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A High Pressure Air-Driven Pump

This design method produced a successful pump made to operate at 60,000 pounds per square inch

Although many problems and considerations are involved in designing for research work a liquid pump for attaining pressures of 60,000 pounds per sq. inch, those discussed here are chosen to illustrate basic approaches applicable to all pressure apparatus.

In developing this pump, it was necessary to subdivide the over-all design into its essential components: type of drive mechanism suitable for safety, adjustability, operational ease, maintenance, and plunger size and design.

These components were based on two major considerations—proper functioning during operation and maximum safety commensurate with good design. Perhaps most difficult was a check valve to provide positive action without leakage at operational pressures.

In designing the plunger and cylinder, available driving forces and adaption to the driving mechanism had to be considered. Then a packing gland had to be devised, capable of sealing against the high liquid pressures involved, which are not constant but rather pulsate from atmospheric to 60,000 pounds per sq. inch with each stroke of the pump. This widely fluctuating pressure produces hazardous pulsating stresses in the cylinder which was partly solved by using a composite cylinder.

Drive Mechanism

Safety was the major factor in selecting the drive mechanism and the air cylinder was chosen for two reasons. First, the output pressure of the pump is directly dependent on air pressure applied to the air cylinder—e.g., if desired output pressure is 50,000 pounds per sq. inch, the air pressure determined from test curves is 94 pounds. Thus, when this setting has been made on the air control valve, the pump will automatically stop when output pressure reaches 50,000 pounds. Of course, a constant output pressure can be maintained by properly adjusting air pressure to the cylinder. The second reason an air cylinder was chosen is because it contains no electrical devices to spark an explosion in the presence of highly volatile substances.

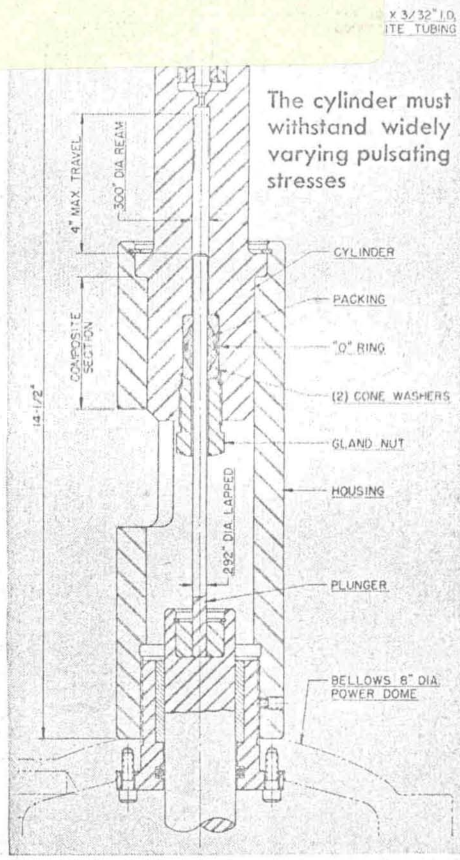
Also, an air drive can easily change the length of stroke and speed of operation without complicated and costly control devices. The cylinders, having a power dome eight inches in diameter and made by the Bellows Co., are readily available and adaptable to high pressure apparatus.

Sizing of Pump Plunger

Air pressure of 100 pounds per sq. inch, readily available in most laboratories, was used to provide an output of 60,000 pounds per sq. inch. The first assumption was used here—from previous experience in pumping at high pressures, an efficiency loss of approximately 20% caused by friction, air leakage, and operational losses from a suitable check valve design was assumed. Using this assumption, the theoretical diameter of the plunger was calculated as 0.288 inch: the effective area of the air cylinder piston is

$$A_a = 0.785\,(D_a^2 - d_a^2)$$

Setting forth basic principles for guidance in designing high pressure equipment is increasingly needed. The problem is attacked here by inductive reasoning—in developing a particular pump, solutions to problems encountered, which can be applicable to all high pressure apparatus, are selected for discussion



The cylinder must withstand widely varying pulsating stresses

$$A_a = 0.785\,(8^2 - 1.5^2) = 48.5 \text{ sq. inches}$$

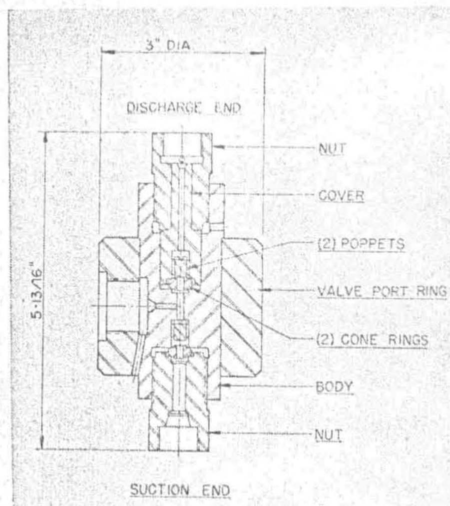
where in square inches A_a effective area; D_a is diameter of piston; and d_a is diameter of shaft, inches.

If the pump is 80% efficient, it can be assumed that 80 of the 100 pounds of pressure applied is actually transmitted to the cylinder.

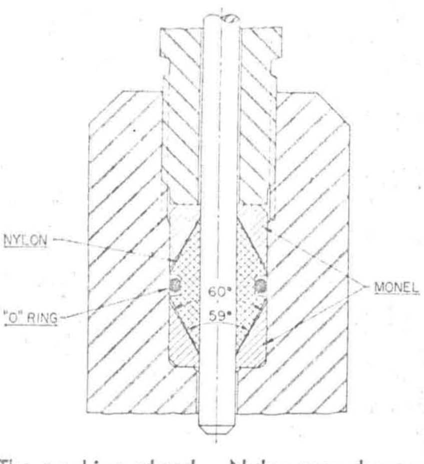
Thus,

$$A = \frac{A_a P_a}{P}$$

$$A = \frac{(48.5)(80)}{60{,}000} = 0.0646 \text{ sq. inch}$$



This check valve presented the most difficult problem in designing the pump



The packing gland. Nylon was chosen as the packing material because it provided sealing with minimum frictional losses

where A is area of plunger, and in pounds per square inch, P is discharge pressure, and P_a is air applied pressure.

The diameter of the plunger is then

$$D = \sqrt{\frac{A}{0.785}} = \sqrt{\frac{0.0646}{0.785}} = 0.288 \text{ inch}$$

For a necessary clearance between the plunger cylinder and bore, a bore diameter of 0.3000 inch was used. Initial tests at 60,000 pounds per sq. inch indicated that reducing this clearance would result in more efficient pumping; consequently, the plunger diameter was increased to 0.292 inch. The 102 pounds of air required to reach 60,000 pounds, as shown on the test curves, indicates that the initially assumed efficiency losses were essentially correct.

Surface finish is an important feature in a plunger for this type of service. A highly polished surface is needed for effective sealing in the packing area. Accurate alignment is needed for the plunger as it progresses through the packing on the pressure stroke. This was accomplished by introducing a predetermined amount of clearance at the connection between the air cylinder shaft and plunger. The plunger is allowed to seek its own path of travel, thereby eliminating extraneous forces which tend to cause eccentric column loading, and consequent plunger wear or even failure by column action. The plunger was made from Type 420 stainless steel, hardened to a Rockwell value of 45 to 48.

Packing Design

Nylon was used as the packing material to provide adequate sealing with minimum frictional losses. Many materials were tried, but not one was as satisfactory as nylon.

By adapting the principle of standard chevron packing, minimum torque on the packing gland nut was obtained to produce maximum sealing. According to this principle, two unequally tapered cylindrical surfaces, forced together in a longitudinal direction, produce resultant forces in the angular or radial direction, that are quite large compared to the axial force applied. Thus, sufficient contact pressure between the pump plunger and packing material could be attained with relatively small torque on the packing gland nut. Leakage is prevented on the exterior of the packing with the O ring.

Material for the back-up rings to produce maximum antisizing tendencies was chosen by actual tests. Monel has the strength and hardness for the bearing and antigalling characteristics required.

Cylinder Design

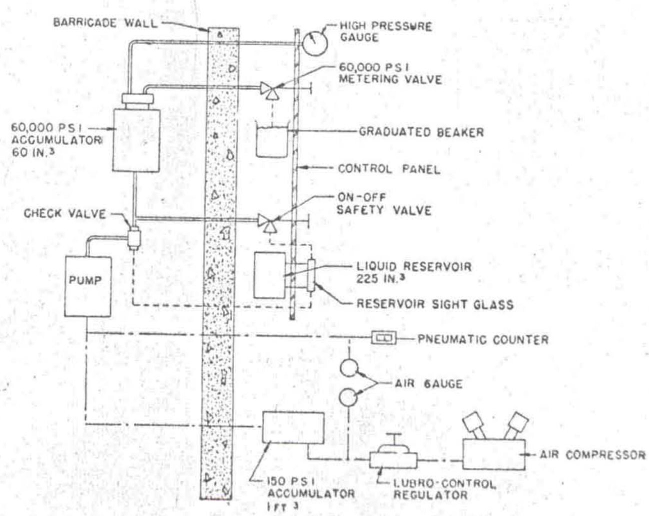
The cylinder head is subjected to pulsating stresses from a maximum at discharge to a minimum at suction. The maximum tangential stress encountered in pumping at 60,000 pounds occurs at the packing section and was calculated in accordance with the Lamé formula as 75,000 pounds per sq. inch. Thus, for a thick-walled cylinder when subjected to internal pressure

$$\sigma t = Pi \frac{b^2 + a^2}{b^2 - a^2}$$

where pounds per square inch, σt is tangential stress; Pi is internal pressure; and in inches, a is inner radius; and b is outer radius.

For the internal pressure of 60,000 pounds per sq. inch the resultant stress on the inner bore of the cylinder will be

$$\sigma t = 60,000 \frac{1.125^2 + 0.375^2}{1.125^2 - 0.375^2} = 75,000 \text{ pounds per sq. inch}$$



Apparatus for testing the pump. An air accumulator gave a constant supply of air to the pump cylinder

═══ Liquid pressure; - - - liquid atmospheric pressure; — · — · air pressure

The cylinder is made from AISI 4340 alloy steel hardened by heat treatment to increase its mechanical properties sufficiently to withstand stresses encountered. Careful consideration was given to high strength requirements and maximum ductility, both of which are important. Based on safe design practice, an elongation of at least 15% is required. Maintaining this elongation would result in a minimum yield strength of approximately 125,000 lb. per sq. inch which means that the resultant safety factor based on yield is 1.78. Cylinder geometry, within consideration of pulsating pressure loading, indicated that a greater safety factor should be provided. This can be done by increasing the outer diameter of the cylinder or by composite design. Lamé's formula indicates that additional wall thickness does not appreciably reduce stresses involved here because of the heavy wall already present. Because the mechanical design of the pump requires the cylinder to be attached to the air cylinder by a suitable housing, a shrink fit was provided between these two components.

As a result of this shrink fit, peak stresses encountered during the compression stroke were reduced from 75,000 to 45,600 pounds per sq. inch as follows:

For a diametrical interference of 0.0025 inch, the contact pressure between the two cylinders is

$$p = \frac{\delta E}{b} \frac{(b^2 - a^2)(c^2 - b^2)}{2b^2(c^2 - a^2)}$$

$$p = \frac{0.00125 \times 30 \times 10^6}{1.125} \frac{(1.125^2 - 0.375^2)(1.75^2 - 1.125^2)}{2(1.125)^2(1.75^2 - 0.375^2)} = 9050 \text{ pounds per sq. inch}$$

where p is contact pressure in pounds per square inch; δ is radial interference, in inches; E is Young's modulus, and c is outer diameter of housing in inches

When a built-up cylinder such as that used here is subjected to internal pressure, the stresses produced are the same as those for a cylinder with a solid wall of thickness c − a. The tangential stress at the inner fibers of the packing section would then be

$$\sigma t = Pi\,\frac{c^2 + a^2}{c^2 - a^2} = 60{,}000\,\frac{1.75^2 + 0.375^2}{1.75^2 - 0.375^2} = 66{,}000 \text{ pounds per sq. inch}$$

The tangential stress at this section produced by the contact pressure is

$$\sigma t = -p\,\frac{2b^2}{b^2 - a^2} = -\,\frac{9050 \times 2 \times 1.125^2}{1.125^2 - 0.375^2} = -20{,}400 \text{ pounds per sq. inch}$$

Superposition of these stresses yields a resultant stress of

$$\sigma t = 66{,}000 - 20{,}400 = 45{,}600 \text{ pounds per sq. inch}$$

as compared to a stress of 75,000 pounds per sq. inch when composite design is not used.

The resultant safety factor based on the above calculation is 2.72 which represents an increase of approximately 53%. Actually, the safety factor is somewhat less than this, because of stress concentration, service, and surface finish factors. However, the design as such was adequate for the service intended.

Check Valve

Design of a check valve was perhaps the most difficult problem encountered in the design of the entire pump. The composite design theory was also used here. The outer ring was shrunk to the main body of the valve, thus reducing the peak stresses to valves similar to those already discussed.

Materials for this valve were selected carefully. More importantly, hardness of the various components affects success of the design. For strength to withstand stresses involved without failure from excessive brittleness, materials need to be hardened with heat. Mostly, this problem was solved with development tests.

For this valve, a poppet seemed most suitable for providing the sealing action required. For best sealing, small contact surfaces are indicated; but for longer operational life, larger contact surfaces are required to reduce high bearing stresses. The poppet design allows for practical adjustment of these stresses.

Using the poppet does, however, have one important requirement—surfaces sealed together must be accurately lapped together. The valve has been designed so that parts requiring lapping are easily removable to facilitate the lapping operation.

Performance and Tests

Newly designed equipment should be evaluated by test methods done under conditions as nearly like those encountered in laboratory operation as possible.

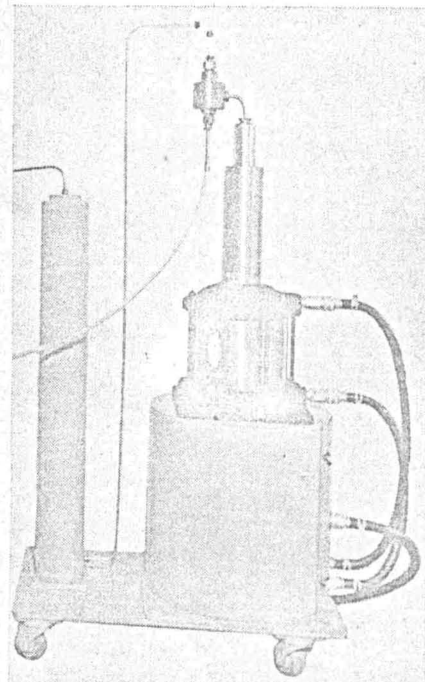
The high pressure apparatus is separated from the necessary control devices by a suitable barricade. To facilitate operation of the pump, the control apparatus has been placed on a panel which can be located on the exterior wall of a barricaded cell.

First it was necessary to have a constant supply of air to the pump air cylinder. This would not ordinarily be necessary during actual use of the pump; however, in determining performance curves it aided in accumulating data and facilitated reproducible results. The addition of the air accumulator as shown on the flow diagram assisted in this requirement and the maximum air pressure variation at any given setting was approximately ± 2 pounds per sq. inch which occurred when the pump piston changed its direction of movement.

Several primary tests were conducted to determine suitable techniques to be used during operation of the test apparatus. These tests indicated that accurate results could be obtained by operating the pump for relatively short periods at any given set of conditions. The discharge pressure could be held constant by hand manipulation of a metering valve to adjust the volume of discharge for the duration of a test run. The average time required for a test run for any given setting was approximately 5 minutes. The maximum variation in discharge pressure ranged from 150 pounds per square inch below and above the desired setting. Higher discharge pressures were more easily controlled.

The fluid pumped was a 25 to 1 mixture of water and a suitable emulsifying oil. The oil was used to some extent for lubrication and for the most part to prevent oxidation of the alloy steel components in direct contact with the fluid being pumped.

All tests were made with a stroke length of 3.750 inches. It was determined during preliminary testing that a change in stroke length had a negligible effect on the amount of fluid pumped during a given period provided air pressure to the air cylinder was constant. It is possible, however, by adjustment of stroke and air pressure to control the volume discharge of the pump more accurately. A shortening of stroke length will provide a more even discharge flow, which will allow a finer adjustment of the total flow required during any given time.



The finished pump. Length of stroke and speed of operation can be varied without complicated and costly control devices

Test runs were made for discharge pressures ranging from 20,000 to 60,000 pounds per square inch. Curves were drawn from these data, plotting the delivery at the various discharge pressures against air pressure applied.

To provide additional versatility, two additional sets of cylinders and plungers were designed. The first was designed to produce a maximum discharge pressure of 30,000 pounds per sq. inch with an air pressure of 80 pounds per sq. inch applied to the air cylinder, and the second was to produce a maximum discharge pressure of 15,000 pounds per sq. inch with an applied air pressure of 80 pounds per sq. inch. Similar tests were completed with these components.

Conclusions

Design of high pressure apparatus requires subdivision into its essential elements which are then designed, keeping in mind the requirements of the complete apparatus. Development and test work must supplement theoretical considerations. The performance and tests indicate that this method was successful.

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