SECTION 9.22
HYDRAULIC PRESSES

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TYPES OF PRESSES

Hydraulic presses, both vertical and horizontal, are used in many industrial technologies. Vertical press applications include forging presses with flat dies, used for hot work to break down ingots and shape them into rolls, pressure vessels (mandrel forgings), forged bars, rods, plates, and so on; forging presses with closed dies, used to process preheated billets into various shapes, such as aircraft bulkheads, engine supports, and main fuselage and wing beams; and upsetting presses, used for the production of items with elongated shafts—long hollow bushings, pipes, vessels, and so on. Horizontal presses are used primarily for conventional hot extrusion. Cold hydrostatic extrusion is finding application in the production of various end products and bulk items, such as very thin wire in long strands.

ACCUMULATORS

Until about 1932, all power systems for hydraulic presses were operated with water. With the advent of reliable, fast, high-pressure oil pumps, a trend of significantly less expensive oil pumps developed.

Installations requiring a relatively uniform and constant supply of hydraulic power are preferably designed with drives drawing the pressurized liquid directly from the pumps. In installations with high power (and liquid) peak demands of short duration, it is in most cases advantageous to arrange for a constant average flow of pressurized liquid from the pumps while providing equipment in which pressurized liquid can be stored in times of low demand and from which liquid can be drawn during demand peaks. These storage

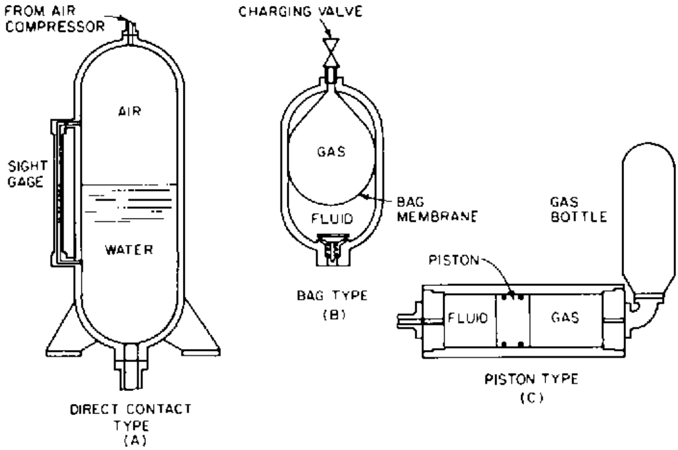

FIGURE 1A through C Accumulators

facilities are called accumulators and are illustrated in Figure 1. A modern accumulator is buffered by a large pressurized gas (mostly air) cushion.

Until about 1960, the direct-contact surface between gas and liquid virtually precluded the use of oil in accumulator installations for larger presses. Gas diffuses into oil under pressure. When oil is discharged from the press at the end of the pressing cycle, the gas begins to bubble out of the liquid; the liquid foams, and because of its large volume, this foam is difficult to handle. This condition exists whether the pneumatic cushion is air or nitrogen. In addition, if air is used as the pneumatic cushion, the oxygen diffused into the oil oxidizes it, producing sludge and gum. (Spring- and weight-loaded accumulator installations have other serious drawbacks and have almost completely disappeared.)

A modification of the accumulator station uses a floating piston to separate oil from the pneumatic cushion (Figure 1C). This design eliminates foaming and oxidizing, thus allowing the inclusion of accumulators into oil systems.

Power plants operated with water are used exclusively in special cases (for instance, where extreme precaution must be taken against fires caused by leaking oil). The increasing viscosity of oil precludes the use of oil at very high pressures.

CENTRIFUGAL VERSUS RECIPROCATING PUMPS

The competition between centrifugal and reciprocating pumps for hydraulic presses has been decisively won by reciprocating pumps. Under full load, centrifugal pumps have higher efficiency than reciprocating pumps (Figure 2). However, in press installations, the power plants are idle a significant portion of the time. The losses during idling are only about 10% of full load for reciprocating pumps and well over 60 to 70% for centrifugal units. The total combined losses in installations with reciprocating pumps are less than half the losses in centrifugal pump plants, as shown in Figures 3 and 4.

Reciprocating pumps for hydraulic presses that use water as the hydraulic medium are generally of the single-acting multiplunger type. Although there are large installations in operation using double-acting pumps, the faster vertical single-acting pump, which requires less floor space, offers a considerable saving in capital expenditure and has proved itself dependable in service. In most cases, this type of pump is used in conjunction with an accumulator, which allows the averaging of demand and thus a reduction in required pump capacity. There are, however, installations in which a vertical single-acting pump drives hydraulic cylinders directly.

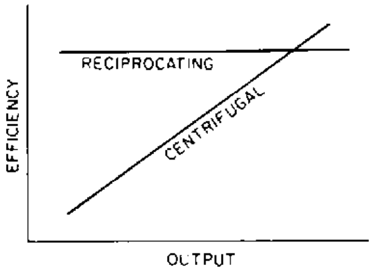

FIGURE 2 Comparison of pump efficiency

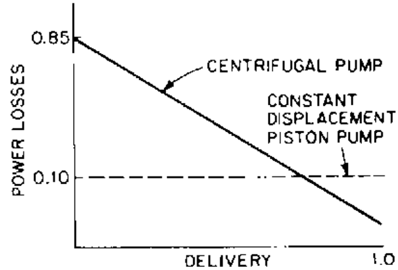

FIGURE 3 Comparison of pump power losses

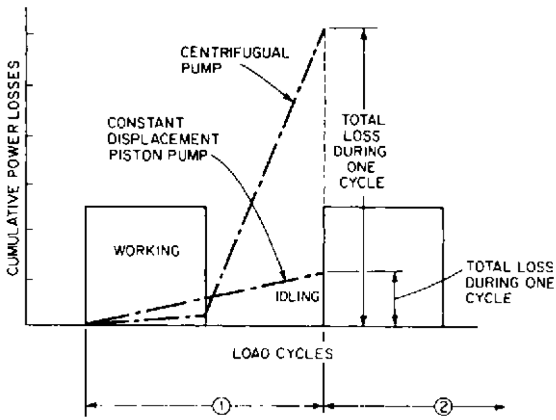

FIGURE 4 Comparison of load cycling losses

Idling is controlled by remotely operated bypass valves or pump suction valve lifters that take command from the operator, accumulator level control, or some other sensor-monitoring press action.

Oil pumps most commonly used for the power strokes of presses are

1. Vane rotary constant-delivery pumps, generally for pressures not in excess of 2500 lb/in^2 (17,200 kPa)
2. Piston rotary constant-delivery pumps
3. Piston rotary variable-delivery pumps

The constant-delivery pumps require a bypass system for idling. This is generally accomplished by an arrangement of directional control valves or by relief bypass valves.

The variable-delivery pump has the advantage of providing efficient press speed control and idling by an adjustment of piston stroke. In cases of multiple pump application, constant- and variable-delivery pumps are often used jointly, their selection depending on the speed ranges required. It is possible to obtain these pumps designed for use with nonflammable liquid as a hydraulic medium.

Generally, both water and oil pumps in press applications are driven directly by electric motors, the motor speed matching the pump speed. Sometimes, on large water pumps, a geared speed reducer is installed between motor and pump.

OPERATING PRESSURES

Operating pressures in hydraulic presses have been slowly moving upward. The most popular level for water hydraulic installations is a rated pressure of 5000 lb/in^2 (34,500 kPa) and an actual operating pressure of 4500 lb/in^2 (31,000 kPa). Intensifiers are used to raise this pressure to between 6750 and 7500 lb/in^2 (46,500 and 51,700 kPa) in large presses, which corresponds to a rated power of from 7000 tons (62,300 N) upward. When used, the intensifiers are installed between the accumulators (and pump) on the low side and the press on the high side. A bypass is always provided. For very large presses, pressures of up to 15,000 lb/in^2 (10,300 kPa) have been suggested. In the former Soviet Union, designers are using 15,000 lb/in^2 (10,300 kPa) regularly.

In oil hydraulic installations, 3000 lb/in^2 (20,700 kPa) is the most popular pressure because of the availability of 3000-lb/in^2 (20,700-kPa) equipment from many reputable firms (pumps, valves, fittings, and so on). However, there are now on the market excellent units for pressures up to 6600 lb/ in^2 (45,500 kPa), and the tendency in new installations is to provide 5000 lb/in^2 (34,500 kPa). Today, oil pumps for pressures as high as 9000 to 15,000 lb/in^2 (62,000 to 103,500 kPa) are being offered.

In the field of what is called ultrahigh-pressure equipment, pumps with direct action up to 225,000 lb/in^2 (1.55 MPa) are available today.

DESIGN PROCEDURE

Considerations and calculations required to dimension the hydraulic power plant for a hydraulic press are given in Table 1. A standardized procedure for determining the basic design features of the power plant for a hydraulic press is offered in the left column. The right column gives an example of how the standard procedure should be used. The press selected for the example is an open die press for forging ingots into billets or bars through gradual reduction of cross section on narrow, flat dies that, at each stroke, penetrate several inches into the hot metal (cogging). The ingot is advanced and rotated, and, when it is close to the desired size, the surface is made smooth by the application of shallow-penetration—$\frac{1}{8}$- to $\frac{1}{2}$-in (0.32- to 1.27-cm)—fast strokes (planishing) of the press (Figure 5).

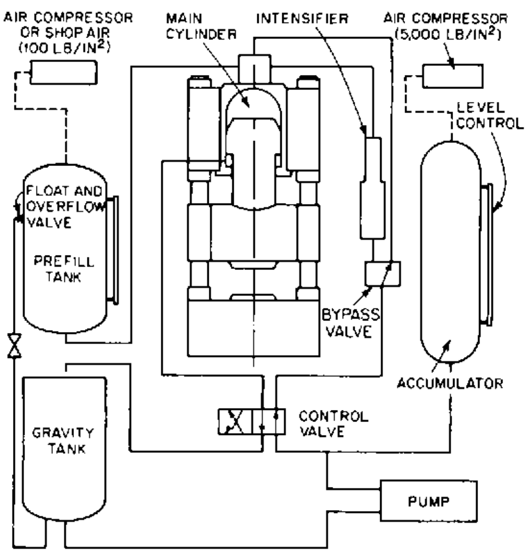

FIGURE 5 Vertical forging press with double-acting cylinder

TABLE 1 Example of power plant design for a hydraulic press

Standard Procedure	Example
1. Water or oil can be used as the liquid in a hydraulic system of a vertical or horizontal press. The decision can be made in accordance with the general discussion in this section. The generation of full pressure during planishing is not required; because of rapid stroking, there would not be enough time to pressurize the liquid fully and expand the cylinder fully. The pull-back cylinders are kept continuously under pressure during planishing, thus reducing the time required for valving. The generation of about 10% of nominally rated pressure is considered sufficient for planishing.	Requirements: Press rating T = 6000 tons (53 MN) Pull-back rating T_R = 600 tons (5.3 MN) Total stroke S_t = 48 in (122 cm) Maximum pressing stroke S_p = 12 in (30 cm) Fast advance speed v_a = 360 in/min = 6 in/s (5 cm/s) minimum Fast return speed v_r = 360 in/min = 6 in/s (15 cm/s) minimum Pressing speed V_p required: At full 6000-ton (53-MN) rating: 120 in/min 2 in/s (5 cm/s) maximum At $\frac{2}{3}$ = 4000-ton (36-MN) rating: 180 in/min = 3 in/s (8 cm/s) maximum At $\frac{1}{3}$ = 2000-ton (18-MN) rating: 360 in/min = 6 in/s (15 cm/s) maximum Rated operating cycles: Cogging: 15 c/min, each cycle consisting of Fast advance—4 in (10 cm) Pressing—4 in (10 cm) Return—8 in (20 cm) Planishing: 80 c/min, each cycle consisting of Advance—$\frac{1}{4}$ in (0.64 cm) Pressing—$\frac{1}{4}$ in (0.64 cm) Return—$\frac{1}{2}$ in (1.3 cm)
	1. Selected: oil hydraulic system
2. Select operating pressure p in pounds per square inch (megapascals) of the hydraulic system.	2. Selected: p = 5000 lb/in² (34 MPa)
3. Determine the required effective pressing area A in square inches (square centimeters) of the main plunger of the press: $$A = \frac{T \times 2000}{p}$$ where T is the rated power in tons (newtons).	3. in USCS units $$A = \frac{6000 \times 2000}{5000} = 2400 \text{ in}^2$$ in SI units $\frac{53}{34} = 1.55 \text{ m}^2 = 15{,}500 \text{ cm}^2$

TABLE 1 Continued.

Standard Procedure	Example
4. Select the number of cylinders N_c comprising the main system of the press and their individual ratings $T_1, T_2, T_3 \dots$.	4. Select three cylinders. Center cylinder T_2 = 4000 tons (36 MN) Two side cylinders $T_1 = T_3$ = 1000 tons (8.9 MN) Operating all three cylinders: T = 6000 tons (53 MN) Operating center cylinder alone: 4000 tons (36 MN) Operating side cylinders alone: 2000 tons (18 MN)
5. Subdivide A in accordance with selected ratings and determine the individual diameters: $$D = \sqrt{\frac{A}{0.785}}$$ If any piston rods detract from the effective plunger area, their area should be added to A before determining D.	5. $A_1 = A_3 = \frac{1}{6} \times 2400 = 400\ \text{in}^2$ $(\frac{1}{6} \times 15{,}500 = 2580\ \text{cm}^2)$ $A_2 = \frac{4}{6} \times 2400 = 1600\ \text{in}^2$ $(\frac{4}{6} \times 15{,}500 = 10{,}300\ \text{cm}^2)$ $$D_1 = D_3 = \sqrt{\frac{400}{0.785}} = 23\ \text{in}^a\ (58.4\ \text{cm})$$ Adjusted area $A'_1 = A'_3 = 415\ \text{in}^2$ (2680 cm) $$D_2 = \sqrt{\frac{1600}{0.785}} = 45\ \text{in}^a\ (114\ \text{cm})$$ Adjusted area $A_2' = 1590\ \text{in}^2$ (10,200 cm^2) Total adjusted area $A' = (2 \times 415) + 1590 = 2420\ \text{in}^2$ $[(2 \times 2680) + 10{,}200 = 15{,}600\ \text{cm}^2]$
6. Using the total stroke S_t and the rated pressing stroke S_p, determine the geometric volumes V_t and V_p corresponding to these strokes.	6. $V_t = S_t \times A' = 48 \times 2420 = 116{,}160\ \text{in}^3$ $(122 \times 15{,}600 = 1{,}903{,}000\ \text{cm}^3)$ For cogging $V_{pc} = 4 \times 2420 = 9680\ \text{in}^3$ per cogging stroke $(10 \times 15{,}600 = 156{,}000\ \text{cm}^3)$ For planishing $V_{pp} = \frac{1}{4} \times 2420 = 605\ \text{in}^3$ per planishing stroke $(0.635 \times 15{,}600 = 9900\ \text{cm}^3)$
7. Using a general compressibility curve, determine the compressibility c of the liquid between atmospheric pressure and rated operating pressure.	7. For 5000 lb/in^2 (34 MPa), the oil compressibility may be assumed at c_0 = 1.5%.

[a]It is customary to select diameters of large plungers in full-inch or $\frac{1}{2}$-inch sizes; this requires adjustment of the area.

TABLE 1 Continued.

Standard Procedure	Example
8. Assume a reasonable approximate level of hoop stresses s_h along the inner surface of the cylinder barrel. The cylinder diameter expansion will be $d_e = \frac{s_h}{E} \times D$ where E = modulus of elasticity, assumed to be 30×10^6 lb/in² (207 GPa)	8. Assume s_h = 30,000 b/in² (207 MPa) Main cylinder $dem = \frac{30{,}000}{30 \times 10^6} \times 45 = 0.045$ in $\left(\frac{207}{207 \times 1000} \times 114 = 0.114 \text{ cm}\right)$ Side cylinders $des = \frac{30{,}000}{30 \times 10^6} \times 23 = 0.023$ in $\left(\frac{207}{207 \times 1000} \times 58.4 = 0.058 \text{ cm}\right)$ where *dem* is the main cylinder diameter expansion and *des* is the side cylinder expansion.
9. On the basis of c_0 and s_h, determine the volume of liquid required to compensate for compression of liquid V_{c1} as well as for expansion of cylinder V_{ec}. During planishing, the pressures rise to only about 10% of rated; therefore, the compressibility of the liquid and expansion of the cylinder may be neglected.	9. Compression of the liquid takes place along the entire length of the cylinder barrel. Assuming the length of the barrel to be approximately $S + \frac{1}{2}S$, or 48 + 25 = 72 in (122 + 61 = 183 cm): $V_{c1} = 72 \times 2420 \times 0.015 = 2614$ in³ $(183 \times 15{,}600 \times 0.015 = 42{,}800$ cm³) Omitting quantities small in the second order, $V_{ec} = 72\,\frac{\pi}{4}\left(2D \times \frac{30{,}000}{30 \times 10^6} \times D\right)$ $= 72 \times 1.57 \times 0.001 \times D^2 = 0113 \times D^2 \; (183 \times 1.57 \times 0001 \times D^2$ $= 0.287 \times D^2)$ For the center cylinder: $V_{ec2} = 0.113 \times 45^2 = 229$ in³ $(0.287 \times 114^2 = 3730$ cm³) For the side cylinders: $V_{ec1} = V_{ec3} = 0.113 \times 23^2 = 60$ in³ $(0.287 \times 58.4^2 = 980$ cm³) $V_{ec} = 229 + (2 \times 60) = 349$ in³ [$3730 + (2 \times 980) = 5700$ cm³] Total additional volume of liquid required. 2614 + 349 = 2963 in³ per stroke (42,800 + 5700 = 48,500 cm³)

TABLE 1 Continued.

Standard Procedure	Example
10. Determine the total amount of liquid V required per cogging stroke.	10. $V = 9680 + 2963 = 12{,}643$ in^3 $(156{,}000 + 48{,}500 = 204{,}500$ cm$^3)$ This value shows what additional burden can be imposed on the power plant by unnecessarily generous dimensioning of the total stroke.
11. Check whether specified speeds are compatible with the required number of cogging strokes. If not, increase the specified speeds (or reduce the number of strokes).	11. Fast advance time: $\frac{4}{6} = 0.67$ s Pressing time: $\frac{4}{2} = 2$ s Fast return time: $\frac{8}{6} = 1.33$ s Valving time (3 switches) $= 0.45$ s Total cycle time $= 4.45$ s This time is too long to allow 15 cogging strokes per minute. Increase fast advance and fast return to 480 in/min = 8 in/s (20 cm/s) and pressing to 180 in/min = 3 in/s (8 cm/s). Then Fast advance time: $\frac{4}{8} = 0.5$ s Pressing time: $\frac{4}{3} = 1.33$ s Fast return time: $\frac{8}{8} = 1$ s Valving time $= 0.45$ s
12. Determine the liquid requirements for the pull-back stroke (Figures 6 and 7). In general, the pull-back cylinders have an area equal to 10% of the area of the main cylinders and twice as long a stroke as the pressing stroke; in general, the required volume is therefore, with sufficient accuracy: $V \times 0.1 \times 2 = 0.2 \times V$	12. $12{,}643 \times 0.2 = 2528$ in^3 $(204{,}500 \times 0.2 = 40{,}900$ cm$^3)$

TABLE 1 Continued.

Standard Procedure	Example
13. Determine the pumping requirements for the pump station with or without an accumulator. Without an accumulator: $\text{gpm} = \frac{V \times 60}{231 \times t_p} \quad \left(\text{m}^3/\text{h} = \frac{V \times 60}{1000 \times t_p}\right)$ where V *is* the volume in gallons (cubic meters) required for one pressing stroke and t_p is the pressing time in seconds. With an accumulator: $\text{gpm (acc)} = \frac{V_{tc} \times 60}{231 \times t_c} \quad \left(\text{m}^3/\text{h} = \frac{V_{tc} \times 3600}{10^6 \times t_c}\right)$ where V_{tc} is the total volume in gallons (cubic meters) of pressurized liquid required during one cycle and t_c is the cycle time in seconds.	13. Without an accumulator: $\frac{12{,}643 \times 60}{231 \times 1.33} = 2469 \text{ gpm} \quad \left(\frac{204{,}500}{10^6 \times 1.33} = 553 \text{ m}^3/\text{h}\right)$ With an accumulator: $\frac{(12.643 + 2{,}528) \times 60}{231 \times 4} = 985 \text{ gpm (acc)}$ $\left(\frac{(204{,}500 + 40{,}900) \times 3600}{10^6 \times 4} = 221 \text{ m}^3/\text{h (acc)}\right)$ It is evident that the use of an accumulator will result in significant savings and in a substantial reduction of the peak load.
14. Determine size of the accumulator required to store the accumulated volume V_s of liquid: $V_s = \text{gpm (acc)} \times \frac{t_c - t_p}{60} - \text{V}_\text{R} \quad \left(\text{m}^3/\text{h} \times \frac{t_c - t_p}{3600} - \text{V}_\text{R}\right)$ where V_R is the volume required for the return stroke in gallons (cubic meters). Air volume required, based on 10% pressure fluctuation for isothermic condition, is $10V_s$. For adiabatic or, more often, polytropic conditions and 10% pressure fluctuation, the air volume is $V_\text{air} = \frac{V_s}{1.11^{1/n} - 1}$ Although $n = 1.4$ is the exponent for adiabatic compression, $n = 1.3$ is considered satisfactory, even for severe and demanding conditions, which require full utilization of press stroke and a fast cycle. Based on $n = 1.3$, $V_\text{air} = 12V_s$	14. $V_s = 985 \times \frac{4 - 1.33}{60} - \frac{2528}{231} = 32.9 \text{ gal} = \frac{32.9}{7.48} = 4.4 \text{ ft}^3$ $\left(224 \times \frac{4 - 1.33}{3600} - \frac{40{,}900}{10^6} = 0.125 \text{ m}^3\right)$ V_air (isometric) $= 1 - \times 32.9 = 329 \text{ gal} = 10 \times 4.4 = 44 \text{ ft}^3$ $(10 \times 0.125 = 1.5 \text{ m}^3)$ V_air (polytropic) $= 12 \times 32.9 = 395 \text{ gal} = \frac{395}{7.48} = 53 \text{ ft}^3$ $(12 \times 0.125 = 1.5 \text{ m}^3)$ Total accumulator volume $= 44 + 4.4 = 48.4 \text{ ft}^3$ $(1.25 + 0.125 = 1.38 \text{ m}^3)$ $= 53 + 4.4 = 57.4 \text{ ft}^3$ $(1.5 + 0.125 = 1.63 \text{ m}^3)$

TABLE 1 Continued.

Standard Procedure	Example
15. Check planishing conditions.	15. Timing: Fast advance: $\frac{1/4}{8} = 0.032$ s Pressing: $\frac{1/4}{3} = 0.084$ s Fast return: $\frac{1/2}{8} = 0.063$ s Valving (2 switches) $= 0.3$ s Total $= 0.479$ s There will be more than enough time for 80 planishing strokes per minute. Only two valve switches are required for planishing because, during planishing, the pull-back cylinders are permanently connected to the pressure source. During planishing, the pressure reaches only about 10% of rated; therefore compressibility and expansion require only 10% of the previously calculated amount. The pull-back system is constantly pressurized and does not require any liquid for compression of liquid and expansion of cylinder. The total requirements are thus For the pressing stroke: $605 + 296 = 901$ in^3 ($9900 + 4850 = 14{,}750$ cm^3) For the pull-back stroke: $605 \times 0.2 = 121$ in^3 ($9900 \times 0.2 = 1980$ cm^3) Total $= 1022$ in^3 ($= 16{,}730$ cm^3) gpm required $= \frac{1022}{231} \times 80 = 354$ m^3/h required $= 16{,}730 \times 10^{-6} \times 80 \times 60 = 80.3$ The selected accumulator and pump power plant will be more than sufficient for planishing. As a matter of fact, 100 or even 110 planishing strokes per minute will be feasible.

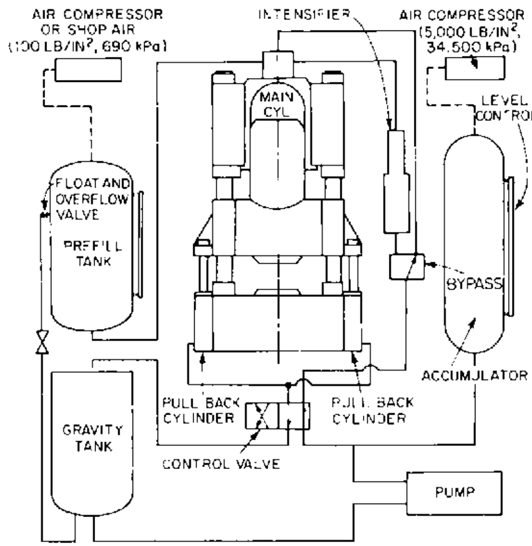

FIGURE 6 Vertical forging press with pull-back cylinders

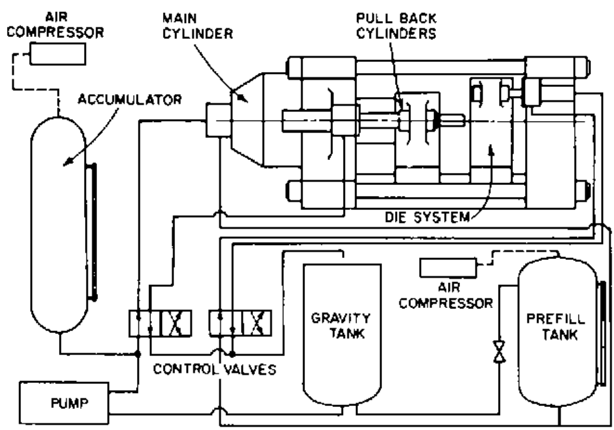

FIGURE 7 Horizontal extrusion press