized by the fact that the high-pressure cylinders have been fitted to frames of classical design, without substantial modification of the existing equipment. Some manufacturers simply allowed for the possibility of difficulties arising from the purely unilateral loading of the crosshead pins—and they generally got into trouble. Others tried to balance the forces by getting additional pistons set under constant or variable gas pressure in the reverse direction; this may work, but is a rather unsatisfactory solution as it is expensive and introduces supplementary wearing elements. The best means of application is to use special high-pressure lubrication pumps, fastened to the crossheads and driven by the rocking movement of the connecting rods, which inject the oil directly into the crosshead bearings, thus lifting the pins against the load. This arrangement is well known from the design of large diesel engines, but as the requirements called for higher delivery pressures and larger capacities, the solutions shown in Fig. 1.6*a* and *b* appeared to be increasingly unsatisfactory. Since it is impossible to use double acting pistons on very high pressures, these designs load the driving mechanism with the full gas pressure (instead of the difference between delivery and suction pressures) and are working only on each second stroke. While this was still admissible for small units, it proved to be uneconomic for larger ones, and there was obviously a need for more specialized constructions.

The widespread design represented in Fig. 1.6*c* is still based on a conventional application of the horizontally opposed reciprocating compressor, but it avoids the above difficulty by having, on each side of the frame, an external yoke which is rigidly connected to the main crosshead by means of solid connecting bars. A pair of opposed pistons (or plungers) is then coupled to each yoke which is shaped as an outboard crosshead. This is not a bad solution because, due to the long flexible connecting bars, the movement of the yoke is not disturbed by any transverse force, and it allows a full loading of the drive. However, the compressor is becoming extremely wide and the accessibility to each second high-pressure cylinder is rather poor.

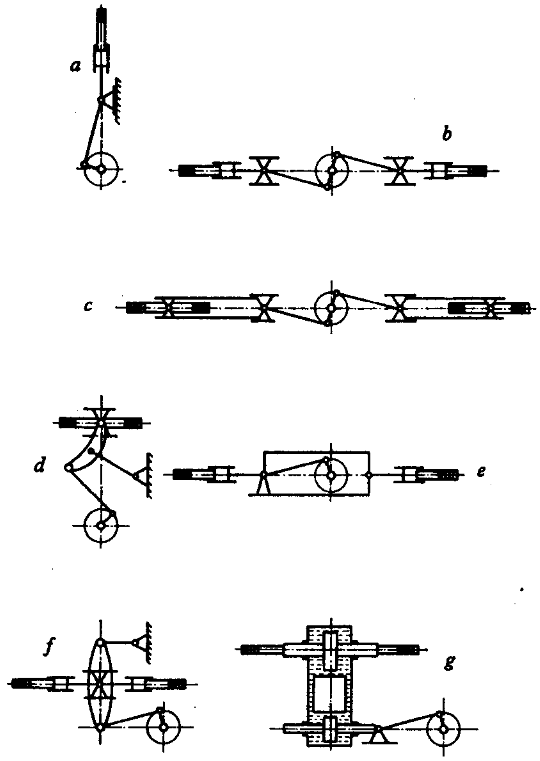


Fig. 1.6. Diagram of different types of driving mechanism

All the other systems illustrated in the figure are specially designed solutions: Fig. 1.6*d* and *f* use a rocking beam bound to a fulcrum by a lever, which gives a linear translation of the rotary movement. Fig. 1.6*e* uses a moving frame surrounding the crankshaft to connect the crosshead to the piston on the opposite side—a solution already applied for more than half a century to high-pressure pumps; and Fig. 1.6*g* is based on the idea of hydraulic transmission of the driving power. It should be noticed that Fig. 1.6*d* may perform a modification of the stroke of the crankshaft in a fixed predetermined ratio, that Fig. 1.6*f* reduces the stroke in a fixed ratio, and that Fig. 1.6*g* can perform a variable reduction of the stroke. Although Fig. 1.6*e* appears to be the best specific design for a large production compressor, the very special solution of Fig. 1.6*g* is worth a further explanation.

Fig. 1.7 shows the basic, greatly simplified diagram of operation. By means of two reciprocating columns of fluid, a double-acting primary piston operates a secondary piston located above it. A pair of opposed high-pressure gas pistons are coupled to the latter. Although the hydraulic transmission of power could theoretically work as a closed system, it is actually necessary to renew the fluid continuously through a forced feed recirculation, both for the purpose of cooling and to compensate for possible seal leaks. Fig. 1.7 shows a low-pressure feeding system; it has also been made as a high-pressure feed. As this transmission may be built as a hydraulic intensifier, it is possible to use a comparatively light primary mechanism at rather high speed, and to reduce the linear speed and increase the forces on the secondary part. Furthermore, by opening a by-pass valve between the two fluid columns, the secondary stroke may be reduced. In this manner, a stepless output control can be achieved down to zero. As the fluid pressures on both sides of the pistons vary according to two opposed indicator diagrams, there are two points on each stroke where they will balance. If such a by-pass is opened wide on the first of these points, the fluid will flow over theoretically without losses, and the secondary piston will stand still until the valve is closed. In fact this is one of the very few ways of realizing a power-

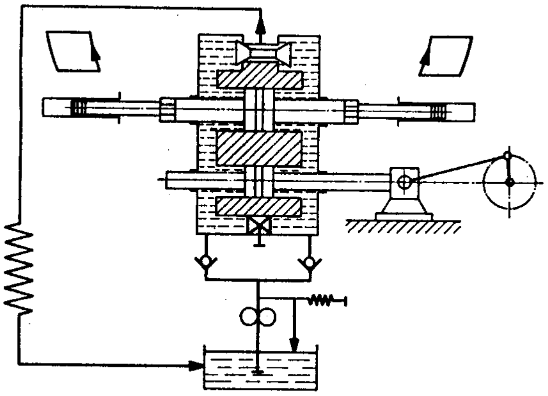


Fig. 1.7. Diagram of hydraulic transmission of power

saving capacity control of reciprocating compressors for very high pressures. This output control can be governed automatically, and being applied separately to each compression stage makes it possible to control exactly the intermediate pressure.

MISCELLANEOUS

The number of problems posed by industrial reciprocating compressors for very high pressures is almost unlimited. Nearly every question of installation or maintenance needs a special study and an original answer, and all elements and accessories require special design and calculation. Only a few of them will be mentioned here.

It is common practice, when designing large reciprocating compressors, to take into account three different kinds of strains for selecting the most favourable crank angle arrangement. First, there are the resulting forces and moments of inertia acting on the foundations; second, the resulting torque diagrams under different conditions of operation (important for the cyclic variations of current consumption of the driving motor); and last, the forces due to pressure pulsations in the gas piping. For most compressors working at lower pressures, this last consideration may be deleted or answered in a summary way at the initial stage of design, because it may be solved by the use of surge drums. In the case of very high pressures, the gas pulsations, which are capable of destroying the piping system, have to be given the first priority in the basic investigations, even if it sometimes leads to acceptance of higher inertia forces.

The most practical way of studying the gas pulsations is to use an analogue computer, which is in fact an electro-acoustical analogical system where every part is individually adjustable or replaceable. The first purpose of the analysis is to avoid any resonance between the active systems (the compression cylinders) and the passive systems (the whole piping network); the second purpose is to reduce the amplitudes of the remaining pressure pulsations as far as possible. Theoretically, the means available are: change of diameter and of length of the gas piping; removal of pipe connections or adjunction of additional piping; use of pulsation snubbers and of orifices at well selected places, etc. In reality the possibilities are restricted because of the high speed of sound in the gas (1000–2000 m/s), because of the very low compressibility of the gas, and because of the very high price of vessels and piping. However, in many cases it has proved easily possible to reduce a dangerous pulsation (for instance,

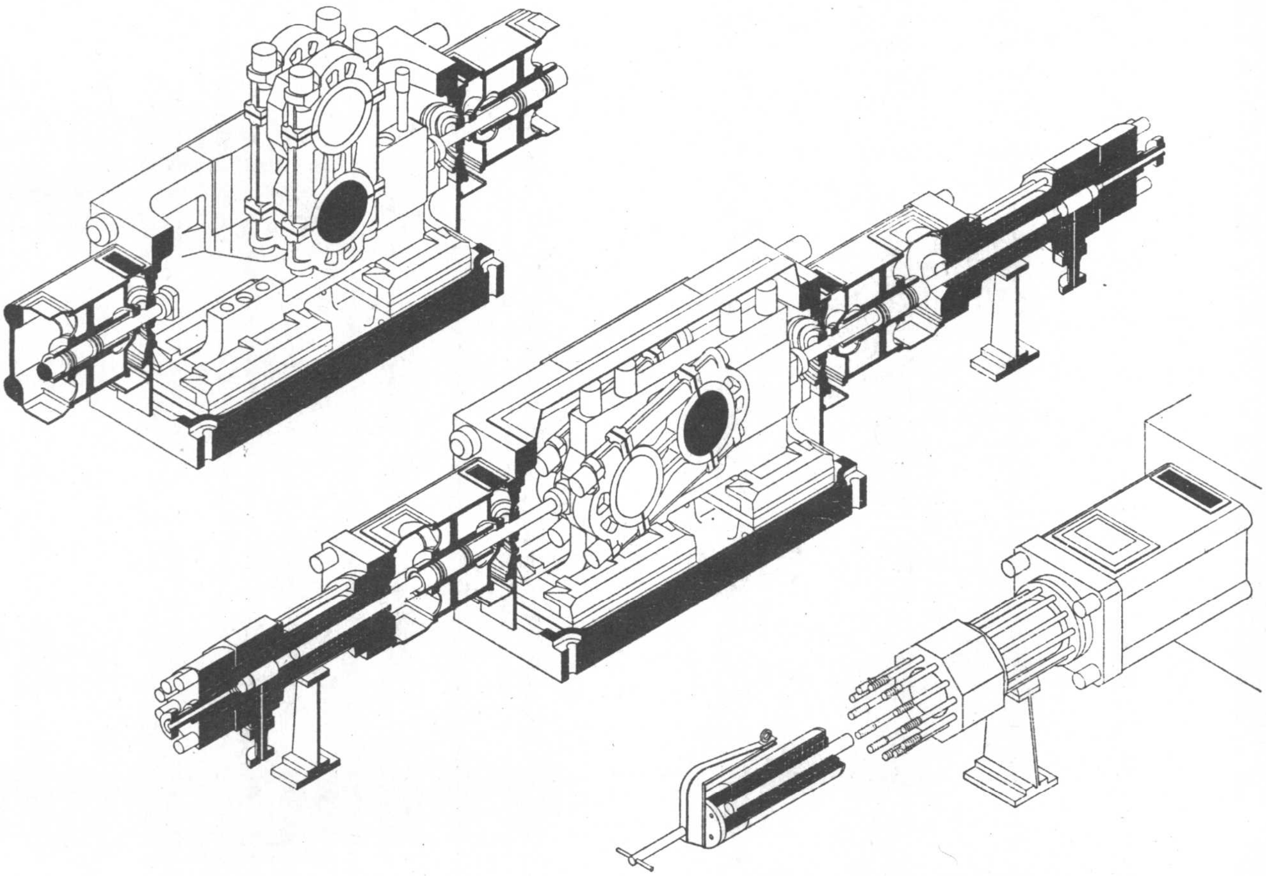


Fig. 1.8. Large mechanically driven compressor for very high pressures

from 25 per cent down to 5 per cent) by cheap and simple means.

Designers dealing with compressors for very high pressures need to keep in mind at least three basic ideas: (1) safety, (2) large forces (how to apply them), and (3) accessibility. The last two are, of course, chiefly economic but they are often combined with the aim of safety. For instance, in the design of large compression cylinders, as shown in Fig. 1.4, the long through-going bolts connecting the base with the cylinder head are an important safety factor: if by chance the gas decomposed in the cylinder, these long bolts, acting as springs, would be elastically lengthened by an appreciable amount without significantly increasing the stresses, and would allow the gas to escape between the liner and the head. They must all be equally pretensioned with a very high force. If done by hand, this would be an extremely tiring and time-consuming exercise, and for this reason a hydraulic piston has been incorporated within the cylinder head which allows, when set under oil pressure, a very quick, easy, and regular tightening and loosening of the bolts. After removal of the outer flange, the whole inside of the cylinder can be removed, with the help of a lifting device, as a closed cartridge, as shown in Fig. 1.8. The same figure also shows how major parts of the driving mechanism may be dismantled without removal of the cylinders.

CONCLUSION

Although closely related to other reciprocating compressors, the industrial compressors for very high pressures require the construction of a separate group of machines, different in many ways, and call for much greater research, development, and calculation than the others. Being compelled to employ all materials very near to their limits of resistance, the designers are bound to keep in close contact with the latest developments of science and technique in many fields.

ACKNOWLEDGEMENTS

The author expresses his gratitude to the Burckhardt Engineering Works Ltd, Basle, for permission to use the illustrations presented in this paper.

Paper 2

EFFECTS OF RECIPROCATING COMPRESSOR VALVE DESIGN ON PERFORMANCE AND RELIABILITY

H. Davis*

The effect of flow through compressor valve restrictions on the efficiency of the cylinder is well known. This paper provides a method of predetermining these pressure losses from the geometry of the valve design and the application conditions of the compressor stage. The model analysed is that of flow through multiple restrictions in series. The end result is a derived quantity, the 'equivalent area', which is the effective area of a single restriction equivalent to the total effect of the actual multiple restrictions. The relationships involving the effects of equivalent area and all other compressor parameters on compressor performance is presented in non-dimensional form. Tests are described which determined actual values of equivalent area for several valve designs and the correlations are presented, which confirm the ability to predict performance of an untested valve from its design and application. Reliability criteria are hypothesized based on considerations of the motion characteristics of the valve elements. Experimental observations of valve motion are described, and the desired characteristics defined from which the quantitative criteria are obtained. Correlations between criteria limits and field experience are presented.

INTRODUCTION

THIS PAPER will describe the approaches to understanding the behaviour of compressor valves. These approaches have formed the guide lines for valve research over the past five years, and some of the significant results will be described.

The emphasis on compressor valve research arises from our intention to gain a full understanding of the factors involving valve reliability and power losses, as they are a function of the valve design variables. It is known that reliability depends on avoiding excessive wear or stresses produced by any phenomena arising from motion, and that power losses are related to the flow properties through restrictions of varying degree, also a function of the motion.

Notation

A	Area of element, in^2.
A_{eq}	Equivalent flow area, in^2.
$\sum A_{eq}$	Total equivalent area for multiple valves, in^2.
A_i	Area of individual flow restriction, in^2.
A_l	Flow area through element lift dimension, in^2.
A_p	Net piston area, in^2.
C_d	Drag coefficient of element.
C_{eq}	Velocity through A_{eq} due to piston displacement rate, ft/s.
E	Modulus of elasticity, lbf/in^2.
F	Lifting force on element, lbf.
G	Mass flow rate, lb/s.
g_c	Gravitational constant (32·2 ft/s^2), ft lb/s^2 lbf.
h	Maximum deflection of feather valve strip, fully open, in.
I	Moment of inertia for bending, in^4.
K_i	Orifice constant of individual restriction area.
k	Spring constant of each spring; plate valve element, lbf/in.
L	Stroke, in.
L_f	Free length of spring, in.
L_1	Compressed length of spring, element closed, in.
L_2	Compressed length of spring, element open, in.
l_c	Connecting rod length between centres, in.
l_s	Length of element, in.
M_c	Dimensionless piston speed parameter, $[=\pi LN/720\sqrt{(n_v g Z R T)}]$.
N	Speed, rev/min.

The MS. of this paper was received at the Institution on 17th February 1970 and accepted for publication on 7th April 1970. 21

* *Manager of Research and Development, Worthington Compressor and Engine International, Buffalo Operations, Buffalo, New York.*

n	Ratio of specific heats.
n_s	Number of springs per plate valve element.
n_T	Isentropic pressure–temperature exponent in $pT^{n_T/(n_T-1)}\|_s = \text{const.}$
n_v	Isentropic pressure–volume exponent in $pv^{n_v}\|_s = \text{const.}$
p	Pressure, lbf/in².
Δp	Summation of pressure drops, lbf/in².
Δp_i	Pressure drop of individual restriction, lbf/in².
Δp_{av}	Weighted average pressure drop, lbf/in².
Q	Instantaneous volume flow rate, ft³/s.
R	Gas constant (= 1545/mol. wt), ft lbf/lb degR.
r	Crank radius, in.
s	Entropy.
T	Temperature, °R.
t	Time, s.
t_s	Thickness of element, in.
V_p	Piston velocity, ft/s.
W	Mass of element, plate valve, lb.
W_1	Energy loss, ft lbf.
Z	Compressibility factor in $pv = 144ZRT$.
β	Area ratio $(=A_{eq}/A_p)$.
γ_g	Density of gas, lb/ft³.
γ_s	Density of material, lb/in³.
η_{int}	Internal cylinder efficiency.
η_{is}	Isentropic efficiency.
η_m	Mechanical efficiency.
η_v	Volumetric efficiency.
$\left.\begin{matrix}\theta_1\\ \theta_2\\ \theta_3\end{matrix}\right\}$	Time quantities measured in crank degrees, related to respective τ by $\theta = 6N\tau$.
λ	Function of volumetric efficiency.
μ	[(Piston velocity)² factor]. Fig. 2.13.
μ_s	Mass per unit length, lb s²/in².
ρ_{ext}	Overall cylinder pressure ratio $(=p_2/p_1)$.
ρ_{int}	Internal cylinder pressure ratio.
τ_1	Time for element to freely close, s.
τ_2	Time in advance of end of stroke when pressure forces equal spring forces on element, s.
τ_3	Time in advance of end of stroke when element opens, s.
ω	Natural frequency, rad/s.

Subscripts for p, T, and Z

0	Reference conditions for standard volume.
1	Suction.
2	Discharge.

Abbreviations in equations or figures

BHP	Brake horsepower.
CED	Crank end discharge.
CES	Crank end suction.
HED	Head end discharge.
HES	Head end suction.
IHP	Indicated horsepower.
SCFM	Compressor flow rate, volume at standard conditions, ft³/min.

STATEMENT OF THE PROBLEM

This paper addresses the problem of relating the power expended as a result of valve pressure losses to the detail design of the valves, and secondly the establishment of a set of rational criteria of reliability which relate the design of the valves to the application of the compressor.

Compressor valves, being self-actuating devices, attain a state of motion depending upon:

(*a*) pressure–time relationships on both sides of the valve; and

(*b*) mass–elastic properties of the moving parts.

The pressure–time relationships depend upon the motion since there is a fluid flow relation between pressures, areas, and velocities for any position of the moving parts. Hence, an explicit statement of pressure conditions as a function of motion, or vice versa, is not possible. The mathematics of each phenomena, valve motion, and gas conditions must be separately stated. The effect of each on the other must be computed step by step using small time intervals with computer techniques.

Properties of compressor valve assemblies

Valve designs under investigation include plate valves (Fig. 2.1*a*) and feather valves* (Fig. 2.1*b*), both of which are commonly used in process compressors.

The plate valves have several separate concentric rings, each with a number of small compression springs. The feather valves have several individual flexing sealing strips with no other springs.

Each of these types of valve elements has a set of mass–elastic properties that can be simply described in terms of spring forces, stiffnesses, and masses. Similarly, each valve configuration has the flow properties of a restriction of varying dimension.

We have measured the motion of the moving parts of these valves in actual compressor service. Typical results, shown in Fig. 2.2, were obtained with special instrumentation using optical fibres to conduct light to and from the moving elements of a valve. Intensity of the light reflection from the moving part can be calibrated in terms of its position.

The same valves were instrumented to measure the pressure change across the valve and some of these data are also shown in Fig. 2.2.

From these records it is apparent that the position of the valve sealing member is not simply described as either fully open or closed, but that the plate or strip can be partially open for a significant time and that the motion is generally complex.

It should be noted that there is a definite coupling between the pressure change across the valve and the valve position, which was, of course, expected.

Further analysis of these data is derived from the relationship that the algebraic sum of the gas forces, elastic forces, and friction forces acting on the moving parts in the system produce the acceleration of the moving masses.

* *Registered U.S. trade-mark.*

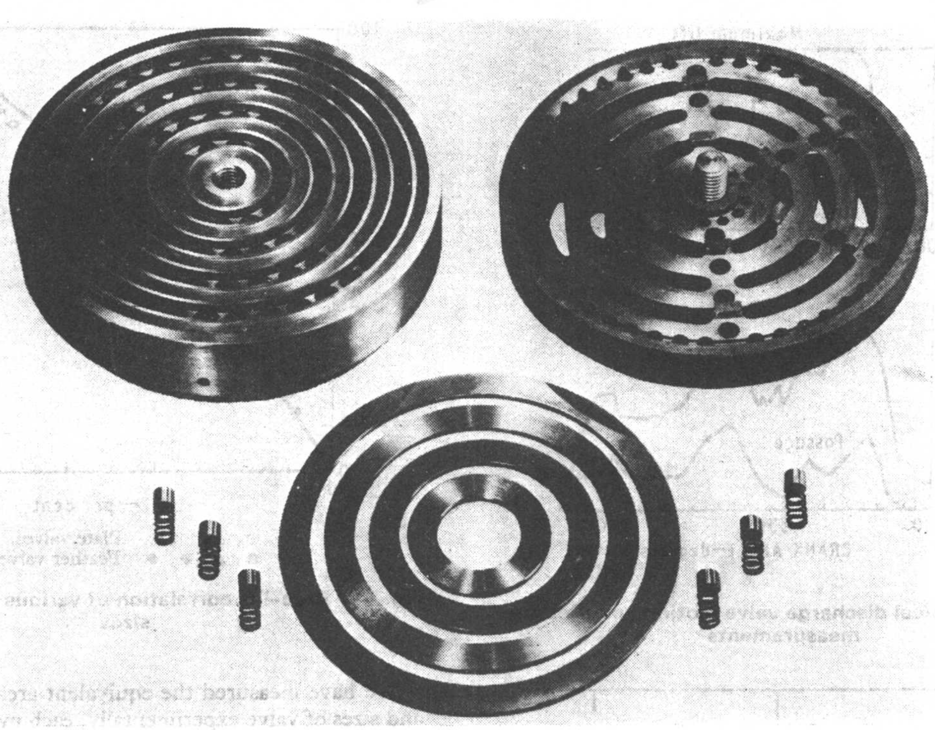


Fig. 2.1*a*. Plate valve components

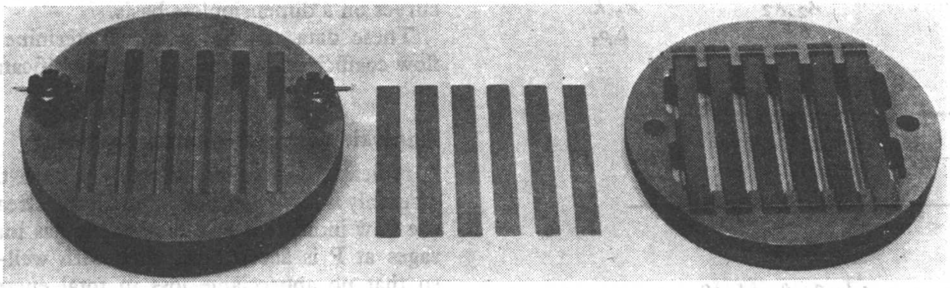


Fig. 2.1*b*. Feather valve components

This, coupled with the dependence of the pressure changes on the mass flow and restrictive areas, provides a mathematical model to describe the phenomena of valve motion. Pressure variations external to the valves are created by the non-steady nature of the flow out of the cylinder. The pressure variations propagate through the system in wave form with local amplitude and velocity, depending on the acoustic properties of the gas and the geometry of the cylinder passages and associated piping.

In order to implement the analysis described, several kinds of experimental data were needed:

(1) A relationship between mass flow, gas properties, and pressure change for any valve and for any degree of opening, or lift.

(2) A relationship between the pressure change across the valve and the force on the sealing member.

The pressure change due to flow through a valve assembly can be considered to be the same as the flow through multiple restrictions in series. This can be analysed from consideration of Fig. 2.3, showing flow through several orifices each with area A_i and orifice coefficient K_i. The coefficient for each orifice is defined by

$$\Delta p_i = 144\,\frac{\gamma_g}{2g_c}\left(\frac{Q}{K_i A_i}\right)^2 \quad . \quad . \quad (2.1)$$

in which Q is the volume flow rate, and γ_g is the density of the fluid.

The summation of pressure drops is

$$\Delta p = \sum \Delta p_i = 144\,\frac{Q^2\gamma_g}{2g_c}\sum\left(\frac{1}{K_i A_i}\right)^2 \qquad (2.2)$$

and the whole series of restrictions can be replaced by the