

THE HYDRAULIC WORLD AS SEEN BY A MECHANICAL HYDRAULIC ENGINEER

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Social psychologists have long emphasized the effects of environment upon the development of individuals. That "environment" may be much more particular than on which side of the tracks one lives is well illustrated by the two engineers who were given the same device to design. The first engineer, having had long experience in rolling-mill machinery, was told to design a reducing gear for a two-thousand-horsepower turbine but to make it as light as possible. The second engineer, an aircraft man, was given the same problem but was told that weight was no object. In due course the first engineer proudly exhibited his design which weighed only five tons. The second engineer apologetically showed his design which weighed nearly half a ton. Similar environmental differences have affected the viewpoints of men in the hydraulic field. The civil, mechanical, and research hydraulic engineers face different

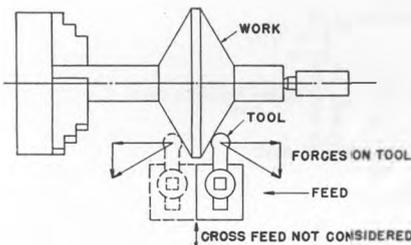


FIG. 1

problems and consequently see different things. The author, believing that mutual understanding which tends to break down the barriers between the branches of hydraulics will have a salutary effect, wishes to present some of the "parameters" of a mechanical hydraulic engi-

neer to the members of this conference. It is hoped that a measure of informality in presentation will clarify rather than obscure the points the author seriously wants to impress.

A typical problem of the mechanical hydraulic engineer is that of applying a hydraulic system to a lathe. There are many things which could be done hydraulically such as longitudinal feed, cross feed, chucking, rapid traverse, and even the rotary drive, but as

an example it is assumed that in this particular case only the longitudinal feed is to be operated by hydraulic means. The most difficult piece of work this lathe is expected to do is something such as shown in Fig. 1. The tool feeds from right to left. When the tool is in the position shown in solid lines, the forces on the tool from the work are as shown inasmuch as a force must be applied to keep the tool moving to the left. However, somewhat later when the tool has reached the dotted position, the longitudinal component of the force on the tool has reversed in direction so now the tool must be held back. The only purpose of this Figure is to point out that even though the motion of the lathe tool is from right to left, the forces applied to keep it moving at a uniform rate may be in either direction.

The simplest possible hydraulic system—a pump, relief valve, throttle valve, cylinder, and a reservoir—is considered first. Since the reservoir is at “ground potential” in the electrical engineer’s language, his symbol may be used for ground instead of running

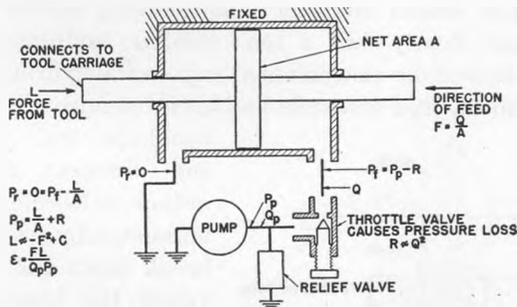


FIG. 2

lines around and drawing the tank. Fig. 2 shows such a system. The rod connects to the tool carriage and the load from the tool is shown applied to the left end of the rod. This is the condition existing with the tool in the position shown in the full lines in the previous figure. The pump draws fluid from the reservoir and discharges to the throttle valve. The relief valve prevents development of excessive pressure by returning unused fluid to the reservoir. The fluid which passes the throttle valve enters the cylinder to the right of the piston (that is, the feed end) and moves it to the left. Fluid in the left end of the cylinder, the return end, es-

capacities to the reservoir. The area of the piston and the working pressure at which the relief valve is set are determined by the maximum force to be exerted under any conditions. The capacity of the pump is set by the maximum rate of tool movement ever necessary. Both maxima seldom occur at the same time but provision must be made for both. It may be assumed that in this problem the tool is fed at half the maximum rate and that L is half of the available maximum. If line losses, mechanical friction, etc. are neglected, P_r , the pressure in the return line, is zero.

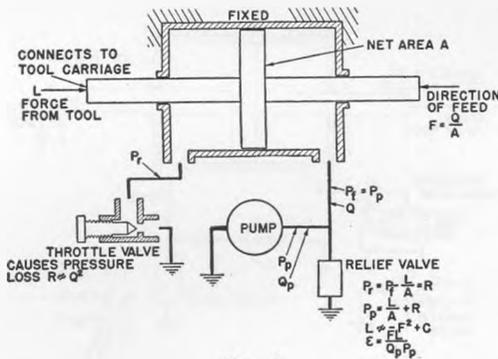


FIG. 3

As long as the force from the tool is in the direction shown everything goes along nicely with the tool feeding no faster than the rate of flow into the cylinder forces it to do. However, when the tool has advanced to the dotted position of Fig. 1, the tool force reverses in direction, and since there is no way to maintain any pressure in the left end of the cylinder, the tool "runs away." (In practice there is never sufficient mechanical friction plus atmospheric pressure less vapor pressure of fluid to hold a tool into any reasonable reverse cut.) It seems foolish to continue with a circuit having obviously bad characteristics so another system will be considered.

The arrangement in Fig. 3 is much the same as before, but the throttle has been moved to the return side of the cylinder. Hence, P_r , instead of being zero, is equal to R , the pressure loss in the throttle, and when the tool tends to run away, a pressure is available to hold it. Here the pump pressure P_p , which is constant because of the relief valve, and the feed pressure P_f are the same

and must equal $L/A + R$. Since the assumed feed rate is half of that provided for, half of the fluid must be going through the relief valve. Also, since the assumed load is half of that provided for, $L/A = 1/2$ and so R must equal $1/2$. Now suppose the tool strikes a hard spot in the work and L increases 50%. What happens? L/A becomes 150% of its former value, or $3/4$ and since P_t is constant, R must decrease to $1/4$. Since the throttle valve is really a fixed orifice, the pressure drop through it varies as the square of the rate of flow. Therefore the change from $1/2$ to $1/4$ in R means

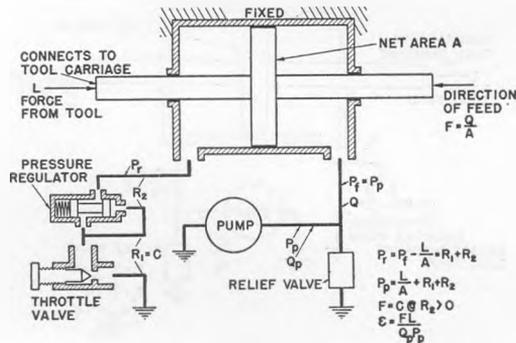


FIG. 4

a decrease in flow from $1/2$ to $1/8$, or simply $1/4$ of the former flow. Thus a change in L of 50% changes the feed rate by 75%. It is important to consider the efficiency of the circuit. Obviously the efficiency equals FL over $Q_p P_p$. At first it was $1/2 \times 1/2$ or 25%, and while the tool was in the hard spot it became $(1/4 \times 3/4) / (1 \times 1)$ or 18.75%. Neither is anything to be proud of but they can be improved.

It may be seen that the feed rate depends upon the rate of flow through the throttle valve. This in turn depends upon the pressure drop across the valve so if a constant pressure drop is maintained, a constant feed rate will result. Fig. 4 shows the same circuit with the addition of a pressure regulator to hold the pressure applied to the throttle constant. Here the fluid coming out of the return end of the cylinder enters the upper port of the regulator, flows through it and to the inlet of the throttle. As pressure builds up in the regulator outlet line, this pressure is transmitted to the right end of the regulator piston valve tending to move it to the left.

This motion, which gradually closes off the inlet port, thus increasing the pressure drop across the valve, is resisted by the spring so the piston comes to rest when the pressure force and the spring force are in balance. If the outlet pressure tends to increase, the piston valve moves over thus further choking off the inlet. Hence, there is now a substantially constant pressure at the throttle inlet which will pass a constant rate of flow. Not only has a constant feed rate been obtained when the tool is resisting movement, but also when it is being held back. Assuming the same feed rate, load, and change in load as before, the efficiency should again be computed. Before the change in load, it is $(\frac{1}{2} \times \frac{1}{2}) / (1 \times 1) = 25\%$, as before. When the tool hits the hard spot, the efficiency becomes $(\frac{1}{2} \times \frac{3}{4}) / (1 \times 1) = 37.5\%$ or twice as good as it was before the regulator was added.

One more change may be made in an attempt to improve the efficiency to complete this illustration. Fig. 5 shows this change. The feed rate control is just as it was; however, the relief valve has been changed to a tricky affair with dual control. Here the

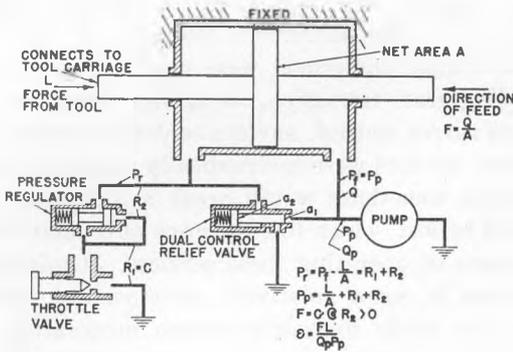


FIG. 5

pump pressure is applied to area a_1 on the end of the plunger and P_r is applied in the same direction to the area a_2 on the head of the plunger. The sum of the two forces is resisted by the spring so that if either tends to increase, fluid is by-passed to the reservoir. If areas a_1 and a_2 are made equal, each will have the same effect. Now $P_p = P_t = L/A + R_1 + R_2$, just as before except that neither side of the equation is constant. P_r , of course equals $R_1 + R_2$ and will be constant because any change will alter the amount of fluid

being by-passed and thus change P_t . With $R_1 + R_2$ constant, any change in L/A will change P_t just enough to compensate, so now the pump only puts out enough pressure to overcome the load. It might be asked "Why is the regulator still necessary?" Actually, it is not as long as the hydraulic system drives the tool, but when the tool must be restrained from running away, regulation of pump pressure will not help, and the pressure regulator is required to take care of this condition.

Again the efficiency may be considered for the same example as before. At first the efficiency is $(\frac{1}{2} \times \frac{1}{2}) / (1 \times \frac{1}{2})$ or 50%,—twice what it was, and with the tool in the hard spot, the efficiency is $(\frac{1}{2} \times \frac{3}{4}) / (1 \times \frac{3}{4})$ or 50%. That the efficiency should remain constant is evident from the fact that the term L/P has become constant. This leaves the efficiency dependent solely on the feed rate.

Of course this system is not complete since in practice directional control valves and other units are required. There are entirely different and better ways of accomplishing the result, but this example serves as a good illustration of a way of thinking.

One more example of a mechanical hydraulic gadget will be given, which was of considerable importance in war time. Large amounts of hydraulic equipment were used on combat aircraft. In general, landing-gear retraction, wing-flap extension, bomb-door actuation, gun-turret control, surface-control boosters, brakes, and sometimes other services were hydraulically operated. Unfortunately, enemy action sometimes would break a hydraulic line and all the fluid would be lost, which left the crew only laborious, separate, emergency means of operating these services. It might be that the broken line was in some relatively unimportant service but its failure made the whole hydraulic system inoperative. Hence, an intensive search was begun for something which would act in a hydraulic circuit as a fuse acts in an electrical circuit. Thus the name "hydraulic fuse" was born before there was anything to which to apply it. The problem was not as simple as it looked. It seemed that valves might be put in the pressure lines which would close if the pressure fell below some predetermined value. However, the valve could not distinguish between normal and abnormal causes of pressure drop. This failure illustrates a common "blind spot". Too many times a device is expected to use more reasoning ability than its designer possesses.

Several different attacks proved equally fruitless until an attack was tried which made use of the fact that in nearly every self-respecting hydraulic system there is a return flow concurrent with the pressure flow. If this return flow failed, there must be a break through which fluid escaped. This attack produced an effective answer and many thousands of such "fuses" were used. Fig. 6 shows how this fuse works. There may be several valves in a parallel

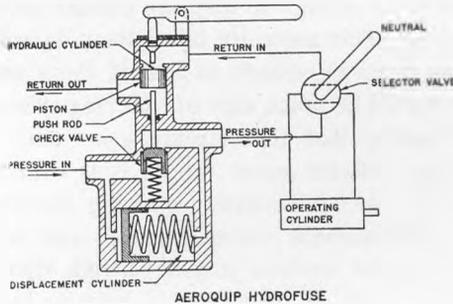


FIG. 6

connection but if all are closed just one of them may be considered. Pump pressure will be maintained in both the "pressure in" and "pressure out" lines since there is no flow. Now if the selector valve is opened, the pressure in the "pressure out" line will fall. This cannot be made up through the poppet valve because it is closed. However, pump pressure is applied to the left end of a piston in a displacement cylinder. The other end of this cylinder is connected to the "pressure out" line so there is a pressure differential across the piston. The piston moves to the right thus displacing fluid to the "pressure out" line. This fluid goes through the selector valve and to the operating cylinder which starts the desired motion. The return fluid from the cylinder goes back through the selector valve and acts on a return piston valve in the fuse. This piston must move downward and uncover the "return out" port before the fluid can escape to the reservoir. In so doing it opens the pressure poppet and flow continues through that to the operating cylinder. Now that the pressure poppet is open the pressures on both sides of the displacement piston are substantially equal and the spring returns the piston so it is ready for the next operation. Now, if one of the lines connected to the cylinder breaks,

the fluid will go overboard and there will be no return flow acting on the return piston. It will move up under the load of the spring plus the force of the pressure in the pressure end acting on the push rod. That permits the poppet to reseal. The displacement piston will advance to the end of its stroke and then flow will cease. Only a very little more fluid than was contained in the displacement cylinder (about 4 cu. in.) has been lost. The displacement piston only moves far enough to displace enough fluid to move the return piston valve. This normally takes about $\frac{1}{4}$ cu. in.; however, the volume is made large enough so that if there were a little air in the system it would be taken care of. Having tried one hydraulic operation and finding that that circuit was "out" the pilot has only to leave that selector valve where it is shut off in neutral position, reset the fuse by manually opening the pressure poppet whereupon the displacement piston returns, and he can then use all the other hydraulic controls just as though there had been no trouble. The only function which would have to be performed by emergency means would be the one which had been damaged.

A more important subject—Research Problems—will be considered next. Just as each branch of engineering has its own particular research problems, the mechanical hydraulic engineer has his. In addition there is the all important problem of basic research into the pure theory behind the work of all fields. The writer can make no suggestions as to the direction this can take, but it is necessary. The other suggestions the writer might make for research projects are related much more closely to his own than to the Civil Hydraulics field.

The characteristics of the best possible hydraulic fluid should be determined, and then one or more closely approaching this ideal should be developed. A lot of work has been done to develop a fluid but the writer feels that most of it has been done by men who are biased, perhaps unconsciously, in favor of one base. Apparently the requirements are not known exactly, and certainly the development should be undertaken by idealists with no axe to grind.

It is also necessary that better packings be developed. Anyone who has had packing experience will admit that of the hundreds of packings now known, none are really good. This problem, in the writer's opinion, requires the attention of men who are not in the rut of conventional thinking.

A study should be made of the suitability of materials to use in hydraulic equipment. There are some factors such as corrosion resistance and homogeneity which are more important here than in most other industries. An outstanding example is that of determining the requirements for surface quality of hydraulic parts, particularly those surfaces traversed by packings. It is known that a rough surface quickly wears out packings, but, strangely enough, too smooth a surface tears them up. This should be correlated with work on production methods of measuring surface quality and maybe even of production methods of reproducing desirable qualities. Much work has been done on this but, so far, most of it has been either too "theoretical" or too "practical" to be of maximum value.

There should be more work on the effects of transient phenomena in hydraulic systems. Strange things occur. For example, the writer has seen tubes blow out of fittings in an airplane, an effect which at first could not be duplicated in the laboratory. It happened repeatedly so could hardly have been charged to a defective tube. Even though extreme measures were taken to have everything right, still it happened. Tubes tested by suddenly applying pressure would burst, but the fitting would stay on. Finally it was shown by means of strain gages, electronic amplifiers, and an oscillograph that shock waves of enormous magnitude were present. The whole thing was over in about 1/100th of a second, but during that time there occurred about five surges having peak pressures three times the bursting strength of the tubing. These hammer blows drove the tube out of the fittings but did not persist long enough to burst it. The effect was duplicated subsequently with a hammer, and after the circuit was changed, the trouble vanished.

Men of imagination and daring could do much to develop new types of hydraulic circuits. Possibly circuits having the characteristics of alternating current will be evolved. Perhaps simpler pumps, smaller lines, hydraulic transformers would eventually result.

There are many problems in large power circuits. If a valve is closed quickly on even a 20-HP circuit, the result is disastrous.

Finally new methods should be found for the critical control of

hydraulic circuits. Much has been done, but no one would seriously contend that perfection has been attained.

Now, in the event that any, some, or all of these projects are undertaken, the most important project of all has not been mentioned. All real men have an inner urge to benefit society—to leave a better world to their children. It may not be a bluntly-stated, consciously-held notion but it is none the less real. Captain Saunders has stressed the need for cooperation between scientists and engineers in order to make progress. This brings up the most important project of all—*Methods of presenting theoretical concepts to the man who will use them should receive serious attention.*

The writer is frequently brought up short by the, to him, completely mysterious meaning of some equation which the research man or mathematician who evolved it blandly assumed any one would know. Many years of experience in the world of business in engineering have convinced the writer that he is not alone in this predicament. It is time that everyone overcome the ostrich-like attitude which leads to complacency. Even though everyone has the feeling that these remarks do not apply to him, the facts should be faced honestly and objectively.

While undergraduates, engineers spoke much the same language, but even then there were two distinct groups and many in between. The first group thought primarily in mathematical terms. The second group in mental motion pictures. Of course, some apparently didn't think in technical terms at all. Where are the two groups now? The first, or mathematical, group is largely in research or teaching and the second, or visualizing group, is in design, development, or application fields.

It is essential that these groups get together. They are about equally important and neither can prosper or be of maximum benefit to society without the help of the other. In general the first group must go more than half way toward this meeting. There are real reasons for this. In the first place, the second group was endowed with a different kind of brain and is incapable of rising to the mathematical level of the first group. Maybe that is an overstatement. Perhaps it is only that they cannot rise to the same mathematical level without years of additional study. In the second place, most of the men in charge of engineering departments or manufacturing concerns, or the men of standing in the consulting

field have been out of school for at least twenty years. During this time they have lost much of the mathematical facility they once possessed. This is to be decried, *but it remains true*. Actually, in the business world there is little occasion to use advanced mathematics and without use it atrophies. There are so many other problems that few men can afford the time to keep up their mathematics as a hobby.

Thirdly, many concepts are accepted and are taught to sophomores today which were unheard of by the student of twenty years ago. Group one, the research men, have been the instigators of these new concepts and consequently are up to date. It should be remembered that the "downtown centers" of today's thought were the "remote outskirts" of knowledge only yesterday.

Lastly, the first group has lived in a world of mathematics or at least a world completely surrounded by mathematical vistas. Their student-day facility has greatly improved.

Yes, facts force the research man to go *more* than halfway to meet his brother in the industrial field.

What can he do to start along the path to better understanding? Several things. First, and far most important, he must make a sincere effort to understand the viewpoint and the limitations of the man in this other world. It may be said that he is working too hard at his own job to take the time and doesn't know too much about the other field. Bunk! If he doesn't know there are thousands of well-seasoned, intelligent engineers he can ask. Any of them can explain a situation in a few minutes, whereas it will take months to reeducate them to the point where they will understand the research men.

Second, less attention should be paid to the interesting parameter which has only obscure, intangible physical meaning. The writer realizes full well the value of dimensionless parameters when intelligently devised. However, the important consideration is their usefulness, and not the mere fact that they are dimensionless. Unless the computations involved in the parameter are much less complex than the problem itself, by all means forget the parameter.

Third, the exact meaning of symbols used must be explained. Group two does not know that in some obscure paper a little-known professor defined p as a particular non-dimensional ratio of lengths.

Fourth, contributions should be made in simple statements easily

applied by the "practical" man. The writer does not advocate omitting the derivation, evidence, theoretical background, or above all, the limitations of an equation. It would be much better to include them and the general form of the equation, but in addition, several frequently occurring, special cases in simple form should be given even at the sacrifice of exactitude in the fifth or fourth, or even third significant figure.

The following example is a childishly simple illustration of the writer's idea, but nevertheless is valuable to the man who must watch costs; time, no matter how spent, is money. Not all men in an engineering department are college graduates; many are not. Frequently some man who never got through high school is the best designer in the place. Such a man may have little formal education, but he grew up in a shop and knows manufacturing processes and limitations. He may have been a draftsman for many years, and hence can be relied upon to do a conscientious job of design and his product can be made at a minimum of cost. As a typical job, this man might be asked to design a hydraulic cylinder having one-inch bore, ten-inch stroke, and able to withstand an internal pressure of 10,000 p.s.i. Weight is very important and the barrel will be made of dural which can be stressed to 30,000 p.s.i. maximum—a simple problem.

Referring to his new handbook, the designer looks up a formula by which to calculate the wall thickness of the barrel, and finds the following three formulas:

Clavarino formula—

$$S = \frac{r^2 P - R^2 P_0 + \frac{4r^2 B^2}{X^2} (P - P_0)}{3 (R^2 - r^2)} \quad (1a)$$

Lamé formula—

$$S = \frac{r^2 P - R^2 P_0 + \frac{r^2 R^2}{X^2} (P - P_0)}{(R^2 - r^2)} \quad (1b)$$

Thin cylinder formula—

$$S = \frac{Pr}{t} \quad (1c)$$

Obviously, he will use the third formula (Eq. 1 c). The cylin-

der was to be light and have a thin wall, in fact an older hand-book would have given only the formula $S = Pr/t$. What do these formulae mean? Starting with the third, the thin-cylinder formula takes no account of the longitudinal load which occurs when the piston bottoms in the front end of the cylinder. Lamé's formula is general, but it gives only apparent stresses. Clavarino's

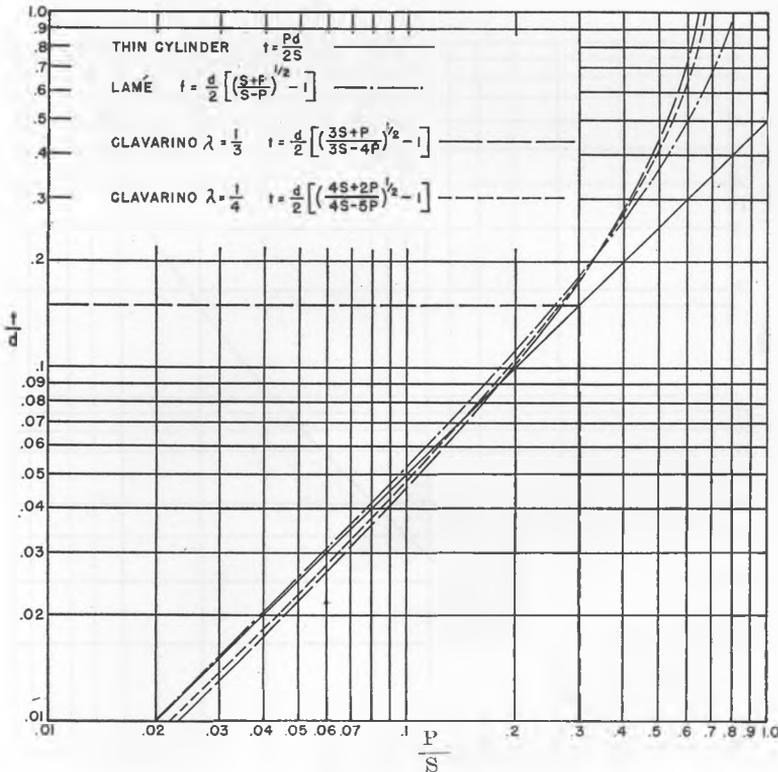


FIG. 7

formula also is general. It takes account of end load, inside pressure, outside pressure, and thickness. It also takes account of Poisson's ratio (the only form given is that corresponding to a ratio of $\frac{1}{3}$), and it gives true stresses.

This design problem is a special case, since there is no outside pressure. However, it is the particular special case which occurs more than 90% of the time and the three formulas have been plotted in Fig. 7 for this case. The fourth curve is for Clavarino's equa-

tion when Poisson's ratio is $\frac{1}{4}$. To make them as easy as possible to use, the coordinates are two real ratios. At the bottom is P/S , that is, internal pressure over allowable stress in the material. The designer knows both these figures so their ratio is very easily obtained. The vertical coordinates are the ratio t/d . The designer knows the inside diameter, so he has only a single multiplication to get his answer. The curves cover only a limited range, but a

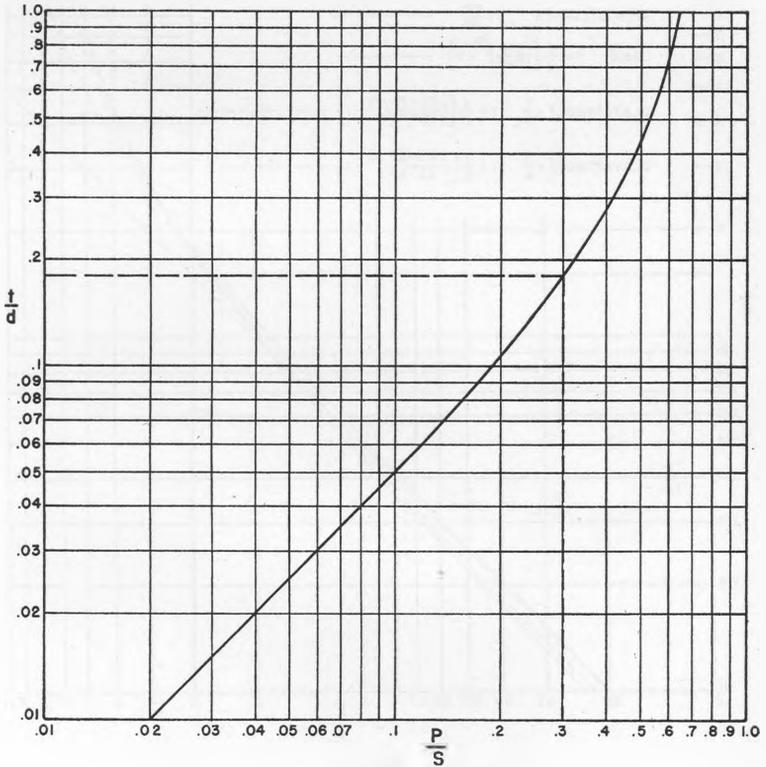


FIG. 8

larger range is unnecessary because smaller ratios of t/d cannot be fabricated readily and larger ratios rapidly approach solid metal with no hole.

Inspection of the curves reveals that the thin-cylinder formula is slightly more conservative than the Clavarino formulae for ratios

of t/d of 0.065 or less. Above that the Clavarino formulae are progressively more conservative (and more exact). From this we see that no wall thicker than $6\frac{1}{2}\%$ of the I. D. of a cylinder can safely be considered "thin". Note that use of the thin-wall-cylinder formula in the example with $P/S = 0.3$ gives a wall thickness of 0.15".

Lamé's formula is of little interest since it does not give true stresses and Clavarino's formulae for Poisson's ratios of $\frac{1}{3}$ and $\frac{1}{4}$ plot very close together. Hence, it is possible to make a single curve and avoid all decisions as to which to use. This is done in Fig. 8. Here the lower part is the straight line given by $t = Pd/2S$. The upper part is the Clavarino formula for $\lambda = \frac{1}{3}$. Since these do not normally become tangent, a short transition curve is used to "fair" one to the other. Use of this curve in the example yields a wall thickness of 0.18"; by working without the curve the designer came out with a cylinder wall 20% too thin.

Fig. 8 is extremely easy to apply. Admittedly it is neither perfect nor general, but it is useful for at least 90% of all problems involving the wall thickness of a pressure-containing member. General formulas are needed for the few general case problems, but the designer needs something simple for the usual special case.

The writer obviously does not have the academic viewpoint. Rather, he has the viewpoint of the engineer who starts out after six years of work in the best technical college in the country with high ambition and rosy dreams of a world run by theoretically correct solutions only to find that engineering processes do not necessarily conform to theory. The useful solution is the one which is economically most feasible, and economics is not restricted to power consumption, but also encompasses interest on the entire investment and maintenance charges.

Actually, the writer is in full sympathy with and wishes to promote the well being of the academician, because he is the man on whom the future depends. Maybe it will be of help to point out that if the research man will take thought of society as a group of individuals with different backgrounds, different aptitudes, and different limitations, he may lead faster and surer and thus enhance his value to the universe.

