EFFECT OF OILS TYPES ON FRICTIONAL TRACTION AT CONTACT AREA FOR LUBRICATED SPUR GEARS

تأثير أنواع الزيوت على الاحتكاك عند مساحة التلامس للتروس العدلة المزيتة

BY

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ملخص البحث:
البحث يبين النتائج النظرية والعملية لعزوم الاحتكاك عند منطقة التلامس للتروس العدلة الزيتة بواسطة الزيت . نجد ان عزم الاحتكاك يتأثر بتغيير أنواع لزوجة الزيت والحمل وكذلك السرعة الانزلاقية .

SUMMARY - Experimental and theoretical results of frictional traction at contact area for lubricated spur gears are illustrated. Frictional traction is influenced with the variety of oils types (viscosities), sliding speeds and loads.

INTRODUCTION

The total frictional traction (1) of test rig consists of two parts one due to the running of gears in the gear box tank oil and the other due to pressurized air bearings. Air viscosity (2) is very small compard with oil viscosity, then the frictional traction through the air bearing can be diminished to zero.

The sliding-rolling friction force is mainly at area (zone) of contact of the gears teeth. The range of contact areas is around pitch point. If the oil is well prepared for gears, then, the teeth at contact area are separated by an oil filmthickness. In fluid friction range, the strength of this filmthickness depends on the loads, sliding - rolling speeds and the characteristics of the oils (3, 4, 5, 6, 7) (e.g: mean viscosities, temperatures, E.P. additives)

In contact area, the fluid friction force is the sum of the resistance to shear for all the elements of oil film area. If the contact oil film thickness and the mean shear rate are constant, then, the mean oil viscosity is a function of friction force (8).

NOMENCLATURE

$N_1\ N_2$	Line of Engagement
OL or g	Length of engagement
R_1 , R_2	Radii of equivalent cylinders
rp_1 , rp_2	Radii of circles of contact
rt_1 , rt_2	Radii of tip circles
P	Pitch point
ψ	Service angle of contact

TEST RIG AND EXPERIMENTALS

The imported GERMAN test - rig (its marked number TM 240) is designed especially for these types of experimentals. These measurements are to study the effect of oils types on frictional traction for involute straight spur gear only, and the other factors are ignored. The materal of the gear is steel.

The test rig is provided by : friction torque blance, speedometer, brake, electric motor, two-step gear box and other cponents. The general diagram of test rig figure 1 is consisted of two step, involute-toothed, straight spur gear-box, which are mounted on swing frame, and has the facility to feed back the out put power to the input shaft, and the variable speed driving motor is only required to supply the friction losses.

The rotation between the gears is either by a fixed shaft between one pair or a variable torsion bar between the other pair of wheels. Variable torsion bar coupling has two flanges with series of holes equally spaced around the outer edge. One flange has one more hole than the other. Flanges are twisted in opposite directions and load the gears, and a pin inserted through appropriate set of holes. The applied load is varied by increasing the twisting angle of one flange of the torsion bar relative to the other flange. This applied load can be found from a calibration graph.

The frame unit completed by the gears are statically balanced about their axis of rotation. The unit is supported on pressurized air bearings, and tubular spring balance fitted at the main frame is used to measure, directly the frictional tractions.

The gear box should be filled so that the gear teeth are dipping into the oil 10 mm. The thermo - meter which records the bulk mean temperature of the oil is positioned in the perspex cover of the gear box at the corner.

The test rig was run at different types of lubricated oils, Table 1. Frictional tractions, temperatures, loads and speeds were measured. For resons of comparison, the measurements were taken at a starting temperature of 27 °C in a definite time before fluid frictional heating could raise seriously the temperature of the oil. After every measurements and before adding another new lubricating oil, the oil box of the gears was cleaned from the oil.

THEORETICAL

I- Kinematic characterization of gear tooth contact

The true kinematic characterization is similar to the contact between two discs of centres (9) N_1 , N_2 , as in Figure 2. However, the radii of the cylinders vary according to the location of the tooth contact on the line of engagement from 0 to L. So the equivalent radius of the pair of the cylinder at point P

$$R_c = r_{p1} \cdot r_{p2} \cdot \sin \cdot \psi / (r_{p1} + r_{p2}) .$$

The engagement of a tooth begins at point 0 where tip circle of the driven gear 2 intersects the line of engagement and ends at L , where the tip circle of the driving gear 1 intersects the line of engagement. The line N_1 , N_2 is the common normal at the points of contacts and is the common tangent to the base circles where the common tangent coincides with the line of engagement. The length of engagement is equal to

$$g = (r_{t1}^{\,2} - (r_{p1} \cos \psi)^2)^{1/2} + (r_{t2}^{\,2} - (r_{p2} \cos \psi)^2)^{1/2} - (r_{p1} + r_{p2}) \sin \psi .$$

During the line of engagement, the normal forces, frictional forces, radii of discs, sliding speed and rolling speed change.

Gear-Tooth surfaces move across each other with (10) a combination of sliding and rolling motions along the common normal line. The sliding motion which is responsible for the formation of an oil film between mating teeth, changes both in magnitude and direction during the meshing cycle. where the sliding motion is a maximum at the first point of contact, reduces to zero at pitch point, changes direction and increases again to a maximum at the last point of contact.

It can be seen that the elemental property of two tooth profiles to transmit motion uniformly is that the common normal at the point of contact passes through the pitch point. The instantaneous centre of motion is the pitch point. The motion of sliding will be zero (the instantaneous motion between the teeth will be one of pure rolling) when the point of contact of the teeth is at pitch point.

II - Frictional tractions of involute spur gears

The viscosity may vary through teeth contact zone. The isothermal case, where the temperature is assumed to be constant will be estimated. The shear stress τ of small element, based on the concept of a Newtonian fluid (11) is :-

$$\tau = \eta \frac{\partial u}{\partial z} \qquad (1)$$

The total friction force is given by integrating shear stress over the contact surface:-

$$F = \iint \tau \; dA \quad \text{hence,}$$

$$F = \iint \tau \; dx\, dz \quad \text{and}$$

$$F = \int_0^L \int_0^{2b} \left(\frac{\eta}{h} (U_1 - U_2) \pm \frac{dp}{dx} \frac{h}{2} \right) dx\, dz \qquad (2)$$

where

L : face length , b : half width of Hertzian zone , η : viscosity

h : filmthickness , U_1 & U_2 : gears speeds, dp/dx : pressure gradient

The rolling friction part emerges when $U_1 = U_2$ that is at the end of the gear approach zone and at the begin of gear recess zone. Out of the last zone the sliding friction predominates most of the meshing cycle and that due to the rolling can be ignored then,

$$F = L \int_0^{2b} \frac{\eta}{h} (U_1 - U_2)\, dx \qquad (3)$$

At higher sliding speeds and loads, an effective viscosity and filmthickness are necessarily introduced, since the actual viscosity and filmthickness at any position will vary with pressures, sliding speeds and temperature. It is usually sufficient, in calculating a values of effective viscosity to assume a Hertzian pressure distribution. The Hertzian distribution differs appreciably from the true hydrodynamic distribution only where the pressures are low. The integration of η gives the effective mean viscosity variation η_m . From reference (12) η_m is as following :-

$$\eta_m = \frac{T_c}{\frac{T_1}{\eta_1} + (U_1 - U_2)^2 \frac{3.5}{8\,k}}$$

Where K : Thermal conductivity of oil which is known, T_c : contact temperature c°

T_1 : inlet temperature c°, η_1 : inlet viscosity

In gear lubrication, the elastohydrodynamic lubrication is capable of giving good result of film thickness. Therefore an elastohydrodynamic mean film thickness h_o must be introduced (13) with the limit of Hertzian zone (0 --- 2b).

$$h_o = \frac{C\,(a \sin \psi)^{1.13}}{w^{0.13}} \left(\frac{\eta_1 \pi N_1}{30}\right)^{0.7} \frac{i^{1.13}}{(i+1)^{1.56}} \qquad \text{ii}$$

where

$a = r_{p1} + r_{p2}$

$i = N_2 / N_1$ speed ratio $= r_{p1} / r_{p2}$

$C = 1.6\ \alpha^{0.6}\ E'^{\,0.03}$

α = lubricant pressure - viscosity coefficient

$$\frac{1}{E'} = \left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right] \cdot \frac{1}{2}$$

E' : effective modulus of elasticity

ν_1, ν_2 : poissons ratio E_1, E_2 : elastic modulii of gears in contact

w : load per unit face length ψ : pressure angle 20°

From i, ii and equation 3 the total frictional traction per unit face length due to the sliding is given by :-

$$M/L = ((R_e \eta_m\, 2b\, (U_1 - U_2)) / ho) \cdot (g/\text{module} \cdot 3.14) \qquad (4)$$

where

$b = (8\, R_e\, w / E' \pi)^{\frac{1}{2}}$

overlap = g/module x 3.14

RESULTS AND DISCUSSION

The frictional tractions were plotted versus sliding speeds. Typical results for eight different types of gear oils are given in figure 3. In this Figure all results refer to the same load of 294×10^5 dyn cm.$^{-1}$

For the curve of the lowest oil number 112, frictional traction varied from 1×10^5 dyn cm^{-1} at low sliding speed up to 2.8×10^5 dyn cm^{-1} at medium sliding speed of 200 cm/s. At the highest oil number 145 the variation was greater because of the greater viscosity. The same figure indicates that if the viscosity decreases the frictional traction decreases, oil 112, In these results, This lubricating oil is the optimum one, but there are another factors (14) which affect gears oil lubrication (wear, welding, pitting, additivies, film strength, rusting, etc,.).

In figure 4 all results refer to the same oil type 140 A. D. but the frictional tractions vary with sliding speeds and loads. It is clear that the four curves show the same trend. It is seen that the no load condition exhibits the lowest possible frictional traction. Increasing the applied load the values of frictional traction become more higher, for the same sliding speeds.

Experimental data for three different types of low oil viscosities are showen in figure 5. This figure shows that with reducing oil viscosity the frictional traction increases. This is due to possible repture of oil film or metal-to-metal contact between gears teeth must be related to low vicosties. These low viscosities conditions even with sump lubrication can lead to boundary lubrication.

Figure 6 shows a comparison between the frictional traction measured by crook (1) and those measured and calculated by the present method. The first curve is measured by crook for roller lubrication. The second curve is for involute gear lubrication. While the third curve is the present theoretical results equation 4. The main feature of this figure is that the frictional traction rises, reaches a maximum and then falls at sliding speed of 380 m/s. This feature is also displayed by the theoretical frictional traction equation 4. There are qualitative similarities between the experimentally values of friction and those calculated theoretically.

The dissimilarity (Fig. 6) may be due to the unknown phesical content of the two types of oils, or the form of contact for the two cases. i.e. the present analysis may have area of contact, while crook's experimental for rollers may have point of contact.

CONCLUSION

For oil 116 and 112 (Fig 3) there are optimum condition where these two oils can working over a wide region of sliding speeds. (i.e. between 150 to 380 (m/s) with minimum friction loss at a difinite load.

For every oil (Fig. 4) there are a difinite loads which give a less friction loss i.e loads from 147×10^5 to 294×10^5 dyn. cm^{-1}

An equation in the form of eq (4)

have been deduced theoretically to calculate the suitable oil for any load and sliding speed.

The influence of increasing oil frictional heat in reducing viscosity at high sliding speeds is illustrated only at low oil viscosities. For well prepared lubricating gears oils, the frictional traction increases with speed, load, and oil viscosity at a definite time.

Conversely, for light lubricating oil, or at low oil viscosity, the frictional traction decreases as the viscosity increasing. The increasing of frictional traction at low oil viscosity, is due to the occurance of metal to metal contact between gears teeth.

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Table (1) Important Physical Properties of the MASR oils.

Oil Type	specific weight at15/4 C°	flash point	REDWOOD at-second	ENGLER at-degrees	S. A. E.
Diesel-30	0.89	455	173	8.5	30
Diesel-40	0.894	465	215	11	40
Diesel-50	0.897	475	320	17	50
Gears-140	0.899	530	635	35	140
Hypoid for Gears 90(A.D.)	.91	405	270/320	15.5	90
Hypoid for Gears 140 (A.D.)	.93	415	600/1200	41	140
Gears-112	--	410	190	10	112
Gears-116	--	415	275	14.5	116
Gears-125	--	425	425	23.5	125
Gears-138	--	430	750	43	138
Gears-145	--	435	1000	62	145

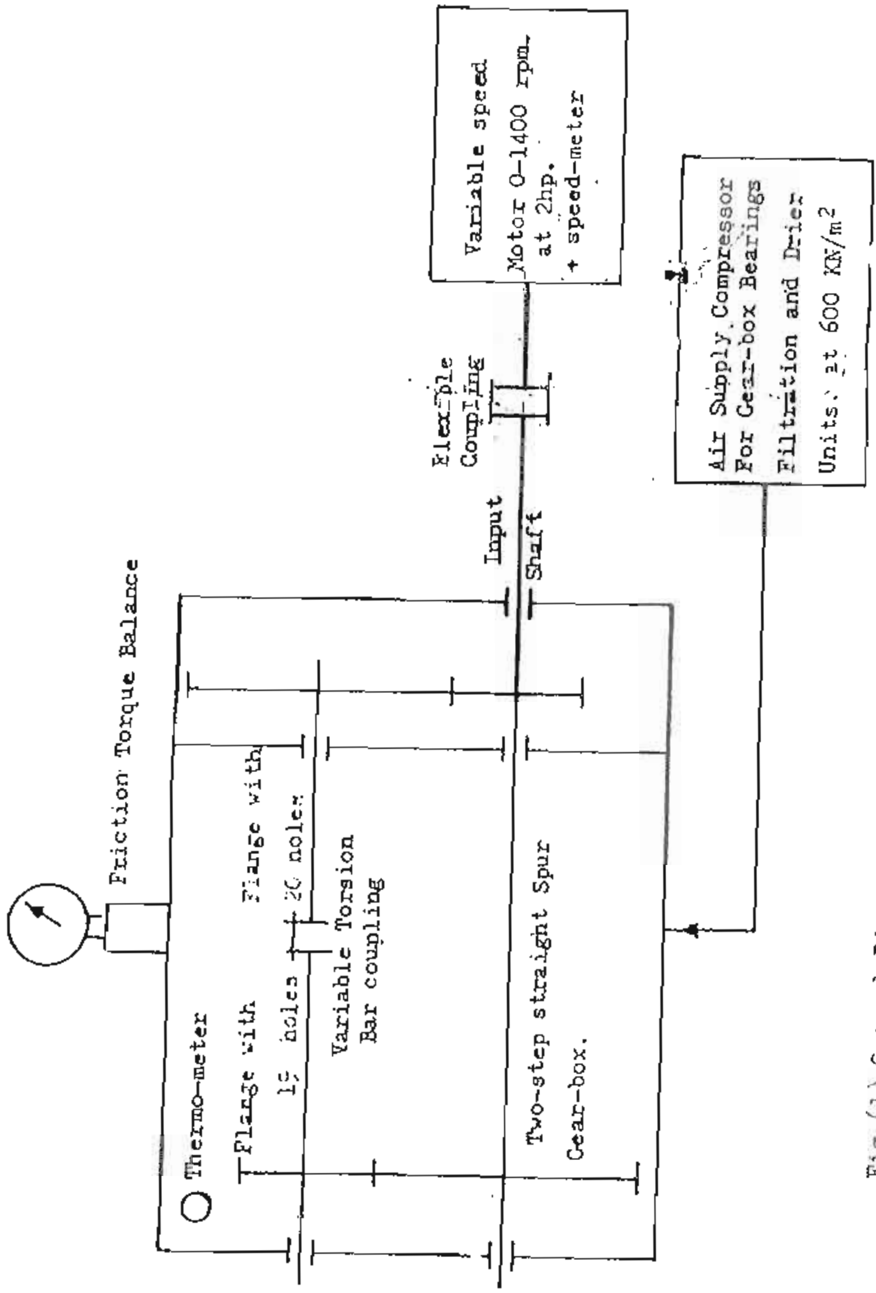


Fig.(1) General Diagram of Test Rig.

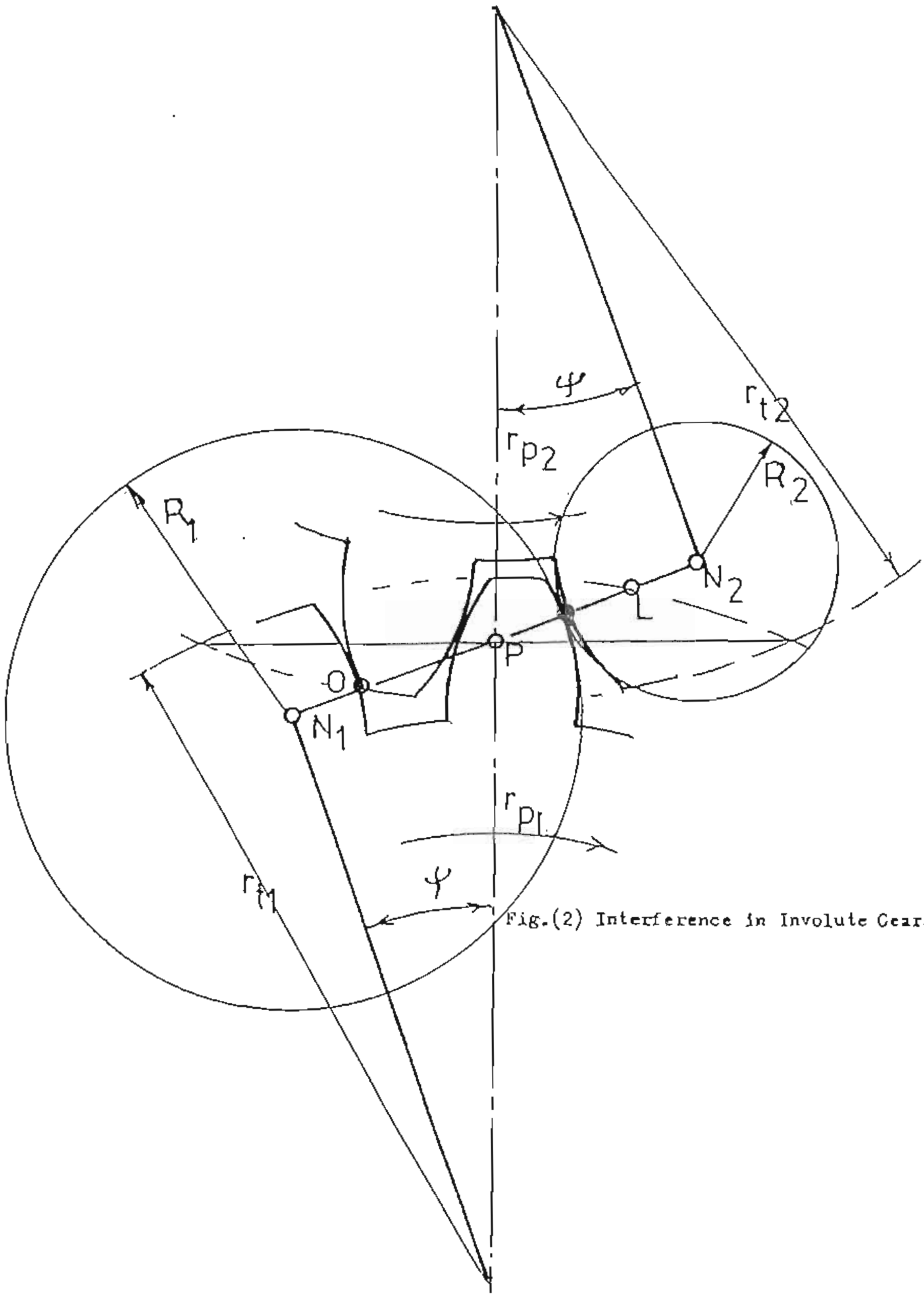


Fig.(2) Interference in Involute Gears

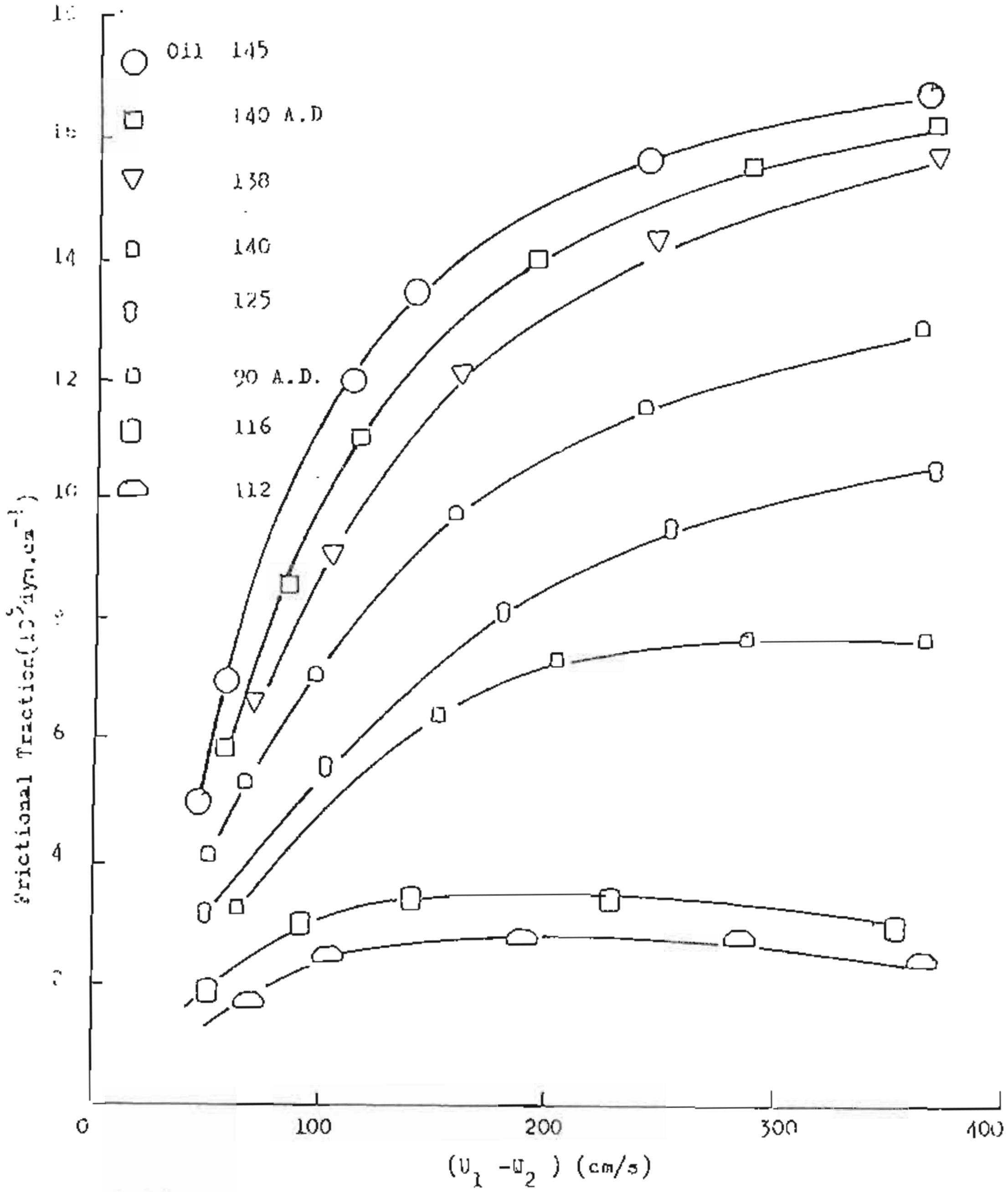


FIG.(3)Frictional Traction as a function of sliding speed at constant Load of 294×10^5dyn.cm^{-1}.and for Different Types of Lubricating Gears Oils.

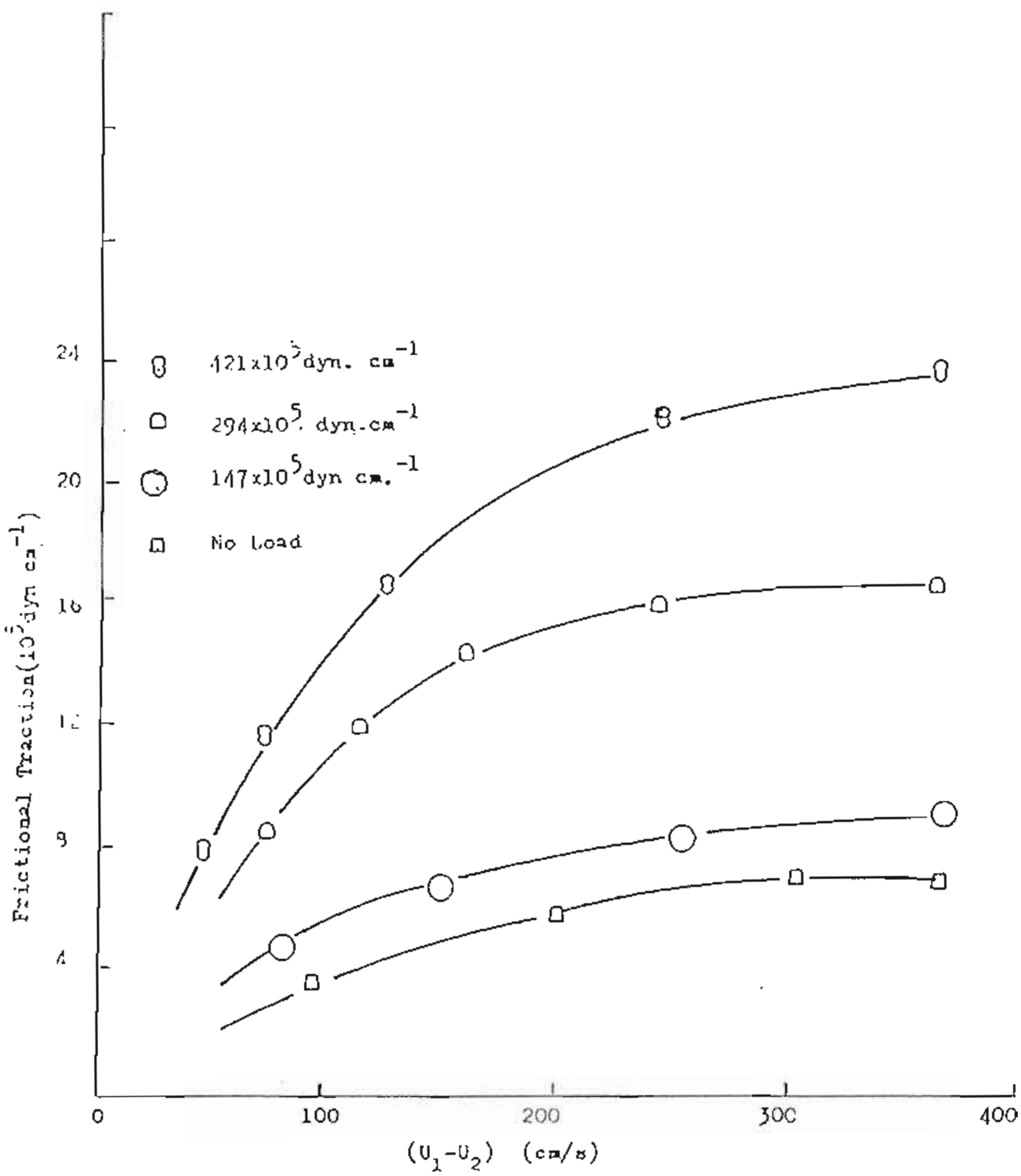
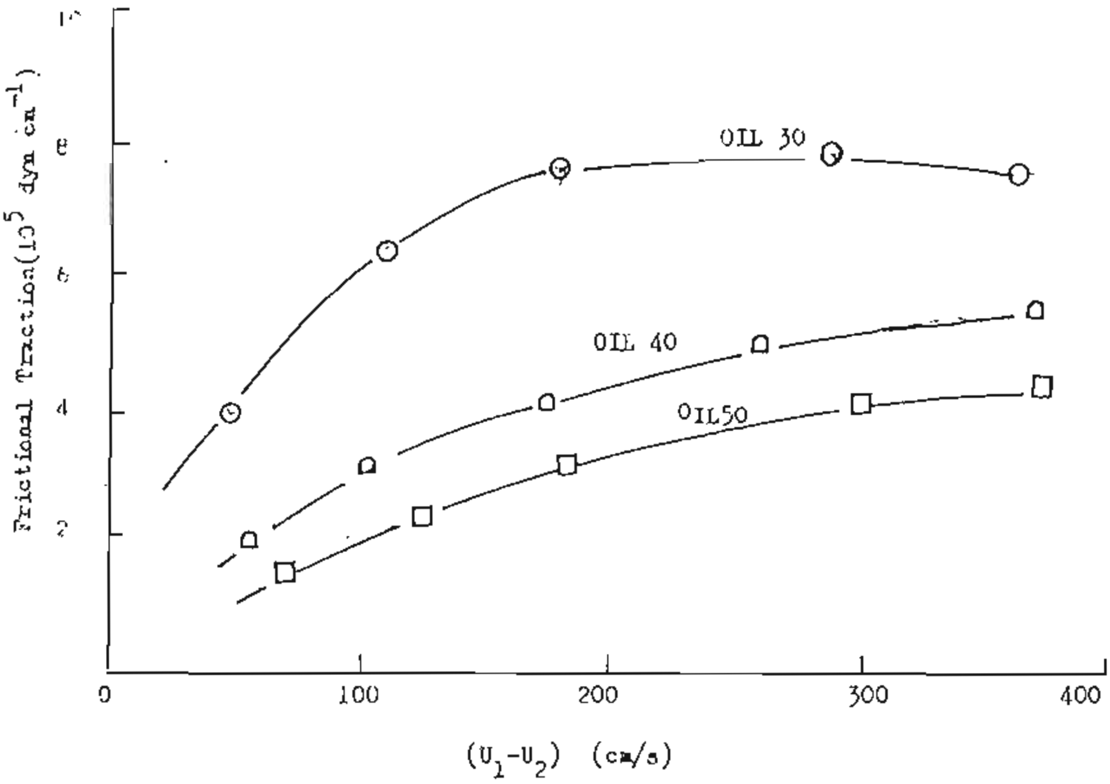


FIG.(4)Frictional Traction as aFuction of Sliding Speeds and Variable Loads For Lubricating Gear Hypoid Oil (140 A.D.)



FIG(5)Frictional Traction Versus Sliding Speed at Constant Load of 294×10^5 dyn cm.$^{-1}$ and for Different Types of Lubricating Oils.

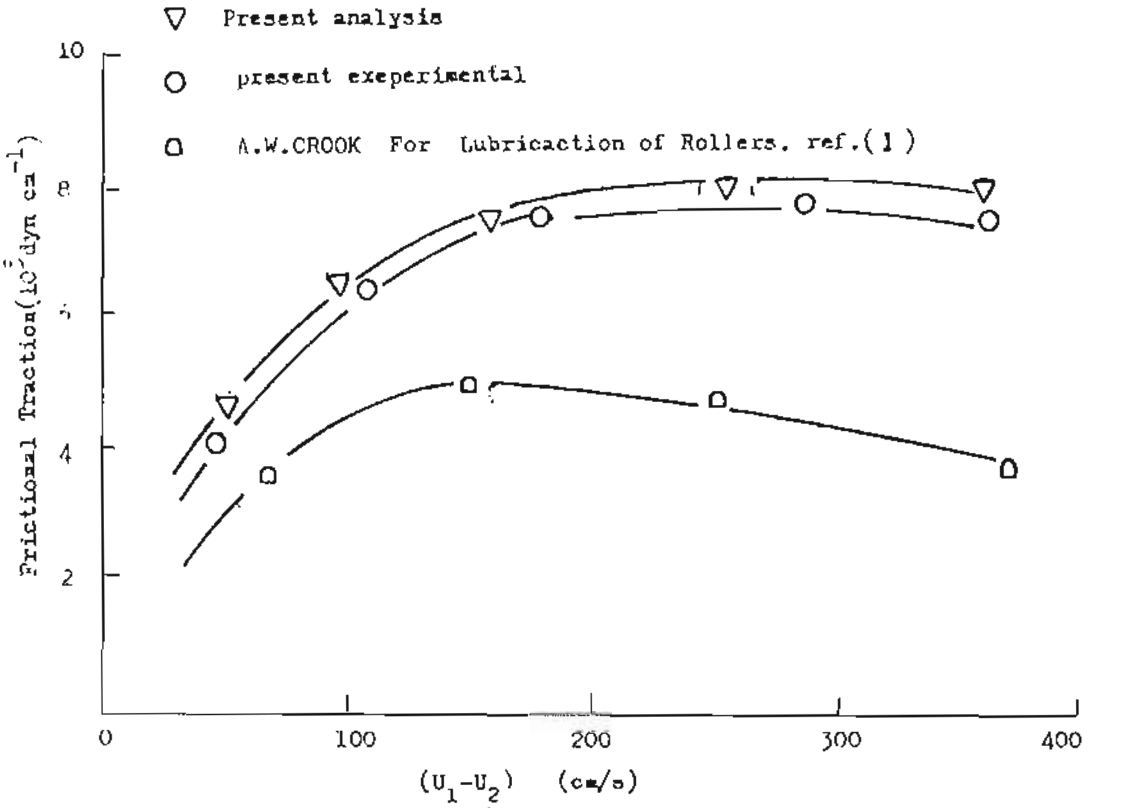


FIG.(6)Comparison Between Calculated and measured Frictional Tractions. For Constant Load of $294x10^5$dyn cm.$^{-1}$ and Lubricating OIL(S.A.E. 30).