Design, Operation, & Maintenance of Pumping Unit Gear Reducers

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DESIGN OF GEAR REDUCERS

Before starting the design the designer of a pumping unit gear reducer must make five important decisions:

1. Peak torque or horsepower to be transmitted.
2. Type of gear teeth to be used — Helical or Herringbone.
3. Minimum Brinell hardness numbers of the gear and pinion steels.
4. Selection of overall reduction ratio and reduction ratio of the individual gear trains
5. Number of hours life on which to select the shaft bearings and the type of bearings to be used.

However, because the American Petroleum Institute has standardized gear reducer sizes, the first decision has already been made for the designer.

The American Gear Manufacturers Association and API have selected seven minimum hardness relationships from which to select the hardness of the gears and pinions (See API Standard 11E). The brinell hardness of the gear and pinion materials controls the wear life and endurance life of the teeth and also influences the face width of the gearing. Selection of low hardness will mean that the gear reducer would be large and balky in physical size, while selection of the highest hardness would result in a reducer of small size but would present problems of manufacture relative to the cutting of the teeth, mounting of shaft bearings and excessive deflecting of the shaft due to small size. Most manufacturers have selected hardness ranges below the maximum but above the average hardness range, and this selection has resulted in reducers with a pleasing physical size, but with a minimum of manufacturing problems.

The overall reduction ratio of the gear reducer is of great importance because of its effect upon the physical size of the reducer but also of more importance because of its effect upon the speed by which a pumping unit may be operated.

If the reducer ratio is too small, the sheaves which may be used to drive the reducer are limited in diameter by the physical size of the reducer, and the resultant overall reduction ratio from prime mover to slow speed shaft is inadequate to operate a pumping unit at speeds as low as six strokes per minute.

On the other hand, if the reduction ratio is too large, the designer is offering the purchaser something which is seldom used and also is increasing the cost of manufacture of the reducer. A reducer reduction ratio of thirty to one is the average supplied by pumping unit manufacturers and seems to meet the requirements of the oil industry.

However, no whole numbers should be used for overall or individual reduction ratios. The possibility of resonance will be decreased and the wear will be more evenly distributed between the pinion and gears if a hunting tooth is used. Thus, it is most economical to divide the overall reduction ratio approximately equal between the slow and high speed gear trains.

The type of bearings selected and the minimum life during which they will operate without failure also influence the physical size of the gear reducer. The designer will select the type bearings which are most economical to purchase, mount, and install. Having made this selection, he will then choose the minimum operating life of the bearings, and this choice will actually determine the size of the types selected. Minimum hours currently used range from fifty to more than one hundred thousand hours based on the weighted mean average loads applied to the bearings.

Design of the gear trains is best accomplished by starting with the slow speed train and calculating each train for wear capacity in turn.

A trial pitch diameter may be calculated from the following equation:

$$PDp = \sqrt[3]{\frac{1}{Cv} \times \frac{2 \times SPM \times PTr}{RPMp \quad \times \quad Kr}} \qquad (1)$$

Where:

PDp = Pitch diameter of the pinion
SPM = Strokes per Minute
PTr = Peak Torque rating
RPMp = Revolutions per minute of the pinion
Kr = Combined factor for materials, tooth form and reduction ratio: Kr is obtained from Fig. 5, API Std. 11E or more accurately by interpolation from Table 2, Page 839, 14th Edition, Machinery Handbook.

$\frac{1}{Cv}$ = Reciprocal of the velocity factor

Reducer Size	$\frac{1}{Cv}$ for Low Speed Train	$\frac{1}{Cv}$ for High Speed Train
25	1.111	1.220
40	1.119	1.245
57	1.130	1.266
80	1.134	1.278
114	1.145	1.301
160	1.153	1.321
228	1.164	1.337
320	1.172	1.361
456	1.184	1.378
640	1.195	1.395
912	1.208	1.422
1824	1.228	1.452

This table is intended to serve as a guide, but actual values will vary from manufacturer to manufacturer. When the trial pitch diameter has been determined it is then necessary to select a diametral pitch and number of teeth for the pinion, a selection which will result in approximately the desired pitch diameter.

$$PDp = \frac{Np}{DP}$$

Where:

Np = Number of teeth on the pinion
DP = Diametral Pitch

According to the Farrel-Birmingham Co., the number of teeth should be related to the peripheral velocity (V) as indicated by the following figures which are intended as a general guide. For velocities below 500 ft per minute, 14 to 25 teeth; for velocities between 500 and 1000 ft per minute, 17 to 27 teeth; for velocities higher than 1,000 ft per minute, 19 to 33 teeth. Fewer than 14 teeth should not be used to avoid teeth undercutting in manufacture and it is suggested that two different combinations of diametral pitch and numbers of teeth be selected, one combination of which will be eliminated.

Since the desired wear capacity is already known, the required face width for the above two selections may be obtained from the following equation:

$$Fi = \frac{SPM \times PTr}{63{,}000 \times KrxDs} \qquad (2)$$

Where:

Fi = Combined factor for face width and in-built factor. Fi is obtained for a given face width on Fig. 4, API Std. 11E or more accurately by the following equations:

Fi = 0.64 x FW for gears 2" wide
Fi = FW(0.667-0.0135 x FW) for gears from 2" to 10" wide
Fi = (0.380 x FW + 1.6) for gears above 10" in width.

Where:

FW = Face width of gear
DS = Combined factor for pinion RPM, pitch diameter and velocity factor, computed by the following equation:

$$DS = \frac{(PDpe)^2 \times Cv \times RPMp}{126{,}000}$$

Where:

PDpe = Enlarged value of pitch diameter $\frac{Np + 0.8}{DP}$

$$Cv = \frac{78}{78+\sqrt{V}}$$

Where:

V = Pitch line velocity in ft per minute

$$= \frac{3.1416 \times PDp \times RPMp}{12}$$

The required face width (FW) for the calculated values of (Fi) may be read directly from Fig. 4, API Std. 11E or calculated more accurately from the (Fi) equations above. The face width calculated from equation (2) should, in practice, not exceed two times the pinion pitch diameter (PDp), but if this excess should occur, it is advisable to increase the pitch diameter and recalculate the train until a satisfactory face width is obtained. On the other hand, a face width narrower than obtained from the following equation should not be used:

$$FW = \frac{2.3 \times 3.1416}{DP \times \text{Tan } h} \quad \text{(for Herringbone gears)}$$

or

$$FW = \frac{1.15 \times 3.1416}{DP \times \text{Tan } h} \quad \text{(for Helical gears)}$$

Where:

h = Helix angle

Sometimes it is more practical, from a deflection standpoint, to widen slightly the face width to maintain the above minimums than to decrease the pitch diameter and thus obtain wider face widths from equation (2).

When the pitch diameter, diametral pitch and face width have been determined by equation (2), the diametral pitch should be checked by the following equation:

$$DP = \frac{63{,}000 \times Ks \times FW \times PDp \times RPMp}{SPM \times PT} \qquad (3)$$

Where:

Ks = 0.03600 for 225 BHN pinion
= 0.04000 for 245 BHN pinion
= 0.04144 for 255 BHN pinion
= 0.04288 for 265 BHN pinion
= 0.04363 for 270 BHN pinion
= 0.04575 for 285 BHN pinion
= 0.04788 for 300 BHN pinion
= 0.05000 for 315 BHN pinion
= 0.05283 for 335 BHN pinion
= 0.05400 for 350 BHN pinion

The diametral pitch which equation (3) calculates is the smallest tooth size which should be used. The diametral pitch to be used is the <u>first whole or half number smaller</u> than the answer equated by equation (3) or the answer itself if it is a whole or half number. Machinery's and similar Handbooks list diametral pitches which are commonly used.

By using the smallest tooth size which equation (3) equates for a given face width and pitch diameter, the largest number of teeth, congruent with good design, are in contact and thus increase the wear above that of larger tooth sizes which might have been selected.

Occasionally it is wise to increase the face width slightly over that calculated by equation (2) to insure that the smallest tooth size permissible by equation (3) may be used and thus render certain that a larger number of teeth are in contact.

As a prelude to calculating the endurance strength of the teeth it is necessary to know the length of the line of tooth contact. It is also convenient to know the tooth contact ratio in the plane of rotation and the face width contact ratio as an index of the wear life of the gear train, for the larger the sum of the contact ratios the better is the wear life.

$$La = \sqrt{ri^2 - r^2} + \sqrt{Ri^2 - R^2} - C \times \text{Sin } p \qquad (4)$$

Where:

La = Length of the line of tooth contact

ri = Outside radius of the pinion $\frac{Np + 2.4}{2 \times DP}$

r = Base radius of the pinion = $\frac{Np}{2xPD} \times \text{Cos } p$

Ri = Outside radius of the gear $\frac{Ng + 0.8}{2 \times DP}$

R = Base radius of the gear = $\frac{Ng \times \text{Cos } p}{2 \times DP}$

C ≒ Shaft center distance = $\frac{Np + Ng}{2 \times DP}$

Where:

p = Pressure angle
Ng = Number of teeth on the gear

The contact rotation in the plane of rotation is determined as follows:

CRp = La/BP (Should never be less than 1.25)

Where: BP = Base pitch = (3.1416/DP) Cos p

The face width contact ratio is:

$CRfw = \frac{FW \times DP \times \text{Tan } h}{2 \times 3.1416}$ (for Herringbone gears)

$CRfw = \frac{FW \times DP \times \text{Tan } h}{3.1416}$ (for Helical Gears)

The total contact ratio is:

CRt = CRp+CRfw = The average number of teeth in contact

The tooth endurance strength capacity of the train is determined by the following equation:

$$T = \frac{63{,}000 \times S \times Yhk \times FW \times La \times Mh \times Dm}{SPM} \quad (5)$$

Where:

S = Allowable or endurance limit stress in bending
S = Sef = 250 x BHN (for 400 BHN and above use
S = 100,000 psi.)
Yhk = Tooth form factor based on virtual number of teeth (Nv) on the pinion and obtained from the chart on Page 125, "Manual of Gear Design", Section Three, by Earle Buckingham.

$$Nv = \frac{Np}{\text{Cos}^3 h} = \frac{Np}{0.6495}$$

$$Mh = \text{Angle factor} = \frac{\text{Cos } h}{\text{Cos } p} = 0.924$$

$$Dm = \frac{PDpe \times Cv \times RPMp}{395{,}000} = \frac{Ds}{3.1416 \times PDpe}$$

Actually the Yhk used above is not accurate but is conservative because of the large total contact ratios encountered in pumping unit reducers.

The values obtained from equation (5) will range from 2.5 to 3 times those obtained from equation (2) and, because of the high starting torques encountered on pumping units, should never be less than twice the rated capacity of the gear reducer. Actually the designer who makes the tooth endurance strength as low as twice that of the wear rating is apt to encounter tooth failures from high stress concentration induced by pitting, scuffing, spalling, etc. And tooth endurance strength as low as twice the wear rating would ignore added stress induced from fluid pound, synschronous speeds of operation, sticking plungers, etc.

Since pumping unit gear reducers frequently operate under heavy loads and shock, the gear hub diameters should have the following minimum relationship to the hub bore:

D = 1.8 x Bore for cast steel gears
D = 1.65 x Bore for forged steel gears

Values of (T) obtained from equation (5) may be used as the torsional capacity of the interference fit between the gear hub bores and shafts. And because starting torques sometimes exceed twice the rated wear capacity, it seems unwise to make the torsional capacity of the interference fit less than the values obtained from equation (5).

The gear hubs will require a medium or heavy force fit with the shaft to achieve the torsional holding capacity required by equation (5). The hub lengths may be determined by the following equation and should never be less than the face width of the gear:

$$L = \frac{T}{0.5 \times 3.1416 \times f \times P \times d^2}$$

Where:

L = Hub length exclusive of chamfers, in in.
T = Torque from equation (5) in in.-lb
f = Coefficient of friction = 0.125
P = Press-fit pressure between hub and shaft in lb per sq in. P is obtainable from Fig. 56, page 924, 4th Edition of Mark's Handbook.

The thickness of metal under the root of the teeth on a gear should never be less than 0.5 x 3.1416/DP, but an equation which gives a more conservative minimum value is:

$$Tr = \frac{1}{DP} \times \sqrt[4]{Ng/2j}$$

Where:

Tr = Minimum metal thickness under the root of the teeth
j = Number of arms. The number of arms for cast steel gears is usually four for gears up to 20 in. pitch diameter, 5 for pitch diameters up to 40 in. pitch diameters and 6 for gears up to 120 in. pitch diameter, and 8 for larger pitch diameters.

The section modulus of the gear arms in bending in the plane of rotation is determined by the following equation:

$$Z = \frac{T}{Sd}$$

Where:

T = Torque from equation (5)
Sd = Allowable stress in bending = Yield stress divided by 3.75

Other proportions of gear arms may be determined by referring to design handbooks or textbooks.

Forged gears are usually designed with webs instead of arms. Thus, when designing forged gears it is best, because of forging problems, to rely upon the recommendations of gear blank forging companies for web shape and size.

The wear capacity of a pumping unit gear reducer at eight strokes per minute is normally about five per cent above the rating at twenty strokes per minute. The designer should provide for this added loading in the strength and rigidity of the shafts.

Backlash of the mating gear trains should be according to the following table:

2-1/2 DP to 4 DP	0.005 to 0.101 in.
4 DP to 6 DP	0.004 to 0.008 in.
6 DP and finer	0.003 to 0.006 in.

From Section 9 of "Recommended Practice", American Gear Manufacturing Association, for the ratings of helical and herringbone gear speed reducers used for oil field pumping units.

Deflection of shafts in a gear reduction unit is of vital importance because of the influence of deflection on tooth bearing pressures and resultant wear. Bearing should be so placed that the bending deformation will be reduced to a minimum congruent with economic design.

The equations for design of shafts may be found in design textbooks and the allowable stresses are found in the AGMA Standards.

Selection and sizing of bearings is beyond the scope of this discussion, but they may also be determined by referring to the engineering manuals of various bearing manufacturers.

Case design and lubrication systems vary from manufacturer to manufacturer and are usually dependent upon the experience of the designer and the machine tools available for manufacture.

<u>Warning</u>

The above equations are based on the American Gear Manufacturer's Association Standard stub tooth involute system. Long and short addendums are used on the pinion and gear respectively. Basic tooth and diameter information is listed below:

Pressure angle = p = 20°
Helix angle = h = 30°
Addendum = 0.8/DP
Defendum = 1/DP
Working Dept. = 1.6/DP
Total Depth = 1.9/DP
Basic Tooth Thickness - at the Pitch Line = 1.5708/DP
Clearance = 0.3/DP
Pitch diameter of pinion = PDp = Np/Dp
Enlarged value of pinion pitch diameter = PDpe = (Np + 0.8)/DP
Outside diameter of pinion = ODp = (Np + 2.4)/DP
Root diameter of pinion = RDp = (Np-1.4)/DP
Pitch diameter of gear = PDg = Ng/DP
Reduced value of pitch diameter of gear = PDgr = (Ng - 0.8)/DP
Outside diameter of gear = ODg = (Ng + 0.8)/DP
Root diameter of gear = RDg = (Ng-3.0)/DP

OPERATION OF GEAR REDUCERS

The unit manufacturers representative is often asked to state what safety factors are used in the design of his company's gear reducer. But what the questioner is really asking for is assurance that no harmful results will be encountered if the gear reducer is overloaded. However, no responsible manufacturer can truthfully give such assurance, for there is <u>only one</u> basic safety factor — that safety factor used to <u>insure a</u> certain total number of life cycles when operating under conditions of uniform load at rated speed for from eight to ten hours per day and conditions of nearly perfect lubrication. It also is assumed that there are more than one hundred teeth in each train and that the ratio of each train is at least eight to one. But except in single reduction type the latter is seldom achieved for pumping unit gear reducers.

Any load imposed in excess to the rated capacity will decrease the wear life of the reducers. Under conditions of one hundred per cent direct overload, heavy shock from the well, moderate shock from the prime mover, and poor lubrication, the involute and wear surface of the gear teeth can be destroyed in as few as one thousand revolutions of the slow speed shaft. This statement <u>does not</u> mean to imply that the gear teeth will fail; instead, <u>it</u> means that the reducer would become increasingly noisy and that tooth failure could be expected relatively soon after that number of cycles has been reached. The tooth failure would be caused by reduced physical size of the teeth and stress concentration caused by pitting, galling, seizing, scaffing, etc. And once these latter detriments have started, the rate to failure is rapidly acceletrated.

Service factors for enclosed speed reducers have been developed by AGMA. These factors shown in Table I should be used when sizing a reducer for and oil well pumping unit. These service factors should be applied to the peak torque which was calculated when sizing the pumping unit by any of the methods currently in use.

TABLE I
SERVICE FACTORS FOR ENCLOSED SPEED REDUCERS

Prime Mover	Duration of Service	Rod String Load Classifications		
		Uniform Ks	Moderate Shock Ks	Heavy Shock Ks
Electric Motor	**Occasional ½ hour per day	0.50	0.80	1.25
	**Intermittent 3 hrs. per day	0.80	1.00	1.50
	8 to 10 hours per day	1.00	1.25	1.75
	24 hours per day	1.25	1.50	1.75
Multi-Cylinder Internal Combustion Engine	**Occasional ½ hr. per day	0.80	1.00	1.50
	**Intermittent 3 hrs. per day	1.00	1.25	1.75
	8 to 10 hours per day	1.25	1.50	2.00
	24 hours per day	1.50	1.75	2.00
Single Cylinder Internal Combustion Engine	**Occasional ½ hr. per day	1.00	1.25	1.75
	**Intermittent 3 hrs. per day	1.25	1.50	2.00
	8 to 10 hrs. per day	1.50	1.75	2.25
	24 hrs. per day	1.75	2.00	2.25

**Maximum momentary or starting load must not exceed 200 percent of the gear reducer rating (rating meaning a service factor of one).

The conditions of operation as shown in Table I are seldom considered when selecting a reducer. The effects of poor lubrication, rough running, slow firing engines, synchronous speed impulses, sticking plungers, plungers bumping bottom, gas and fluid pounds, overload from poor counterbalance and intentional overload are of little concern to most users of pumping units, but when failure does occur the manufacturer is usually condemned for his lack of foresight in not building the reducer as big as a barn initially. Were he to do so, the manufacturer could not build economically reducers, nor could the user afford them. Further, service factors are generally neglected in actual practice but do affect the life of the reducer. Thus lack of application of the above service factors can only result in decreased life of gear trains.

All manufacturers of pumping unit therefore design to a service factor of one and rightfully expect the user of their equipment to apply the service factor which the particular situation to which the unit will be fitted.

Wear factors reflecting conditions of lubrication cannot be applied directly to the basic reducer capacity; instead they must be applied as part of the life factor. The effect of indifferent lubrication can be more severe than a fifty percent overload. Table II lists lubrication factors applicable to pumping unit gear reducers.

TABLE II
LUBRICATION FACTORS FOR WEAR

Condition of Lubrication	Lubrication Factors, Kg
Oil of proper viscosity & character, filtered annually & moisture drain monthly	1.15
Oil of proper viscosity & character, filtered every three years & moisture drained bi-annually.	1.25
Oil of poor viscosity & character, seldom filtered or moisture seldom drained - Indifferent lubrication	1.35

The above service and lubrication factors may be utilized to develop life factors which can be used to determine the approximate wear life of gear trains. For this determination we must revert to the very basic equation for wear of the gear train, because the wear life of the gear train is related to the shear stress imposed on the gear teeth. The resulting equation (6) is shown below.

$$Kl = Kg \sqrt{\frac{PTa \times Ks \times SF}{PTr}}$$

Where:

Kl = Life Factor
Kg = Lubrication Factor
Ks = Appropriate Service Factors
PTa = Actual Peak Torque on the Reducer
PTr = Rated Peak Torque of the Reducer
SF = Speed Factor obtained from Table III

TABLE III
SPEED FACTORS

Speed of Operation	Speed Factor	Speed of Operation	Speed Factor
6 spm	0.939	22 spm	1.007
8 spm	.950	24 spm	1.013
10 spm	.960	26 spm	1.019
12 spm	.969	28 spm	1.025
14 spm	.978	30 spm	1.030
16 spm	.986	32 spm	1.036
18 spm	.993	34 spm	1.041
20 spm	1.000	36 spm	1.046

When appropriate values have been selected and equated by equation (6), the resultant life factor (Kl) may be used with the following nomigraphs to determine the approximate wear life of the gear train.

Problem 1

Find the approximate wear life in revolutions of the slow speed shaft of a 160D gear reducer operating under the following conditions (315 BHN):

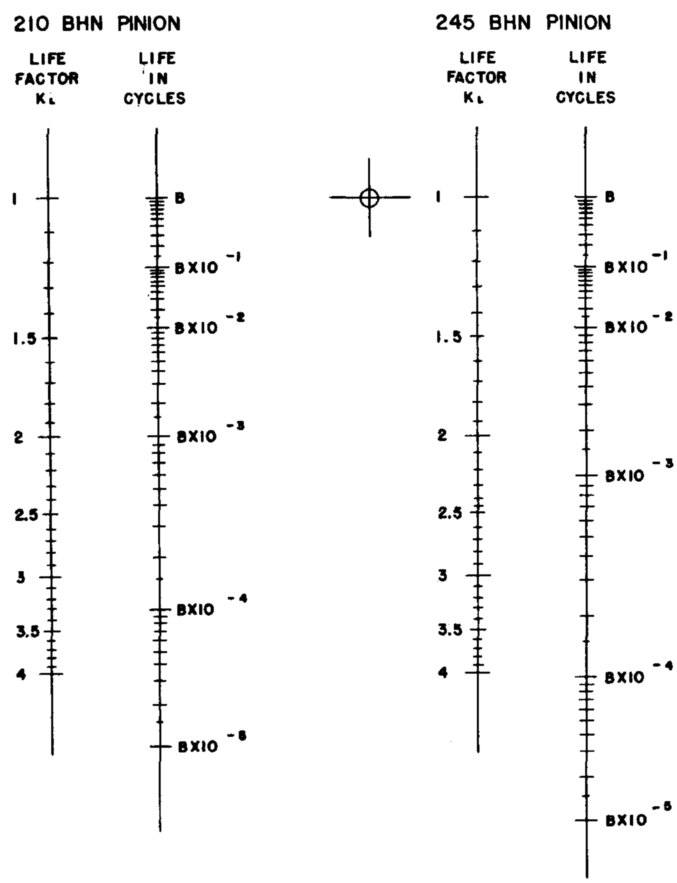
a. Indifferent lubrication.
b. Operating speed = 16 spm for 24 hr
c. Rated peak torque = 160,000 in.-lb
d. Actual Peak Torque = 246,000 in.-lb
e. A rough running slow firing single cylinder engine
f. Heavy shock caused by a severe fluid pound at 3/8ths of the down stroke.

$$Kl = 1.35 \sqrt{\frac{246{,}000 \times 2.25 \times .986}{160{,}000}} = 2.49$$

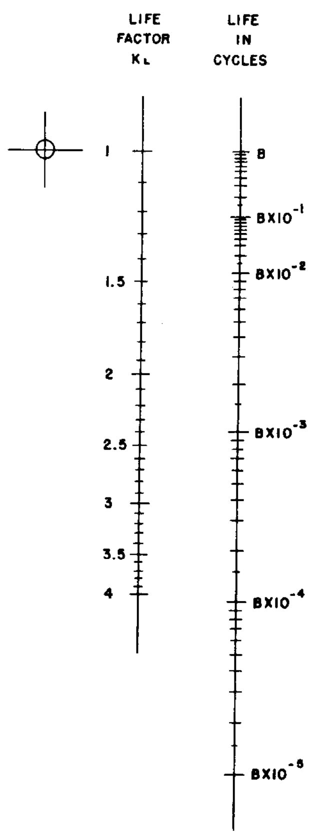
A straight line drawn from the zero point through 2.49 on the 315 BHN pinion nomigraph shows the life to be about one thousandths of four per cent of the base life (B). This estimate does not mean to imply that the reducer will become inoperable; rather, a progressively noisy reducer may be expected almost from the day the unit was installed in the field. However while tooth failure may be expected any time after the wear surfaces and involute of the teeth have been destroyed, the length of time will be dependent upon the size teeth the designer selected: small diametral pitch numbers result in poor wear life but great tooth endurance, while large diametral pitch numbers result in good wear life but poor tooth endurance.

The various methods used to size pumping units give consideration to only two of the conditions actually required in selecting the right size: the speed of operation and the loads imposed by the rod and fluid load and depending upon the purpose which the purchaser intends, a reducer selected on the basis of these two considerations may be too small or too large.

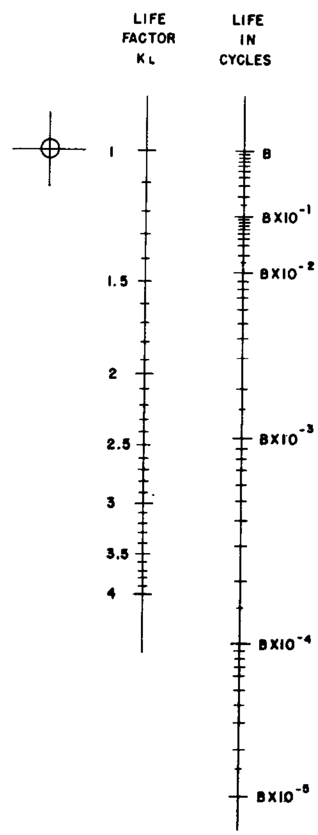
If the reducer is required to operate twenty-four hours a day every day for several years and is operated with a rough running, slow firing single cylinder engine with scant attention paid to proper lubrication, the reducer may be inadequate for the job for which it is intended. Thus, the purchaser may be faced with repair bills which were totally unexpected because only two conditions of operation were considered at the time purchased. The result would be higher lifting costs per barrel because of higher repair bills and down time. Such a situation can cause resentment toward the manufacturer who actually is not responsible for lack of consideration of operating conditions.



265 BHN PINION



285 BHN PINION



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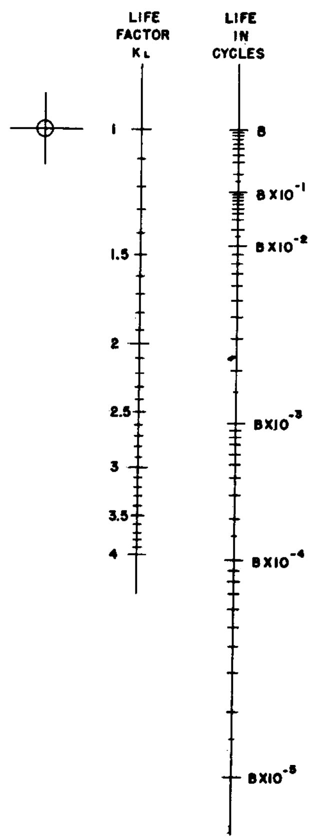
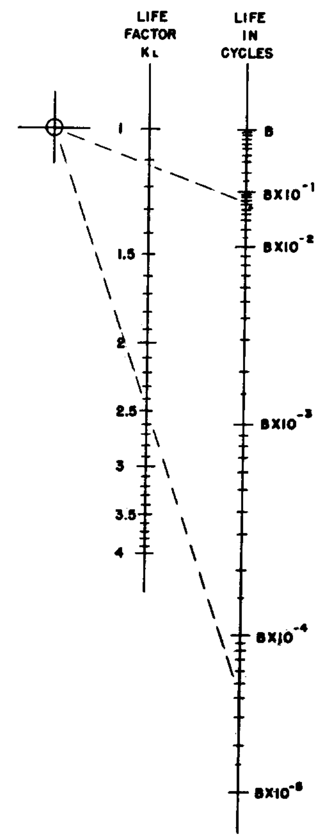


315 BHN PINION



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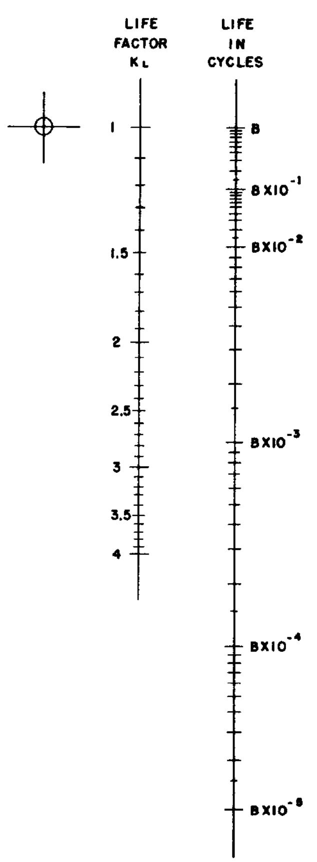
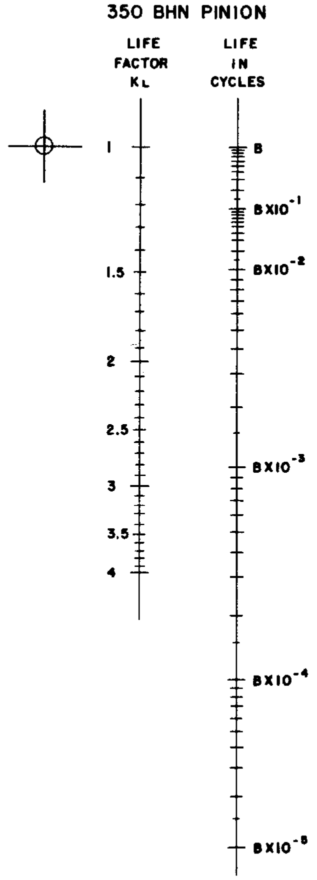


350 BHN PINION



On the other hand, a reducer may operate for as little as three to six hours per day only nine to fifteen days of each month and may be powered by an electric motor with proper attention paid to lubrication. In this case the equipment life may exceed by years the amortization period which the purchaser has assigned to the unit. However, the lifting cost per barrel would again be higher than they should be because the reducer was actually larger than needed to pump the required fluid.

Problem II

A producer has a stripper well which he desires to pump intermittently three hours per day, at ten strokes per minute, 365 days per year. The unit will be operated by an electric motor controlled, by a time clock, at intervals which will insure sufficient fluid at the pump to eliminate any possibility of fluid pound. He has also determined that proper maintenance, counterbalance, and lubrication will be employed. He plans to amortize the cost of the unit over a ten year period and expects an increase in repair bills after the ten years. His method of unit sizing shows that 20,000 in.-lb of torque are required but he is tempted to install a 16D reducer rather than a 25D. Is the 16D reducer sufficiently large to meet his needs and last the ten year period?

Total cycles of operation in ten years = 3 x 60 x 10 x 365 x 10 = 6,570,000

K_g = 1.15
K_s = 0.80 for intermittent 3 hr per day
BHN = 315 (must be obtained from unit manufacturer).

If B on the 315 BHN pinion nomigraph is assumed to be one hundred million cycles the resulting life factor for 6,570,000 cycles is:

$Kl \leq 1.12$

Formula (6) may be rearranged and solved to determine if a 16D reducer is sufficiently large:

$$PTr = \frac{PTa \times Ks \times SF}{\left(\frac{Kl}{Kg}\right)^2} = \frac{20{,}000 \times 0.8 \times 0.96}{0.948}$$

$PTr = 16{,}200$ in.-lb.

Since the resultant peak torque is only slightly above the 16D rating, it would undoubtedly be satisfactory.

It is important to state that the lubrication factors, life factors, and nomigraphs shown in this section are not approved by the API or AGMA organizations. The nomigraphs were developed by the author from information found in API and AGMA publications and have been found on the basis of his experience to be reasonably accurate.

MAINTENANCE OF GEAR REDUCERS

It is quite obvious that the conditions under which a pumping unit is operated are directly related to the maintenance costs. The conditions of operation which affect maintenance cost are as stated previously:

a. Speed of operation of the unit
b. Hours of operation per day
c. Overload from all causes; overloads from well conditions improper counterbalance, crooked holes, etc. All have an appreciable effect on life.
d. Shock loads from all causes. Synchronous speed impulses, sticking plungers, plungers bumping bottom, fluid pound, gas pound; and type, speed and condition of prime movers all cause shock loads.
e. Condition of gear lubricant

The API allows manufacturers to advertise only the reducer rated capacity at 20 strokes per minute as set forth in API Standard 11E. However, the equations for rating the gearing will calculate higher capacities at slower speeds of operations and lower capacities at higher speeds. But experience indicates that the capacities calculated at speeds other than twenty strokes per minute are valid and that no real reason exists for not using them. However, manufacturers who use API monograms must not represent them as being any rating other than the API capacity.

Members of the AGMA have found that just as with continuous operation of an internal combustion engine, continuous operation of a gear reducer is detrimental to the life of gear trains. Since the original basis of design was for operation up to ten hours per day, the capacity must be derated if operation is to be continuous. Probably the most important reason for the derating is the passage of metal particles, worn from the gear teeth and suspended in the gear lubricant, through the gearing. This action would occur many times more than it will when operating up to ten hours per day and thus accelerate the rate of wear.

Further unintentional overload, next to poor lubrication, is the largest offender in shortening the life of a gear reducer.

Methods of sizing pumping units are empirical equations which by experience have been found to be reasonably accurate. However, because so many variables exist the loading can at any time be materially different than the calculated values. For that reason the use of portable test units and dynamometer card analysis is the most accurate way of predicting what loads will exist on the rod string and unit. However, even when this type of program is used for sizing equipment the results can still be bad if little attention is paid to proper counterbalance after the unit is installed. The author has found that in the largest share of gear failures that little attention has been given to proper counterbalance. In fact, overloads of more than 100 per cent have been found due to improper counterbalance. But even in cases where proper counterbalance would not have brought the reducer loading under the rated capacity the life expectancy would have been greatly enhanced by proper counterbalance from the date of installation.

Shock loads are the third largest offender relative to reducing gear life. Generally speaking, economical solutions to shock loading exist. For instance, synchronous speed impulses can be eliminated by changing speeds; sticking plungers can be eliminated by repair; fluid pound can in some cases be eliminated by intermittent pumping; gas pounds and gas locks may be eliminated by installation of proper equipment. Further, the effect of engine impulses may be offset by speeding up the engine and sheaving down to the desired strokes per minute, and rough running engines can be repaired. There are few valid excuses for shock loading of a gear reducer, and continued operation under impact may cost an operator a repair bill or even a new reducer. However, it seems to be a general practice to pound fluid on water flood projects, and this practice should be considered by application of a proper service factor when the units are selected.

The character of the gear lubricant is, next to overload, the most critical condition of operation. When supplying lubricants to customers, pumping unit manufacturers generally follow the recommendations of API Standard RP11G. However, in certain areas conditions exist, and only experimentation by the producers can determine proper lubrication practices. The manufacturer, not being aware of individual practices, will follow those recommended by API or his own recommendations.

A large number of pumping unit operators pay scant attention to the condition of the gear lubricant, even though their lubrication practice for structural bearings may be excellent. Thus, under the loads which pumping unit gear reducers operate conditions of border line lubrication exist. Carelessness in relationship to the character and cleanliness of the gear lubricant can result in decreased life and permanent damage to the gear teeth.

The following is an abstract from Gears and Their Lubrication, published by Socony Mobil Oil Company, Inc. of New York and presents excellent information on factors influencing lubrication and types and causes of gear failures:

FACTORS INFLUENCING LUBRICATION

To prevent metal-to-metal contact between meshing gear teeth, a thick oil film is required in the area of contact. The character of the oil employed has a major effect in establishing and maintaining this film. When selecting the oil for enclosed gears, the following factors of design and operation require consideration:

(a) Type of gear
(b) Pinion speed
(c) Ratio of reduction
(d) Operating temperature
(e) Imput horsepower
(f) Nature of load (uniform or shock)
(g) Type of drive (motor or reciprocating engine)
(h) Method of oil application
(i) Water contamination

On spur, helical, herringbone, bevel and spiral-bevel-gear teeth, the line of contact between engaging teeth sweeps swiftly, without side slide, over the entire working surfaces of each tooth; hence contact at any specific point endures for only an instant. Pressure, therefore, at such a point is applied and released so quickly that there is comparatively little time to squeeze out the fluid film of lubricating oil. Moreover, the direction of sweep in relation to the line of contact acts to create a fluid film. However, if an effective oil film is to be formed, it is important that the oil be of proper viscosity. Too light an oil would be squeezed from between the gear teeth and metal-to-metal contact would occur; too heavy an oil would result in needless fluid friction, and higher operating temperature would, therefore, result.

The higher the speed of meshing gears, the higher will be the sliding and rolling speeds of individual teeth. So even though sliding between two teeth varies from maximum to zero to maximum during the meshing period, a thick film tends to form during zero slide because of the squeeze-film action.

When an ample supply of oil is available, speed assists in forming and maintaining a fluid film. At high speed, more oil is drawn into the pressure area, and, in addition, the time available for oil to be squeezed from between the meshing teeth is exceedingly short. Therefore, comparatively light bodied oils may be used, for despite their fluidity there is insufficient time to squeeze out the oil film. At low pinion speeds, however, more time is available for oil to be squeezed from between the teeth and less oil is drawn into the pressure area. Therefore, the lower the speed of the pinion, the heavier should be the body of the lubricating oil.

With multiple-reduction gear sets containing two or more successive steps of reduction, it is necessary to consider the low as well as the high-speed reductions. In so far as lubrication is concerned, the first reduction in such sets can be compared with a single-reduction gear set. However, because of the lower speed of the last reduction, the lubricating oil requirements are different.

Where the reduction of speed is less than 10 to 1, a single reduction gear set is usually used and the oil is then selected on the basis of the speed of the driving pinion. But, when the reduction of speed is greater than 10 to 1, a multiple-reduction gear set is usually employed. In this case, oil should be selected to meet the requirements of the low-speed pinion of the last reduction, instead of the requirements of the high-speed pinion of the first reduction. Therefore, when the ratio of speed is greater than 10 to 1, heavier bodies oils are required.

The temperature, under which different gears may operate, constitutes another important factor in the selection of correct oil. At ordinary room temperature the body of the oil in the barrel or container may be quite different from its body at operating temperature, for during operation, the heat generated by friction and by churning of the oil will increase the temperature of the oil in base of the gear set. In a fully loaded spur, helical, herringbone, bevel or spiral bevel gear set, the temperature rise will be approximately 50° F. This rise is affected to some degree by the power input to the unit; thus, when the exterior surroundings of the gears are at ordinary room temperature (50° F. to 100° F), the maximum operating temperature with spur, helical, herringbone, bevel, spiral bevel gears should not exceed 150° F.

When gear sets are located in hot places or subjected to heat from outside sources, the final operating temperature may be considerably higher. In such cases, it is necessary to compensate for the increased temperature by using an oil of increased body.

On the other hand when gear sets are located in cold places, the final operating temperatures may be much lower than given above. Furthermore, when the operation starting the temperature of the gear set may be so low that the oil ceases to flow properly. If the oil in a splash-lubricated set becomes too viscous, the gears will channel a groove through it with the result that practically no oil at all will be carried to the point of mesh. Under cold conditions, therefore, it is necessary to select an oil which will be amply fluid for starting purposes, but which will also be heavy enough to maintain protecting oil films when heated to the operating temperature.

Further, the load on the meshing teeth of a gear acts to squeeze the separating oil film from between the contact areas. Tooth pressure, therefore, is a factor to be considered in selecting proper body. The greater the pressure, the more viscous the oil must be to resist the squeezing action and thus to maintain an effective oil wedge. On the other hand, where gear-tooth pressures are light, a lighter-bodied oil will provide protective oil films that keep fluid friction at a minimum and assure low power loss.

Basically, the total force acting on gear teeth depends on the power transmitted and on the diameter and speed of the pinion. Formulae for the design of gears have been pretty well standardized, and according to these formulae, the maximum permissible unit loading for tooth surfaces largely depends on the Brinell hardness of the gear metals.

Gears of low hp rating are ordinarily constructed with narrow face, small diameter, and small tooth size; those of high hp rating are of wider face, larger diameter, and larger tooth size. Because the tooth profiles of the larger gears are formed by longer radii which results in wider bands of contact for similar loads per inch of tooth width, the teeth of the larger diameter gears will be able to carry greater loads per in. of tooth width than can be carried by the teeth of smaller gears even though both are designed to carry the same pressures per square inch of contacting surface. Therefore, with gears of similar type, the larger sizes (greater hp units) are usually designed and constructed to take advantage of the increased width of contact line and thus to operate with higher linear tooth loadings, i.e., higher loads per in. of tooth width.

Input power also has an effect on lubrication because of the heat which is generated between gear teeth.

If one considers two reduction-gear sets of the same type and with the same ratio of reduction and driven at the same rotational speeds, but designed for different horsepower ratings, he will find that the gears designed for greater hp may be either of wider face or of larger diameter, or of both.

Larger diameter gears may have either a greater number of teeth or the same number of teeth but of larger cross-sectional size. In either case, the total area of contact surface is increased. This contact surface is also increased when the gears are made wider and the cross-section of the teeth left unchanged.

Inasmuch as gears of similar type and hardness are constructed to operate with approximately the same pressures per sq in. of tooth contact area, any increase of contact area will result in a corresponding increase in friction. Thus, irrespective of whether the increased power is taken care of by additional gear width or greater diameter or both, the result is increased friction. Furthermore, in either case there will be increased churning of the oil at the point of mesh. Where splash lubrication is used, there will also be increased churning in the bottom of the gear case. This condition would be aggravated by high oil level or heavy-bodied oil.

In both small and large units, the power lost in friction and in churning oil is converted to heat. The relative area of radiating surface per unit of heat generated is usually less in a large unit than in a small one. Hence, larger gear sets, transmitting greater amounts of power, tend to run hotter than smaller sets. Inasmuch as heat thins the oil, it is advisable to offset this condition by using heavier bodied oils for the larger and hotter units.

The nature of the load on any gear set has an important influence on the selection of a lubricating oil. If this load is uniform, the torque (turning effort) and the pressure carried by the teeth will also be uniform. However, excessive tooth pressures because of shock loads tend momentarily to rupture lubricating films that if the load were more uniform, would otherwise separate the gear teeth. Therefore, where this shock-load factor has not been considered in the design or selection of the gear set, an oil of heavier body is required to prevent film rupture.

Of course, in some operations, the conditions may be still more severe because of overloads or because of a combination of heavy loads and extreme shock loads — for example with fluid pounds or with gears starting under heavy load. In such cases, it may be impossible to maintain an effective oil wedge. Hence during a considerable part of mesh only boundary lubrication exists: thus, there is only a microscopic lubricating oil film between the rubbing surfaces. Excessive wear then results unless a special oil is used that possesses unusual film strength characteristics.

Occasionally, however, there are instances when gears are loaded so heavily that it is very difficult to maintain an effective lubricating film between rubbing surfaces. When operating under this condition of extreme pressure, metal-to-metal contact is so severe that even the highest film strength will not completely solve this problem. Wear, therefore, cannot be avoided. However, it can be controlled by using oils that possess special properties which prevent the meshing tooth surfaces from welding together under the intense, localized pressures and spot temperatures. It is this welding or fusing that roughens, tears, and scores the rubbing surfaces, and thus rapidly destroys the teeth. By chemically preventing welding, rapid and drastic tooth failure can be prevented; only slow wear of a smooth and controlled character will then take place. Lubricants for this purpose are known as EP (Extreme-pressure) lubricants.

When the power transmitted by gears is developed by electric motors, the uniform turning effort of this prime mover throws no additional load on the teeth of the transmission gears. The variable torque developed by reciprocating engines, however, is accompanied by a corresponding variation in tooth loading. Therefore, when transmission gears are driven by multicylinder engines, single cylinder engines, etc., heavier-bodied oils may be required to assure effective oil films at all times. <u>But heavier oils are not necessary where this has been considered and compensated for in the design or selection of the gear set.</u>

When lubricating oil is applied to gear teeth by means of a splash system, the formation of an oil wedge between teeth is, generally speaking, less effective than when the oil is circulated and sprayed directly onto the meshing surfaces. This generalization is particularly true in the case of low-speed splash-lubricated units when only a limited amount of oil may be carried to the meshing area. However, heavier bodied oil is needed to offset this condition, since the heavier the body the greater will be the quantity of oil that clings to the gear teeth.

With a gear set lubricated by a pressure-circulation system, there is usually a better opportunity for dispersal of heat than with a splash-lubricated system, with a circulation system, the oil is thrown against all internal surfaces of the gear case and is thus more readily cooled by these heat radiating surfaces. On the other hand, with a splash system — particularly on a low-speed unit — the oil may dribble over only a small internal surface of gear case and thus restrict heat dissipation. As a result, the splash-lubricated unit usually runs hotter and, therefore, requires a heavier bodied oil.

The lubricant in a circulation or splash-oiled gear set is subjected to very severe service, for it is thrown from the gear teeth and shafts in the form of a mist or spray. In this atomized condition, it is exposed to the oxidizing effect of air, and fluid friction, and in some cases metallic friction, generates heat which raises the oil temperature.

The violent churning and agitation of the oil by the rotating gears of splash-lubricated sets also increases temperature. This increased temperature speeds up oil oxidation which, however, can be minimized by using oils especially manufactured to resist this action.

Oxidation is a chemical action which takes place more rapidly when the oil is broken into a mist and thus offers a large surface to the attacking oxygen in the air. When oil oxidizes, it forms sludge: some oils have little resistance to this oxidizing action, i.e., are not chemically stable; and with such an oil in use, sludge particles soon start to form. These particles are soluble in the oil up to a certain limit (saturation point), which is higher for hot oil than it is for cold oil. Therefore, when the temperature of oxidized oil drops either in cooler weather or after a shutdown, some of the soluble products of oxidation may precipitate as sludge. If these particles deposit on the case wall, they form a heat-insulating film that interferes with cooling. This formation results in a further increase of oil temperature and is accompanied in turn by an increased rate of oxidation. Eventually, failure of lubrication and destruction of gears may result.

Some of these deposits may collect in idle spaces on the sides or in the bottom of the gear case, or deposits may collect as a sticky mass in oil pipes and oilways. When these deposits collect oil flow is restricted, and, if the gears receive insufficient lubrication, damage or destruction of tooth surfaces takes place. Further, if the bearings receive insufficient lubrication excessive wear may result and cause misalignment of the shafts and gears. This misalignment may impose excessive loads on gear teeth, and the loads may rapidly destroy their surfaces.

Too, water sometimes finds its way into the oiling systems of enclosed gears and may result from condensation of moisture in the atmosphere. In this case, it is often an indication of inadequate ventilation of the gear case and oil reservoirs. It is apt to occur in gear sets operated intermittently when warm periods of operation alternate with cool periods of idleness. This possibility of moisture contamination makes it necessary to use an oil of high demulsibility, i.e., an oil that separates quickly and completely from water.

Every effort should be made to exclude water from gear cases by preventing its entrance, by ventilating properly, and by removing the water with filters or dehumidifiers. If water is allowed to remain in a gear case, rusting may occur and consequently damage the gear-tooth surfaces.

Water and rust also act to hasten deterioration of the oil. Water separates slowly, or not at all, from oil that has been oxidized or has been contaminated with dirt. In this respect, iron rust is a particularly objectionable form of contamination, for water in severely oxidized or dirty oil usually forms a permanent emulsion. Such an emulsion may cause excessive wear of gears and bearings by creating deposits that restrict the amount of oil flowing through pipes and oilways to the gears and bearings. That oxidized oil promotes the formation of permanent emulsions constitutes another reason for re quiring an oil to maximum chemical stability in enclosed gears. Obviously then, to protect gear-tooth surfaces and bearings, an oil must not only separate quickly from water when new, but must also have the high chemical stability necessary to maintain a rapid rate of separation even after long service in a gear case.

Since only a very thin lubricating-oil film stands between efficient gear operation and failure, it is of great importance that the oil used should be carefully selected to meet service requirements. Reliable, economical lubrication can be attained only through the use of oils having proper characteristics, namely:

1. Correct Fluidity (Body): a characteristic which assures the distribution of oil to all rubbing surfaces and the formation of protective fluid films at the prevailing speeds, pressures and temperatures.
2. High Chemical Stability: a characteristic which (1) enables oils to resist the destructive oxidizing influence of continued circulation and agitation in the presence of air and (2) largely determines the durability of an oil and can be attained only in those oils especially refined for this purpose.
3. Good Demulsibility: a characteristic which assures quick separation of oil from water and, thereby, protects against the formation of emulsions which might be responsible for clogging of oil-ways and, thus, shutting off the supply of oil.
4. High Film Strength and Lubricity: characteristics required under conditions of boundary lubrication which reduce friction and protect against wear.
5. Anti-Rust Qualities: which protect against the formation of rust products that promote wear and increase the rate of oil oxidation.

In addition to the above, there is another characteristic that is necessary in the oil used for lubricating gears under extremely heavy operating pressures, namely:

6. Anti-Weld Properties: a characteristic of special EP oils which, on steel-to-steel gear teeth, act to prevent welding and destruction of rubbing surfaces.

GEAR TOOTH FAILURES

When two loaded surfaces slide over each other, as is the case with meshing gear teeth, there is always frictional resistance opposing the motion. Wear and loss of power result unless friction is minimized by interposing a film of lubricating oil between sliding surfaces. Such a film substitutes the flow fluid friction of the oil for the high metallic friction of dry surfaces.

A film of the oil-wedge type may be of sufficient thickness to completely separate rubbing metallic surfaces with a measurable and comparatively thick layer of fluid oil, i.e., hydrodynamic film. With such a film, friction is entirely due to the fluid friction within the oil which in turn depends on oil body (viscosity).

When an oil film between two surfaces is thinner than the combined heights of the opposing surface irregularities, metal-to-metal contacts will occur, a condition referred to as boundary lubrication. In this case, the protecting oil film is so microscopically thin that the oil ceases to act as a fluid and assumes a more rigid structure, because of the strong bonding action of the oil molecules with the surface molecules of the metal. Wherever this thin adsorbed film protects the metal, friction losses are fairly low, but nowhere are they nearly as low as when the film is thick enough to act as a fluid. However, at those points where the surface irregularities puncture the boundary film, the metal-to-metal contacts result in high friction losses.

Gears and their bearings are usually designed to function satisfactorily during the entire life of the machine of which they are a part, and their failure to do function satisfactorily indicates some unusual condition of operation. However, regardless of gear design, size of construction, correct lubrication is essential to assure minimum wear, quiet operation, and long service life. During manufacture, tooth shapes are very accurately developed; nevertheless, no matter how accurately and smoothly the teeth are finished, there will always be microscopic surface irregularities which cause frictional resistance if they are allowed to meet. With the gears in proper mechanical condition and with correct lubrication, friction is reduced; wear is practically eliminated; and gears should operate efficiently for years. However, even with correct lubricants properly applied, certain mechanical or operating conditions can cause wear and destruction of teeth. Destruction seldom takes the form of tooth breakage, but usually shows up as damage to the rubbing surfaces.

Such surface failure may rapidly destroy the original tooth contour, and, once this occurs, smooth, quiet operation can no longer be expected. Surface failure may be directly attributed to deficient lubrication, but more often lubrication is in no way responsible. In fact, even though the correct lubricant for normal service is in use, tooth surfaces may still be ruined by overloading, overheating, shock, abrasives, chips, improper alignment, loose bearings, or deflection of shafts or housing. These conditions lead to various types of surface failures.

To graphically describe the appearance of damaged tooth surfaces employed are many terms: pitting, abrasion, scratching, spalling, galling, scoring, scuffing, wiping, cutting, gouging, seizing, burning, burnishing, peening, rolling, ridging, rippling, fish-scaling and flaking. But none of these readily lend themselves to classification according to the fundamental causes of tooth damage. However, the first five listed above are the major types of failures under which all the other failures can be grouped as sub-divisions. Furthermore, it can be shown that these five types of failures have different basic causes. Of course, from the lubrication point of view, the cause of the failure, rather than the appearance of the surfaces, is of major interest. For the purpose of this publication, therefore, gear-tooth failures are classified under the headings of "pitting", "abrasion", "scratching", "spalling" and "galling".

Pitting

The tooth surfaces of new gears, although smooth to the eye, are actually comparatively rough and in addition to roughness, there may be variations in the hardness of the surface metal. Under conditions of boundary lubrication both roughnesses and variations in hardness cause uneven distribution of load across the tooth surfaces. As teeth pass through mesh, the load is concentrated on local high spots or high spots or hard spots. At each turn of the gear, the heavy localized stresses are repeated, and metal fatigue eventually occurs. As a result, minute particles break away and leave small pits where high points or hard spots have been (Fig. 71).

The type of pitting described ordinarily occurs only where there is a low ratio of slide to roll. Therefore,

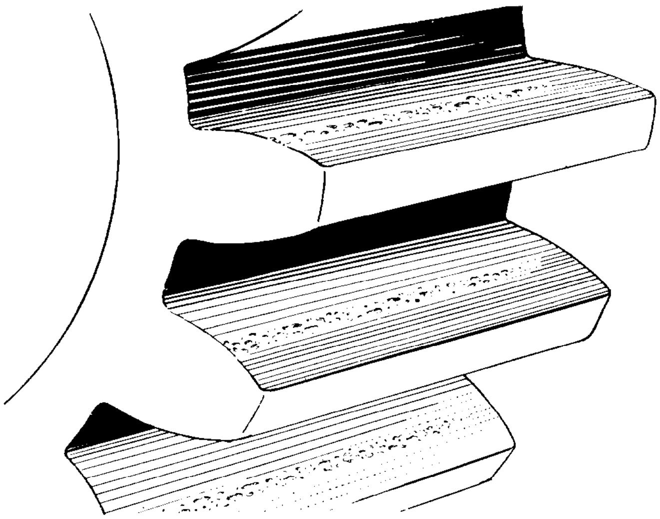


Fig. 71 . . . *Incipient Pitting.* Repeated stresses on the high or hard spots of gear teeth cause local fatigue failure of the metal. Small pieces or particles of metal break out at or slightly below the pitch line, leaving small craters or pits. After the high spots have broken out, further pitting may cease and normal wear may eventually polish out the pits.

in spur or helical gears, the action occurs at or near the pitch line, where sliding is at a minimum. On other areas of the tooth surface the sliding action wears away the high spots before pitting can occur.

Although pitting is not a lubrication failure, there is some experimental evidence that oil plays a part in the following manner. As a result of fatigue, microscopic surface fatigue cracks start, and these cracks become filled with oil; then, under the contact loads existing between the teeth, hydraulic pressure is developed. This pressure tends to extend the cracks and eventually to push out small particles of metal. This hydraulic action would occur with any fluid.

With spur and bevel gears, as each tooth passes through the center of mesh, the entire load is momentarily concentrated on the pitch line. If the area along the pitch line has already started to pit, the load is further concentrated on the remaining undamaged metal, and pitting is likely to increase progressively until the tooth surfaces are destroyed or severely damaged.

On the other hand, with helical, herringbone and spiral bevel gears, there is less likelihood of destructive pitting, because each tooth during mesh makes contact along a slanted line extending from root to tip. This line cuts across the pitch line, and, although pitting may have roughened the area along the pitch line, the line of contact always extends beyond this roughened surface; thus the load is carried on undamaged root and tip areas. Under such circumstances, pitting may cease as soon as the few, isolated high spots along the pitch line have been removed.

Pitting is more likely to occur on wide-than-narrow-faced gears because of the greater difficulty in obtaining true and uniform contact across the entire width of the wide teeth. After high points and hard spots break away, the load may be sufficiently distributed well over the rest of the tooth surfaces so that the gears may then operate without further damage. With correct lubrication, normal wear may ultimately polish the surfaces to a smooth, unbroken finish. This sort of pitting is called corrective or incipient pitting and is not serious.

When pitting becomes progressively worse (Fig. 72), a reduction of load on the gears will decrease the localized stresses and may prevent further destruction of tooth surface. Lubricants should not be expected to eliminate pitting, although, in some instances, a borderline case of pitting may be improved by using a heavier-bodied oil which tends to spread localized spot-loads over slightly larger areas. However, when continued pitting cannot be stopped, it becomes necessary to replace the offending gear with one better suited to the conditions.

Abrasion

Another type of surface destruction results when abrasive materials enter between meshing teeth. These abrasives may consist of dust or dirt from external sources, rust from within a gear case, or in some instances, metallic particles that have been pitted or scored from the gears. When gritty particles are larger than is the thickness of the oil film between the teeth, abrasion occurs and may result in a more or less uniform wearing away of tooth surfaces. Abrasion (Fig. 73) is usually characterized by numerous fine, short scratches that extend in the direction of slide on an otherwise smoothly worn surface. Naturally the rate of wear is greatly influenced by the hardness of the tooth surface and the nature of the abrasive material; in some instances, it may take place rapidly and destroy not only the surface but also the tooth shape.

Where the abrasive is of fine nature, the surfaces may show a high polish although excessive wear and destruction of tooth controur occur. On the other hand, with very coarse contaminating material the surfaces may be roughened and deeply scratched, or when the contaminating particles are metallic, the abrasion sometimes may take the form of dents impressed into the tooth surfaces as the particles are trapped between meshing teeth and are crushed or squeezed down to flat flakes.

Abrasion cannot be prevented by oil, but it can be controlled by eliminating the materials that cause

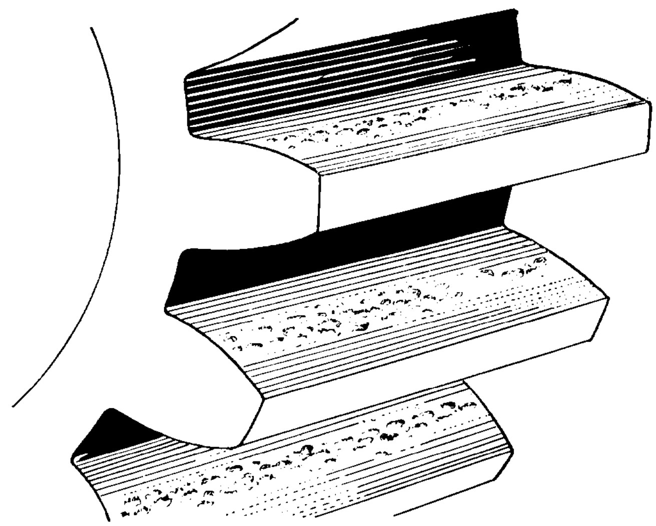


Fig. 72 . . . *Destructive Pitting.* Rough tooth surfaces may have many high spots or hard spots and may pit so badly that too much of the load-carrying surface is rendered ineffective. When this occurs, the increased loading of the remaining surface causes further pitting until the working areas are destroyed.

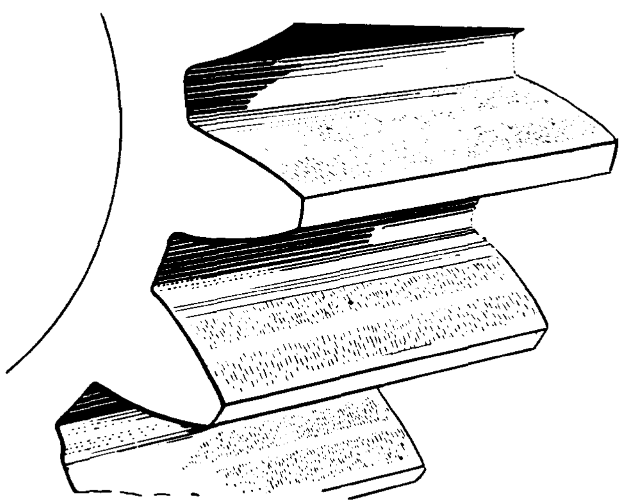


Fig. 73 . . . *Abrasion.* When foreign material of an abrasive nature enters between the meshing teeth, the resulting lapping or grinding action may either polish the surfaces or scratch them. In either case, there is abnormal wear.

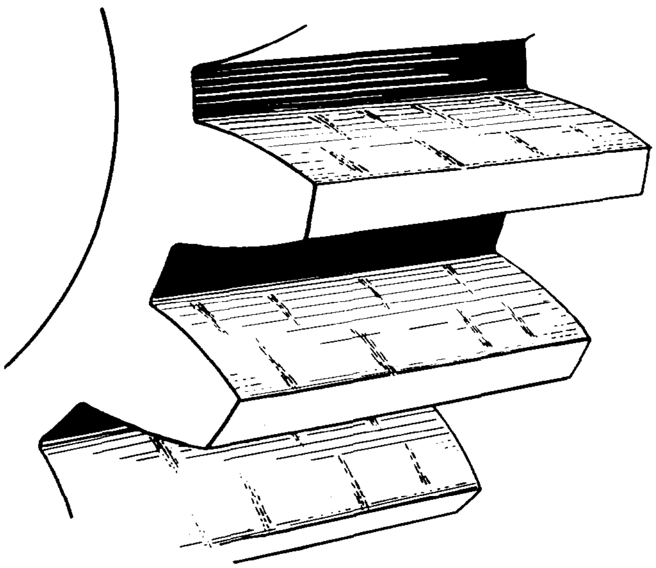


Fig. 74 . . . *Scratching.* When sharp projections on the surfaces of gear teeth pierce the oil film, they gouge or score the surface of the mating teeth. Rough finish, pitted surfaces, or misalignment may be the cause.

abrasion. For example, on gear sets lubricated by oil splash or by oil circulation, abrasion can be reduced and often prevented by systematic filtering of the oil to remove foreign abrasive material.

Scratching

Generally, scratching (Fig. 74) results from conditions unrelated to lubrication and occurs when an abnormal projection on a tooth is of sufficient height to puncture the oil film and scratch or gouge short furrows into the teeth of the mating gear. These teeth, in turn, may transfer similar damage back to other teeth of the original gear. Therefore, all teeth of both gears may eventually be scratched as the result of a single projection on one tooth; and abnormal projections such as those that would cause scratching may be the result of rough finish or of mechanical damage by metal carried through mesh. Roughnesses capable of causing scratching may, however, develop through other forms of surface failure such as aggravated pitting, spalliig or galling.

Spalling And Other Metal Failures

The condition known as spalling is a form of metallurgical fatigue failure that results in the removal of relatively large pieces of metal from gear-tooth surfaces. The surface metal of a meshing gear tooth tends to deform elastically under the transmitted load and to form waves which will roll ahead of and behind the line of contact. It is easy to visualize such rolling waves in the case of loaded rubber rolls, but similar action actually occurs when metal surfaces are involved, for the metal is subjected to tensile, compressive and shear stresses and the last reaches a maximum value at some distance below the surface. Ordinarily these stresses are within the limits contemplated by the designer; however, under some conditions, such as misalignment or over load, the subsurface stresses may exceed the endurance limit of the metal. As a result, subsurface fatigue cracks develop, and eventually pieces of metal break away from the surfaces and leave fairly large pits (Fig. 75). Spalling most often occurs in a relatively small "flank" area below the pitch line where, in this area of the pinion, surface stresses are highest because fewer teeth are in mesh. In addition, this same area is, because of beam loading, later subjected to high tensile stresses. With proper lubrication but with overloaded metal, the flanks of the teeth of one gear may be spalled while the faces of the teeth of the mating gear may be undamaged. Spalling is distinctly a metallurgical fatigue failure, and it results from the inability of the gear metal to carry the imposed load for the length of time intended by the designer.

Spalling, as previously described, commonly occurs with hard metal. Under similar conditions with soft metal, thin flakes may be rolled from the working surfaces and leave the teeth in fairly smooth condition. With this type of failure, which is often called flaking,

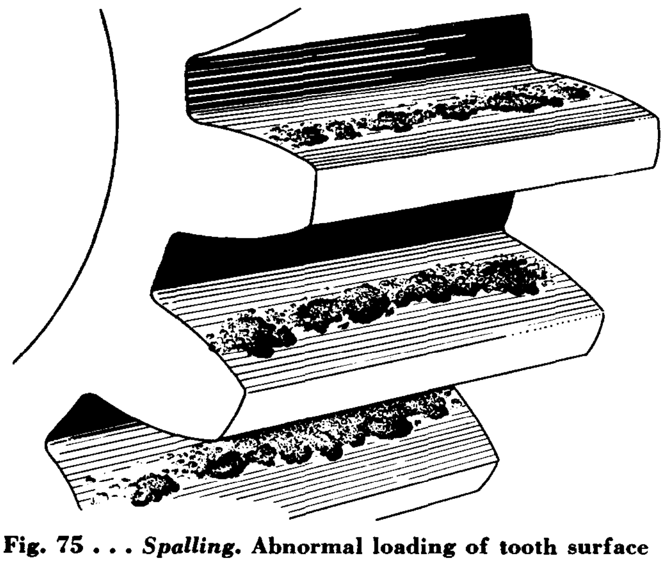
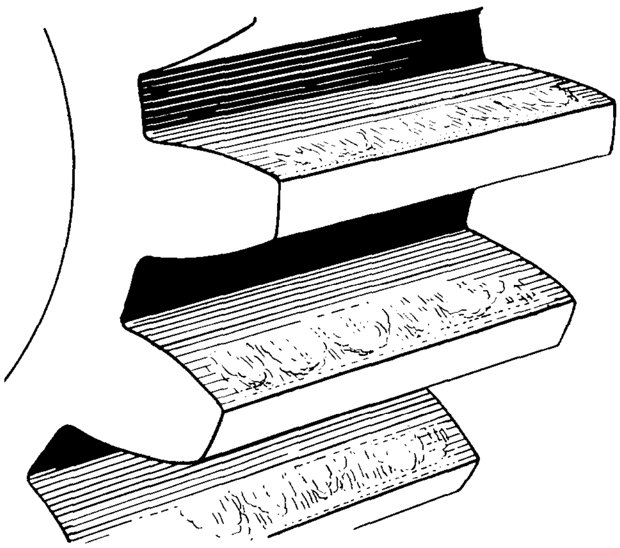
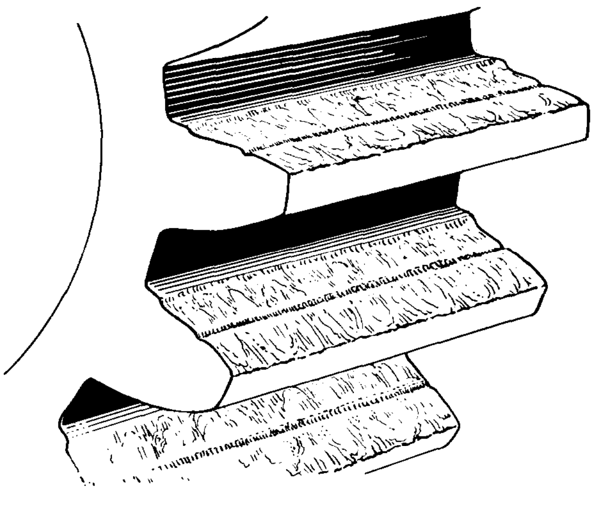


Fig. 75 . . . *Spalling.* Abnormal loading of tooth surface may overstress the subsurface metal until large chips or flakes break away from the teeth.



DRIVING PINION

Fig. 76 . . . *Mild Galling.* When full-fluid films fail, the first signs of wear occur above the pitch lines of the teeth. The teeth then usually show evidence of a yielding and sliding of the surface and subsurface metal. This yielding then progresses toward the tips of the driving teeth and below the pitch line of the driven teeth.



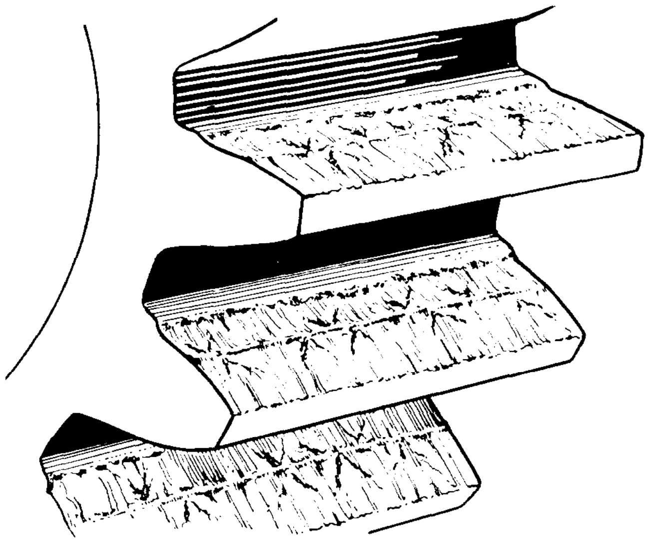
DRIVING PINION

Fig. 77 . . . *Advanced Galling.* Consistent failure of the lubricating film may cause not only localized yielding and displacement of the metal, but also a pressure-welding or seizure between the engaged surfaces. When such welding occurs, chips and scales of metal tear from the teeth, and the working surfaces become roughened. Scoring, abrasion and excessive wear then follow.

small flat flakes of metal may be visible in the oil. Sometimes, metal is rolled beyond the tooth tips to form a feather-edge. Again with soft material, heavy or shock loads may squash or hammer metal from tooth ends. This last type of failure is called peening. Metallurgical failure as well as abrasion and scratching may occur on any type of gear. These are faults that can, in most cases, be traced to mechanical or operating conditions, and lubricants should not be expected to prevent these types of gear troubles.

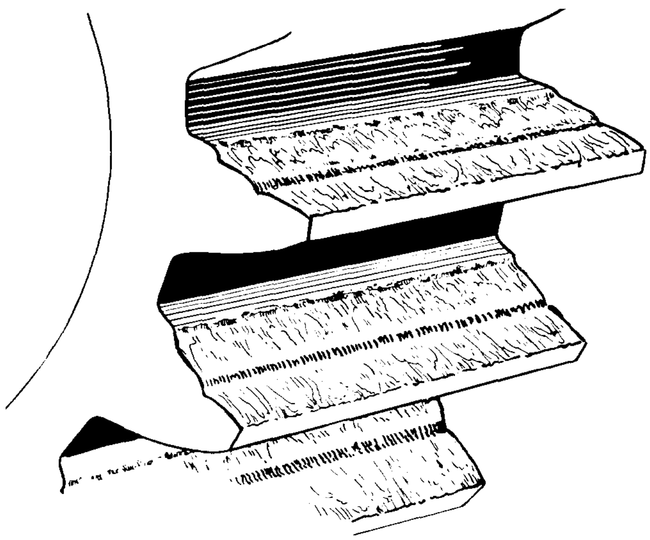
Galling

The term galling is used to indicate another type of failure that removes metal from tooth surfaces. In the milder or initial stages, this type of failure is very often referred to as scoring. Galling is definitely due to failure of the oil film to carry the load, either because the



DRIVEN GEAR

Fig. 78 . . . *Severe Galling.* The direction of slide on the teeth of a driven gear is always toward the pitch line. Thus, where simple galling occurs, the movement of metal is always toward the pitch line. Eventually this plastic flow of metal creates a hump or ridge across each tooth.



DRIVING PINION

Fig. 79 . . . *Severe Galling.* On a driving gear (pinion) the direction of slide is always away from the pitch line. Thus where simple galling occurs, the plastic flow of metal tends eventually to create a hollow or groove across the face of each driving tooth.

operating conditions were abnormally severe or because the oil was incorrectly selected. Under such circumstances, metal-to-metal contact occurs, and the tooth surfaces are worn, dragged, or torn in varying degrees. Excessive but smooth wear sometimes results, but, more commonly, the surfaces are left in a roughened condition.

In most gear sets, the loading of tooth surfaces is such that a condition of boundary (i. e., thin-film) lubrication results. When boundary films prevail, there is a limit of combined pressure (loading) and velocity (sliding) beyond which it is impossible to go without extensive metal contacts at the high spots. These contacts weld together, then break apart as meshing of the teeth proceeds; and this extensive welding and attendant dragging of metal is the mechanism of galling.

Because sliding is at maximum at tooth tips, boundary-film failure tends to occur first in the tip areas. In border-line situations, mild galling is these areas, or tip scuffing, may be the first indication of trouble.

Further, if operating conditions become more severe and the oil less able to maintain a boundary film, the entire working surface of the teeth may become galled (Fig. 77). Then extensive welding or seizure and plastic flow of metal occur, and surface destruction will extend from root to tip of both gears.

Sometimes when gear teeth are galled, metal is dragged over the tooth tips and creates a feather-edged appearance. Feather-edging, then, can result from lubrication failure as well as from the failure of soft metal. Therefore, when feather-edging is present, the cause of failure must be determined by examining other features of the tooth surfaces (Figs. 76, 77, 78, 79).

During the interval of approach, the direction of sliding on the contact surfaces is toward the pitch line on the driven gear and away from the pitch line on the driving gear. At the pitch line, the direction of slide reverses, so, during the interval of recession, it is still toward the pitch line on the driven gear and still away from the pitch line on the driving gear. Therefore, when surfaces gall, there is a tendency for metal to be wiped toward the pitch line on the driven teeth and away from the pitch line on the driving teeth. As a result, a ridge may develop at the pitch line of driven teeth and a groove at the pitch line of driving teeth (Figs. 78, 79).

Combination Failures

So far the various types of gear-tooth surface failure have been discussed as occurring separately, but this is not always the case in practice, since two or more types of failure may occur during the same period and, in fact, one type may be the cause of another type. For example, pitch-line pitting may progress to such an extent that overloads are created at the edges of the roughened areas, and, as a result, subsurface fatigue and, ultimately, spalling occur. In other words, pitting may lead to spalling. In other instances spalling and galling may be present and show both metal-fatigue and lubrication failure. Too, as noted previously, abrasion may be caused by metal particles that were spalled, pitted or gouged from tooth surfaces, and scratching may be preceded by severe pitting or galling. Combination failures must be carefully studied together and with mechanical and operating conditions in order to determine the most probable initial cause of failure.

BENEFITS OF CORRECT LUBRICATION

Successful operation of gears, either open or enclosed, depends largely on the completeness as well as the character of the lubricating oil films that are maintained between the meshing gear teeth. To resist rupture of the films when the meshing teeth slide over each other, the lubricant must not only be applied correctly, but must also be suited to the particular circumstances of load, speed, temperature, etc. In other words, correct lubrication implies the application of the right lubricant — at the right place and in the right way. The resulting benefits are reflected in four important factors:

1. Continuous production
2. Low maintenance costs
3. Low power consumption
4. Low lubrication cost.

The proper application of correct lubricants assures protection of the rubbing tooth surfaces, despite adverse conditions of speed, pressure, shock and temperature. Too, wear and tooth failure attributable to lubrication are minimized; reliability is enhanced; and the machinery depending on the gears is kept in continuous production.

Wear of gear teeth is minimized, and galling is prevented when protective films of oil are maintained on the tooth surfaces. In this way, correct lubrication greatly extends the useful life of gears. Moreover, the teeth retain their true contour; and smooth, quiet operation is assured. Under these conditions, repairs or replacements are minimized, and by contributing to smooth quiet operation and by extending gear life, correct lubrication materially reduces the cost of maintenance.

Correctly lubricated teeth slide in and out of mesh with minimum friction, hence, with minimum loss of power. Since the teeth maintain true contour and smooth surfaces, the gears transmit power efficiently, smoothly, uniformly and quietly. Furthermore, because of fluid friction resulting from churning of oil between the teeth or in the bottom of the gear housing, lubricants correctly selected for the operating conditions cause minimum power loss and minimum heating.

Thus, correctly selected lubricants, correctly applied, render efficient service for long periods, and low consumption and low oil cost, therefore, result. The cost of the oil, however, is not the real cost of lubrication. To measure the true overall cost of lubrication, the benefits of correct lubrication in terms of maximum reliability, low maintenance costs, and low power consumption should be contrasted with the penalties of incorrect lubrication in terms of shutdowns, reduced production, increased repairs and maintenance, increased depreciation, more frequent renewals, and unnecessary power consumption.

SUMMATION

1. All manufacturers of pumping units design from the same basic equations, and all design to a service factor of one.
2. There exists in the design of pumping unit gear reducers only one safety factor: that which assures a certain number of life cycles under conditions of uniform loading and perfect lubrication up to ten hr per day.
3. The conditions of operation greatly influence the wear life of gear reducers. It is possible to determine what effect the conditions of operation have upon the life of a reducer and in general to predict the wear life of a reducer under certain conditions of operation.
4. Preventitive maintenance by good conditions of operation can increase the life of gear reducers and cut repair costs.

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