Economy in Rod String Design from Simple Graphs

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In today's present market one must consider savings wherever possible. In past practices, the balanced rod string has been most widely accepted. We should like to offer in this paper a simple graphical method of retaining all the advantages of a balanced rod string design and still obtain maximum economy through full utilization of allowed rod stresses. These graphs will give the maximum footage of minimum sized rods and result in the most economical design within the allowed working stresses. This economy will be accomplished without fixing or dropping any of the primary variables used in designing sucker rod strings.

DESIGN VARIABLES

Primary Variables

In sucker rod design there are six primary variables: depth, plunger size, stroke length, strokes per minute, rod size, and allowed rod stress. Each of these variables, if changed, will cause changes in one or more of the others. And at this point it should be worthwhile to show how each variable affects the pumping system when all others are held constant.

First a straight string of 5/8 in. carbon manganese rods at 4500 ft and use 25,000 psi as a safe working stress should be considered. It is assumed that a 54 in. surface stroke pumping unit is operating at 14 strokes per minute and an inch and one quarter plunger. This assumption gives a fluid load of 2650 lbs and a rod load of 5100 lbs for a total peak load of 7750 lbs which imparts a 25,300 psi stress in the rods.

This rod string is now operating at its optimum, but to show the effect of merely speeding up the unit to 18 strokes per minute, the rod load is increased to 5600 lbs, the peak load to 8250 lbs and the rod stress to 27,000 psi which is beyond the working stress for this rod. This error is one of the most common made.

A much more critical type error is to increase plunger size. In this case an inch and one half plunger is used, while all other conditions remain the same. The fluid load then increases to 3450 lbs the peak load to 8600 lbs, and the rod stress to 28,200 psi. It is interesting to note that increasing speed or stroke length alone can be as critical as is an increase in plunger size. And without further examples, one can readily see that depth also is critical.

Here it should be emphasized that whenever any of the variables — depth, stroke length, speed, or plunger size — is to be changed, its effect should be considered prior to the causing of costly errors. The design charts take into consideration all these variables and can be used to check any or all and determine their effects on the system.

Another error is rod stress which is a result of the other variables and is dependent on changing only by changing the other five primary variables. It should actually be classified with the secondary or dependent variables but for convenience it is grouped here for it is the basis for which rod strings are designed.

Secondary Variables

The secondary variables are stretch, overtravel, and load range. These cannot be changed independently but are again only results of the primary variables.

These will not be considered in the basic rod design. Although all rod designs are for lifting fluid and these must be considered, a basic design must first be made from which a final acceptable design including stretch, overtravel, and load range is considered.

By using a practical allowed working stress, these secondary variables can be fairly well controlled. For instance, if a grade of sucker rod is recommended for maximum stress up to 33,000 psi, a general good rule under normal well conditions (if there is such a possibility) would be the design of an allowed working stress of, perhaps, 25,000 to 28,000 psi. By such a design, load range and stretch will be held to within good practices and not become dominant factors.

ALLOWED ROD STRESSES

All manufacturers advertise a recommended maximum allowed stress for each type of rod. These recommendations are for easy pumping conditions with the understanding that, when other than ideal conditions (including

ROD STRESS DUE TO OWN WEIGHT
FOR ANY UNIFORM ROD STRING

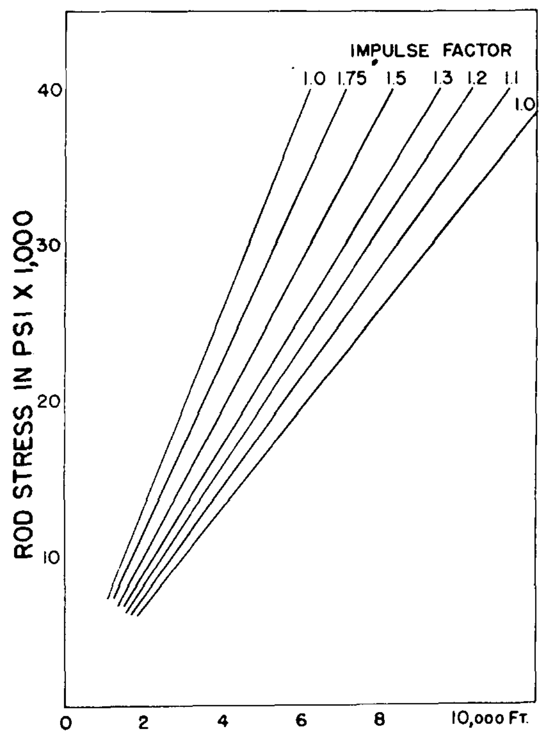

FIG. 1

no corrosion) exist, corresponding reductions in load should be considered.

Table I is a general classification of sucker rods with manufacturers' maximum recommended loads and generally accepted stresses or allowed stresses for average conditions.

TABLE I

General Classification of Sucker Rods	The average Maximum Recommended Working Stress	The Average Recommended Allowed Stress for Average Well Conditions (No Corrosion)
Carbon Manganese	30,000 psi	25,000 psi
Medium Alloy	35,000 psi	30,000 psi
Hi-Tensile	40,000 ± psi	35,000 + psi
*Special	35,000-40,000 ± psi	30,000-35,000 ± psi

*The special rods are generally in the hi-tensile category with certain restrictions on speed, load range, corrosion, etc.

Table II is a comparison of increased cost for increased allowed stresses of sucker rods. Using Carbon-Manganese rods as 100 per cent, the additional cost for additional allowed stress pretty much stays in line. The difference in cost of 135 per cent of alloy over carbon to an increase in allowed stress of only 120 per cent is offset somewhat by the added advantages of alloys for corrosion resistance and other benefits which are not always directly measurable.

TABLE II

General Classification of Sucker Rods	Average Per Cent Increase in Allowed Stress	Average Per Cent Increase in Cost
Carbon Manganese	100	100
Medium Alloy	120	135
Hi-Tensile	140	140
Special	125-140	120-125

Much controversy exists on the advantages of medium alloy rods over carbon, but over the years purchases of alloy rods have continually increased percentagewise, the increase indicating that the users over-all feel that these benefits of additional stress and corrosion characteristics are worthwhile. However, the suggestion has many times been presented that cost savings could be realized in using smaller high-tensile rods instead of larger carbon or alloy rods.

Table II points out that the added load advantages are equalized by the added cost. Even though beam loads may be decreased, increased surface unit sizes for stroke length to offset increased rod stretch and an increase in gear size to offset increase in torque would tend to offset any savings. These factors generally limit this type of installation.

COMBINING VARIABLES

In the past, sucker rod design charts have had to hold either one or more variable so that a chart simple enough for practical usage could be designed. In designing the graphs for this paper some of the variables will be combined but will still maintain their total effectiveness. Fortunately, both speed and stroke length affect the rod weight by modifying the acceleration on the up and down stroke. Without losing the related effect of either one can combine these two factors and consider their influence on rod weight because of acceleration.

ROD STRESS DUE TO FLUID LOAD ONLY FOR ANY UNIFORM ROD STRING

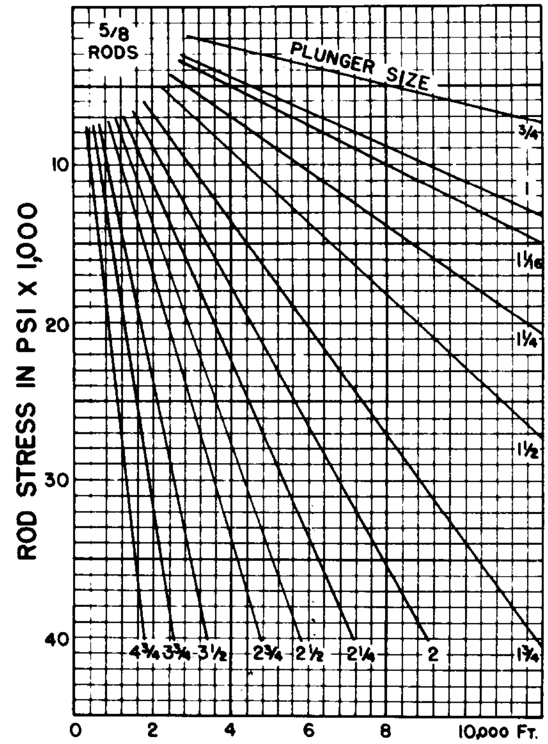

FIG. 2

One other variable has an interesting relationship that one can also take advantage of in simplifying the graph. That variable is that any rod, regardless of size and when suspended to the same depth, will stress its top fibre the same. As an example, if one would multiply the area of any rod by an equal stress, say 30,000 psi, and divide it by its buoyant weight he would find that any or each rod would have a length of 9300 ft + 100 ft or so. This slight variation is due to the upsets and couplings. One would further find some variation if he took 25 ft rod lengths and 30 ft rod lengths. But, for all practical purposes, any rod suspended by its own weight to the same depth will impart exactly the same stress at its top member. Further, if any such string of rods, freely suspended, were accelerated by equal stroke and speed one would find that its stress on the top member would increase exactly the same in each size rod. Therefore, the first graph can now be drawn, and Fig. 1 shows the stress at any depth for any uniform rod string for different impulse factors.

A second series of graphs is needed to show the weight per foot of fluid for each size plunger plotted against stress for that size rod. For instance, using again 30,000 psi x the area of a 3/4 in rod (.442 sq in) equals 13,260 lbs. If we then divide that 13,260 lbs by the weight per foot of fluid of a 1 3/4 in. plunger (1.040 lbs. per ft) it will equal 12,800 ft. This is the fluid load equal to 12,800 ft of a 1 3/4 in. plunger which would impart a stress equal to 30,000 psi in a 3/4 in. rod. Now the second graph (Fig. 2) can be drawn, and it will include all plunger sizes and note that the 1 3/4 in. plunger crosses the 30,000 psi line at 12,800 ft. This graph will only be for 3/4 in. rods.

Now a third series of graphs are needed. Fortunately, one can again simplify the graphs because the difference in loads for any given stress remains constant between

ROD STRESS & ROD LENGTH FOR UPPER SECTION OF TAPERED STRINGS

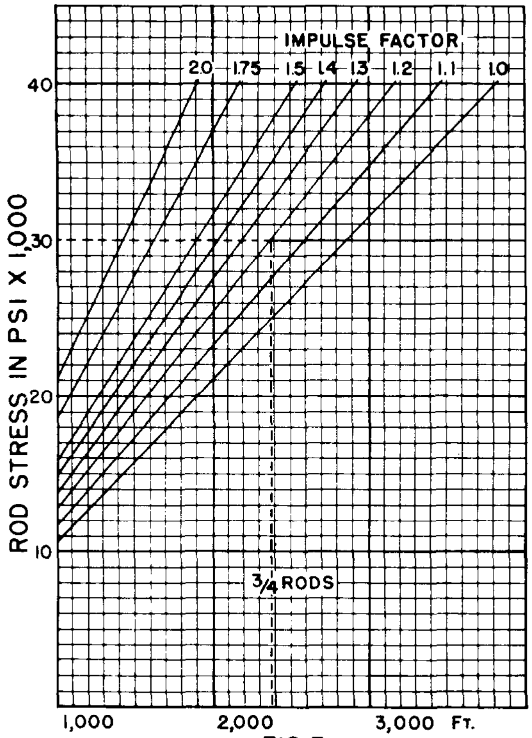

FIG. 3

any two rod sizes.

30,000 PSI x .442 (3/4-in. rod) = 13,260 lbs.

30,000 PSI x .307 (5/8-in. rod) = 9,210 lbs.

4,050 lbs.

Or, any string of rods using 5/8 in. as the lower section and 3/4 in. above, designed at 30,000 PSI will always have 4,050 lbs of 3/4 in. rods. One can convert the 4,050 lbs. to feet and include impulse factor — this computation gives us our third series of graphs (Fig. 3).

This graph shows for any stress the footage of 3/4 in. rods that can be used as an upper section only. Now the three graphs that have been developed can be used to design a rod string. The rod design for a 1 1/4-in. pump at 6,000 ft. can be found, using 5/8 in. rods as the bottom section. The surface unit will be a 54-in. stroke running at 16 strokes per minute and giving the desired production.

First, stroke and strokes per minute are combined into the impulse factor, which is 1.20. Further, a medium alloy type rod with an allowed wroking stress of 30,000 psi from Table II will be used.

Fig. 1 should be set over Fig. 2 so that the base line of Fig. 1 is on the 30,000 psi line (see Fig. 4). One should read directly up the 6,000 ft. line to the intersection of the 1 1/4 in. plunger line, then read across to the left to the intersection of 1.20 impulse factor, then straight down to find 5050 ft of 5/8 in. rods. What one has found is that with a 1 1/4 in. plunger at 6,000 ft and a 54 in. surface stroke at 16 strokes per minute, he can run 5100 ft of 5/8 in. rods and that the top rod will have a 30,000 psi stress. Actual calculation shows 5050 ft. These graphs are consistently accurate within one or two rods or $\pm$ 100 psi stress.

One next goes to Fig. 3 and reads directly below the intersection of the 30,000 psi line and the 1.20 impulse factor for 2375 ft of 3/4 in. rods (actual calculation is 2350 ft). Adding together these figures, one obtains 7475 ft. But only 900 ft of 3/4 in. rods would be used and thus give the most economical string for the allowed rod stress.

These graphs can also be used to find maximum depth to which a pump can be lowered, the maximum plunger size at the depth being pumped, etc.

For example: What depth can a string of 5/8 in. rods at 4000 ft be safely lowered with a 1 1/4 in. plunger, operating at fourteen 54 in. strokes per minute and a maximum allowed rod stress of 25,000 psi?

To find the answer one must set the base line of Fig. 1 on the 25,000 psi line of Fig. 6 for 5/8 in. rods and read directly up the 1 1/4 in. plunger line to its intersection with the 1.15 impulse factor line (impulse factor for 14-54 in. strokes per minute) and then straight down to 4600 ft.

What is the maximum size plunger at 4000 ft at which this same system may operate safely? With the base line on the 25,000 psi line, one must read directly up the 4000 ft line to its intersection with the 1.15 impulse factor. This point lies half way between the 1 1/2 in. and 1 3/4 in. plunger sizes. Therefore, the 1 1/2 in. plunger should be used. If one desires to use a 1 3/4 in. plunger, then he must move the slide down till the 1 3/4 in. plunger intersects this same point. He will then have found that the stress would be 28,700 psi and that these rods would not be safe. However by moving the slide up till the 1 1/2 in. plunger line, 4,000 ft line, and 1.15

DETERMINING LENGTH OF BOTTOM SECTION OF TAPERED ROD STRING

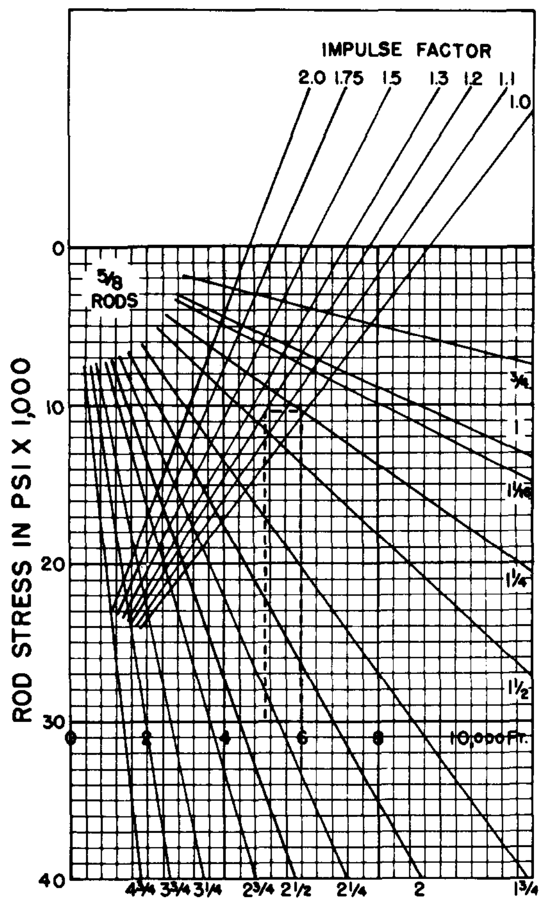

FIG. 4

CHARTS FOR STRIGHT STRINGS AND
BOTTOM SECTION OF TAPERED STRINGS ONLY

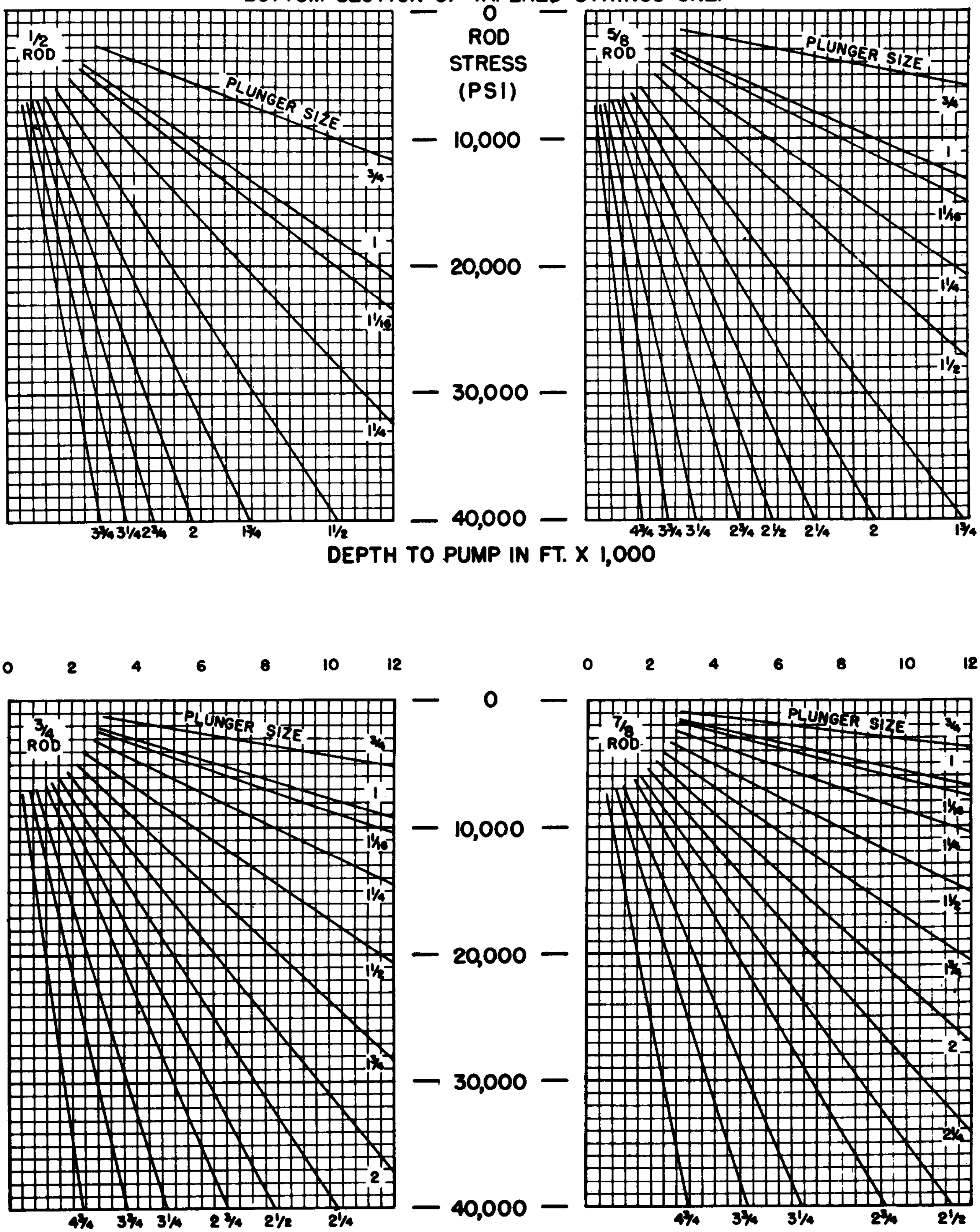

FIG. 5

UPPER ROD SECTIONS FOR TAPERED STRINGS ONLY

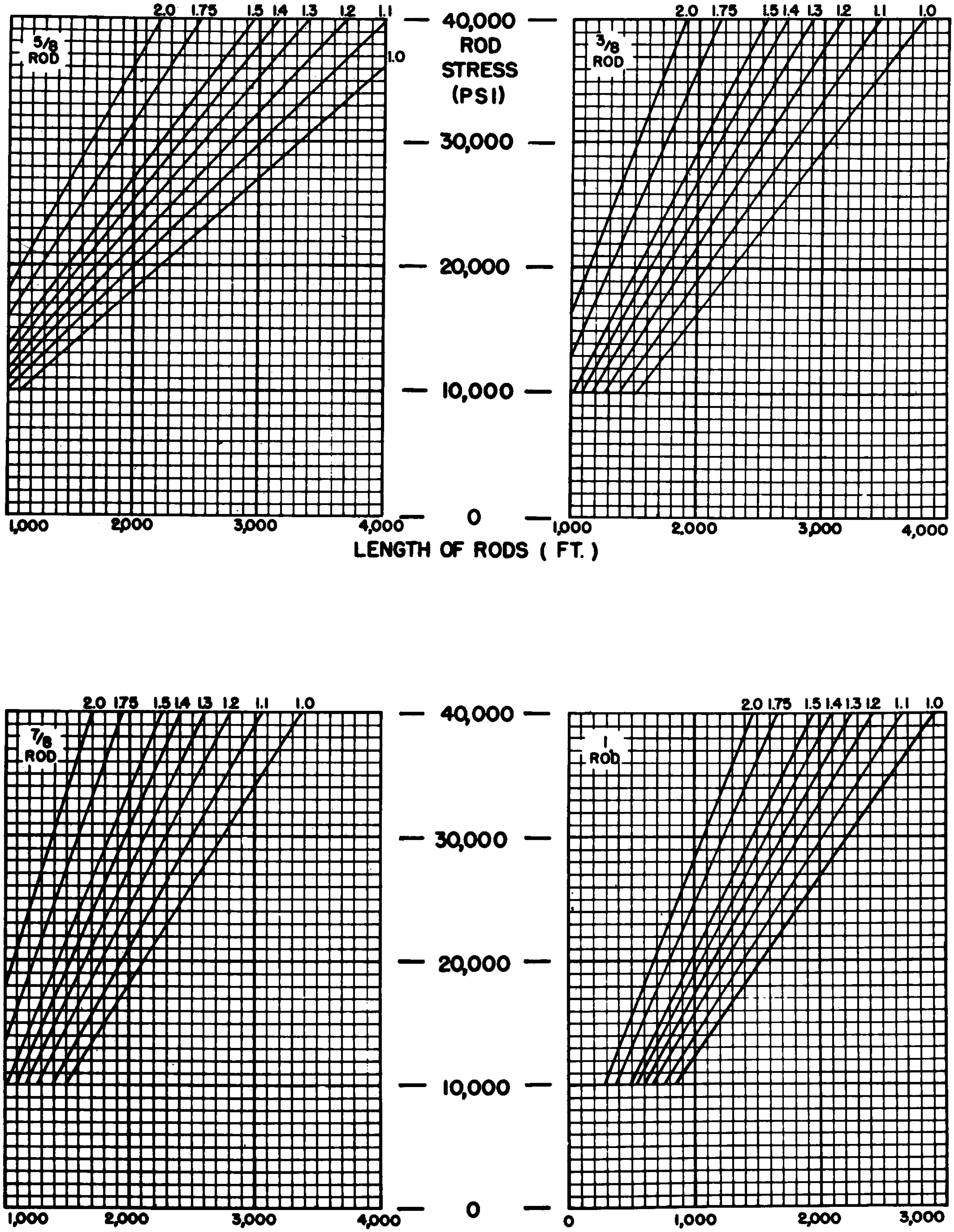

FIG. 6

impulse factor line intersect, one would find that the actual stress for the safe condition is only 23,800 psi and satisfactory service life should be attained.

A balanced design for tapered strings can also be found readily. In this instance one assumes a 7600 ft. well with a 1 1/2 in. plunger operating at twelve 120 in. strokes per minute and desires to use medium alloy rods of 30,000 PSI allowed stress.

The base line of Fig. 1 should be set over the 3/4 in. rod graph on Fig. 5 at 30,000 psi. Next, one should read up the 7600 ft line to the 1 1/2 in. plunger and then across to the 1.20 impulse factor and on straight down to 4500 ft of 3/4 in. rods. One reads then from Fig. 6 at 30,000 psi 2300 ft of 7/8 in. and 1950 ft of 1 in. for 8650 ft. The same procedure should be repeated for 27,500 psi and 3750 ft. of 3/4 in., 2175 ft of 7/8 in., and 1675 ft of 1 in. for 7600 ft should be found. This string is balanced at 27,500 psi rod stress in the top of each rod section.

If the most economical design were desired then one should use the footages found at 30,000 psi of 4500 ft of 3/4 in., 2300 ft of 7/8 in. and should use only 800 ft of 1 in. instead of 1850 ft. At present prices, the balanced rod string would cost approximately $5,640.00, and the most economical design on 30,000 psi would cost $5,310.-00 for a dollar savings of $330.00 for almost 6 per cent reduction in cost.

CONCLUSION

An over-all savings in rod strings can be obtained by utilizing the stress of the rods purchased up to their safe working loads for the particular well conditions.

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