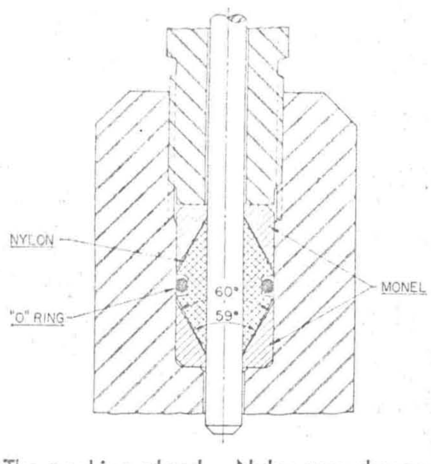


The packing gland. Nylon was chosen as the packing material because it provided sealing with minimum frictional losses

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 pounds per sq. inch

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:

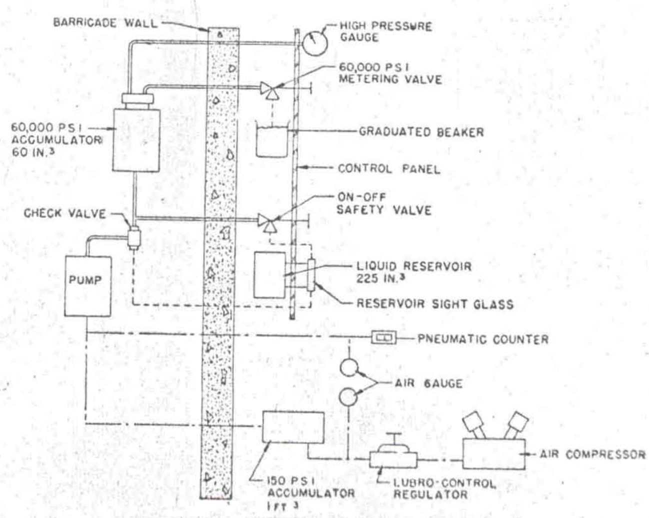
For a diametrical interference of 0.0025 inch, the contact pressure between the two cylinders is

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

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

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



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