EXPERIMENTAL ASSESSMENT OF FRICTIONAL CHARACTERISTICS OF LINE CONTACT UNDER EHD LUBRICATION REGIME

F.A. Soliman, M.I.A. Mokhtar and H.E. Elkateb
Mechanical Design and Production Department, Factlty of Engineering
Cairo University, Egypt.

ABSTRACT

This work represents an experimental investigation into the effect of the operating conditions on the value of friction at the lubricated rotating cylindrical surfaces under line contact. A special disc machine has been designed and constructed with appropriate measuring technique to identify the values of the frictional forces between contacting bodies under different ratios of sliding and rolling movement. Records of friction traction during operation have shown that the coefficient of friction decreases up to certain value of load and relative slip beyond which the value of the friction coefficient starts to increase approaching the behavior under pure sliding at relatively high load. It was also found that the higher lubricant viscosity gives rise to the coefficient of friction.

Keywords: Assessment, Frictional characteristics, Line contact, Lubrication.

INTRODUCTION

Some efforts were exerted to investigate the frictional behavior under combined effects of sliding and rolling situations where the contact resembles a lubricated Elastohydynamic line or point contact [1].
The simulation of the contact mechanism was achieved by the construction of multiple discs at high contact pressur. condition [2,3,4,5].

The rolling friction arises primarily from the shear associated with the curvature of the velocity profile in the convergent entry region of the contact, while the shear of high viscous oil in the main pressure zone initiates a sliding friction component which contributes to the possible scuffing at high pressure [6]. Extended work on the four discs machine [7] allowed the evaluation of the friction traction between discs as function of slip. The rolling friction was then evaluated by extrapolating the graphs to zero value of slip, the value of rolling friction was in the order of 10. Other techniques to measure the frictional behavior of line contacts under Elastohydrodynamic regimes during pure sliding was presented [8] to evaluate the oil film behaviour under such conditiön which typifies many applications.

The high slip condition is a typical gear contact case in which possible scuffing failure is experienced, it is mainly dependent on load and the relative velocity, while the rolling friction was found to be independent of load and proportional to the oil film [9,10]. This led some investigators to study the oil film behaviour under the combined effects of rolling and sliding [11,12].

A general agreement was always reached that the frictional behaviour under high pressure contact is not a straight forward problem since the mechanical, physical and chemical variables involved in the contact including contact area, film thickness, oil viscosity, oil temperature, etc. are interacting in a complex way. Accordingly, any numerical analysis for such a contact problem must essentially be supported by experimental findings covering the effects of all possible parameters. This will enable high pressure contact elements designers and users to accurately define the safe range of operation elements at line contact under different conditions of loads and sliding to rolling ratios. The effect of the parameters involved in such contact on the friction traction between the contacting element will be studied to provide a better understanding of

Elastohydrodynamic lubrication contact.

Experimental set-up

The experimental scheme is set-up to determine the traction between two lubricated discs running under Elastohydrodynamic lubrication situation of sliding or sliding and rolling subjected to different loading conditions.

A testing machine was specially designed for this purpose. It consisted of two contacting discs simulating the two elements under line contact. The discs are connected to a driving, loading, and friction measurement devices as shown in Figure (1).

Two seperatly driving Motors have been coupled to the two main shafts on which the two discs are mounted. The upper disc is manufactured from (CK60 DIN 17006) heat treatable steel, having 99 mm diameter, 10 mm thickness, of surface roughness Ra = 0.1 hm and a surface hardness of 32 Rc. The lower one is a compound disc composed of a rim manufactured from copper alloy of 103 mm diametre and 20mm width while the surface roughness Ra = 0.09 hm and a surface hardness of 13 Rc. The rim is mounted on a specially adapted hub to measure the angular shift due to friction.

The traction force measuring system is built on the lower compound disc as shown in Figure (2). The outside rim and inside hub of the disc are interconnected by leaf springs on which strain gauges are adhered. The relative motion between the rim and the hub due to friction traction deflects the leaf spring this deflection can be recorded by the strain gauge bridge. The gauges readings are calibrated to give the friction traction. The strain gauge signal is transmitted to the recording system by a wire passing inside a longitudinal hole in the shaft carrying the disc. The wires are led through the centre of rotation and soldered to slipless rings incorporated in the rotating mechanism. The rings are submerged in a mercury pool which transmits the current to stationary terminals. The mechanism is shown in Figure (3). This mechanism insures the complete transmission of resistance changes of the strain gauges to the measuring system. The friction traction between the two discs can then be continuously recorded under a wide range of normal load at combined rolling and sliding speeds.

The condition of pure sliding was achieved by keeping any of the discs stationary. The lower disc is partially submerged in oil sump to insure flooded lubrication mechanism.

The oil viscosity change from 0.126 Nm Sec. to 0.345 Nm Sec.

The loading mechanism consisted mainly of a loading arm on which the upper disc shaft is supported by means of two deep groove single row ball bearing. The loading arm is hinged at one end through a ball bearing, while the the other end is loaded by a calibrated compression helical spring which gives a load range between 200 to 1400 N. The driving system permitted the change in the discs speeds through a V belt drive to obtain the rolling to sliding ratio.

The upper disc speed ranges from 697 R.P.M. to 3165 R.P.M. where as the lower disc speed varies between 495 R.P.M. to 3195 R.P.M.

RESULTS AND DISCUSSION

The variation of the measured friction traction force and the associated coefficient of friction with the normal load at pure sliding and different ratios of rolling to sliding speeds is shown in Figures (4 to 13).

The results show that the friction traction increases as the load increases while the coefficient of friction is behaving oppositely for certain speed range. The variation of speed for both discs from zero to maximum value was meant to gradually provide condition of pure sliding or sliding to rolling of different ratios. The friction is shown to be increasing with higher oil viscosity especially at high loading conditions.

In case of pure sliding Figure 4(a,b) the friction traction increases as the load increases while the friction coefficient showed a decrease as the load increases up to certain limit beyond which it starts to increase. This effect is pronounced when oil with higher viscosity is used as shown in Figures (4.5). As the load increases beyond certain limit the friction force increases at a faster rate than before giving rise to the coefficient of friction. This latter phenomenon could be attributed to the effect of load increase on the generated hydrodynamic pressure which in turn influences the lubricant velocity distribution within the clearance between the two discs and hence affects the oil shear rate, also the possible heat generated. The change in the shear rate would ultimately affect the viscous shearing of oil giving rise to friction at high rate.

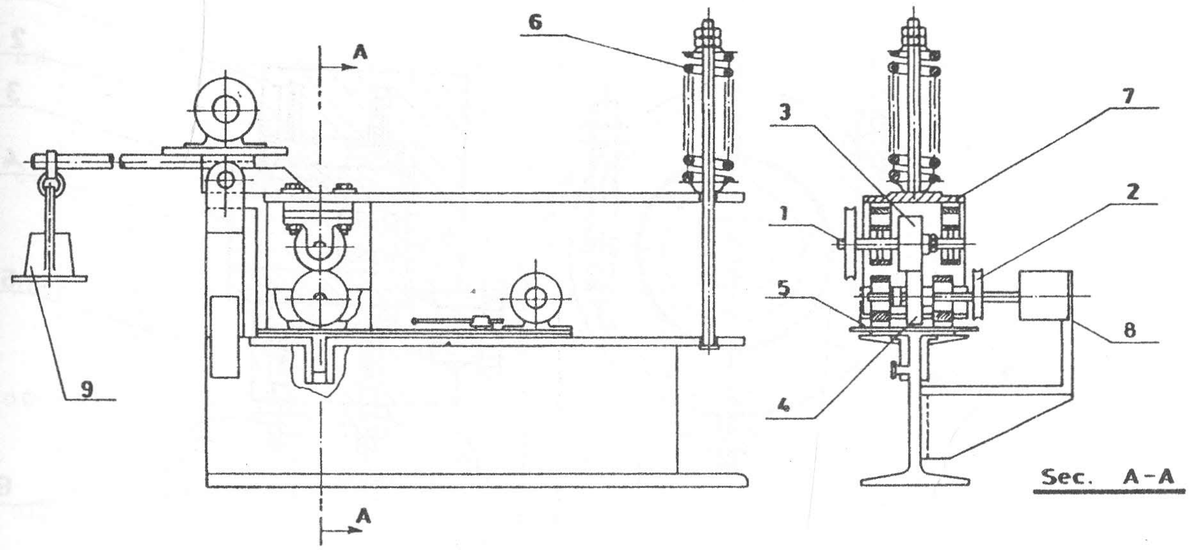

Figure (1) Schematic arrangement of test rig

1. Upper driving shaft
2. Lower driving shaft
3. Upper cylindrical disc
4. Lower cylindrical disc
5. Oil sump
6. Loading spring
7. Loading arm
8. Mercury pool slipless ring
9. Balancing weight

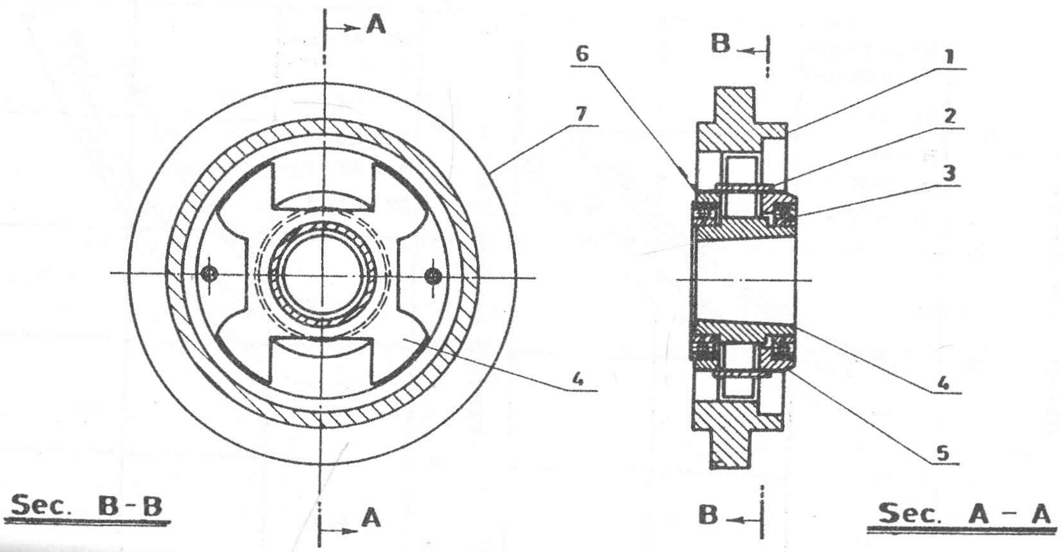

Figure (2) Lower cylindrical disc

1. Disc cover
2. Strain gauge
3. Ball bearing
4. Hub
5. Leaf spring
6. Output terminal
7. Friction disc

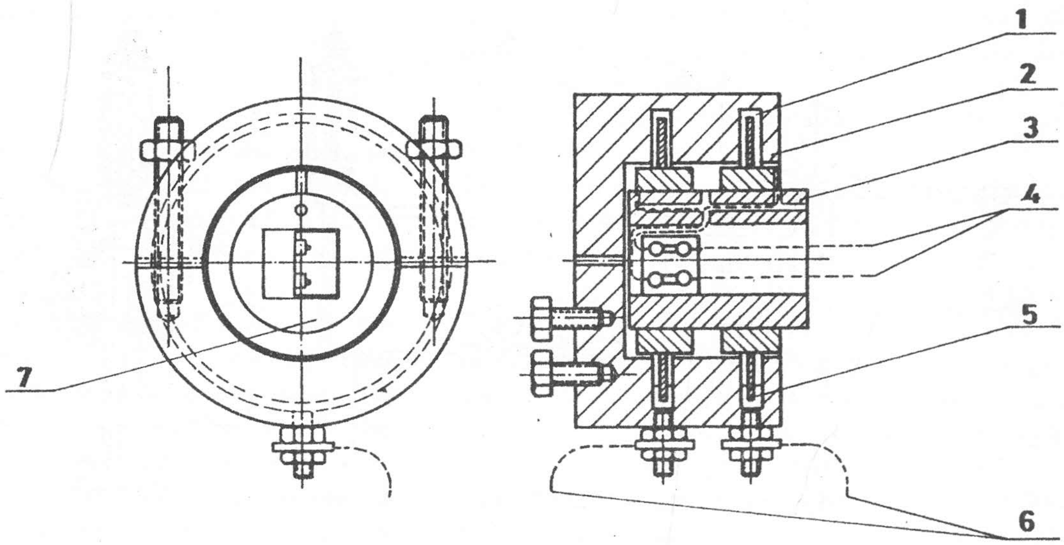

Figure (3) Mercury pool slipless ring

1_ COPPER DISC
2_ HOUSING
3_ INSULATING RING
4_ INPUT TERMINAL
5_ MERCURY
6_ OUTPUT TERMINAL
7_ SQUARE SHAFT

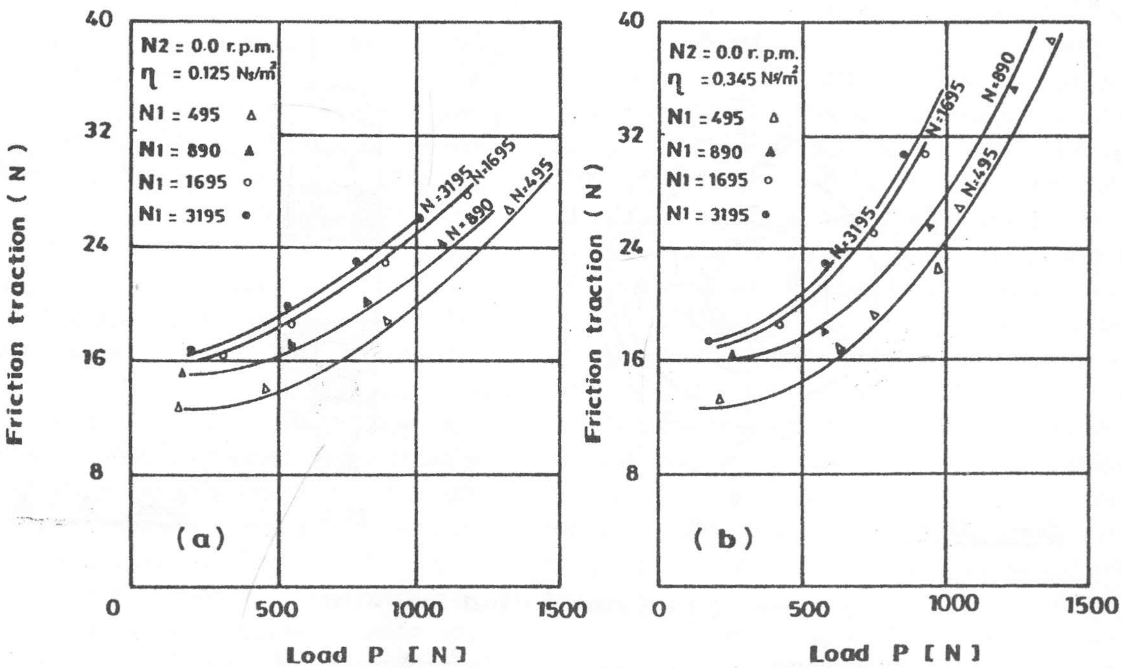

Figure (4) Variation of traction force with load at different speeds for two types of oils at condition of pure sliding

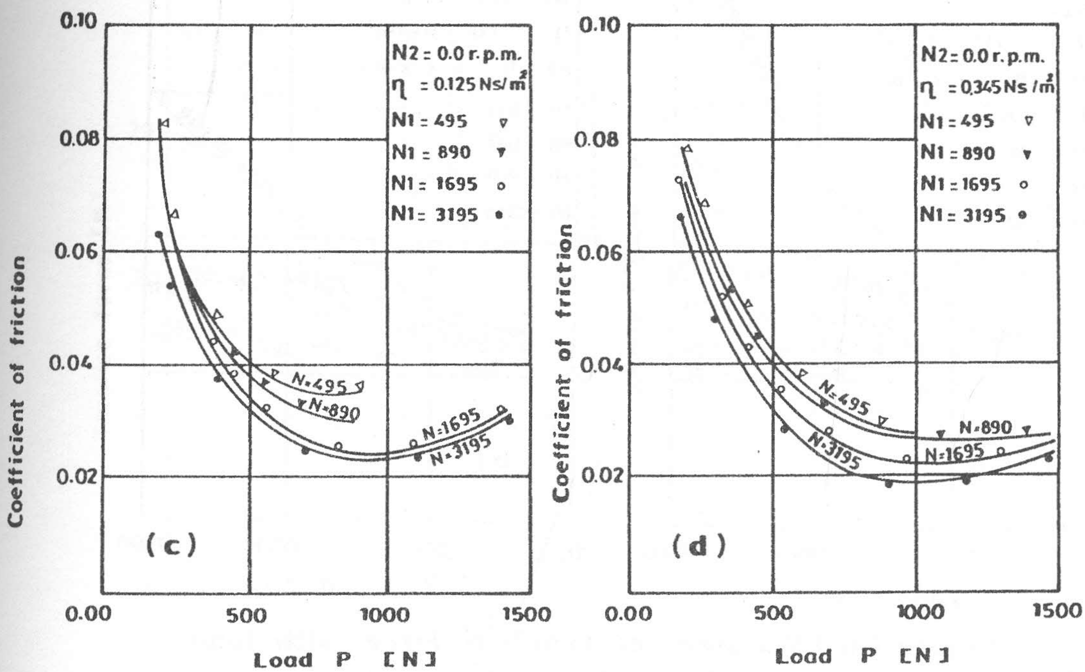

Figure (5) Variation of coefficient of friction with load at different speeds for two types of oils at condition of pure sliding

In case of combined sliding and rolling, the friction coefficient shows a unique behaviour Figure (6-13) where the coefficient of friction decreases up to certain value of relative slip (sliding to rolling ratio) and load. Beyonthis limiting value the coefficient of friction starts to increase approaching the behaviour under pure sliding at relatively high load. The previously mentioned condition is expected to be a stage leading to possible partial Elastohydrodynamic lubrication at which the EHD regime can no longer initiate the enough pressure to support the load. At this stage further increase in the coefficient of friction due to increase in the applied load and relative speed will increase oil temperature and consequently decrease oil viscosity. This condition may give way to scuffing.

The effect of viscosity on friction is shown in Figures (4-13). The coefficient of friction increases when increasing oil viscosity due to increase in oil shear force, while the viscous oil provides better hydrodynamic behaviour from the point of view of load support. Therefore, the choice of the proper oil is always a compromise between its ability to load support and the amount of shear developed due to its viscosity.

CONCLUSIONS

The obtained results of the experimental work carried out in this investigation could lead to the following conclusions :

1- In case of pure sliding the friction traction increases while the coefficient of friction showed a decrease with the increase of load up to certain limit beyond which it starts to increase. This effect is pronounced when oil of higher viscosity is used.

2- In case of combined sliding and rolling (relative slip) the friction coefficient decreases up to a certain value of load and relative slip beyond which the value of coefficient of friction starts to increase approaching the behaviour under pure sliding at relatively high load.

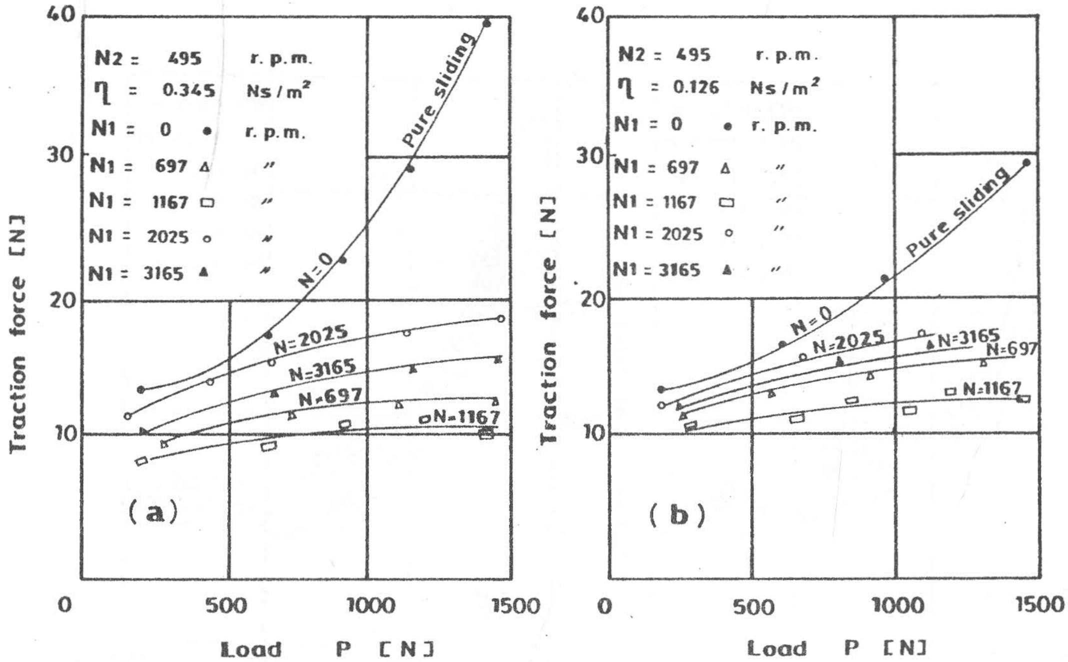

Figure (6) Variation of traction force with load at combined sliding&rolling of different ratios and different oil viscosities

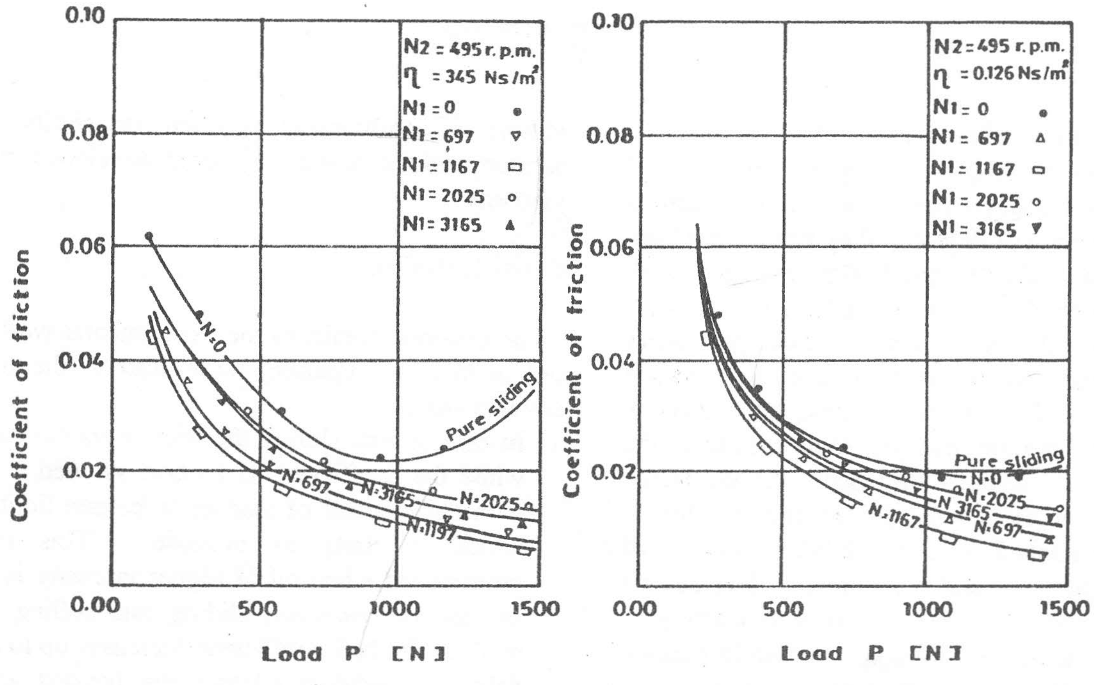

Figure (7) Variation of coefficient of friction with load at combined sliding and rolling of different ratios and different oil viscosities

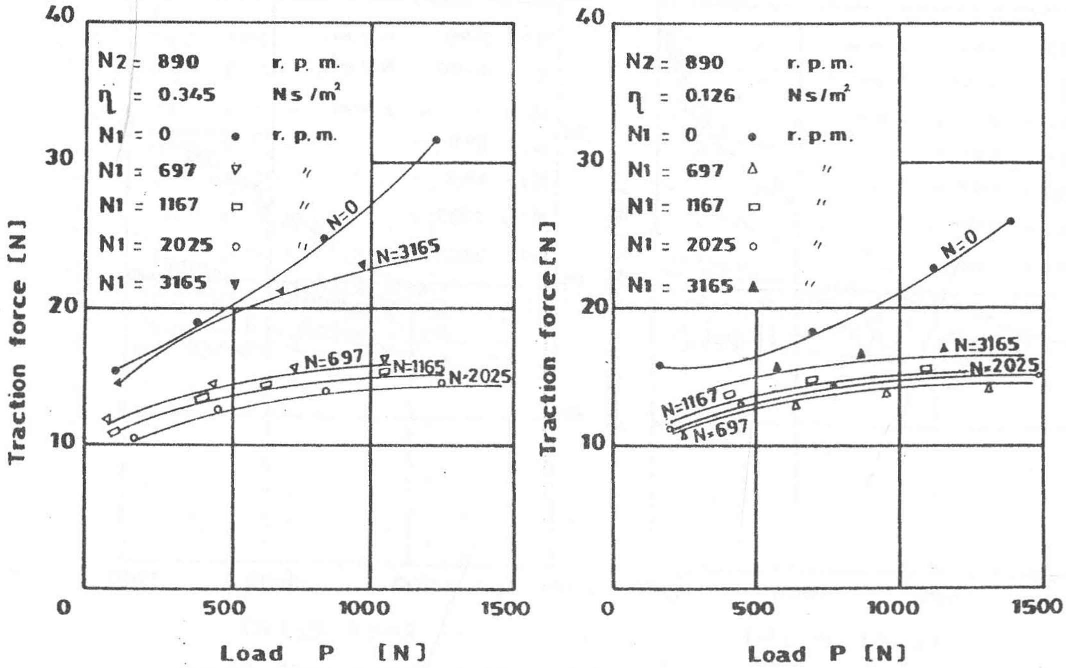

Figure (8) Variation of traction force with load at combined sliding&rolling of different ratios and different oil viscosities.

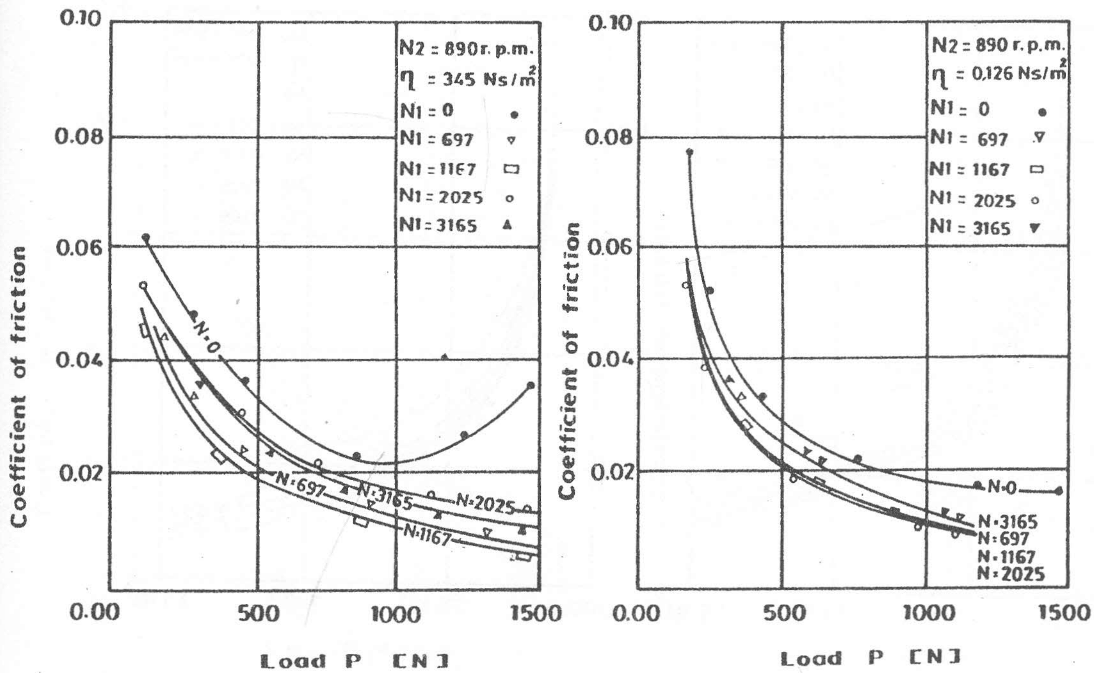

Figure (9) Variation of coefficient of friction with load at combined sliding and rolling of different ratios and different oil viscosities

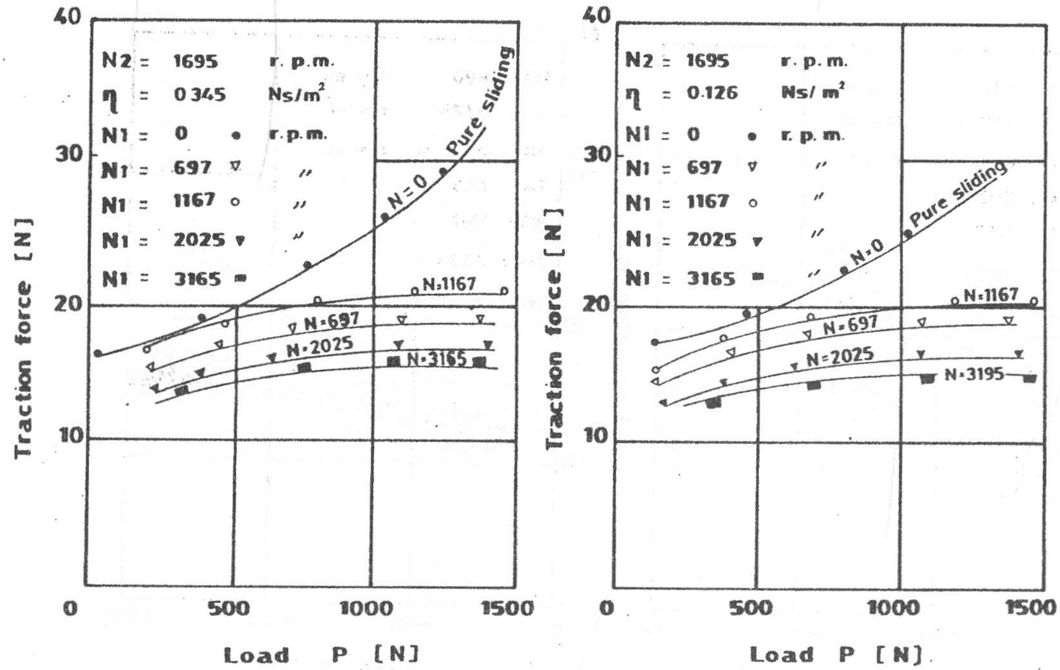

Figure (10) Variation of traction force with load at combined sliding rolling of different ratios and different oil viscosities

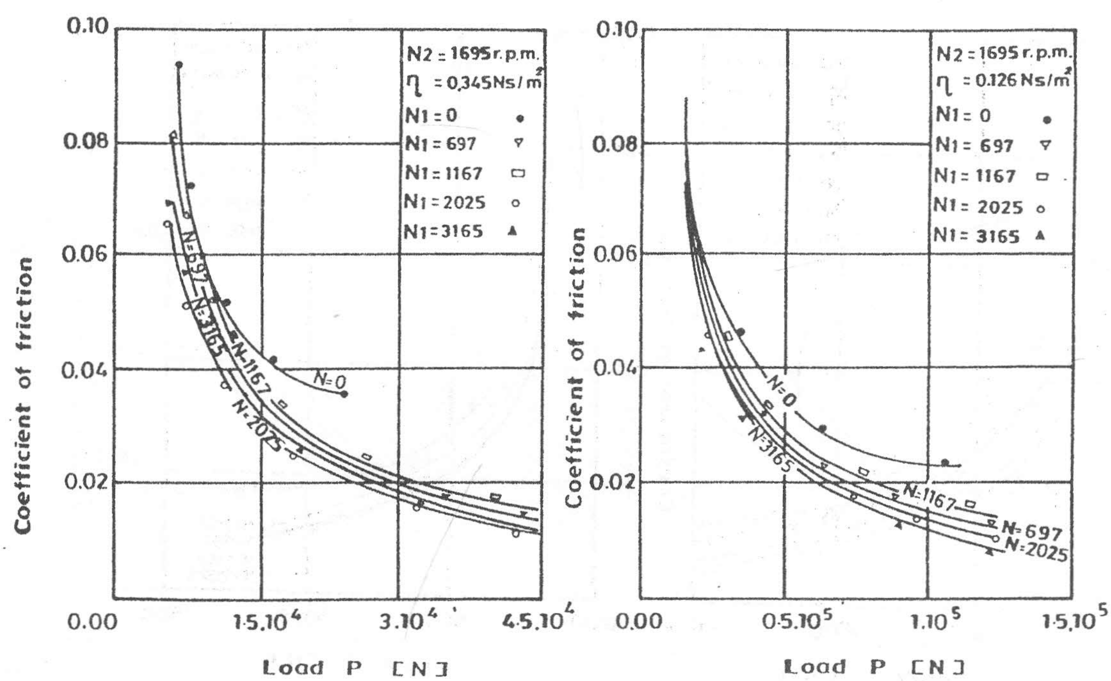

Figure (11) Variation of coefficient of friction with load at combined sliding and rolling of different ratios and different oil viscosities

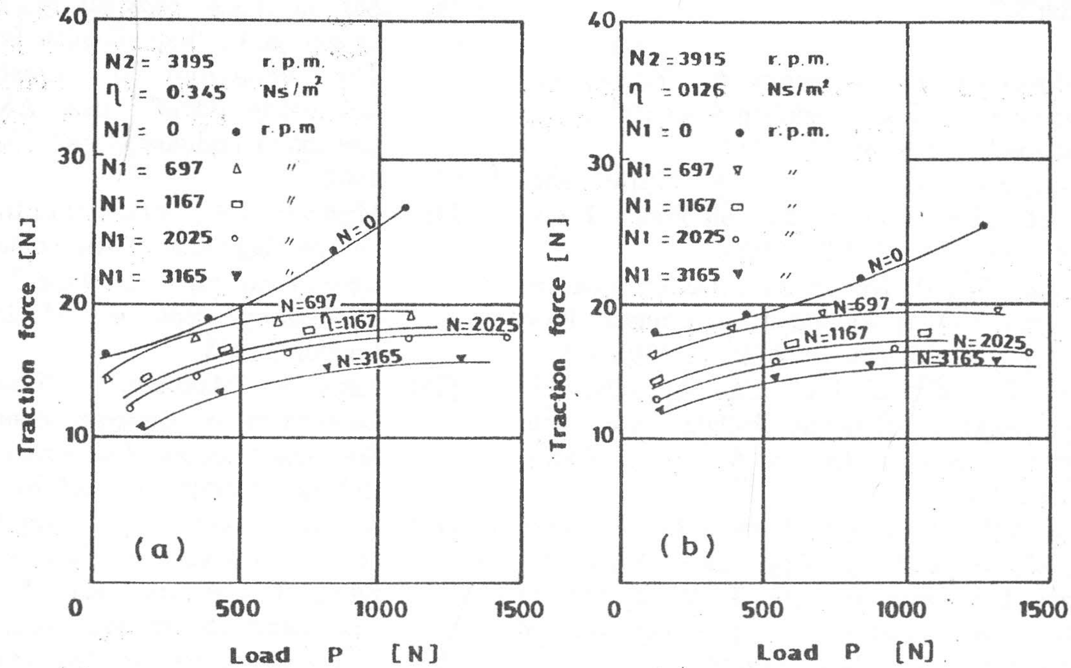

Figure (12) Variation of traction force with load at combined sliding and rolling of different ratios and different oil viscosities.

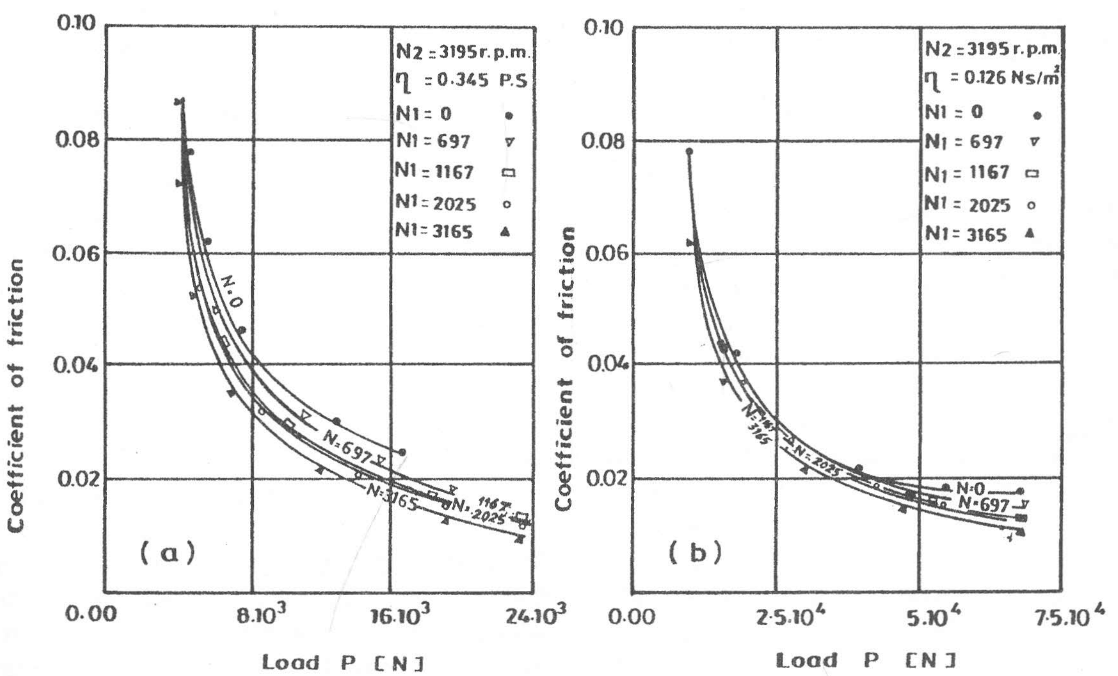

Figure (13) Variation of coefficient of friction with load at combined sliding and rolling of different ratios and different oil viscosities

REFERENCES

[1] O' Donoghue, J.P., Cameron, A., "Friction and Temperature in Rolling /sliding Contacts", ASLE Tarns., vol. 9, pp. 186-194, 1996.

[2] El Sisi, S.I., Shawki, G. S. A., "Performance Characteristics of Lubricating oil Film". Trans., ASME. series D, 82, 1,P,19,1960.

[3] Hamilton, G. M., Moore, S.L., :Deformation and Pressure in an Elastohydrodynamic Contact". Proc. Roy. Soc., London, A., 322,PP.313-330,1971.

[4] Parker, R.J., Kannel, J.w., "Elastohydrodynamic Film Thickness Between Rolling Discs with Synthetic Parafinic Oil to 600 úF", NASA, TND-6411,1971.

[5] Crook, A.W., "Elastohydrodynamic Lubrication of Rollers", Nature, 190,pt. 2,pp. 1182-1183, 1961.

[6] Eugane, E.S., "How to predict Efficiency of Gear Trains". Advanced Gear Engineering, General Electric Co., Lyon, Mass., 244-245.

[7] Crook, A.W., :The Lubrication of Rollers - IV Measurement of Friction and Effective Viscosity", Phil-Transmission Series A,pp.181-255,1962.

[8] Shawki, G.S.A., Mokhtar, M. O. A. and Abdel Ghany, A.A., "Experimental Investigations Into The Behaviour of Elastohydrodynamic Lubricating Films" Trans. ASME, Journal of lubrication Technology, vol. 104,No. 1,pp.91-98, 1982.

[9] Misharin, G.A., "Influence of friction Conditions on The Magnitude of The Friction Coefficient in Case of Rolling With Sliding", Proc. Int. Conf. on Gearing, Paper 40, 159(Inst. Mech Eng., London), 1958.

[10] Sanae, W., Masafumi, T., "Elastohydrodynamic Lubrication of Squeeze Film, Part 2, Two Spherical Bodies Lubricated with Grease". Bulletin of JSME. vol. 23, No.179, May 1980.

[11] Mokhtar, M.O.A., and Abdel Ghany, A.A., "Elastohydrodynamic Behaviour of Rolling Elliptical Contacts" Part I : Pressure and temperature Distribution Trans. ASME, Journal of Tribology, vol. 107, No. 3,pp.343-351, 1985.

[12] Mokhtar, M.O.A. and Abdel Ghany, A.A., "Elastohydrodynamic Behaviour of Rolling Elliptical Contacts" Part II : Oil film Thickness and Contact Profile, Trans ASME, Journal of Tribology, vol. 107, No. 3,pp.343-351, 1985.