New Data for Calculating Lowest Annual V-Belt Drive Cost

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INTRODUCTION

Each year more and more wells go on the pump. *The Oil and Gas Journal* predicts that in 1957 over 29,000 wells will be placed on artificial lift. In addition, this publication comments on a trend of a 7% increase in the number of wells put on the pump each year.

The decision of whether or not to place a particular well on the pump is based entirely on the economics involved. This evaluation includes estimates of the expected production, the cost of the equipment needed, and the expected annual maintenance costs.

Many of the components on a pumping unit have been studied and annual costs have been estimated, based on loads, bearing size, gear-face width, and so on. One component which has previously defied attempts at prediction of annual costs has been the V-belt drive.

This condition has occured not only because of the complex nature of belt wear but also because of previously existing methods of designing V-belt drives from standard horsepower ratings. If one designs so that his actual drive is more conservative than the standards would recommend, then he knows that his drive will be more than adequate. But he does not know how much more than adequate it will be. The design engineer, because of the lack of information, has been forced to accept a partial answer in many situations where a more complete answer is needed.

We can suppose, for example, that a drive checks out to require 3.5 belts. Will the additional service given by going to a four-belt drive offset the lower cost of dropping back to a three-belt drive?

Or the engineer may have a drive on which the driver sheave must be changed. Will this change affect the expected service life of the drive? If so, how much? Would it be advisable at this time to go to premium-quality V-belts instead of standard-quality V-belts?

Or perhaps a number of possible pumping units are being considered, each using a different V-belt drive. Which drive is best? How much longer service could reasonably be expected from the best drive?

Any of these problems illustrates the fact that a horsepower rating for a drive is only a partial guide. The true measure of a V-belt drive is not its horsepower rating; it is the service which the drive will give out on the lease.

Until the present time, no data have been published which could be used to predict the relative service of a V-belt drive. With recognition of this problem, the factors affecting V-belt service and horsepower capacity have been studied to provide a quantitative measure of their inter-relation. It is now possible to analyze any V-belt drive and to predict its relative service based on the drive conditions and the load that the belts will experience.

The results of these studies have been compiled into a series of design manuals. These manuals are readily available, but no attempt will be made here to explain how to use them. Instead, the engineering fundamentals behind these manuals will be discussed and some of the ways that they can be used will be reviewed.

ENGINEERING BACKGROUND

When the study of V-belt drive characteristics was undertaken, the qualitative factors which affect belt life were known. For example, all other factors being equal, belt life is shortened when higher horsepower is transmitted. The use of smaller-diameter sheaves also shortens the life of a belt; and both belt length and belt speed enter into the computation of the life and the horsepower rating of a belt. At this point, a major objective was to develop an engineering approach to horsepower ratings for V-belts.

To solve this problem laboratory facilities were needed and tests were carried out on many types of equipment.

One of the simplest types of testing equipment is a deadweight tester in which the belt is suspended from a driving sheave with weights supported by the driven sheave (Fig. 1). The belt is then run under the tension developed by the weights until it fails, transmitting no horsepower. This type of test is often used for screening purposes when comparisons are made of various constructions, materials, and so on. However, the results are such that they require interpretation by experienced personnel and are not as precise as results from dynamometer testing are.

One type of dynamometer is a water brake shown in Fig. 2. In this type of dynamometer the belt drives a perforated disc in a chamber in which a flow of water is maintained at constant, but adjustable, level. The resistance to the rotation of the disc is then calibrated in terms of horsepower load. On a test of this type, studies can be made on the effect of drive geometry, belt speed, unequal diameters, and so on. In addition, tests can be made involving two or more loaded sheaves.

In another type of dynamometer (Fig. 3) the V-belt being tested couples an electric motor to a generator suspended in a cradle. The horsepower output and input can be measured very accurately with equipment of this type. With this equipment the effects of shock loads, fluctuating loads, diameters, tension, speed, and so on can be studied under very closely controlled conditions.

The dynamometer shown in Fig. 3 is equipped with an electronic gear which enables rather severe shock loading to be applied. The oscillograph of Fig. 4 shows a typical shock load cycle, running from a minimum of 4.8 horsepower to a maximum of 28.4 horsepower in approximately one-half second. Speed is maintained constant within 0.5% throughout the cycle.

With the results from a number of test programs, it was possible to analyze the effects of the various factors which control performance. To illustrate what happens to a belt on a drive, we shall consider a typical drive, Fig. 5. The 1160-rpm motor drives a driven machine with a C-195 premium-quality V-belt on 12-in.-diameter sheaves. As we see the drive in Fig. 5, the machines are in motion but carrying no load. The belt tension is 105 pounds in each strand, giving a total tension of 210 pounds in the belts. Because of the motor speed and sheave sizes, the belt speed is 3650 feet per minute.

In Fig. 6 we see the situation after we have applied a load to the driven machine until there is a tension of 175 pounds on the tight side of the belt. The total belt tension is still 210 pounds, leaving 35 pounds on the slack side of the drive. The ratio of the tight-side tension to the slack-side tension is now 5 to 1, which is the value normally used for a V-belt-drive design.

The horsepower transmitted to the driven machine depends

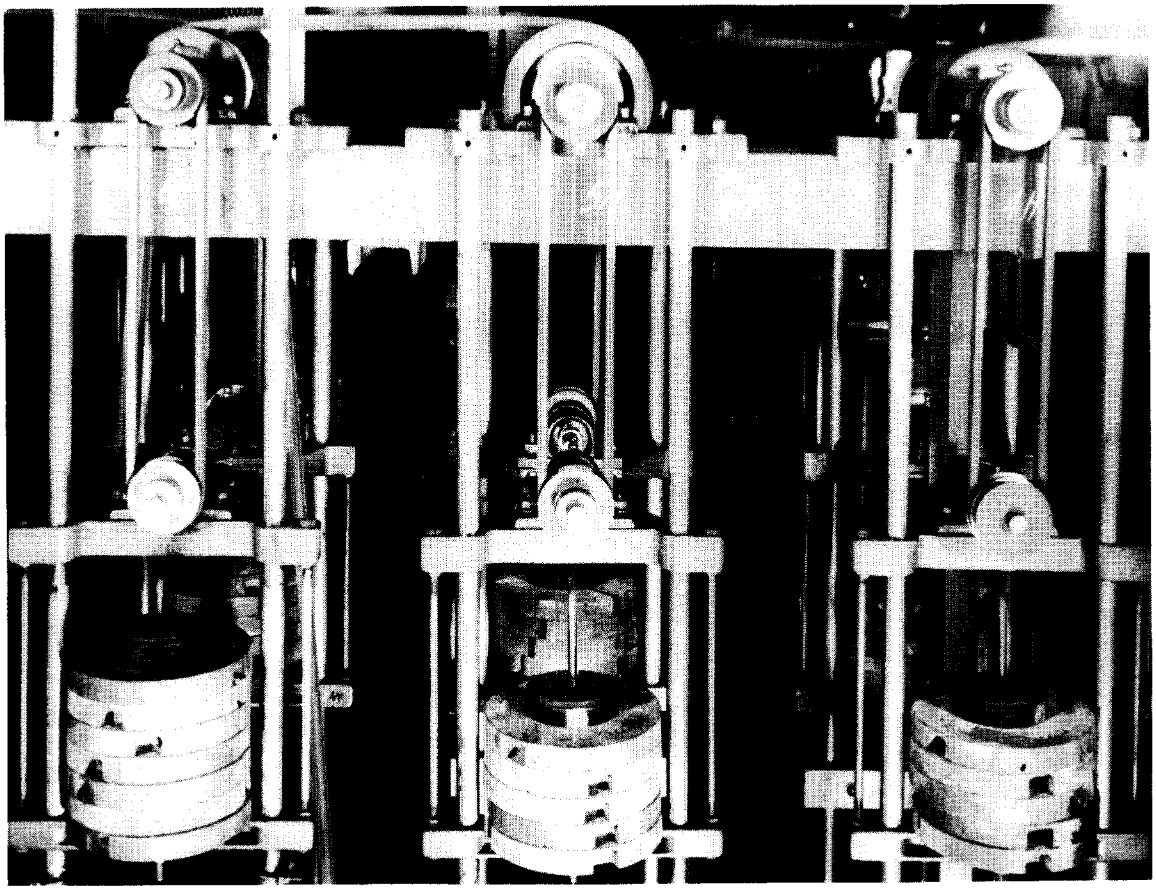


Figure 1

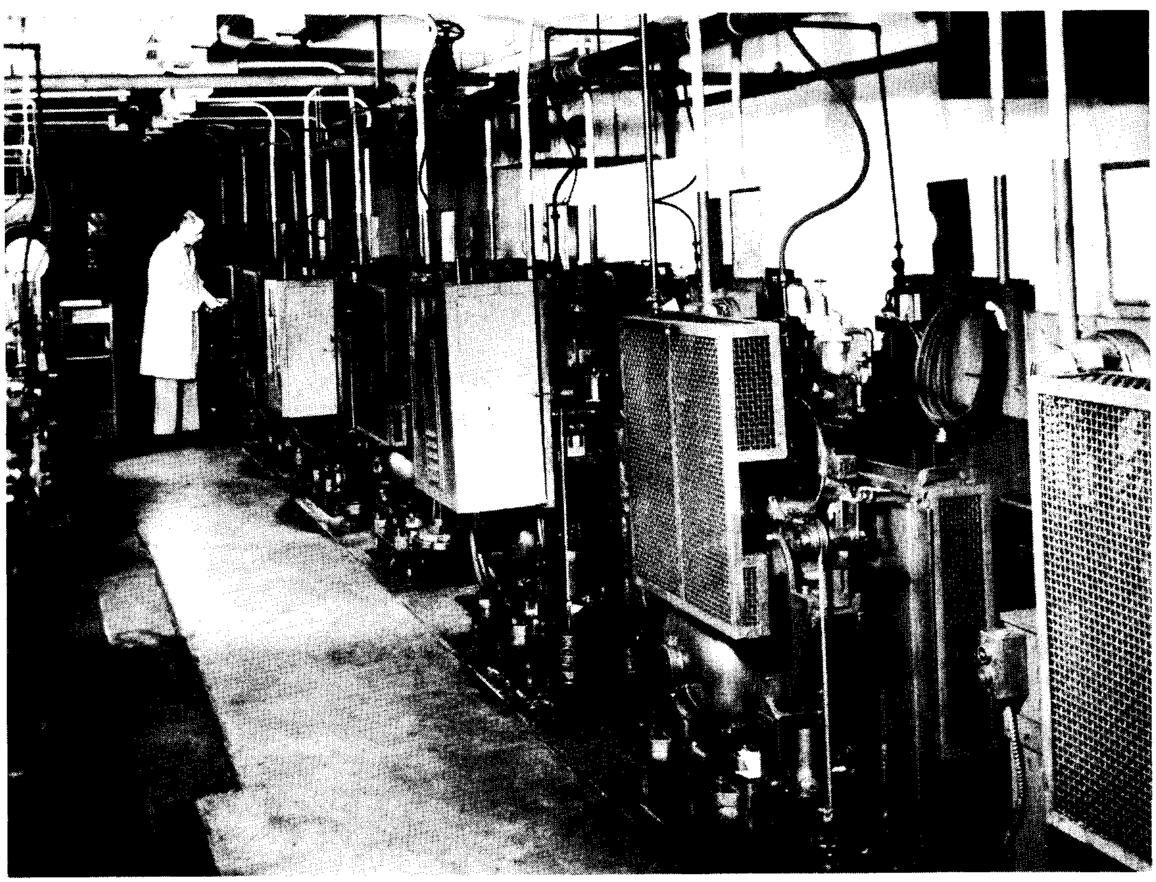


Figure 2

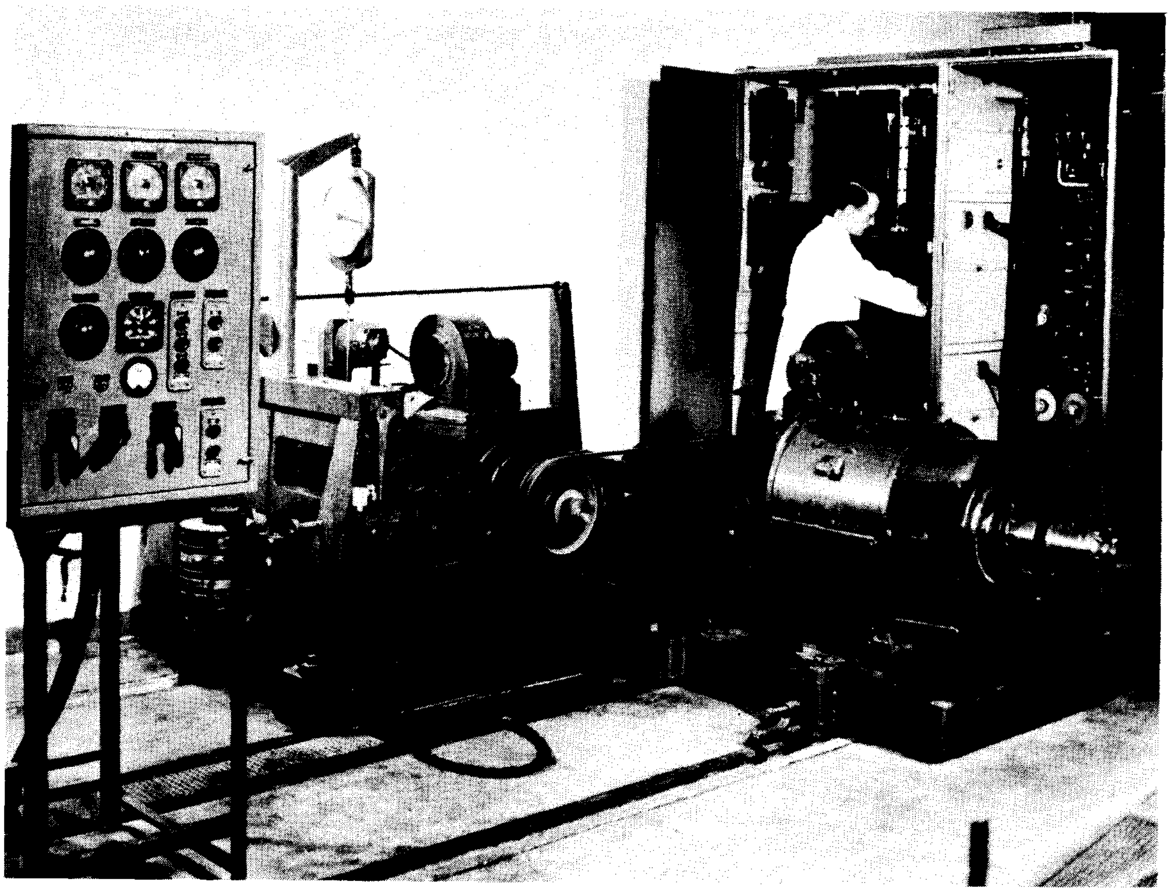


Figure 3

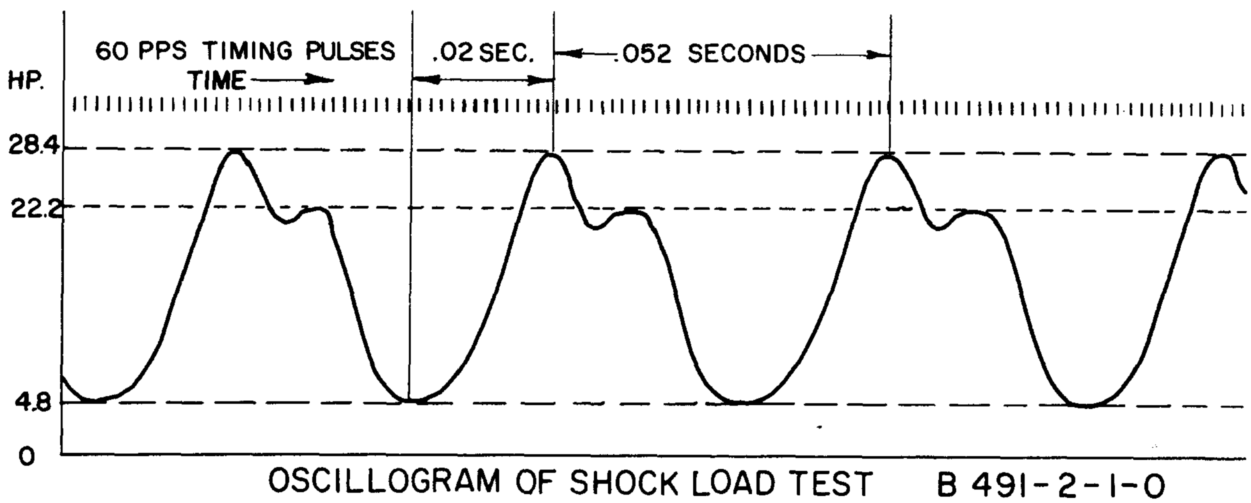
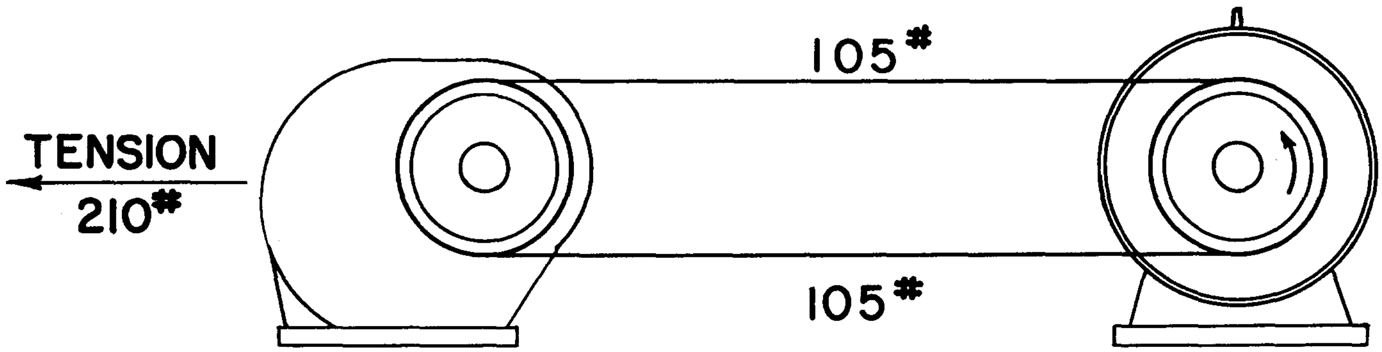


Figure 4



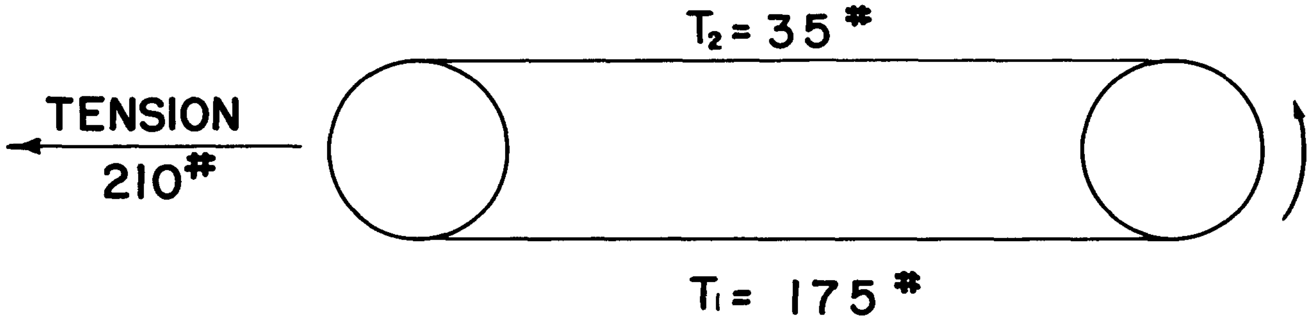
MOTOR 1160 RPM

SHEAVES 12"PD

BELT C195S (P.L.= 197.9")

BELT SPEED 3650'/MIN.

Figure 5



TENSION RATIO = $T_1/T_2 = 5$

$T_1 = 5T_2 = 5 \times 35 = 175$#

NET PULL = $T_1 - T_2 = 175 - 35 = 140$#

HORSEPOWER = $\frac{140 \times 3650}{33{,}000} = 15.4$ HP

Figure 6

on the difference between the tight- and the slack-side tensions. In this case, the difference is a net pull of 140 pounds. Using this value and the belt speed of 3650 feet per minute, we see that the belt is transmitting 15.4 horsepower.

To see what actually happens to the belt in traveling around the drive, we can follow a section as it makes one revolution (Fig. 7). The drive is shown at the top with various positions lettered. At the bottom of the figure is a diagram of the pattern of the stresses set up in the cords above the pitch line. Because the belt is moving at 3650 feet per minute, there is a centrifugal force set up which tends to throw the belt away from the sheaves. This force is resisted with an equal but opposite force in strands BE and CD. In the example, this centrifugal force equals ?5 pounds.

Besides the centrifugal stress, at point A the belt is also subjected to the slack-side stress of 35 pounds. The belt experiences the sum of these two stresses until it reaches point B. At B the belt bends onto the sheave and develops an additional tensile stress in the cords above the pitch line. This bending stress is inversely proportional to the diameter. It is represented in Fig. 7 by the vertical stress, T_B. In our example, T_B is equal to 130 pounds.

In going from B to C on the sheave, the belt goes from the slack side to the tight side of the drive. At point C, the stress in the belt reaches a peak which is made up of three components: the bending stress of 130 pounds, the tight-side stress of 175 pounds, and the centrifugal stress of 25 pounds. The total peak stress is 330 pounds. Leaving point C, the bending stress is relieved, and the belt travels from C to D under a tension equal to the sum of the tight side plus the centrifugal stress. At point D, the belt again bends onto the sheave and experiences a second peak stress equal in size to the one at point C. In going from D to A, the belt reverses the stress cycle experienced in going from A to C.

The peak stress that the belt actually sees (330 pounds) is considerably more than the tight-side stress of 175 pounds or the net pull of 140 pounds, although these are the two values which can be directly related to the horsepower transmitted.

Fig. 7 shows the stresses in the tensile section. But other components of the belt also undergo cyclic stress patterns as the belt travels around the drive. Some of these are shown in Fig. 8. For example, the bottom of the belt experiences compression instead of tension when the belt is bent onto the sheave. At the pitch line no stress due to bending occurs, because the pitch line is defined as that plane in the belt which is unchanged in length as the belt bends.

Stress patterns similar to those of Fig. 8 could be developed for each of the other components of the belt. But they would only serve to emphasize two important points: (1) Each portion of the belt is subjected to a cyclic stress pattern as it travels around the drive and (2) each portion experiences a peak stress once each time that it travels around the sheave.

We have all seen a piece of baling wire break after it was bent back and forth at the same place. We know that the wire failed in fatigue because of repeated stressing below its ultimate strength. In a similar manner, the components of the V-belt fail in fatigue when subjected to repeated cyclic stress patterns.

From a large number of tests, the relation between the average number of cycles at failure to the value of peak stress has been determined. This relationship may be plotted as a "fatigue" curve. The curve shown in Fig. 9 represents a Gates "C" section Super Vulco Rope.

On this diagram, the peak stress of the example drive results in an average of 663 million cycles before failure. The shape of this curve is similar to that of the "S-N" fatigue curve for steel. It differs in that an "S-N" curve for most metals has a "knee" at a certain value of peak stress. If the metal is stressed continuously below this critical value, it will not fail in fatigue; but above this value, it will fail at a predictable number of cycles. No such knee has been found in the fatigue curves for V-belts.

A curve such as that in Fig. 9 allows a prediction of the service life of any drive similar to the one used in the example. From the characteristics of the drive, the tight side (T_1), the bending (T_B), and the centrifugal (T_C) stresses can be computed. From these, the total peak stress ($T_1 + T_B + T_C$) can be derived. This value, when used with the fatigue curve, gives the average number of cycles before belt failure. The number of cycles, obviously, is related to the length of the belt, to the belt speed, to the number of sheaves, and to the average service given by a V-belt before it fails.

While the peak stress-fatigue life concept is basically quite simple, a number of complicating situations arise on nearly every practical drive. These make the application of the stress-fatigue idea complex. We have been forced to study ways to reduce the complicating factors to equivalent simple factors.

As an example, the stress pattern of Fig. 9 assumes a constant horsepower load. But on a pumping unit it is well known that the load is far from steady. Fig. 10 shows the load diagram for one stroke of a pumping unit. The fluctuation of the load is shown as following a sine curve. It is recognized that the actual load seldom follows a smooth curve such as this. However, if a sine curve with a maximum value equivalent to the peak torque derived from a dynamometer card is used, the work area under the sine curve will equal the work done on a perfectly balanced well at the same peak torque and will be somewhat greater than the actual work if the well is unbalanced. Therefore, when a sine curve based on actual peak load conditions is used for analysis, the results are on the conservative side.

Under the fluctuating load condition shown in Fig. 10, the stress pattern experienced by the belt also fluctuates. Fig. 11 shows the stress pattern for the example drive of Figs. 5 and 6. We notice that even when no load is transmitted, the belt still is subjected to centrifugal stress, bending stress; and the 105 pounds no-load tension. If the number of cycles to failure of the belt is computed from the fatigue curve based on the peak stress at peak load conditions, the number will be less than the actual number of cycles of the belt on the given drive. Similarly, the number of cycles predicted from the peak stress under no-load condition would be greater than that of the actual drive.

To relate the actual belt service to the fluctuating load pattern, a method is needed for arriving at an equivalent steady load which will give the same life as that of the actual fluctuating load. To do this, a multiplier called the service factor is used with either the average horsepower or the peak horsepower. This will result in steady horsepower, known as the design horsepower, which will be equivalent, from a fatigue life standpoint, to the actual fluctuating horsepower. The design horsepower will be greater than the average horsepower, but less than the peak horsepower. Fig. 12 shows two methods of arriving at the design horsepower for an API 57 reducer. We note that either method gives the same results. API Standard 1 calls for the use of the second method, that of multiplying the horsepower corresponding to the peak torque rating of the reducer of 0.89.

As another example of complications experienced in applying the simple stress-fatigue concept to actual drives, we can consider the fact that on a typical pumping unit drive the driving sheave is considerably smaller -- and the bending stress correspondingly higher -- than it is on the reducer sheave. This means that the peak stresses are no longer equal (as they are in Fig. 9 for the example drive). Thus finding a way to determine a pair of equal-diameter sheaves which would be equivalent, from a fatigue life standpoint, to the unequal diameters of the actual sheaves has become necessary. The diameter of these equivalent sheaves is known as the index diameter.

With the understanding that "horsepower" refers to design horsepower and that "diameter" refers to index diameter, we can then say that the stress-fatigue relationships are as shown in Fig. 13. We note that the peak stress experienced by

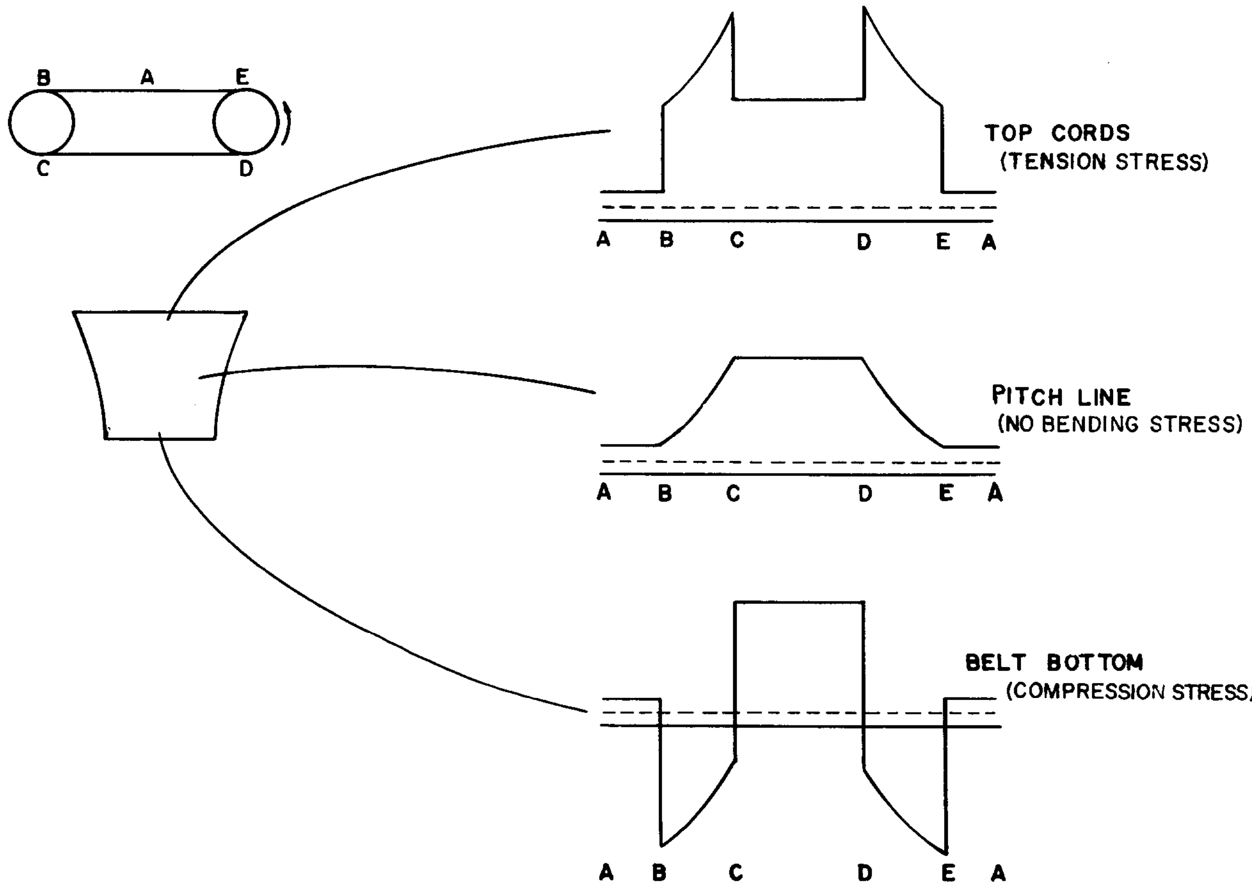


Figure 7

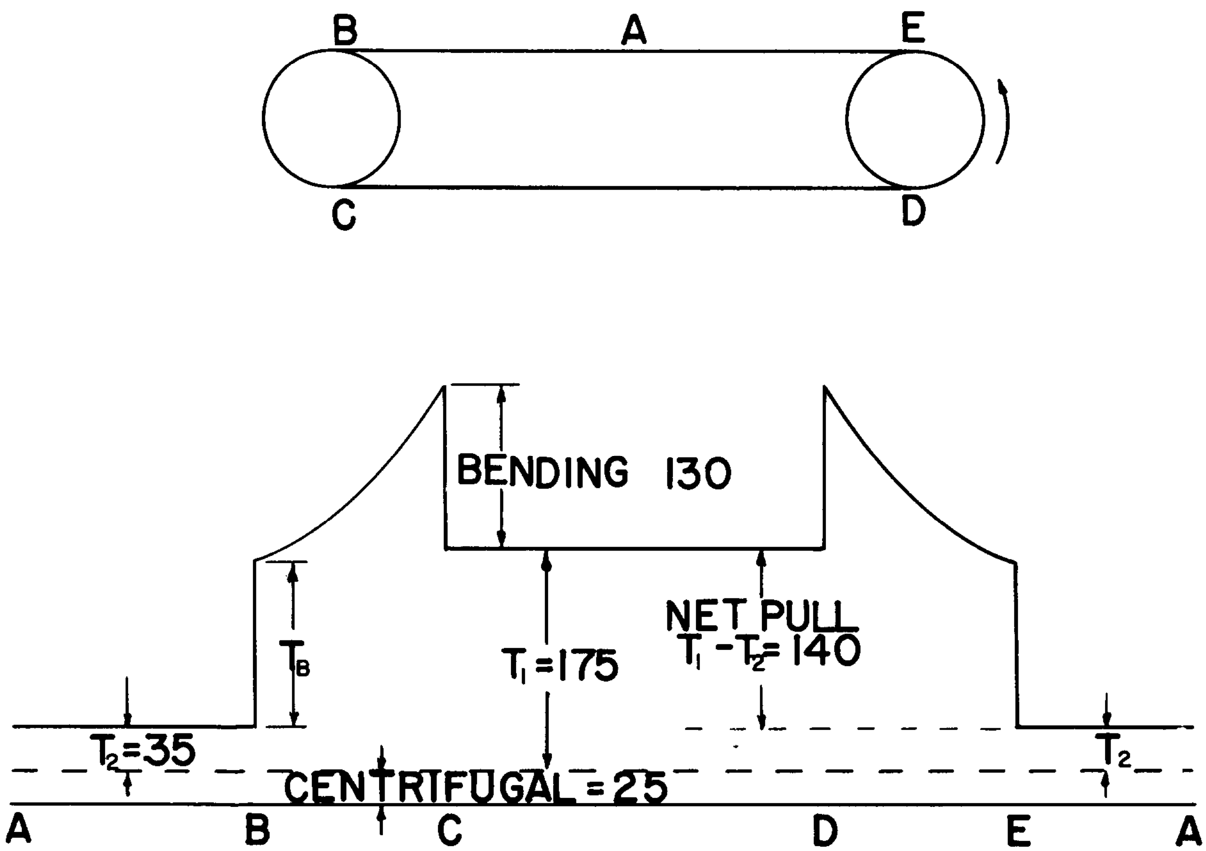


Figure 8

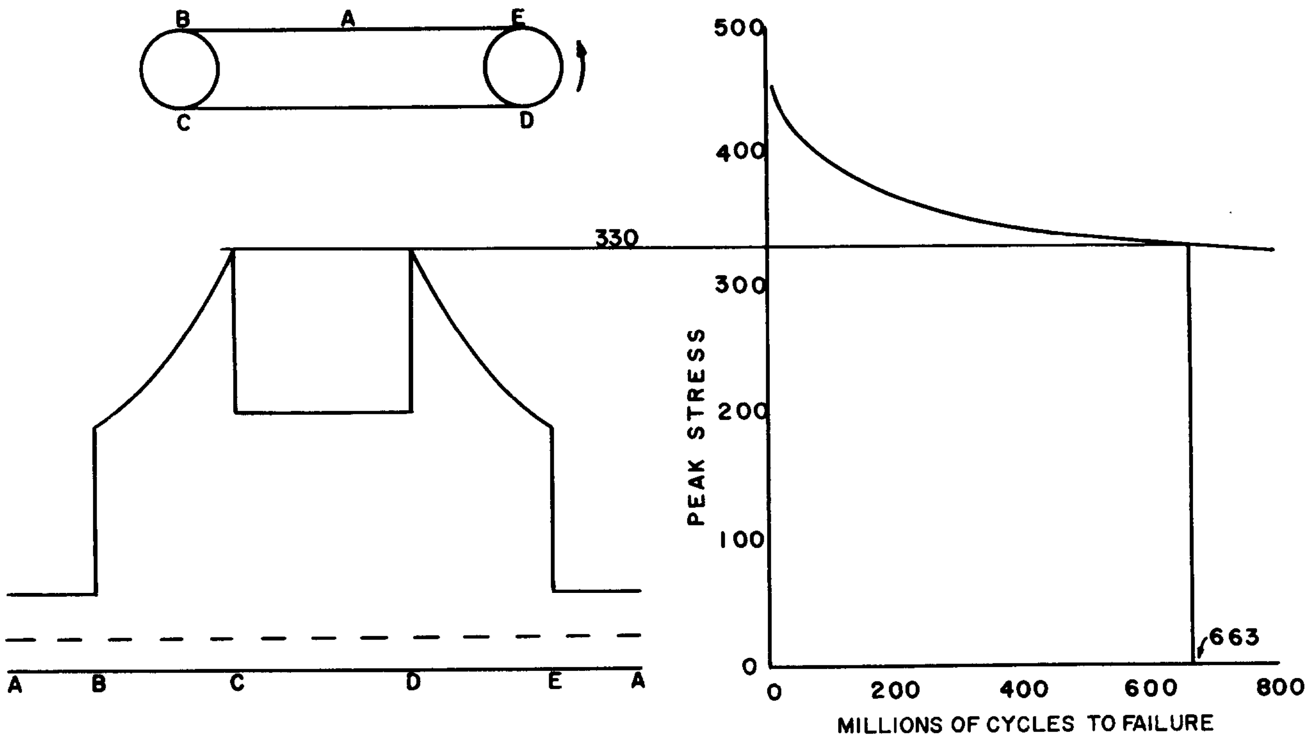


Figure 9

LOAD DIAGRAM-ONE STROKE OF PUMPING UNIT-

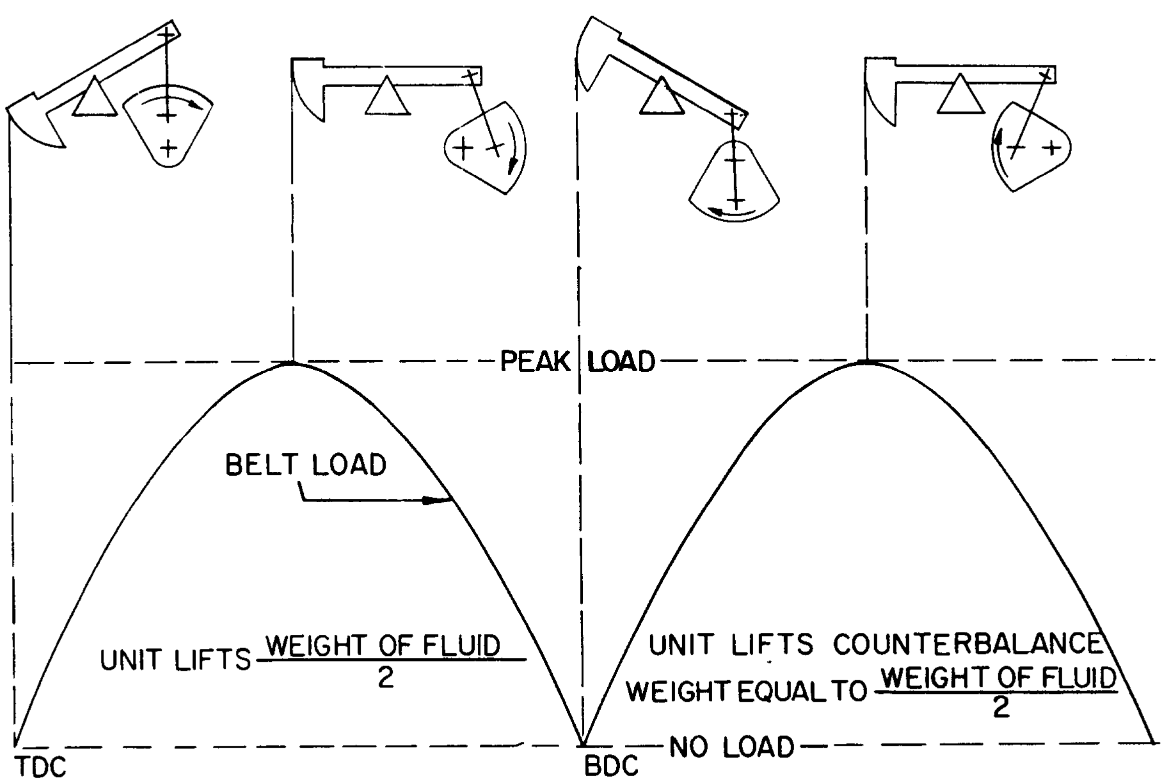


Figure 10

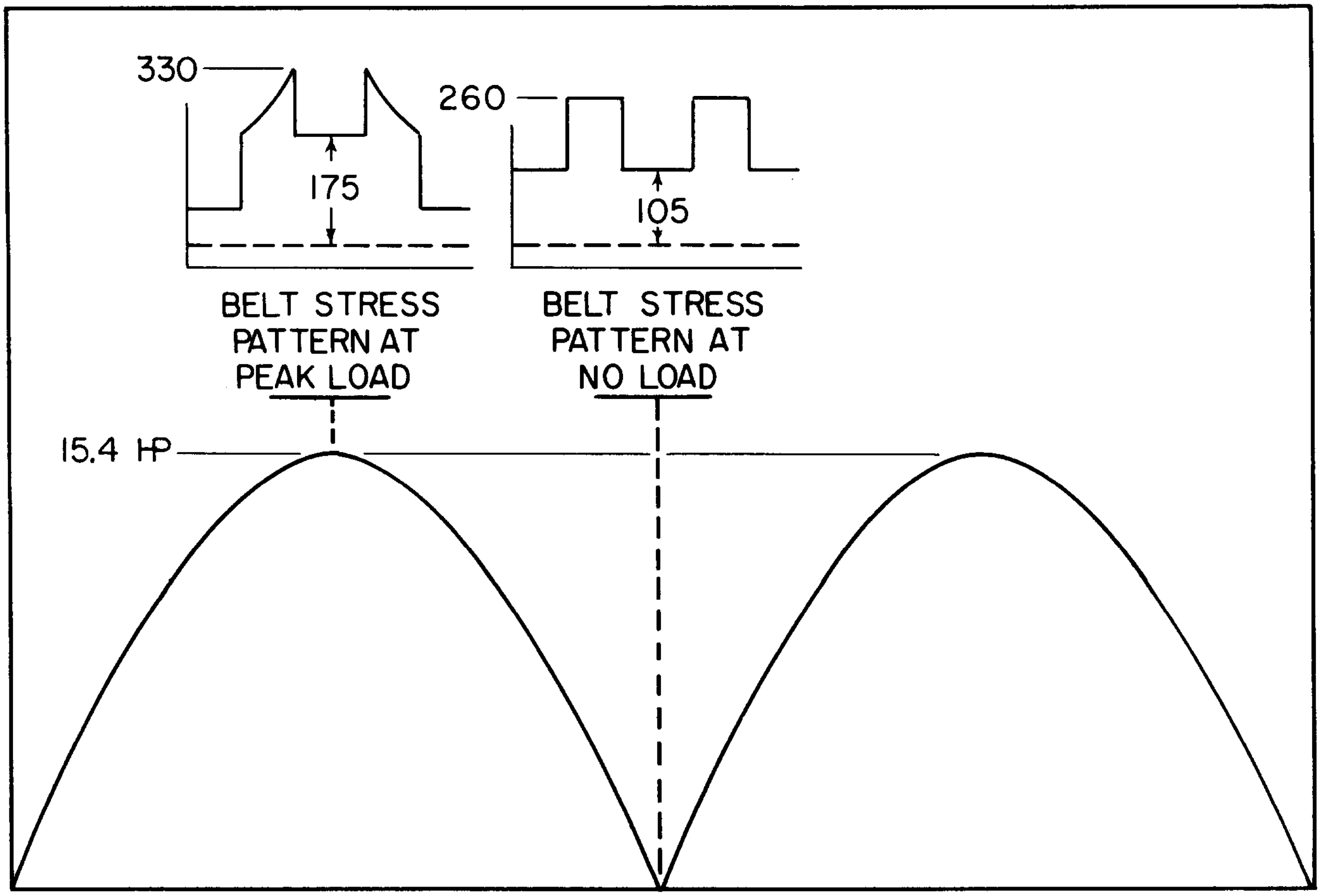


Figure 11

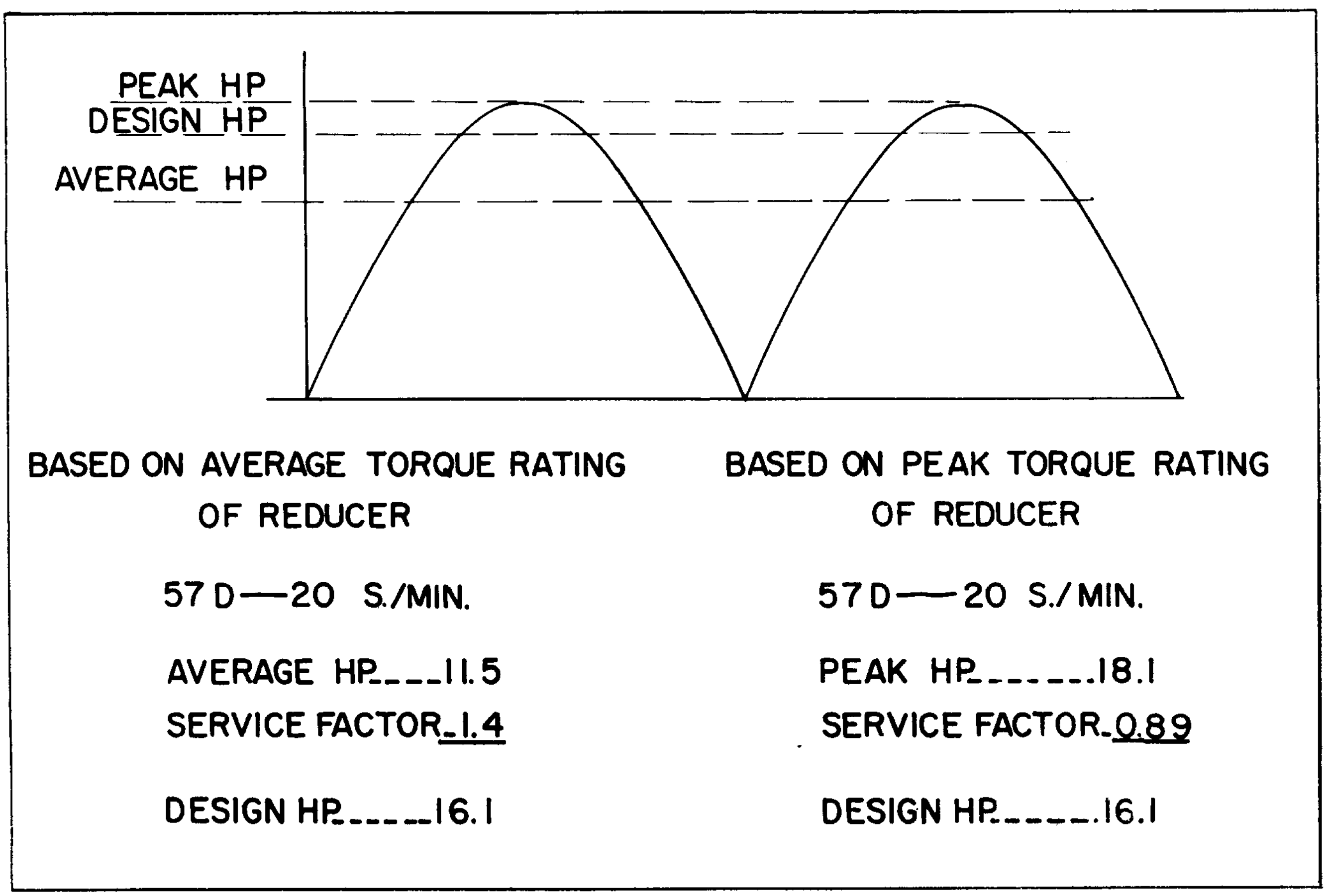


Figure 12

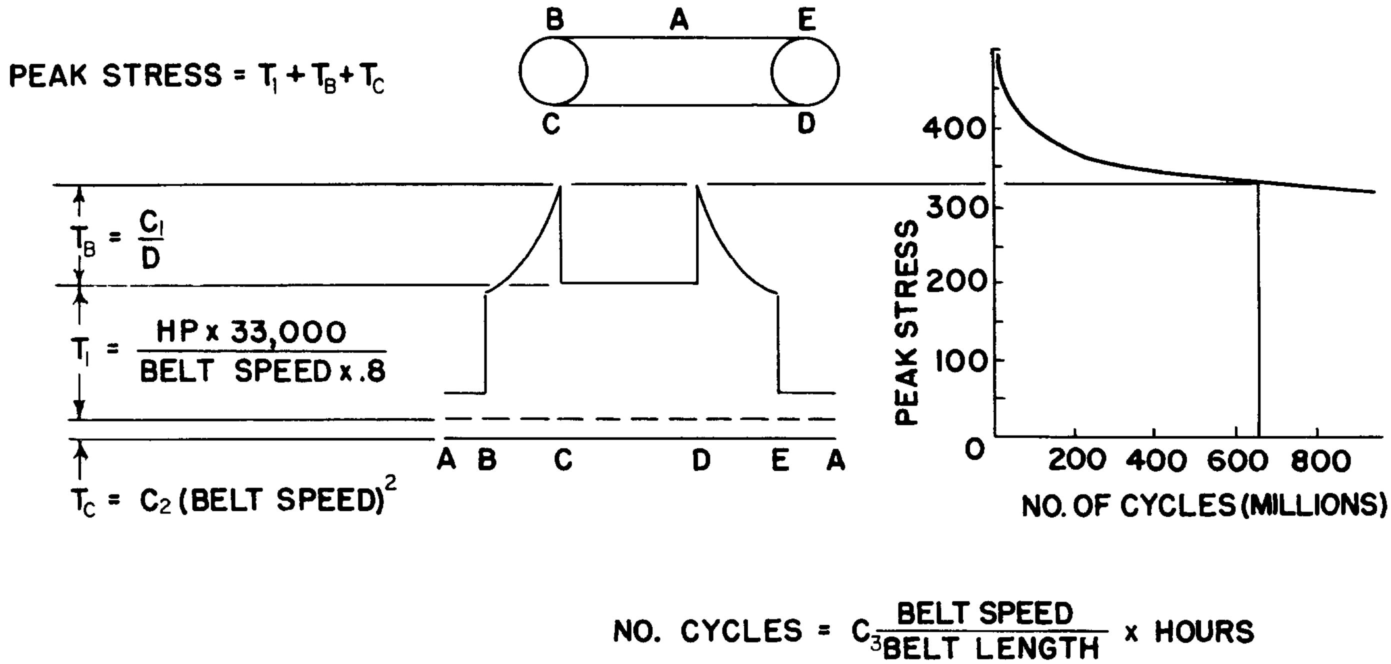


Figure 13

the belt is made up of three components: tight-side stress, bending stress, and centrifugal stress. The first equation shows that the bending stress is inversely proportional to the index diameter; the second equation shows the relationship between the design horsepower and the tight-side stress; and the third equation shows that the centrifugal stress is a function of the belt speed squared. In these equations C_1 is related to the thickness and the stiffness of the belt and C_2 is related to the mass of the belt per unit length. Adding these three values, we can determine the peak stress. Then, from the fatigue curve, we can predict the average number of cycles that the belt will withstand before failure. The equation at the bottom of Fig. 13 shows the relationship between the number of cycles and the belt speed, the belt length, and the hours of service. C_3 converts the various measurements to the same units and adjusts for the number of sheaves on the drive.

The fundamental mechanism of V-belt fatigue, as outlined above, will apply to all V-belts. The specific values for the various constants will, of course, depend upon the belt, whether standard or premium-quality, and so on.

From the stress-fatigue relationship we see that a horsepower "rating" for a V-belt must either state or imply some average service for the belt. We see further that there is no such thing as <u>the</u> horsepower rating for a V-belt. Instead, we could compute any number of ratings, each corresponding to a different fatigue life. Conversely, we could choose any fatigue life and compute a corresponding rating. In order to set up some standard for comparison between drives, the new design method discussed here takes the service level of a drive designed exactly to the horsepower rating of API Standard 1, adopted in 1956, as its standard. Then, instead of working directly with the fatigue curves, or with their complicated equations, this new method expresses the service level of any drive as a percentage of that standard service given by a drive designed exactly to API standard ratings.

When an engineer sizes a unit to a well, he bases his choice of the beam rating and the reducer for a pumping unit on a series of assumptions. In so doing, he knows he may occasionally finish up with an oversized or undersized unit. The engineer must also use some of these assumptions in choosing or analyzing a V-belt drive. With the allowance for the usual, unavoidable differences between the assumptions that the designer must make and the conditions actually seen and with recognition that unpredictable factors in any well may cause even more drastic discrepancies, a service level of 100% can be considered generally to correspond to three to five years' service.

THE NEW METHOD USED FOR ANALYSIS OF DRIVE COSTS

To show some of many ways that this new design method can be used, two typical problems have been solved. The first deals with the problem of the most economical drive and the second, with the economics of premium belts versus standard belts on a maintenance replacement basis. (The calculations have been made with data found in recently published manuals (1) and they further assume that 100% service level corresponds to exactly three years of service.

We shall assume that an API size 114 pumping unit is to be installed in an electrified field. We shall operate it at 20 strokes per minute. The unit selected has a double reduction gear box and is available at the same cost either with a 4-C-19.25-in.-unit sheave or a 3-C-27.0-in.-unit sheave. The motor speed in 1160 rpm and the center distance is about 60 in. Evaluating possible drives for a 10-year period, we raise two questions:

1. What is the lowest first-cost drive?
2. What is the most economical drive?

Since the driven sheave is part of the unit, it is not con-

TABLE I. POSSIBLE SOLUTIONS

Unit	Sheave	Sheave	Belts	Cost	Service Level	Belt Sets	10-Year Cost	Cost Per Year
114D	4-C-19.25"	3-C-9.8"	3C-162	$60.23	32%	10.3	$359.60	$35.96
114D	4-C-19.25	4-C-9.8	4C-162	76.38	170	1.96	117.58	11.76
114D	3-C-27.0	3-C-13.0	3C-180	70.35	115	2.90	138.75	13.88
114D	3-C-27.0	3-C-13.0	3C-180S	83.79	300 +	1	83.79	8.38

sidered in our cost analysis. Starting with the 19.25-in.-pumping-unit sheave, it is impossible to design a 100% service level drive, because 3.6 standard quality C-162 V-belts would be required. Since we cannot put on a fraction of a belt, we shall have to use either 3 or 4 belts. The additional belt and groove on the motor sheave will cost $16.15. Is this additional expenditure justified?

Designing the drive with the 27-in.-pumping-unit sheave shows that 3C-180 standard V-belts will give 115% service level (about 3 1/2 years). The question is raised, "Will the additional cost of $13.44 to use 3C-180S premium V-belts be justified in actual service?" We also wish to know which of the four drives considered will give the lowest cost.

In TABLE I these four possible solutions have been tabulated. The first five columns give details of each drive and its first cost. On a first-cost basis alone, the 3C-162 standard V-belt drive would be the cheapest at $60.23. However, the second question of the most economical drive requires a look at service level.

When we consider the first two drives, we see that the 3C-162 standard belts will give a service level of 32% or almost 1 year of service per set of belts. In 10 years the drive would need 10.3 sets of belts at an annual cost of $35.96.

However, adding one additional belt and one groove on the motor sheave (additional cost $16.15) results in a service level of 170% (about 5 years per set of belts). This drive, then, will show an annual cost of $11.76 or a savings of $24.20 per year (more than $2.00 per month) over the first drive.

Now we shall consider the second pair of drives. The 3C-180 standard belts will give a service level of 115% (about 3 1/2 years) at an annual cost of $13.88. Replacing the standard belts with premium-quality belts (additional cost $13.44) will result in a service level of over 300%. Therefore, 3C-180S premium-quality belts will run the entire 10 years at an annual cost of $8.38, a savings of $5.50 per year in annual costs. This is a return of 40% on the $13.44 additional investment.

In the comparison of all four drives, the 3C-180S premium-quality belt drive has the lowest annaul cost. Besides, this set of belts will perform the entire 10 years, if a 100% service level drive would last 3 years as assumed. This would eliminate downtime costs for belt replacement and lost production, which, while they have not been included in the above analysis, are often very real costs.

As a matter of fact, the costs of lost production may be considerably more than the annual belt replacement cost. For example, we shall consider a well with a 60-barrels-per-day allowable, producing 40-deg gravity oil. We assume the following conditions:

1. The company realizes $2.25 per barrel.
2. When the belts fail, the well is shut down for eight hours before the crew arrives to replace the belts.
3. It takes two hours to put the new belts on and to start up again.

Under these conditions the company would lose production worth $56.25. Discounted at 6% over a well life of 20 years, this means an actual loss of $40.00.

The second typical problem deals with comparison of annual belt replacement costs. For this comparison we shall assume a pumping unit operating at 12 spm with a 7-groove 32 in. reducer sheave. The design horsepower is 62.0 and the unit is now equipped with 7C-180 standard quality V-belts.

Here are the questions to be answered:

1. Would the additional cost of replacing the standard belts with premium-quality belts be justified over a 10-year period?
2. Could the unit operate with 6 standard V-belts instead of 7?

A comparison of four possible complements of belts is shown in TABLE II. The 6C-180 and 7C-180 standard belts are compared with 5C-180S and 6C-180S premium-quality belts. Tabulated are the cost per set of belts, the service level, the service per set of belts (still assuming 100% = 3 years), the number of sets of belts in 10 years, the total belt cost, and the annual belt cost.

TABLE II. ANNUAL REPLACEMENT COST COMPARISON

Belts per Set	7C-180	6C-180	5C-180-S	6C-180-S
Cost per Set	$84.00	$72.00	$82.40	$98.88
Service Level	100%	40%	100%	290%
Years of Service *	3.0	1.2	3.0	8.7
Sets of Belts in 10 Years	3.3	8.3	3.3	1.15
Total Belt Cost	$277.20	$597.60	$271.92	$113.71
Annual Belt Cost	$ 27.72	$ 59.76	$ 27.19	$ 11.37
Motor Sheave Cost (for comparison)	$ 57.76	$ 48.06	$ 41.47	$ 48.06

*(Based on 100% = 3 years)

First, we note that dropping one standard belt more than doubles the annual cost. Second, we note that the cost per set of belts and the annual cost for 5C-180S premium-quality belts is approximately equal to those for 7C-180 standard belts. Last, we note that the annual cost for 6C-180S premium-quality belts ($11.37) is less than half that for 7C-180 standard belts and less than one-fifth that for 6C-180 standard belts.

If the sheaves have become worn and need replacement or if a change in speed requires a change in sheaves, the premium-quality belt drive allows less expensive sheaves. This sheave-cost saving is in addition to the belt-cost saving. For comparison, the motor sheave costs for each drive are given on the last line.

SUMMARY

The true measure of a V-belt drive is not its horsepower rating. Instead, the true measure is the service that the drive will give under conditions it actually encounters in operation. An extensive study of the inter-relation of drive service with belt size and quality, load, speed, diameters, and so on has resulted in a rational, quantitative, fatigue concept of belt service. This concept has been used to develop new drive design methods which for the first time allow the engineer to consider the economic aspects of V-belt drives.

As to the uses of this method, only a few have been shown. Any time that answers are needed to the questions of V-belt drive service and cost, this new method is applicable. In most cases, the conservative drive is the most economical. However, each company's own rules on payout, maintenance cost, cost controls, and others will govern the decision on drive economics. The method presented simply allows the engineer to compare numerically the economies of several possible courses of action and guides him in spending his money wisely for new installations and in saving maintenance costs.

LIST OF REFERENCES

(1) E. G. Nelson. V-Belt Drives for Pumping Units and V-Belt Drives for Heavy Machinery. Denver, Colorado: Industrial Division, The Gates Rubber Company.