621.833-72: 621.89

The Lubricant Churning Loss in Spur Gear Systems*

By Yasutsune ARIURA**, Taku UENO***, Teruo SUNAGA****,
and Shigemi SUNAMOTO☆

Due to the greater power loss caused by higher operating powers and speeds the operating temperatures of gear systems in use nowadays are higher causing the strength of tooth surfaces to decrease. Usually, the power loss consists of (1) the tooth friction, (2) the lubricant churning loss and (3) the bearing loss. In this paper, the churning loss is discussed. Essentially, lubricants are used for decreasing tooth friction and for dissipating the heat generated, and in order to know the best method of lubrication the amount of churning loss and its cause should be determined.

In the present experiment, the churning loss of a spur gear system with jet lubrication is measured, and several analyses are proposed. From the results thereof, it becomes clear that the churning loss consists of the power required to trap the oil between the mating gears and to accelerate the oil in the tooth space. And it is considered that at high speeds the churning loss does not become so large regardless of the viscosity of the lubricating oil.

1. Introduction

In gear systems, the power loss usually consists of: (1) the tooth friction, (2) the lubricant churning loss, and (3) the bearing loss. All of these losses change into heat, and they dominate the temperatures of all the parts and lubricant under the cooling effect of the surroundings and lubricant. Nowadays, due to the greater power loss caused by higher operating powers and speeds, the operating temperatures of gear systems in use become higher and as a result, the oil film thickness between the tooth surfaces decreases as its viscosity decreases. After all, these phenomena cause the tooth surfaces not only to be scored, but also to become weak in the fatigue strength which is concerned with pitting.

Among the power losses abovementioned, (1) the tooth friction and (3) the bearing loss are frictional and they have been discussed many times before. And these losses may decrease to a certain degree with the improvement of machining accuracy and the proper lubrication. But, as for

(2) the lubricant churning loss, there seems to have been few publications on this and this has not been yet clarified enough. Essentially, lubricants are used for decreasing tooth friction and for dissipating the heat generated, but using the oil with high viscosity and supplying a great amount of oil for these purposes, on the contrary, cause a larger lubricant churning loss, resulting in a temperature rise.

Therefore, in order to select the lubricant and its amount for increasing the load carrying capacity of gears, the amount of these power losses must be estimated. Recently a method⁽¹⁾ is applied of considering the whole gear system as a thermal network which consists of heat sources and heat resistances and estimating the temperature at all the parts and lubricant of the system, and for this purpose the intensity of heat sources, that is, the amount of power losses must be known.

In the present paper, the lubricant churning loss is discussed. On this question, only a few papers^{(2)~(4)} have been reported in the case of oil bath type lubrication before, but almost all of them have been discussions mainly on the basis of the experimental works.

From the observation of the running gears by means of a stroboscope, the causes of the lubricant churning loss are considered as follows:

(1) the power required to accelerate the oil in the tooth spaces,

* Received 8th May, 1971.

** Lecturer, Faculty of Engineering, Kyushu University, Higashi-ku, Fukuoka.

*** Professor, Faculty of Engineering, Kyushu University.

**** Associate Professor, Faculty of Engineering, Kyushu University.

☆ Seikosha Co.

(2) the power required to trap the oil between the mating gear teeth and to deform it..... in this paper this power loss will be called oil trapping loss,

(3) the power to splash the oil in the oil bath (especially in the case of oil bath type lubrication),

(4) the power to bend the oil flow by gear teeth when oil goes into the meshing zone (especially in the case of jet lubrication)

(5) the power to induce an air draft near the circumference of running gears.

In the present experiment, the amount of churning loss of the spur gear system with jet lubrication is measured, the effect of gear size, the viscosity and the rate of oil flow on the amount of loss are investigated, and further, several analyses are proposed by considering a few causes abovementioned.

2. Experiment

2.1 Test apparatus and method

The test apparatus is shown in Fig. 1. Two test gears are mounted on the one-supported shafts and the torque of the driving shaft is measured by means of a torque meter which consists of strain gauges and slip-rings. In the high speed test, the bearings and supportings of their shafts are designed to be sufficiently stiff, and a fly-wheel is mounted on the driven shaft so that the vibration does not occur because of pitch errors and profile errors of gears. The state of lubrication of gears can be observed through the transparent plate in front of the gear box. The temperature of lubricant is controlled to be constant by means of a heater and a radiator, and its temperature

Fig. 1 Outline of test apparatus

just before oil jet is supplied can be checked by a thermocouple inserted into the nozzle. And these experiments are made under no load, as the lubricant churning loss is considered to have little to do with the magnitude of load.

In the range of low speeds, the churning loss is measured in such manner that the bearing loss which has been measured previously is subtracted from the total loss when some amount of oil is supplied. Therefore, as the loss measured includes the tooth frictional loss, the latter is estimated in another way. And in the high speed range, the power losses with some amount of oil supplied and with no oil (but only a few droplets) are compared when the mating gears are running at a constant speed. This method is reasonable considering that only a little oil is sufficient for the formation of an oil film on tooth surfaces, and that the tooth frictions in both cases are equal whether a large amount of oil is supplied or not, because these experiments are made under no load. Figure 2 shows that the total loss of torque, which consists of bearing loss, tooth frictional loss and lubricant churning loss, is measured at a few constant speeds when decreasing the rate of oil flow, and from this figure the total loss approaches constant values near no oil flow. Therefore, as the loss at no oil flow is considered the sum of the loss of bearings and tooth friction, the increment of the loss when some amount of oil is supplied represents the lubricant churning loss.

2.2 Test condition

Two test apparatuses are used for both low speed and high speed ranges, but they are the same basically. The dimensions of gears used for both tests are indicated in Tables 1 and 2. The

Fig. 2 Relation between rate of oil flow and various losses (An example of module 2.0 and No. of teeth 62)

material of gear blanks is S55C (H_B 200) and they are hobbled carefully. Both driving and driven gears have the same number of teeth. Table 3 shows the lubricants used and their viscosities. The size of gear box (for high speed range), the position of jet and the direction of rotation of gears are illustrated in Fig. 3. The rate of oil flow ranges from 10 to 200 cc/sec and the diameter of the nozzle exit is 6 mm.

2.3 Test result

Examples of the result for low speed range

Table 1 Dimensions of test gears for low speed

Kind of gears	I	II	III	IV
m (Module)	2.5	4.0		8.0
z (No. of teeth)	61	38		19
b (Face width)mm	40	40	20	40

α (Pressure angle) are all 20° .

Backlash can vary by changing center distance.

Table 2 Dimensions of test gears for high speed

Kind of gears	V	VI	VII	VIII
m (Module)	3.0		5.0	
z (No. of teeth)	42		25	
b (Face width)mm	20	40	20	40
Backlash mm	0.12	0.11	0.15	0.16

α (Pressure angle) are all 20° .

Center distance 125.0 (fixed)

Table 3 Kinds of lubricants

Kinds of lubricants	Viscosity (Temperature) cSt $^\circ\text{C}$	
Mobil DTE Oil Extra Heavy	280	(23)
Mobil DTE Oil Heavy Medium	36	(50)
	73	(35)
	124	(25)
Cylinder oil 680W	330	(40)

Fig. 3 Size of gear box and direction of rotation of gears

(within 24 m/sec of pitch line velocity) are indicated in Figs. 4 and 5, and for high speed range (from 13 to 52 m/sec) in Figs. 6~9. The ordinate indicates the value after the bearing loss is subtracted from the total loss. Figures 6 and 7 indicate the difference of viscosity, Figs. 7 and 8 the difference of the face width, and Figs. 6 and 9 the difference of module ($m=5.0$ for Fig. 9).

From these figures, the following facts are recognized:

(1) In a certain range of low speeds (about 5 to 10 m/sec), the lubricant churning loss has a peak, and this phenomenon is more obvious as the module is small. And it seems to be concerned with the viscosity of oil (c.f. Figs. 4 and 5, and Figs. 6 and 7).

(2) In the high speed range (about 30 to 50 m/sec), the churning loss increases as the speed increases, and it has nothing to do with the module and the viscosity of oil but it is approximately

Fig. 4 Lubricant churning loss

Fig. 5 Lubricant churning loss

proportional to the speed (c.f. Figs. 6, 7 and 9).

(3) The higher the rate of oil flow, the larger the churning loss is.

(4) The larger the face width, the larger

the churning loss is (c.f. Figs. 7 and 8).

And also by means of the stroboscope, the following are observed:

(5) When oil is supplied at more than a certain jet speed from the nozzle, all of the oil goes into the meshing zone. Otherwise, that is, if the jet speed is not so large, the oil is often blown off by the air draft induced by the running gears and does not reach the meshing zone.

(6) When the teeth of both gears mesh again after one revolution, they still have rather a large amount of oil on their surfaces.

3. Analyses and discussion

In the introduction, some causes of churning loss are mentioned, but in the analyses, mainly the power required to accelerate the oil and the power required to trap the oil between the mating gear teeth are considered.

3.1 The power required to accelerate the oil in the circumferential direction

This power loss can be considered from the view point of momentum change. As indicated in Fig. 3, assuming that the oil supplied at the jet velocity v from the nozzle placed just over the meshing zone goes into the clearance of the zone and that all the amount of oil is flung off at the pitch line velocity U in the circumferential direction, the force acting on oil F , which is transmitted from the gears, is given as follows according to momentum change:

$$F = \rho Q(U - v) \quad (1)$$

where, ρ is the density of oil, and Q the rate of oil flow supplied. If the diameter of the nozzle is d , $v = 4Q/(\pi d^2)$, and hence the torque is

$$T = \frac{\rho Q D}{2} \left(U - \frac{4Q}{\pi d^2} \right) \quad (2)$$

Fig. 6 Lubricant churning loss

Fig. 7 Lubricant churning loss

Fig. 8 Lubricant churning loss

Fig. 9 Lubricant churning loss

where D indicates the pitch circular diameter.

The dotted lines in Figs. 8, 10 and 11 indicate the results calculated from Eq. (2) and they are compared with the experimental results. In Fig. 10, the churning losses measured agree well with the calculated values by Eq. (2), and therefore in this case it can be considered that the churning losses are caused mostly by the power required to accelerate the oil. And Fig. 8 shows the result under the same condition as Fig. 10 besides the modules, that is, in Fig. 8 the module is smaller, and also in this case the losses nearly agree with the values calculated by Eq. (2) in the high speed range, but in the low speed range the phenomenon of oil trapping occurs, which will be treated in the next section.

Further, Eq. (2) indicates that the power of acceleration is independent of module, face width and the viscosity of oil, but it depends upon the rate of oil flow Q if the pitch diameter D and the nozzle diameter d are constant. But when comparing Figs. 10 and 11, it seems that the face width influences the churning loss undoubtedly and

Fig. 10 Lubricant churning loss

Fig. 11 Lubricant churning loss

that it can not always be explained simply by Eq. (2).

3.2 The power loss required to trap the oil in the meshing zone

By using a stroboscope, it is observed how oil is pressed out from both sides of the meshing zone at considerable speed (Fig. 12). This is due to the oil being compressed between the teeth in the meshing zone of both gears because of the change of the area trapped between them. Therefore, the work done by this process is the power loss.

The change of meshing is so complicated that the actual analyses are almost impossible. Then, by considering a very simple model as shown in the next subsection, some analyses will be made and compared with the experimental results.

3.2.1 Model and basic equations

Assuming that a plate with the width h is placed in a rectangular channel having the same width and that a certain amount of oil exists between them, as shown in Fig. 13, the viscous resistance, $P' (=2P)$, which acts on the plate, may be calculated when it is forced to move vertically at a certain speed. And in the calculation the thickness of oil w , that is, the distance between the plate and the bottom of the channel is assumed to be small, because the larger the viscous

Module 4.0, No. of teeth 38, Face width 40mm, Revolutions 300rpm, Rate of oil flow 120cc/sec, Viscosity of oil 280cSt (Photograph by using stroboscope, exposure time 1/40 000sec)

Fig. 12 Spout of oil from the sides of meshing zone

Fig. 13 A model of trapping in tooth groove

resistance concerned, the smaller the thickness is. Further, it is assumed that oil is a perfectly viscous fluid, its viscosity is constant everywhere because the temperature is constant, the boundary between oil and air remains plane and the effect of both sides of rectangular channel on the resistance is neglected.

Under these assumptions, considering the equation of the balance of force and the continuity of oil^(*), the pressure distribution of oil is given by solving the following equation:

$$\frac{d^2p}{dx^2} = \frac{12\eta}{w^3} \frac{dw}{dt} \quad (5)$$

where p indicates the pressure at any place of oil, w is the thickness of oil as shown in Fig. 13, t is time, and η is the viscosity of oil.

Considering that its distribution is symmetrical to the central plane, Eq. (5) may be integrated under the condition that $p=0$ at $x=l$ as follows:

$$p = \frac{6\eta}{w^3} \left(-\frac{dw}{dt} \right) (l^2 - x^2) \quad (6)$$

where, $2l$ is the length of oil which spreads in the channel at a certain time t .

Therefore, the total pressure P along the half channel ($x \geq 0$) is given as follows:

$$P = h \int_0^l p dx = \frac{4\eta h l^3}{w^3} \left(-\frac{dw}{dt} \right) \quad (7)$$

and, further the work, E , which is done by oil is as follows:

$$E = \int_{w_i}^{w_e} P(-dw) \quad (8)$$

where, w_i and w_e are the thickness of oil when oil begins to be trapped and that when trapping finishes respectively. The area trapped in the meshing zone of the actual gears once becomes minimum and then increases again, but in the remaining half period, the work concerned is neglected because the tooth space has a little oil.

And as the face width b of the actual gears is finite, the two processes must be considered that the face width of the tooth space is filled with the oil supplied, that is, $2l=b$, and, after that, oil is pressed out from both sides of the meshing zone of gears. Therefore, assuming that

(*1) As the shearing stress of τ is given by $\tau = \eta(\partial u / \partial y)$, the equation of motion to the small element of oil is as follows:

$$\frac{\partial^2 u}{\partial y^2} = \frac{1}{\eta} \frac{dp}{dx} \quad (3)$$

and, the equation of continuity is given as follows:

$$-\frac{dw}{dt} = \int_0^w \frac{\partial u}{\partial x} dy \quad (4)$$

Therefore, by integrating Eq. (3) with the boundary conditions ($u=0$ at $y=0, w$) and substituting this equation into Eq. (4), Eq. (5) can be obtained.

the amount of oil $2V$ is supplied in the channel, and that the thickness of oil is w_0 just when that amount spreads over the length b of the channel, from Eqs. (7) and (8) the viscous work acting on oil is given as follows:

$$E = \frac{4\eta V^3}{h^2} \int_{w_i}^{w_0} \frac{1}{w^6} \left(\frac{dw}{dt} \right) dw + \frac{\eta h b^3}{2} \int_{w_0}^{w_e} \frac{1}{w^3} \times \left(\frac{dw}{dt} \right) dw \quad (9)$$

3.2.2 The application to actual gears

First, the thickness w may be expressed as trapped area S of one tooth space in the meshing zone. The backlash is small, so oil cannot move through it but only flows in the axial direction. Therefore, each groove works independently. In this connection the paper⁽⁵⁾ shows that in trapping oil of gear pumps, their gears should be considered to have no clearance between their teeth hydrodynamically even when having usual backlash in reality or rather pressure difference of oil. Therefore, the assumption abovementioned may be considered reasonable because the oil in the usual gear sets can move more easily. Area S is measured as a parameter of revolution angle θ (deg) of gears by engaging both gears on a special apparatus which can set actual center distance and by enlarging them with a projector. As it is known that the correlation of S and θ is parabolic theoretically, the coefficients of the following equation are approximated by measuring areas,

$$S = S_{\min} + a\theta^2 \quad (10)$$

where $S_{\min}$ and a are coefficients. Figure 14 indicates the correlations of S and θ of the gears used in the experiment. By using this area S , the oil thickness w is given as follows^(*):

$$w = S/h \quad (11)$$

Then, h corresponds to the width of trapped area between teeth in the meshing zone, and it must be the effective space width when oil is trapped. In actual gears, the space width varies with meshing position, so h is approximated as follows. As shown in Fig. 15, the space width when trapped area becomes minimum (the position of follower II) is $h_2 (\overline{A_2 B_2})$, and the position when trapping starts is assumed to be the place where $\overline{A_1 C}$ is equal to the backlash just before both working surfaces begin to contact (the position of follower I) and at that position the space width is

(*2) As h is assumed to be constant as described later, from Eqs. (10) and (11) dw/dt is given as follows:

$$\frac{dw}{dt} = \frac{2a}{h} \theta \frac{d\theta}{dt}$$

where $d\theta/dt = -180\omega/\pi$ and ω is angular velocity of gear.

$h_1 (\overline{A_1 B_1})$. Then, it is assumed that h can be given as follows:

$$h = (h_1 + h_2)/2 \quad (12)$$

The viscous work of oil abovementioned (E) can be converted to the loss of torque T as follows (*3):

$$T = 2Ez/\pi \quad (13)$$

where z is the number of teeth.

3.2.3 The comparison with measuring values

Figures 16~19 indicate examples of these calculations with measuring results. In these figures, measured values contain the tooth friction. Therefore, it is estimated by considering the rolling friction of two cylinders assuming that the oil thickness between tooth surfaces is equal to half of backlash because of tests being done under no load (*4), and the calculation in these figures indicates the sum of the loss in accelerating the oil (described in the last subsection), trapping loss, and tooth friction loss.

Fig. 14 Trapped area

(*3) To convert the energy into the loss of torque, they are easily considered in the form of the power. Therefore, the power for the whole face width and for the whole tooth spaces is given as follows:

$$T\omega = 2E \times 2z \cdot \frac{n}{60}$$

where ω and n are respectively angular velocity and revolutions per unit time (rpm). By the expression $\omega = 2\pi n/60$, Eq. (13) can be obtained.

(*4) When two cylinders with radii of curvature R_1 and R_2 rotate at speeds U_1 and U_2 respectively and with oil film thickness h_{min} , the viscous friction forces per unit width of these cylinders, F_1 and F_2 , are given as follows (Soda, N., Bearings (in Japanese) (1964), p. 80):

$$F_{1,2} = 3.254\eta U \sqrt{\frac{R}{h_{min}}} \left(1 \pm 0.619 \frac{U_1 - U_2}{U} \right)$$

where $U = (U_1 + U_2)/2$ and $1/R = 1/R_1 + 1/R_2$

Assuming that h_{min} is half of backlash and considering the contact ratio ϵ , the torque on pitch circle is given as follows:

$$T = \epsilon b (F_1 R_1 + F_2 R_2)$$

Figures 16~18 indicate the loss of torque for module 2.5, and Fig. 19 for module 4.0. Figures 16 and 17 show the difference of the amount of oil, Figs. 16 and 18 the difference of backlash and Figs. 16 and 19 the difference of module, where only the effect of backlash is not indicated, be-

Fig. 15 Decision of width of tooth space (Observed on driving gear)

Fig. 16 Comparison of measured and calculated values

Fig. 17 Comparison of measured and calculated values

cause the backlash varies by changing the center distance and at the same time the trapping area varies.

These figures show the following:

(1) All the calculated values of the loss are smaller than the measured values.

(2) All the peaks of calculated values appear at a lower speed than those of measured values.

(3) The larger the module and backlash are, and the less the rate of oil flow is, the larger the differences between the calculated values and the measured values are.

As a result, there are some differences between the calculated and measured values, and as for the gears used in the present experiment the following conclusions can be drawn.

(1) Within a certain speed range (10~15 m/sec) the churning loss is almost caused by the power required to trap the oil, and the peak of the loss indicates that trapping is most effective.

(2) Over a certain speed, the effect of trapping becomes smaller, and then the loss consists of oil acceleration and tooth friction.

Fig. 18 Comparison of measured and calculated values

Fig. 19 Comparison of measured and calculated values

Concerning these conclusions, it may be considered that the differences between measured values and calculated values are due to the following reasons:

(1) In the calculation, the complexity of the actual tooth space is not considered, and the assumption that the thickness of oil w is small does not hold when the module is large.

(2) The estimation of the tooth friction is insufficient.

(3) According to the observation with a stroboscope, the teeth go into the meshing zone with oil on the tooth surfaces after one revolution, so the amount of oil into the meshing zone actually increases and, therefore, the peak of calculated value will get larger, and move at a higher speed range.

(4) When oil from the nozzle goes into the meshing zone, the oil is bent by the teeth (Fig. 20).

Among the reasons given above, the oil clinging to the tooth surfaces after one revolution will be treated in the next section.

3.3 The effect of the oil clinging to the tooth surfaces on the trapping

Now, we define the following equation:

$$\phi = Q/q \quad (14)$$

and ϕ is named "the sufficiency of oil" in the tooth space in meshing zone. In this equation, Q is the amount of oil supplied per unit time, and q is the amount of oil which the tooth spaces can carry in the meshing zone and it is given as follows:

$$q = 2S_{\min} bzn/60 \quad (15)$$

Module 8.0, No. of teeth 19, Face width 40 mm, Revolutions 440 rpm, Rate of oil flow 40 cc/sec, Viscosity 280 cSt (Photograph by using stroboscope, exposure time 1/40 000 sec)

Fig. 20 Bending the oil by gear teeth

where S_{min} is the minimum area of trapping in a tooth space, and n is the revolution per unit time of the gear (rpm).

By this definition, when $\phi=1$, the trapping volume is just filled up with the amount of oil supplied. And when $\phi>1$, it means that an excessive amount of oil spouts from both sides of gears in the meshing zone.

The dotted line in Fig. 21 shows a calculated example of gears with module 3.0, number of teeth 42, and face width 40 mm. This curve indicates that the spout of oil does not occur at 13 m/sec (2000 rpm). Besides the amount of oil supplied, the increment of oil must be added because the oil clinging to the tooth surfaces goes into the meshing zone after one revolution. The oil clinging to the tooth surfaces is very effective to form an oil film between the tooth surfaces, but it has much influence on the trapping of oil.

In order to estimate this increment of oil which resists the centrifugal force and clings to the tooth surfaces, the equation of the oil film thickness which the authors have obtained before is applied. This equation indicates the oil film thickness on the tooth surfaces after supplying oil to gears and is a function of time, and it is assumed that it can be applied to the time of one revolution of gears.

By this equation, the oil film thickness on the tooth surfaces of a spur gear is given as follows [c.f. Eq. (32) in the reference (6)]:

$$T_H = 1.4\sqrt{\nu/\omega^2 t z} \quad (16)$$

where ν is the kinematic viscosity, ω is the angular velocity of gear, and t is the time after supplying oil, and therefore $t=2\pi/\omega$ after one revolution of gear. And then assuming that oil film with the thickness T_H clings to both tooth surfaces, the increment of volume per unit time ΔQ is given as follows:

$$\Delta Q = 4HbT_H zn/60 \quad (17)$$

Fig. 21 Sufficiency of oil in tooth space of meshing zone

where H is the whole depth of gear.

And, we redefine the sufficiency of oil as follows:

$$\phi = (Q + \Delta Q)/q \quad (18)$$

and show it by a full curve in Fig. 21. This curve shows that ϕ is nearly equal to 1.0 at the velocity 13 m/sec, and then it can be considered that a spout of oil from both sides of gears occurs at this speed. Judging from the fact that the oil spouts also at higher velocity, the above estimated increment is insufficient yet. But considering that the increase of oil viscosity increases the sufficiency ϕ as indicated by Eq. (18) and that the increase of pitch line velocity decreases ϕ , the state of trapping can be predicted rather quantitatively by ϕ .

3.4 The arrangement in high speed range by the dimensional analysis

In this section we will try to arrange the experimental data in high speed range by the method of dimensional analysis.

Now, eight physical properties are considered. They are the loss of torque T , module m , number of teeth z , face width b , and pitch line velocity U to gears concerned, and density ρ , kinematic viscosity ν , rate of oil flow Q to oil concerned.

Fig. 22 Relation of T and QU

Fig. 23 Relation of T and QU

Using these properties, the following result is obtained by means of the dimensional analysis:

$$\frac{T}{\rho \nu^2 m} = f\left(z, \frac{b}{m}, \frac{mU}{\nu}, \frac{Q}{m\nu}\right) \dots\dots\dots (19)$$

where f indicates a certain function. To a certain pair of gears, that is, fixing m , z and b , in order to investigate the effect of U and Q on T , from the measured data the following equation can be derived in the high speed range:

$$\frac{T}{\rho \nu^2 m} \propto \left(\frac{U}{\nu}\right)^{1.0} \left(\frac{Q}{\nu}\right)^{1.0} \dots\dots\dots (20)$$

As ρ is almost constant in the range of these experiments, it is reduced to:

$$T \propto QU \dots\dots\dots (21)$$

therefore, the loss of torque T is proportional to the product of Q and U . In Fig. 22 ($m=3.0$) and Fig. 23 ($m=5.0$), the measured values are re-plotted by taking QU as abscissa and T as ordinate, and as a result they show that Eq. (21) is applicable in the high speed range. And in the case of the same face width, both data of module 3.0 and 5.0 can be arranged by the same equation in the high speed range.

In the case of module 3.0 (Fig. 22), the separation from Eq. (21) in the low speed range indicates the phenomenon of trapping abovementioned, and therefore, other physical properties (viscosity etc.) are concerned with this phenomenon.

And also assuming that the oil jet speed v can be neglected compared with the pitch line velocity U in Eq. (2) which is described in Section 3.1, Eq. (2) has the same form of Eq. (21) qualitatively.

4. Conclusions

In the case of jet lubrication in spur gear systems, the lubricant churning loss is mainly caused by two phenomena as follows:

(1) Trapping of the oil in the tooth spaces

This phenomenon dominates in the low speed range, and has a considerable peak in the loss of torque in the case of gears with small module. By the analysis in which tooth space is replaced by a simple model, this phenomenon can be explained though not perfectly. Further, the oil clinging to the tooth surfaces after one revolution must be considered in this phenomenon.

(2) The acceleration of oil by the gear teeth

This dominates in the high speed range, and

the loss is almost proportional to the pitch line velocity. By the analysis based on the momentum change of oil, the relation of loss with other variables is established but the experimental data can not always be explained completely only from this relation. And by the method of dimensional analysis the loss of torque can be arranged by the product of the rate of oil flow and the pitch line velocity.

From the results abovementioned, it is now clear that in the high speed range the loss of trapping which seemed to be large is rather small, and that almost all of the loss in the high speed range is caused by the acceleration of oil. On the contrary, the trapping becomes more controversial in the low speed range. For example, if attention is paid to starting a gear system when the oil viscosity is high at low temperature, the characteristics of lubrication is expected to be better in the high speed range. But it is necessary to pay some attention to the loss caused by bending the oil flow in the high speed range.

And, when oil jet is supplied with a certain rate of flow from the entry of meshing zone in high speed range, almost all of the oil enters the meshing zone.

Essentially, the lubricant churning loss abovementioned must be compared with other losses in the gear system. Judging from the experimental data, the churning loss may become larger under the conditions of pitch line velocity, the viscosity of oil and the rate of oil flow supplied. And even if the churning loss is smaller than the tooth friction, much attention must be paid to it when gear systems are designed as the churning loss changes into a part of heat generated.

References

- (1) Blok, H., *The Present Status of the Theory of Gear Lubrication*, (Lectured in Japan in 1968).
- (2) Blok, H., *De Ingenieur* (Holland), Vol. 63, No. 39 (1951), p. 53.
- (3) Niemann, G. und Ohlendorf, H., *VDI-Z*, Bd. 102-6, Nr. 6 (1960-2), S. 216.
- (4) Fujita, K. and Miyanishi, K., *Jour. Japan Soc. of Lub. Engrs.* (in Japanese), Vol. 13, No. 12 (1968-12), p. 680.
- (5) Ishibashi, A. and Muta, T., *Trans. Japan Soc. Mech. Engrs.* (in Japanese), Vol. 33, No. 251 (1967-7), p. 1155.
- (6) Ueno, T., et al., *Trans. Japan Soc. Mech. Engrs.* (in Japanese), Vol. 34, No. 265 (1968-9), p. 1589.

Discussion

M. KAWAMURA (Nagasaki University)

(1) Some good results have been obtained in these experiments under no load condition. It is considered that the state of the oil under any load may be almost the same as that under no load, but as the complicated factors like the deformation may be concerned with this phenomenon, is it not better to test under some load by a proper device and compare them in order to investigate the serious failures of gear such as pitting and scuffing?

(2) From data in Fig. 2, the churning loss in the low speed range becomes large with an increased oil flow, and it can not be neglected compared with the losses of the tooth friction and the bearings. In this connection, are they considered to be constant? Why has such a fact been overlooked before? Are there any other causes of the difference between the measured and calculated value in Figs. 16~19?

(3) It may be considered that less amount of oil can enter the meshing zone, as the pitch line velocity gets larger, and further, these experiments show that only a more pressurized oil jet is insufficient in preventing the noise (which may be caused by another reason) and the failure of gear systems, and that more oil must be supplied and be forced to enter the meshing zone, or another oil supplying device must be prepared.

A. TOYAMA (Tokyo Institute of Technology)

(4) In the case of test with extremely large backlash in the low speed range, what result will be obtained?

(5) In Figs. 16~19, the experimental and calculated values are compared including the loss of tooth friction. I think these comparisons should be made between the values which do not include the loss of tooth friction for the following three reasons:

(a) The loss of tooth friction is not the main subject of this study.

(b) The method of calculation for estimating the tooth friction is inadequate.

(c) From the result in Fig. 2, the losses of tooth friction and bearing may be separated from all the losses in the low speed range.

Y. TERAUCHI (Hiroshima University)

(6) As in the high speed range the phenomenon of tooth beating appears easily under no load or a small load, can it be considered that the loss of torque by compressing and spouting the oil in the tooth spaces increases at a particular

speed? For example, it seems that T increases near 30~40 m/sec in Figs. 7 and 8, and near 32 m/sec in Fig. 9.

Authors' closure

(1) As it is pointed out, it is better to test under some load, but then it is difficult technically to measure the loss of bearing and tooth friction because they are much influenced by the load. On the other hand, as the lubricant churning loss may be little influenced by the load, it can be measured easily under no load. Therefore, in present experiment we assume that the effect of deformation is secondary, and we do not consider it in the lubricant churning loss. But in the near future we intend to test under some load in order to examine the ratio of the churning loss to other losses and to relate it with the failures of gears.

(2) Figure 2 indicates whether the measuring method in the present test is proper or not. Under no load it is considered that the losses of bearing and tooth friction are constant, and as a result the lubricant churning loss is obtained by subtracting them from the total loss when supplying some amount of oil.

As the churning loss has never been separated from the losses of tooth friction etc. before, we consider that the phenomenon such as in the observation with a stroboscope has not been grasped.

In Figs. 16~19, the amount of oil clinging to the tooth surfaces is not taken into consideration in the calculation of trapping, but it really must be considered, and then a more complicated calculation will be needed.

(3) It is certain that more of oil must be supplied in high speed gear systems because of great amount of heat generated, but the question is how much of all the oil works effectively. For, there is such a case that oil is blown off by the centrifugal force before getting to tooth surfaces or that oil does not get to the gears on account of an air draft induced by running gears.

According to the observation with a stroboscope in the present experiment, if a higher speed jet oil (the more pressurized oil) is supplied, almost all of the oil can get to the meshing zone even in the high speed range. For example, by using the nozzle with small exit diameter, that is, by supplying the oil at higher speed the oil may be expected to work more effectively in the meshing zone.

(4) Figures 16 and 18 indicate the difference of backlash, but the difference of trapping effect

is especially remarkable, as the center distance of gear set is changed, and therefore trapped area also varies. In a recent experiment*, some changes of backlash with the same clearance have little influence on the churning loss. But in the case with extremely large backlash the peak in the low speed range is not considered to appear, as the phenomenon of trapping does not occur.

(5) In the present experiment, the question is how to eliminate the loss of friction (at the bearings and tooth surfaces).

In the results of the experiment in the low speed range only the loss of bearing is eliminated. In these cases we assume that the loss of tooth friction is small because of test being done under no load, and that the oil film exists between the contacting tooth surfaces if their backlash is not so large and we consider that the tooth friction can be estimated by the equation of hydrodynamic friction of two cylinders.

Of course, it is insufficient as pointed out. It is only an approximation, but judging from experimental data the calculated values are not considered to be extremely different from these data. By these facts we are able to explain the lubricant churning loss qualitatively.

* Ariura, Y. and Ueno, T., *Preprint of Japan Soc. Mech. Engrs.* (in Japanese), No. 718-2 (1971), p. 125.

In the high speed range, on the other hand, as the temperature rise of all the parts is large and the error in the method abovementioned becomes larger, the churning loss is obtained by measuring the difference when no oil is supplied and when some amount of oil is supplied to the engaging gears. Therefore, the measured value is the net value of churning loss.

By using this method, the churning loss can be measured also in the low speed range, and we intend to use this method in the future.

(6) With a little oil such a phenomenon is observed. In order to avoid it and to eliminate the vibration, the test apparatus has a flywheel provided on the follower shaft as is shown in Fig. 1. Using of such a flywheel seems to be effective on this phenomenon.

And, in the range of the present experiment any particular peak of loss does not seem to appear under the resonance of tooth beating and tooth engagement. For, instead of considering that the losses increase in a certain range of speeds in Figs. 7~9, it is better to consider that they decrease in the high speed range. As their trends are obvious with lesser amount of oil, the oil is thought to be blown off by the air draft induced by running gears and can not get to the meshing zone.