

59-10788

SUPERPRESSURE HYDRAULIC COMPRESSOR

L.F. Vereshchagin

A description is given in this article of the construction of a hydraulic compressor of very high pressure, working at a pressure of up to 5000 atm., with an output of 1.7 liters of liquid per hour. The mechanism of the compressor's work and the influence of the idle space are analyzed. The volumetric coefficient of the compressor's efficiency is calculated.

A multiplier is usually used as a pressure generator in investigations connected with the study of any phenomena occurring at very high pressures. This apparatus represents a device the construction of which is based on the use of the so-called "differential piston", that is, a steel piston consisting of two pistons with different diameters. The large piston works in a low-pressure cylinder and the small piston works in a high-pressure cylinder. The ratio of their areas gives the coefficient of multiplication in a given apparatus.

Bridgeman writes in his book "Physics of High Pressures" in the description of the multipliers used by him: "The apparatus is limited by the possibility of one stroke, and the compressibility of the transmitting liquid gives a very definite limit to the possible capacities. At the same time, the decrease in the volume of the transmitting liquid at a pressure of 12000 atm. fluctuates from 25 to 30%".

The initial pressure of 2000 or 3000 atm., the purpose of which is the attainment of the maximum pressure for one stroke of the piston with a larger capacity of the vessel, is produced by a separate auxiliary pressure generator.

If large volumes of liquid must be compressed to a pressure of 5000 or 6000 atm., then the repeated displacement of the multiplier piston is necessary with corresponding overlappings of the vents or valves, which leads to the rapid wearing out of the material of the sealing gaskets. These circumstances make the

multiplier a rather unpleasant machine, slow working, requiring careful attention and frequent overhauling, and therefore finding itself a place in industry with difficulty.

The demand for an apparatus which would guarantee the convenient production of very high (higher than 1000 atm.) pressures in large volumes exists both in the industry manufacturing armaments (1) (for self-loading of barrels), and in fine chemical technology where it is necessary to use pressure (2).

The English firm, Imperial Chemical Industries, for example, uses two compressors for getting the necessary pressure for the production of polythene (polymer of ethylene). The first one is a gas compressor of the usual type for a pressure of 250 atm.; the second one is the so-called "press-compressor", and is a multiplier with a differential piston and an oil-mercury seal. The latter device is an especially bad disadvantage of the press-compressor. (p.670)

The presence of a mercury seal in a very high-pressure part must cause trouble, if we take into consideration the possible discharges of the mercury in the form of a finely pulverized mist in the case of possible ruptures in the packing. This causes contamination of the working space and poisoning of the attending personnel. The very high-pressure hydraulic compressor developed by the author of this article is free of these disadvantages. At first this compressor was intended for the compression of liquids.

Specially conducted experiments showed, however, that the compressor also compresses gas satisfactorily if the latter is compressed to a pressure of 400-600 atm. before it enters the compressor. It is easy to see the reason for this if it is taken into consideration that the gas is already in a highly condensed condition under this pressure.

Five models of the machine were made with gradual improvement in their construction. Their operation during a period of six years showed a high perfection of construction as compared to a multiplier and reliability in work. Since it is unusually compact and easy to attend (starting and stopping are controlled by two buttons), this machine is a very convenient generator of very high pressure both for laboratory and industrial purposes.

The basic difficulty in building a hydraulic compressor of very high pressure, as well as a gas compressor of very high pressure, was the development of the packing construction of the moving piston. Leakages thru the packing of the piston at a pressure of up to 5000 atm. must be reduced to zero in connection with the small volume of liquid or compressed gas drawn in with one stroke of the piston. This is an essential condition of the compressor's operation due to the low compressibility of the liquid or of the preliminarily compressed gas. The above consideration makes it very hard to solve the problem of making a very high-pressure compressor, which explains the lack of this type of machine up to the present time in spite of all the attractiveness of the idea of using a compressor instead of the multiplier which is so inconvenient in operation.

This problem was solved by making the packing construction of the moving piston of the hydrocompressor so that it would not permit any noticeable leakages of the delivered gas or liquid. This led to the development of a construction and the building of a hydrocompressor for a pressure of 5000 atm. and an output of 1.7 liters per hour, and subsequently for 6000 atm. and 15 liters per hour. Experience showed that the problem of building a compressor of very high pressure for industrial purposes can be solved. There is also the question of the possibility of conducting certain chemical reactions (first of all the polymerization reactions of certain organic compounds) at very high pressures.

Description of the Hydraulic Compressor

The inside construction of the machine is shown in Fig. 1 and the whole view in Fig. 2. The liquid is drawn in and subsequently compressed in its return stroke by the piston made from the ball-bearing steel ShKh15. After it is made, the piston is tempered and drawn to a hardness of 40-45 Rockwell units (C scale). After thermic treatment the piston is polished. Its diameter is 6 mm. when finished. It is set in motion by a 5.8 kW motor thru a reducer which reduces the number of revolutions of (p.672) the motor from 1500 to 40 per minute. The rotating movement of the reducer shaft is transformed into a progressive movement of the piston with the help of a small link gear a (Fig. 1).

Fig. 1. Inside construction of the compressor.

Fig. 2. Whole view of the compressor.

Fig. 3. Construction of the link gear

We developed the link gear for the hydraulic compressor in a series of constructions and at present it is a totally reliable part of the machine. Its general appearance is shown in Fig. 3. The link a is connected with the pin of the eccentric b by means of the polished steel ring v. The cross piece is provided with two polished steel strips which can be tightened in the link by means of the screws d. The strips are designated in the drawing by the letter e. The link is connected with the piston by means of the ball bearing i. The eccentric is rigidly connected with the shaft of the reducer. The entire link gear is connected with the cylinder of the compressor by means of the coupling hooks b (Fig. 1). The purpose of the latter is to relieve the bracing of the reducer from the strains arising in the progressive movement of the piston.

Fig. 4 shows the construction of the packing of the hydro-compressor piston. Virtually, the idea of building a hydraulic compressor is solved in this construction.

In general, when it was necessary to increase the pressure in the pump, with the help of which it was intended to deliver the liquid somewhere under pressure, the designer proceeded to increase the length of the packing and the degree of its compression. The entire packing gland structure which packs the moving piston, if it is not self-packing in its construction, requires the preliminary formation of a

pressure in the packing of the gland which is somewhat higher than that at which the pump or compressor will work. There are two conditions which must be observed first of all to have the packing gland device of the moving piston work satisfactorily. The first condition is that a pressure must be maintained in the packing of the gland which exceeds the value of the pressure in the medium being compressed by a certain percentage. The second condition is that a sufficiently low friction must be assured in the packing. It is usually very difficult to observe both of these conditions at the same time, since the necessity of maintaining a sufficiently high pressure in the packing of the gland makes a very high friction develop in the gland which, in its turn, makes it wear out rapidly and causes the pressure to fall in the packings. It is therefore difficult to build a compressor which works satisfactorily at a pressure above 1000-2000 atm. when using glands of a non-self-packing type. (p.673)

The problem of building a continuously working hydraulic compressor is even more difficult if it is intended to compress a liquid to a pressure of several thousand atmospheres.

The diameter of the piston must be selected so as to avoid great strains on the piston of a very high-pressure compressor. It should be remembered that when the pressure is 5000 atm. and the diameter of the piston 6 mm. the strain on the piston is equal to 1420 kg. without taking into account the friction in the packing; when the diameter is 9 mm. the strain is equal to 3180 kg., and when the diameter is 12 mm. this strain attains the imposing figure of 5660 kg.

These few figures show that the length of the gland packing cannot be greatly increased when there are such big strains on the piston, since this will have an immediate effect on the stability of the piston. The latter circumstance also prevents an increase in the piston stroke.

We therefore find it necessary to work with a suction volume of liquid of 1-2 cc. In view of the low compressibility of the liquid (water decreases 12% in volume at 5000 atm. and glycerin 8%) it is necessary to guarantee complete absence of leakages of liquid in the pressure cycle.

The above considerations resulted in a number of experimenters, among them Bridgeman, Poulter, Newitt (3, 4, 5) and others, generally refusing to use a gland arrangement for packing a moving piston in their very high-pressure apparatus for producing very high pressures on the order of 3000, 5000 atm. or even more.

Altho the packings used by them for this purpose retain the liquid compressed to a very high pressure in the pressure cycle, nevertheless all of them, without exception, suffer from the disadvantage that they wear out rapidly and require frequent replacement (usually after 10-15 operations). This means that very high-pressure apparatus (so-called multipliers) are not continuously operating machines, and attempts to make them continuously operating by using valves do not give a positive result due to the rapid wearing out of the rubber packing.

After we had succeeded in developing and putting into practice the new (p.674) construction of the packing of a progressively moving piston, we succeeded in building a continuously operating machine - a hydraulic compressor of very high pressure.

Fig. 4 shows the arrangement of the piston packing of the hydro-compressor. Strictly speaking, the packing device consists of two packings - an upper "labyrinth" packing and a lower, section packing.

The labyrinth packing is a system of alternating steel and leather rings. The steel rings have a diameter which is 0.2 mm. bigger than the diameter of the piston. The purpose of the described labyrinth packing is to create during the movement of the piston a gradient of pressure at the beginning of the progressive movement of the compressor piston, when a sufficiently high pressure has not yet been produced for the second packing, the cylindrical section, to go into action. The action of the section as a packing device is based on the establishment of a gradient in the progressive movement of the piston along its axis. The gradient being established will be the greater the higher the pressure. This happens because the cylindrical section in which the piston moves is mounted in such a way that the pressure acts

on its outer surface as well as on the ring-shaped space between the piston and the cylinder. The outside pressure compresses the cylinder around the piston and thus prevents, inasmuch as this is possible, the expansion of the ring-shaped space. Fig. 5 shows the distribution of pressure along the axis of the section for an analogous arrangement (a similar type of packing with the help of a section found application in the construction of manometers of the free piston type (6, 7).

$\frac{\text{kg}}{\text{cm}^2}$

Insofar as I know, we were the first to carry out the idea of using a cylindrical section as a packing device for a moving piston in a very high-pressure compressor.

At present the described construction of the packing can be considered as a fully developed part of the very high-pressure compressor. Years of work of hydrocompressors equipped with this packing have shown that this part is exposed to very little wear and tear.

Fig. 5. Distribution of pressure along the section.

We use steel of the brand 18KhMA for making this packing. The part is not tempered after it is made. Only the sides need to undergo thermal treatment by means of a high-frequency current. This is desirable to avoid denting of the sides with a wrench when the packing is screwed into the cylinder.

The question of the necessary thickness of the walls of the section has not (p.675) yet been clarified. The thickness of the walls is 5 mm. when the diameter of the piston is 6 mm. This thickness of the walls cannot be considered the optimum one, however. This question will be studied in greater detail in the near future.

Since the compressibility of the liquid forced by the piston reaches almost 20% at a pressure of 5000 atm., the so-called "idle space" in the compressor must be reduced to a minimum. The indicator diagram shows that it is very high in a hydraulic compressor. For example, the value of the idle space attains 35% of the suction volume of the liquid for the described model of compressor.

Fig. 6. Indicator diagram of compressor.

Fig. 7. Function of volumetric efficiency of the hydrocompressor in relation to pressure at a varying value of m .

Normal pentane:

The effect of the idle space in a hydraulic compressor on the work of the compressor is substantially less than in a high-pressure gas compressor.

The size of the idle space in gas compressors is usually from 5 to 10% of the volume circumscribed by the piston. In addition to this, the lower the pressure produced by the compressor the higher the permissible value of the idle space.

It is known that there is a gradual transition from the equations of thermodynamics to the equations of hydraulics as the ratio $\frac{P_2}{P_1}$ decreases, the latter equations being the ones that must be dealt with in studying the work of ventilators and, especially, of low-pressure air blowers. (p.676)

In our case we encounter a reverse phenomenon, namely, we have to go from hydrodynamics equations to thermodynamics equations on analyzing the work of the compressor, due to the increased change in the volume of the liquid as a result of the increase in the working pressure in the compressor to a super-high pressure. This permits us to introduce the name "hydraulic compressor".

We will examine the work of the hydraulic compressor. Fig. 6 shows an indicator diagram of a small model.

Fig. 8. Function of volumetric efficiency of the hydrocompressor in relation to the value m for normal pentane.

$$P = 6000 \text{ atm. } \frac{V_L}{V_0} = 1.3$$

The ratio of the actual suction volume FS_1 to the theoretical FS is

Fig. 9. Function of volumetric efficiency of the hydrocompressor in relation to the value m for water.

$$P = 6000 \text{ atm. } \frac{V_u}{V_0} = 1.15$$

is called the "volumetric coefficient of efficiency of the compressor", in which F is the area of the piston, S_1 is the stroke of the piston during which suction of the liquid begins and occurs, and S is the entire stroke of the piston.

Returning to the indicator diagram shown in Fig. 6, it should be pointed out that $S_0 = \frac{V_0}{F}$, in which V_0 is the volume of the idle space, F is the area of the piston, and $S_2 = \frac{V_2}{F}$, in which V_2 is the volume obtained as a result of polytropic expansion of the volume of the idle space during the return stroke of the piston.

We can then write that the suction volume

From this the volumetric efficiency:

(p.677)

Using this expression for λ_0 , we can calculate:

$$\text{in which } m = \frac{V}{V_2}$$

The results of these calculations are shown in Figs. 7, 8 and 9. The data on compressibility obtained by Bridgeman for normal pentane and water (3) were used in the calculation of the curves.

Fig. 7 shows the curves of the function of λ_0 in relation to the pressure for normal pentane. It can be seen from the diagram that the idle space already plays a very noticeable role at a pressure on the order of 5000 atm. It can be seen from curve 3, for example, that λ_0 falls to a value of 0.44 when the pressure is 5000 atm. Figs. 8 and 9 show the function of λ_0 in relation to m when $P = 6000$ atm. for one and the same compressor. In the first case the working liquid was normal pentane, in the second case it was water. It can be seen from these diagrams that λ_0 attains zero at a high value of m . This is easy to understand if it is taken into consideration that water is almost two times less compressible than normal pentane at the same pressure.

The idle space cannot be made small on constructing the cylinder of a hydraulic compressor for the following reasons, however.

In order to make it sufficiently small the volume of the liquid for suction must be increased. This can be accomplished either by increasing the diameter of the piston, which will immediately affect the strain on the piston, or by increasing the stroke of the piston. This method is also full of after-effects for the designer in view of the fact that the length of the piston is increased simultaneously and its resistance to bending force decreases. Nevertheless, it is more expedient to take this course than to increase the diameter of the piston. It must be considered that an extremely high working pressure causes restrictedly high tensions of the material in the basic parts of the compressor even with static action. With dynamic action the tensions increase so rapidly with increase in the diameter of the piston that it is impossible to guarantee their necessary strength even when using high-quality steels. This involves very great difficulties in the construction of a hydraulic compressor of very high pressure and of high output.

In the construction of a hydraulic compressor with an output of 20 liters per hour at a pressure of 10,000 atm., for example, we encountered a strain of approximately 12 tons on the rod when the diameter of the piston was 12 mm. At this strain, the normal tension in the cross section of the piston attains a value of 11400 kg. per sq. cm. and the tangent approximately 5000 kg. per sq. cm. The application of a hydraulic transmission instead of the crosshead or the link gear will probably be a further improvement in the compressor.

The cylinder of the compressor (Fig. 1) is equipped with two valves of the disk type, a suction and a pressure valve (c and d). The suction valve is sealed with a red copper bushing (Fig. 10). The construction of the pressure valve is shown in Fig. 11. The disk type valve is screwed into the cylinder of the hydrocompressor by means of ^{the} thread on it. The valve is sealed by means of a copper little ring at its lower end. Little force is required to screw it in since the little ring has to resist the pressure drop caused in the pressure cycle by the force of the tightening of the (p.678) valve plate by the valve spring (several atmospheres). The valve plug a is also sealed with a bushing. This type of seal was also found very satisfactory in other very high-pressure apparatus. The seal is a bushing made of red copper, placed in a specially cut groove in an obturator, as can be seen in the drawing. The ends of the bushing must be sharpened when it is first used so that the bushing does not bulge inward when the obturator is screwed in. This is no longer required in subsequent dismantling of the obturator.

Fig. 10. Inside construction of the suction valve.

Fig. 11. Inside construction of the pressure valve.

This construction requires only a clean surface, unlike the wave-shaped obturator proposed by the firm of Imperial Chemical Industries which requires very careful polishing of the obturator seat. The valves of the hydrocompressor represent its most vulnerable part. For a number of years we have been working on the problem of making a reliable valve construction. It seems to me that at present the construction of the valves used by us is substantially better than that which we used at the beginning of our work on the hydrocompressor at the Ukrainian Physico-technical Institute. The first valves used in it were of the type used by the Zhenevsky Society of Scientific Instruments in pumps of the Callette type.

In his book "Physics of High Pressures" Bridgeman writes: "There is also the question of valves, namely the inlet and outlet valves of the chamber in which the work of the plunger (of the piston) occurs. I have never had a valve with which I have been completely satisfied. The valve which came with the pump of the Zhenevsky Society has a conical shape and is made of ebonite. It works well for a given length of time but after continued use it gradually catches at the lower end where it jams and finally breaks." (p.679)

Instead of the ebonite valve I use a conical or spherical copper valve which has also proved to be entirely satisfactory. It is probable that everyone working with a pump of this type has his own experience with the construction of the valves and has undoubtedly had a whole series of valves which seemed to solve the problem satisfactorily for a long period of time, and then, for some inexplicable reason, the valve went out of order for several days.

Apparently the worse enemy of the valve is the contaminated condition of the liquid delivered by the pump."

Ball valves with a ball diameter of 12-14 mm, made from steel (from ball bearings), work well on a new seat only at first.

Then the ball begins to wear out (get out of shape) on the edge of the seat and the ball immediately stops working as soon as it turns slightly. The selection of balls for valves must be very careful since balls are far from always actually round. This shows up when a pressure of 3000-4000 atm. is attained. The suction valve is a particularly bad source of trouble.

The pressure valve, ~~which is also~~ a ball, frequently goes out of order in the hydrocompressor.

It gradually breaks down in the ~~middle~~ and barely noticeable annular dents are formed in it which are sufficient to make the valve stop working however.

The frequently inexplicable freakish behavior of ball valves made us decide not to use them in the hydraulic compressor in our later constructions.

We find that disk valves are substantially more reliable than ball valves. The disks and seats must be polished with an accuracy of up to 2 or 3 Newton rings.

There should not be a strong spring on the suction valve. Otherwise it opens on the suction cycle. If delivery of a liquid free from contamination can be assured, it works entirely satisfactorily. It should be noted that the requirements for the pressure valve must be especially high. This can be concluded if only from the fact that a suction valve of the same construction as that of a pressure valve, made from one and the same kind of steel, goes out of order far more rarely than the pressure valve. We have spent a great deal of time on the study of the work of the pressure valve during our work with hydrocompressors and yet, at present, we cannot say with certainty that we completely understand all the processes which apparently occur during the work of the pressure valve in the suction and pressure cycle. On the basis of the many experiments conducted in this direction however, it can be assumed at present that the size of the opening of the valve does not play any role and does not have any important effect on the value of its coefficient of volumetric input.

The ball valve must be given up definitely for pressure feeding. - We believe that the disk valve can be considered best for hydraulic compressors of very high pressure at present.

In conclusion, I would like to express my gratitude to I.G. Gurvich (p.680) for the interesting discussions in connection with the theory of the work of the hydrocompressor and for assistance in the work.

Literature

1. De La Chaise. Self-Loading of Artillery. Oborongiz, 1940.
2. I.C. Swallow, Endeavour, III, 9, 1944.
3. P.V. Bridgeman. Physics of High Pressures, 1931.
4. T.S. Foulter, Phys. Rev., 35, 297, 1930.
5. D.M. Newitt, The Design of High Pressure Plant and the Properties of Fluids at High Pressures, 1940.
6. P.W. Bridgman, Proc. Amer. Acad., 44, 201, 1909.
7. L.F. Vereshchagin and B.S. Alexandrov, J. of Tech. Physics (USSR) 6, 1938.