*International Refereed Journal of Engineering and Science (IRJES)*
*ISSN (Online) 2319-183X, (Print)* 2319-1821
*Volume 1, Issue 2 (October 2012), PP.17-24*
*www.irjes.com*

# Study of Tribological Parameters In High Kinematic Pairs

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***ABSTRACT :*** *This paper is designed to provide familiarization and perspective for planning to pursue on any of the topics in the collective name of higher kinematic pairs. There are two pervading objectives: (i) to develop an understanding of the basic concepts of concentrated contacts; (ii) to develop a facility with the analytical techniques for predicting and assessing the behaviour of concentrated contacts which are typical for higher kinematic pairs. The information contained in this paper can be used to solve a number of problems common for all higher kinematic pairs. First, problems associated with contact between two non-conforming surfaces are discussed. They include the force transmitted at a point of contact, surface tractions, Friction, lubrication and wear in higher kinematic pairs, elastic hysteresis during rolling, rolling friction, and the lubrication of rollers. Next, film thickness under isothermal elastohydrodynamic conditions, inlet viscous heating, regimes of line contact lubrication are presented. Finally, contact problems in rolling element bearings, gears,and cam-follower systems are reviewed and equations to evaluate required minimum film thickness are discussed.*

***Key words –*** *Hertz function, hysteresis losses, Tension and compression cycle.*

## I. Introduction:

The tribological parameters like Friction, lubrication, wear and tear study in high kinematic pairs is done. It is known in theory of machines that normals to three points of restraint of any plane figure have a common point of intersection , motion is reduced to turning about that point. For a simple turning pair in which the profile is circular, the common point of interaction is fixed relatively to either element, and continuous turning is possible. A pair of elements in which the Centre of turning changes its position at the completion of an indefinitely small rotation, i.e. the new position is again the common point of intersection of the normal at three new points of restraint. For this to be possible the profiles will, in general, have differing geometric forms, and are then referred to as a higher pair of elements. Again, since the elements do not cover each other completely as in lower pairing and are assumed to be cylindrical surfaces represented by the profiles, contact will occur along a line or lines instead of over a surface. Relative motion of the elements may now be a combination of both sliding and rolling. In higher pairing, friction may be a necessary counterpart of the closing force as in the case of two friction wheels in ontact. Here the force on the wheels not only holds the cylinders in contact but must be sufficient to prevent relative sliding between the circular elements if closure is to be complete. In certain cases it is essential that force closure of higher pairs shall do more than maintain contact of the functional surfaces.

## II. Loads acting on contact area

In this section loads acting on a contact area and the way they are transmitted from one surface to another shall be considered. The load on the contact can be resolved into a normal force P acting along the common normal and a tangential force T opposed by friction. The relationship between W and T is given by

$$T \leqslant fW, \quad (1)$$

where f is the coefficient of limiting friction.T can be resolved into components Tx and Ty parallel to axes x and y. In a purely sliding contact the tangential force reaches its limiting value in a direction opposed to the sliding velocity. The force transmitted at a normal point of contact has the effect of compressing solids so that they make contact over an area of finite size. As a result it becomes possible for the contact to transmit a resultant moment in addition to a force. This is schematically shown in Fig.1. The components of this moment Mx and My are called rolling moments and oppose a rolling motion but are small enough to be neglected. The third component Mz, acting about the common normal, arises from friction within the contact area and is referred to as the spin moment. When spin accompanies rolling, the energy dissipated by the spin moment is combined with that dissipated by the rolling moments to make up the overall rolling resistance. Free rolling is defined as a rolling motion in which spin is absent and where the tangential force T at the contact point is zero. This is the condition of the unpowered and unbraked wheels of a vehicle if the rolling resistance and the friction in the bearings are neglected. It is in marked contrast with the driving wheels or the braked wheels which transmit sizeable tangential forces at their points of contact with the road or rail.

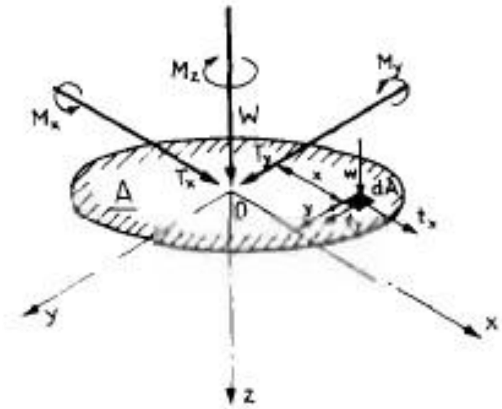


**Figure-1**

## III. Traction in the contact zone:

Traction in the contact zone: The forces and moments discussed above are transmitted across the contact interface by surface tractions at the interface. The normal traction (pressure) is denoted here by w and the tangential
traction (due to friction) by t, shown acting on the lower surface. For overall equilibrium

$$W = \int_A w \, \mathrm{d}A,$$

$$T_x = \int_A t_x \, \mathrm{d}A, \qquad T_y = \int_A t_y \, \mathrm{d}A.$$

**(2, 3)**

With contacts formed by the convex surfaces the contact lies approximately in x-y plane. Therefore

$$M_x = \int_A wy \, \mathrm{d}A, \qquad M_y = -\int_A wx \, \mathrm{d}A$$

and

$$M_z = \int_A (t_y x - t_x y) \, \mathrm{d}A.$$

**(4, 5)**

When the bodies have closely conforming curved surfaces, as for example in a deep-groove ball-bearing, the contact area is warped appreciably out of the tangent plane and the expressions for Mx and My, eqn (4), have to be modified to include terms involving the shear tractions tx and ty.

## IV. Hysteresis losses:

Some energy is always dissipated during a cycle of loading and unloading even within the so-called elastic limit.This is because no solid is perfectly elastic. The energy loss is usually expressed as a fraction a of the maximum elastic strain energy stored in the solid during the cycle where a is referred to as the hysteresis loss factor. For most metals, stressed within the elastic limit, the value of a is very small, less than 1 per cent, but for polymers and rubber it may be much larger.

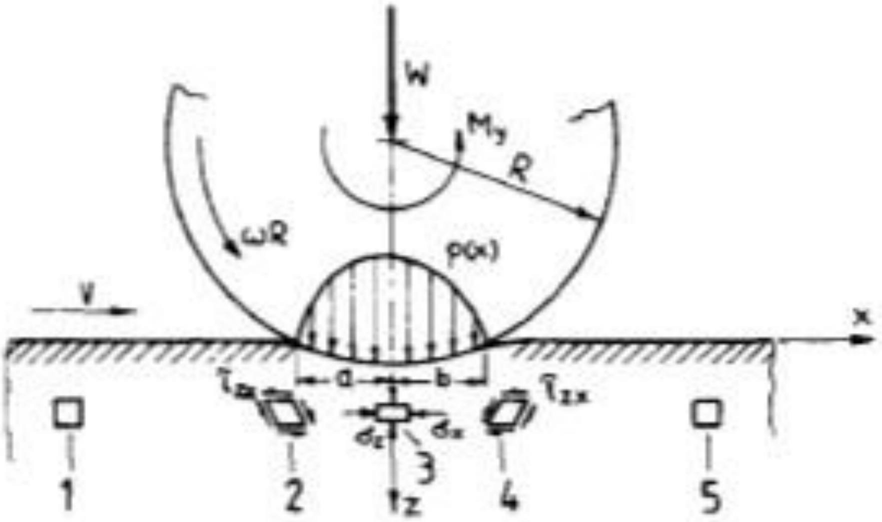


**Figure 2**

In free rolling, the material of the bodies in contact undergoes a cycle of loading and unloading as it flows through the region of contact deformation (Fig.2). The strain energy of material elements increases up to the centre- plane due to the work of compression done by the contact pressure acting on the front half of the contact area. After the centre-plane the strain energy decreases and work is done against the contact pressures at the back of the contact. Neglecting any interfacial friction the strain energy of the material arriving at the centre-plane in time dt can be found from the work done by the pressure on the leading half of the contact. For a cylindrical contact of unit width

$$dP = \omega\, dt \int_0^a p(x) x\, dx,$$

Where = V/R is the angular velocity of the roller. Taking p(x) to be given by the Hertz theory

$$\dot{P} = \frac{2}{3\pi} W a \omega. \qquad (7)$$

Where W is the contact load. If a small fraction a of this strain energy is now assumed to be dissipated by hysteresis, the resultant moment required to maintain the motion is given by equating the net work done to the energy dissipated, then

$$M_y \omega = \alpha \dot{P} = \frac{2}{3\pi} \alpha W a \omega \qquad (8)$$

Or

$$f_r = \frac{M_y}{WR} = \alpha \frac{2a}{3\pi R} \qquad (9)$$

where f is defined as the coefficient of the rolling resistance. Thus the resistance to rolling of r bodies of imperfectly elastic materials can be expressed in terms of their hysteresis loss factor. This simple theory of rolling friction is due to Tabor. Using the same calculation for an elliptical contact area given the result

$$f_r = \frac{M_y}{WR} = \alpha \frac{3}{16} \frac{a}{R}$$

where a is the half-width of the contact ellipse in the direction of rolling. For a sphere rolling on a plane, the effective rolling resistance

$$F_t = My/R \qquad (10)$$

should be proportional to $W^{\frac{4}{3}} R^{-\frac{2}{3}}$.

.

This relationship is reasonably well supported by experiments with rubber but less well with metals. There are basically two problems with this simple theory. First, the hysteresis loss factor a is not usually a material constant. In the case of metals it increases with strain (a/R),particularly as the elastic limit of the material is approached. Second, the hysteresis loss factor in rolling cannot be identified with the loss factor in a simple tension or compression cycle. The deformation cycle in the rolling contact, illustrated in Fig.2, involves rotation of the principal axes of strain between points 2, 3and 4,with very little change in total strain energy. The hysteresis loss in such circumstances cannot be predicted from uniaxial stress data. The same deformation cycle in the surface would be produced by a rigid sphere rolling on an inelastic deformable plane surface as by a frictionless sphere sliding along the surface. In spite of the absence of interfacial friction the sliding sphere would be opposed by a resistance to motion due to hysteresis in the deformable body. This resistance has been termed the deformation component of friction. Its value is the same as the rolling resistance F given by eqn (9).

## V. Rolling friction

Rolling motion is quite common in higher kinematic pairs. Ideally it should not cause much power loss, but in reality energy is dissipated in various ways giving rise to rolling friction. The various sources of energy dissipation in rolling may be classified into:

(i) Those which arise through micro-slip and friction at the contact interface;
(ii) Those which are due to the inelastic properties of the material;
(iii) Those due to the roughness of the rolling surfaces.

Free rolling has been defined as a motion in the absence of a resultant tangential force. Resistance to rolling is then manifested by a couple My which is demanded by the asymmetry of he pressure distribution, that is, by higher pressures on the front half of the contact than on the rear. The trailing wheels of a vehicle, however, rotate in bearings assumed to be frictionless and the rolling resistance is overcome by a tangential force Tx applied at the bearing and resisted at the contact interface. Provided that the rolling resistance is small (Tx <^ W) these two situations are the same within the usual approximations of small strain contact stress theory, i.e. to first order in
(a/R). It is then convenient to write the rolling resistance as a non-dimensional coefficient fr expressed in terms of the rate of energy dissipation P,

$$f_r = \frac{M_y}{WR} = \frac{T_x}{W} = \frac{\dot{P}}{WV}. \qquad (11)$$

thus The quantity P/V is the energy dissipated per unit distance travelled.

## VI. Energy dissipated due to micro-slip

Energy dissipation due to micro-slip occurs at the interface when the rolling bodies have dissimilar elastic contacts. The resistance from this cause depends upon the difference of the elastic constants expressed by the parameter ß (defined by eqn (11)) and the coefficient of sliding friction

$$\beta = \frac{1}{2}\left[\frac{[(1-2\nu_1)/G_1]-[(1-2\nu_2)/G_2]}{[(1-\nu_1)/G_1]+[(1-\nu_2)/G_2]}\right]. \quad (12)$$

The resistance to rolling reaches a maximum value of

$$f_r = \frac{M_y}{WR} \approx 15\times 10^{-4}\beta\left(\frac{a}{R}\right) \quad (13)$$

when ß/f=5 Since, for typical combinations of materials, ß rarely exceeds 0.2,the rolling resistance due to micro-slip is extremely small. It has been suggested that micro-slip will also arise if the curvatures of two bodies are different. It is quite easy to see that the difference in strain between two such surfaces will be second-order in (a/R)
and hence negligible in any small strain analysis. A special case is when a ball rolls in a closely conforming groove. The maximum rolling resistance is given by

$$f_r = \frac{M_y}{WR} = 0.08f\left(\frac{a}{R}\right)^2\left(\frac{b}{a}\right)^2. \quad (14)$$

The shape of the contact ellipse (b/a) is a function of the conformity of the ball and the groove; where the conformity is close, as in a deep groove ball bearing, b > and the rolling resistance from this cause becomes significant. In tractive rolling, when large forces and moments are transmitted between the bodies, it is meaningless to express rolling resistance as Tx or My/R. Nevertheless, energy is still dissipated in micro-slip and, for compar ison with free rolling, it is useful to define the effective rolling resistance coefficient fr = P/VW. This gives a measure of the loss of efficiency of a tractive drive such as a belt, a driving wheel or a continuously variable speed gear.

## VII. Energy dissipated due to plastic deformations

In the majority of cases, resistance to rolling is dominated by plastic deformation of one or both contacting bodies. In this case the energy is dissipated within the solids, at a depth corresponding to the maximum shear component of the contact stresses, rather than at the interface. With materials having poor thermal conductivity the release of energy beneath the surface can lead to high internal temperatures and failure by thermal stress. Generally metals behave differently than non-metals. The inelastic properties of metals,and to some extent hard crystalline non-metallic solids, are governed by the movement of dislocations which, at normal temperatures, is not significantly influenced either by temperature or by the rate of deformation. The rolling friction characteristics of a material which has an elastic range of stress, followed by rate- independent plastic flow above a sharply defined yield stress, follow a typical pattern. At low loads the deformation is predominantly elastic and the rolling resistance is given by the elastic hysteresis equation (8). The hysteresis loss factor as found by experiment is generally of the order of a few per cent. At high loads, when the plastic zone is no longer contained, i.e., the condition of full plasticity is reached, the rolling resistance may be estimated by the rigid-plastic theory. The onset of full plasticity cannot be precisely defined but, from the knowledge of the static indentation behaviour, where full plasticity is reached when
W/2a=2.6 And Ea/YR= q100, it follows that GW/kR=300,
where

k is the yield stress in shear of the solid.

## VIII. Energy dissipated due to surface roughness

It is quite obvious that resistance to the rolling of a wheel is greater on a rough surface than on a smooth one, but this aspect of the subject has received little analytical attention. The surface irregularities influence the rolling friction in two ways. First, they intensify the real contact pressure so that some local plastic deformation will occur even if the bulk stress level is within the elastic limit. If the mating surface is hard and smooth the asperities will be deformed plastically on the first traversal but their deformation will become progressively more elastic with repeated traversals.

## IX. Analysis of line contact lubrication

In this section line contact lubrication is presented in a way which can be directly utilized by the designer. The geometr y of a typical line contact is shown in Fig. 6.5. The minimum film thickness occurs at the exit of the region and can be predicted by the formula proposed by Dowson and Higginson for isothermal conditions

$$\frac{h_{\min}}{R} = 2.65 \frac{G^{0.54} V^{0.7}}{W^{0.13}} \quad (15)$$

where G = Ea is the dimensionless material parameter,

$$K=[\mu_o (V, +V2)]/2ER \quad (16)$$

is the dimensionless speed parameter, W=w/ERL is the dimensionless load parameter, a is the pressure-viscosity coefficient based

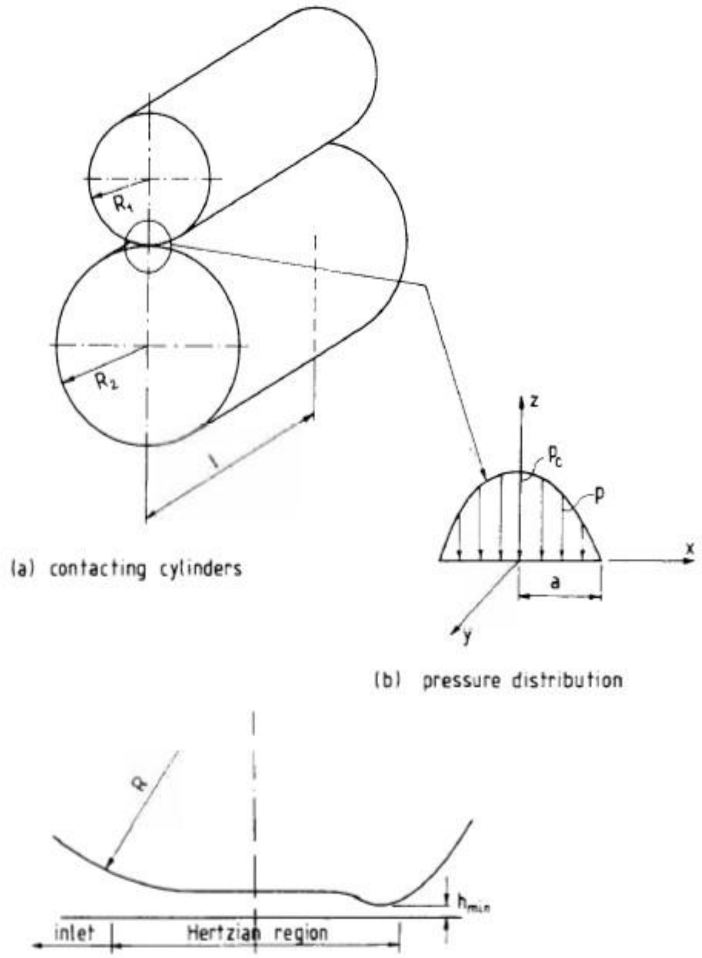


Figure 3

on the piezo-viscous relation $\mu = \mu_o\, e_{xp}$ and reflects the change of viscosity with pressure

$$1/E = \tfrac{1}{2}\left[\frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2}\right], \quad (17)$$

$\mu_o$ is the lubricant viscosity at inlet surface temperature, $V_1V_2$ are the surface velocities relative to contact region

$$R = \frac{R_1 R_2}{R_1 \pm R_2} \quad (18)$$

where the plus sign assumes external contact (both surfaces convex) and the minus sign denotes internal contact (the surface with the larger radius of curvature is concave), w is the total load on the contact and L is the length of the contact. The viscosity of the lubricant at the temperature of the surface of the solid in the contact inlet region is the effective viscosity for determining the film thickness.This temperature may be considerably higher than the lubricant supply temperature and therefore the inlet viscosity may be substantially lower than anticipated, when

based on the supply temperature. Usually the inlet surface temperature is an unknown quantity in design analyses. The solution to this problem is to use the lubricant system outlet temperature or an average of the inlet and the outlet temperatures to obtain an estimate of the film thickness

## References

[1.] F. F. Ling. Surface Mechanics. New York: Wiley, 1973.
[2.] A.W. Crook. The lubrication of rollers,Phil. Trans.,A250 (1958), 18-34.
[3.] A. W. Crook. The lubrication of rollers, II and III.Phil. Trans. A254 (1961), 141-50.
[4.] A. W. Crook. The lubrication of rollers, IV. Phil. Trans., A255 (1963), 261-9.
[5.] B. Jacobson. On the lubrication of heavily loaded cylindrical surfaces considering surface deformations and solidification of the lubricant. Trans. ASME, J. Lub. Technoi, 95 (1973), 176-84.
[6.] D. Dowson and G. R. Higginson. Elastohydrodynamic Lubrication. New York: Pergamon Press, 1966.