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Hydraulic Ram Design for Modern High Pressure Devices

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Equations are developed for designing hydraulic rams of maximum thrust for a given external cylinder diameter. For push type rams, the optimum ratio of outside to inside wall diameter w is 1.554, and the maximum fluid pressure P to be used in 0.4142 times the allowable hoop stress f_t in the cylinder. For pull type rams in which the allowable tensile stress f_b in the pull bar (cylinder rod) and the allowable hoop stress f_t in the cylinder wall are equal, w = 1.414 and P = 0.333 f_t. When these allowable stresses are not equal, more complicated relationships exist and are discussed in the text.

WHEN I first became interested in high pressure research about 15 years ago, a 1000 ton hydraulic ram was rather rare and bigger than a house. Such rams also cost more than most houses. A 1000 ton press purchase by General Electric for diamond research in 1954 was three stories high and cost about $100,000. The advent of modern day high pressure devices such as the belt,[1] the supported piston,[2] and the tetrahedral press[3] created a need for high tonnage presses of greatly reduced size and cost. This need has been partially met during the intervening years by the appearance on the market of several compact hydraulic jacking rams for use in the construction industry. For example, 200 ton capacity rams only about 25 cm in diameter by 60 cm long and weighing around 180 kg may now be purchased for under $1000. Such rams have been used to provide the driving thrust for high pressure devices.

Through the years, I have endeavored to discover some principles for designing the smallest possible hydraulic rams and for lowering their cost. A multipiston ram which somewhat successfully meets these objectives has previously been discussed.[4] At least one manufacturer has adopted the multipiston design in a commercial press recently offered for sale. It is the purpose of the present paper to develop some equations that are useful in designing simple push or pull type hydraulic rams of minimum external dimensions. These equations are developed in problem form. The symbols used and their meanings follow:

D_0 is the outside ram (cylinder) diameter;
D_i the inside ram (cylinder) diameter;
w the ration of outer diameter of cylinder to inner diameter;
T the ram thrust;
P the hydraulic fluid pressure acting on piston;
f_t the maximum circumferential hoop stress allowable in cylinder (occurs at inner wall for stresses in elastic range);
D_b the diameter of pull bar in pull type ram; and
f_b the maximum allowable tensile stress in pull bar of pull type ram.

PROBLEM 1, PUSH TYPE RAMS

[1] H. T. Hall, Rev. Sci. Instr. **31**, 125 (1960).
[2] F. R. Boyd and J. L. England, Yearbook Carnegie Inst. **57**, 170 (1958).
[3] H. T. Hall, Rev. Sci. Instr. **29**, 267 (1958).
[4] H. T. Hall in Progress in Very High Pressure Research, F. P. Bundy, W. R. Hibbard, Jr., and H. M. Strong, Eds. (John Wiley & Sons, New York, 1961), pp. 1-9.

Given the fixed quantities T and f_t for a simple push type hydraulic ram (Fig. 1), find the values of P, D_i, and w that make D_0 a minimum.

Solution: The thrust developed by the ram is given by

$$T = \tfrac{1}{4}\pi D_i^2 P. \qquad (1)$$

It is assumed that the piston and piston rod have adequate strength to transmit this thrust. This is normally not a problem in push type rams. The ratio of outer to inner cylinder diameter is

$$w = D_0/D_i. \qquad (2)$$

The relationship between the maximum hoop stress in the cylinder, the oil pressure, and the wall ratio is given by the familiar Lame equation,[5]

$$f_t = [P(w^2 + 1)]/(w^2 - 1). \qquad (3)$$

The condition of minimum outside diameter requires that

$$dD_0/dw = dD_0/dD_i = dD_0/dP = 0 \qquad (4)$$

Noting from (2) that w = D0/Di, combine (1) and (3) to eliminate Di and obtain

$$D_0 = [4T\,(f_t + P)/\pi P\,(f_t - P)]^{½}. \qquad (5)$$

Differentiate (5) with respect to P and set dD_0/dP = 0 according to (4) and obtain

$$P = [f_t \pm (f_t^2 + 4T)^{½}\,]/4T. \qquad (6)$$

Equation (6) gives the hydraulic fluid pressure P in terms of the known quantities f_t and T. This value of P may be substituted in (5) to find D_0 and in (1) to find D_i. When this has been done and the ratio of D_0 to D_i is taken, the value of w is found to be 1.554 and is independent of P, f_t, and T. Additional relationships may now easily be found, and some are listed below,

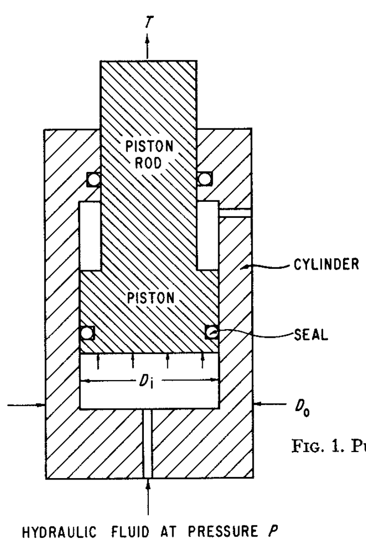

FIG. 1. Push type hydraulic ram.

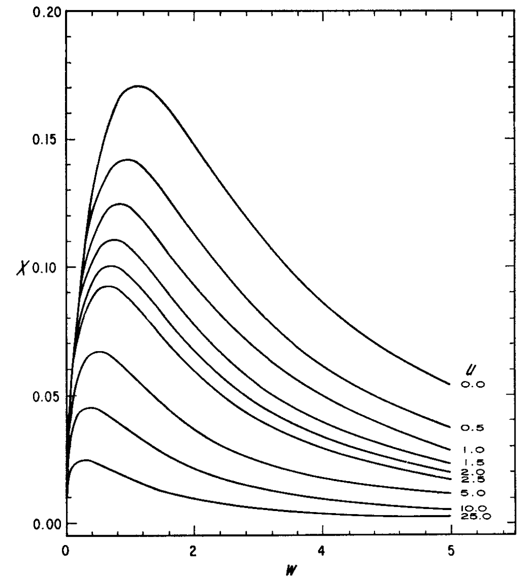

FIG. 2. Thrust efficiency factor X as a function of cylinder wall ratio w. The parameter u is the ratio of allowable stress in the cylinder f_t to the allowable stress in the pull bar f_b of a pull type ram.

w=1.554, (7) P=0.4142 f_t (10)

D_0=2.725 $(T/f_t)^{½}$ (8) T=0.1347 $D_0^2 f_t$, (11)

D_i=1.753 $(T/f_t)^{½}$ (9) f_t=2.414 P. (12)

It would be illuminating to know how, for a given D_0 and f_t, departures from the optimum value of 1.554 for w affect the ram thrust T. The thrust in terms of D_0, w, and f_t is given by

$$T = \tfrac{1}{4}\pi\,(D_0^2/w^2)\,[(w^2 - 1)/(w^2 + 1)]f_t$$
$$= \tfrac{1}{4}\pi\, D_0^2 f_t X, \qquad (13)$$
$$X = w^{-2}[(w^2 - 1)/(w^2 + 1)].$$

Equation (13) was obtained by eliminating P between (1) and (3). The factor X may be regarded as the thrust efficiency factor and when plotted against w yields the uppermost curve shown in Fig. 2. The remaining curves in the figure are discussed later when pull type rams are considered. It is to be noted that the maximum value of X (maximum ram thrust) occurs at w = 1.554 as previously determined and falls off rather steeply each side of the maximum.

In general, values of w larger than the optimum are not of interest because of the increased hydraulic pressure required (Eq. 3). Smaller values of w, however, require lower fluid pressures for a given f_t and may be of value

Table I. Effect of decreasing cylinder wall ratio w below the optimum value on the relative efficiency and fluid pressure for push type hydraulic rams (u=0).

Relative efficiency	w	Relative fluid pressure
1.00	1.554 (optimum)	1.00

[5] S. Timoshenko, *Strength of Materials* (D. Van Nostrand Company, Inc., New York, 1956), Pt. II, 3rd ed., p. 208.

0.95	1.380	0.75
0.90	1.313	0.64
0.85	1.275	0.58
0.80	1.240	0.52

under some circumstances. Analysis of the upper curve of Fig. 2 for values of *w* slightly less than optimum indicates the relative efficiencies and fluid pressure requirements given in Table I. Note that a reduction in efficiency of only 5% results in a 25% reduction in the necessary fluid pressure.

PRACTICAL RAMS

Low volume, reliable, hydraulic pumps, particularly of the air driven variety, are readily availble for pumping hydraulic fluid to pressures of 2000 bar. These pumps may be purchased for around $500. Tubing, fittings, and gauges are also readily available for 2000 bar use. Equation (12) indicates that the maximum hoop stress developed in an optimum design cylinder operating at 2000 bar oil pressure would be 4800 bar. Now, the Lame equation (Eq. 3) for the maximum tangential stress is based on an analysis of a cylinder in which the entire inner surface is subjected to fluid pressure. The stroke (piston travel) of a hydraulic ram used in multiple anvil or belt like devices need be only a few centimeters. Consequently, only a short length is strengthened by a section of cylinder not exposed to hydraulic pressure and may also be strengthened by the closure (bottom of cylinder) which is often designed to act as an integral part of the cylinder. Therefore, the actual maximum stress developed in the cylinder wall is nominally only about half of 4800 or 2400 bar. Allowing a safety factor of 3, the steel of which the cylinder might be constructed should have a minimum yield point of 7200 bar. Steels of this strength level, of good workability and reasonable cost are readily available. Therefore, the full 2000 bar capability of the hydraulic pump may be utilized to keep the outside diameter of the ram as small as possible. When the ram diameter is small the length may be correspondingly small, and other associated components of the complete high pressure system may also be reduced in size to give an apparatus that in over-all physical dimension is substantially smaller than would be the case of the optimum design ram were not used.

A simple O-ring seal (buna N rubber with a shore durometer hardness of 90) with a leather backup washer on each side is adequate as a piston seal when the above ram is used to actuate gasket type high pressure devices (Bridgman anvils, belts, tetrahedral presses, etc.). In these devices, the actual piston travel of the hydraulic ram while under heavy load is less than a centimeter—just that required to compress and extrude the gaskets and accommodate for stretching and compressing of the apparatus components. More elaborate seals, designed to follow the "breathing" of the cylinder (radial expansion and contraction under loading and unloading), would be necessary for longer strokes under heavy loads.

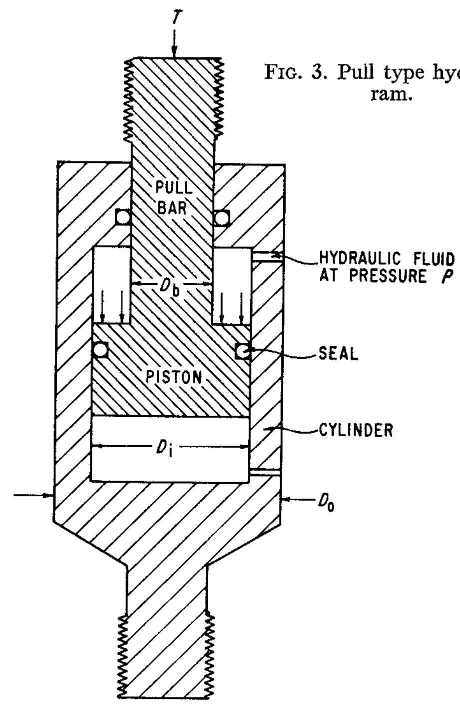

FIG. 3. Pull type hydraulic ram.

PROBLEM 2, PULL TYPE RAMS

Given the fixed quantities T, f_t, and f_b for a simple pull type hydraulic ram (Fig 3) find the values of P, D_i, w, and D_b that make D_0 a minimum.

Solution: In this ram, it is assumed that the connections to the pull bar are stronger than the pull bar. The pull bar diameter is set by the allowable tensile stress f_b in the equation

$$D_b = (4T/\pi f_b)^{1/2}. \qquad (14)$$

The thrust (pull) is generated by hydraulic pressure acting on the annulus defined by Di and Db and given by

$$T = (\pi P/4)\,(D_i^2 - D_b^2). \qquad (15)$$

The quantities w and f_t are the same as given before in (2) and (3) and the minimum

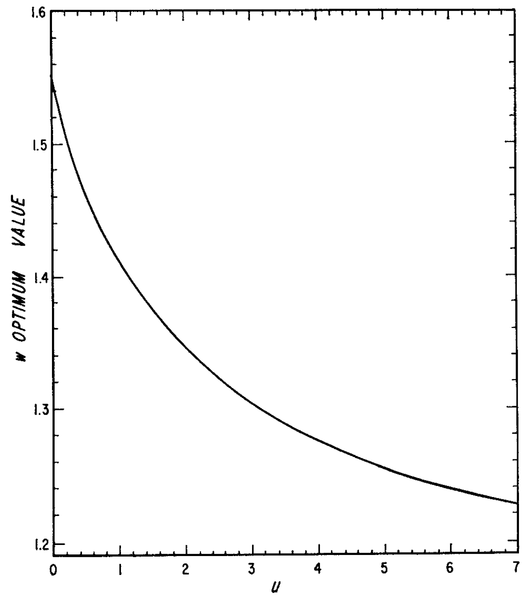

FIG. 4. Effect of allowable stress ratio u on the optimum wall ratio w of a pull type ram.

conditions given by (4) also hold. It is convenient to define a new quantity, the ratio of the maximum allowable hoop stress in the cylinder f_t to the maximum allowable tensile stress f_b in the pull bar,

$$u = f_t/f_b. \qquad (16)$$

By manipulating the above equations in a manner similar to that followed in the case of the push ram, the following relationships were discovered for a minimum D_0.

$$w^2 = [2u + (2 + 2u)^{1/2}]/[2 + 2u – (2 + 2u)^{1/2}]; \qquad (17)$$

$$D_i^2 = (4T/\pi f_t)\{(1 + 2u)/[(2 + 2u)^{1/2} -1] + u\}; \qquad (18)$$

and

$$P = f_t\,[(2 + 2u)^{1/2} - 1]/(1 + 2u). \qquad (19)$$

In these equations note that w depends only on u. A plot of w vs. u is shown in Fig. 4. The o.d. of the pull ram D_0 is given by the square root of the product of (17) and (18). For the special case in which u = 1; i.e., $f_t = f_b$, the above three equations reduce to

$$w^2 = 2, \qquad (20)$$

$$D_i^2 = 16T/\pi f_t, \qquad (21)$$

$$P = \frac{1}{3}\, f_t. \qquad (22)$$

When u = 0, the equations become identical with those developed for a push type ram. This represents the hypothetical situation in which the pull bar is a fine wire of infinite strength. The upper curve of Fig. 2 is labeled with the parameter u = 0. The effect of increasing the parameter u is shown by the lower curves. It is unlikely that u would ever exceed 2.5 for a practical ram design.

PRACTICAL PULL RAMS

As was mentioned for push rams, it may also be desirable at times to utilize a value of w slightly less than optimum for pull rams. A slight reduction of w again leads to a substantial reduction in the operating fluid pressure. To illustrate this point, relative efficiencies, fluid pressures, and wall ratios are given in Table II for u = 1 and u = 2. It is to be noted that the relative fluid pressures for a given efficiency are about the same for u = 2, 1, or 0 (push ram case for which data were previously given).

Table II. Effect of decreasing cylinder wall ratio w below the optimum value on the relative efficiency and fluid pressure for pull type hydraulic rams in which the ratio of hoop stress to tie bar tension u=1 and 2.

Relative efficiency	w	Relative fluid pressure
	u=1	
1.00	1.414 (optimum)	1.000
0.95	1.280	0.726
0.90	1.235	0.619
0.85	1.200	0.540
0.80	1.175	0.479
	u=2	
1.00	1.348 (optimum)	1.000
0.95	1.235	0.715
0.90	1.195	0.606
0.85	1.165	0.528
0.80	1.145	0.465

In general, the optimum ram would use the same high strength steel for both the pull rod and the cylinder. However, when the cylinder wall is strengthened by an adjoining section of unpressurized cylinder and by end closures, the allowable hoop stress f_t to be utilized in the above equations may be as much as twice as large as the allowable tensile stress f_b in the pull bar.

ACKNOWLEDGEMENT

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