Proceedings of the STLE/ASME 2010 International Joint Tribology Conference
IJTC2010
October 17-20, 2010, San Francisco, California, USA

IJTC2010-41101

Experiments on Grease Performance in Aircraft Landing Gear Pin Joints

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ABSTRACT

A pin joint simulation machine has been built to test a real landing gear pin joint under realistic loading and reciprocation conditions. The pin is loaded hydraulically using a hydraulic actuator to apply a fixed displacement cycle whilst measuring the reactive torque. The machine was used to measure the torque cycle (and hence friction coefficient) required to operate the joint.

In this work a method of evaluating different formulation greases has been proven. This involved measuring their frictional torque and also evaluating performance using a Sommerfeld type approach that displays the different lubrication regimes in the joint for different conditions. Measured friction coefficients were in the region of 0.02 to 0.12 depending on the joint load and articulating speed. In actual gear the surface sliding speed is low and so the joint operates in the boundary regime. The required torque and coefficient of friction have been related to the lubrication mechanisms occurring as a function of articulation angle, reciprocal frequency and applied axial load for lubrication starvation in a reciprocating journal bearing.

INTRODUCTION

This study focuses on the design of pin joints in aircraft landing gear. The articulation in landing gear systems is achieved by the use of a system of pin joints and members.

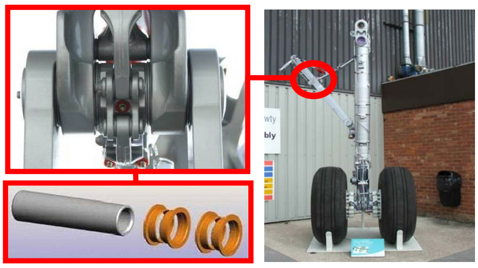

Figure 1: Photograph of landing gear and typical pin joint

The pin used throughout are from actual landing gear lower side-stay pin joints. They are hollow and manufactured from hard chrome plated 300M steel. The hard chrome provides a wear and corrosion resistant surface that also has micropores that can trap lubricant. The bush configuration consists of four aluminium bronze sleeves with the pin and bush contact lubricated by an aerospace grade grease. Each of the bushes have a circumferential groove to allow distribution of grease. In this instance landing gear articulation is ±50º during normal operation with a relative rotation of 12 deg/s, giving a surface velocity of 0.07m/sec.

The pin joint bears a close similarity to a conventional journal bearing. The science behind the lubrication of such configurations is well understood. The pressure is usually expressed simply as the load divided by the projected bush area. This presupposes the contact area extends over half of the bush (i.e. the wrap angle is 180º) and the pressure is uniformly distributed. This is not the case and there are several models to predict the wrap angle between a pin and bush and the resulting pressure distribution [1,2]. This is important because it is this pressure distribution and wrap angle that will affect the lubrication mechanism, and in turn the reflow time for the lubricant. The design and durability of landing gear pin joints therefore depends on the load capacity and torque requirement to articulate the joint. These correspondingly depend on the pressure distribution, load bearing capacity of the grease, and the friction between surfaces. This paper seeks to provide experimental measurements of the torque required to rotate the joint, its coefficient of friction, and relate this to the lubrication mechanisms provided by the grease.

TEST RIG

The pin joint function test rig was designed to simulate the loading and articulation of a pin and bush assembly. An adapted tension-torsion apparatus (Schenck POZ 0921) was used with a bespoke test head designed to load the joint. This apparatus used a double fork arrangement geometrically similar to the pin joint found on the landing gear upper to lower side-stay pin, as highlighted in Figure 1 with Figure 2 showing a schematic of the test head assembly. An Enerpac low height hydraulic cylinder provided the lateral loading. The pin was rotated using a torsional hydraulic actuator connected to a rotational spline drive which enabled articulation. ±40 degree rotation was the maximum rotational range limit of the rig.

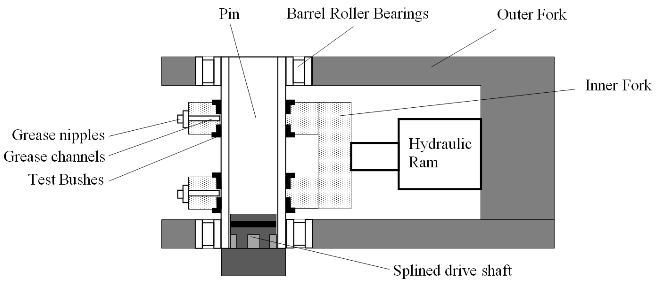

Figure 2: General assembly diagram of pin housing

Instrumentation

The torsional actuator was fitted with a strain gauge based internal torque sensor. A sinusoidal drive signal was inputted via a function generator. This provided the input signal for the displacement controlled reciprocal rotation of the pin.

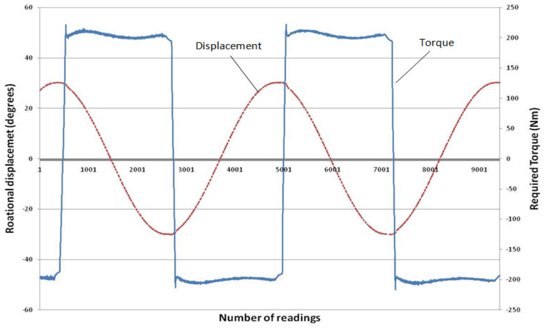

Figure 3: Raw torque and displacement data recorded for a pin loaded at 45kN and articulating at ± 30 deg/sec.

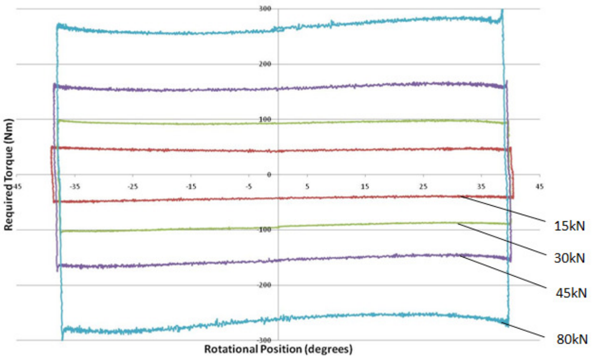

Figure 4: Recorded torque cycles for a range of loads rotating at 0.07 m/s ±40 degree rotation

Instrumentation

Torque cycles for different applied loads are shown in Figure 4. As the lateral load increased, the resistive torque increases. For the compilation of the data used in this study, the measured values were averaged across each test and load/speed condition, with three repeats completed for each condition.

Friction coefficient

The required torque *T,* has been converted into friction coefficient, μ according to

$$T = \mu P \frac{D}{2} \qquad (1)$$

Where *P* is the normal force and *D* is the pin diameter.

It should be noted that equation (1) is a simplified relationship. References [3] and [4] describe in detail a version of equation (1) that properly takes into account the fact that the contact pressure acts normal to the surface and therefore at an angle to the load application direction.

RESULTS: Figure 5 shows the plot of coefficient of friction against mean sliding velocity. The friction coefficient is not constant as the lubrication regime is changing with load and speed, with the higher axial loads squeezing the grease out of the contact resulting in an increase in torque and friction coefficient. Also shown is the reduction in required torque at higher reciprocating frequencies and therefore surface speeds. At higher speed, a thicker lubrication film can be created by the grease resulting in a lower required torque for these conditions.

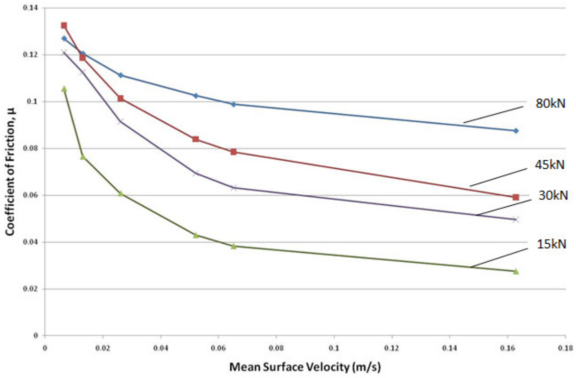

Figure 5: Coefficient of friction against mean surface velocity, for ±40 degree rotation.

Comparison of Different Lubricants

A comparative study of four different aerospace greases was performed. Each lubricant is used commonly in aircraft landing gear applications. Table 1 shows the physical properties of these greases. Each lubricant was tested using a range of loads from 3-80kN and rotational frequencies 0.017-0.417 Hz. Between each test the system was fully degreased to prevent any form of cross contamination.

	Thickener	Mean Hertz Load, kg	Oil Viscosity @ 98.9^{o}C
Lubricant A	Microgel	57	5.8
Lubricant B	Lithium Complex	45	3.4
Lubricant C	Organo-clay	38	5.7
Lubricant D	Clay	32	6

Table 1: Properties of test greases

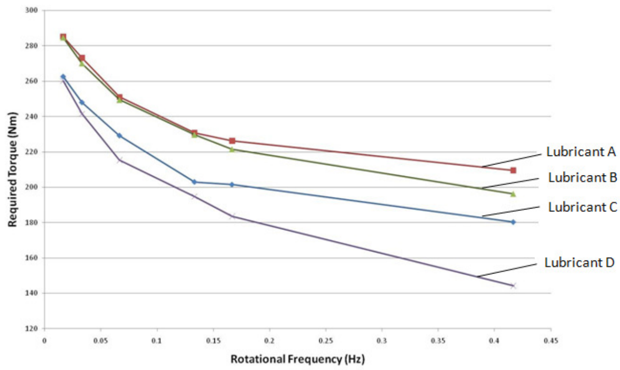

Figure 6: Torque curve for various lubricants at 80kN, for ±40 degree rotation

For the same load speed operation there is a significant difference between the four lubricants. The principal difference is in the rate of which the required torque and friction falls with speed. This will be a combination of the base oil viscosity, thickener consistency, and the rate of which grease can reflow back into the contact. All lubricants show an increase in friction with load, as the grease is squeezed out of the contact under high load.

Lubrication/starvation conditions

In an attempt to investigate possible starvation/ lubrication conditions in the reciprocating pin joint, an experiment was conducted keeping the mean surface speed constant throughout through changing the angle of rotation and rotational frequency accordingly to produce the desired speed. The articulation angles used were ±20, ±30, ±33 ±35 and ±40 degrees. This can be seen in Figure 7.

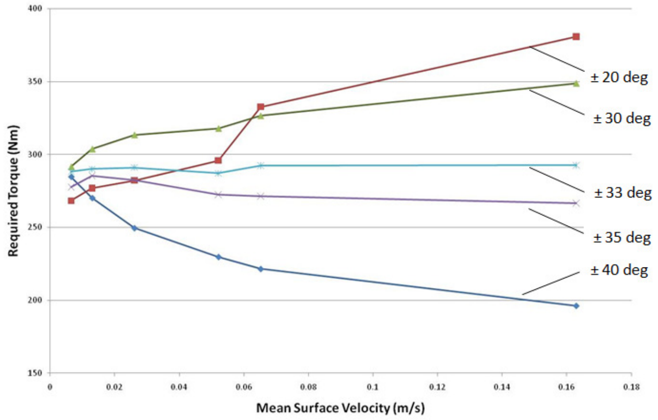

Figure 7: Torque Curve plot against Rotational Frequency for a range of angular rotations under 80kN load.

Visible in Figure 7 is the transition between starved and non-starved lubrication conditions relating to the change in articulation angle. The transition point, shown by constant required torque value for all velocities occurs at around ±33 degrees. It was found that at slower rotational speeds the torque and friction coefficient do not differ much between the articulation angles, however as the rotational frequency increases, the spread in friction values for different degrees of rotation increases dramatically.

As the mean surface velocity was kept constant throughout; with the larger slower stroke of the ±40 degree rotation, more grease was able to be drawn into the surface contact with each rotational cycle of the joint, therefore reduced torque was recorded at higher speeds. With the faster shorter stroke of the ±20 degree rotation, it was not possible to draw in the required grease into the contact. As the rotational frequency increased, it is believed that the short fast 'rubbing' movement caused a dispersal of the grease from the contact, reducing the lubrication available, and in turn higher μ and required torque. It is believed this behaviour is caused through starvation conditions in the joint. It must also be remembered that as the joint operates largely in the boundary condition, it is unlikely that it will be possible to generate a full film.

Angle of rotation (deg)	Radians	Result
20	0.349	Starved
30	0.524	Starved
33	0.576	Starvation Boundary
35	0.611	Lubricated
40	0.698	Lubricated

Table 2: Angle of rotation for reciprocating journal bearing under 80kN load

Figure 8 demonstrates keeping the articulation angle and rotational frequency constant, and varying the load applied. It can see seen that the change in applied load also provides a transition between starved and lubricated bearing conditions, in this instance this is occurs at approximately 60kN.

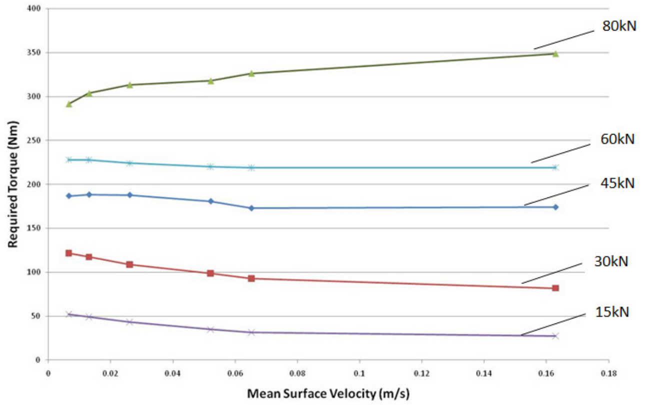

Figure 8: Torque Curve plot against Rotational Frequency for ±30 deg rotations under a range of loads keeping mean surface velocity constant.

CONCLUSION

It is believed that the amount of lubrication drawn into the reciprocating joint in each cycle is a function of the articulation angle ϕ, the surface speed u, and also the load applied to the joint, P.

$$\mu = f(\phi, u, P) \qquad (3)$$

It has been demonstrating at higher speeds; under lubricated conditions the required torque is reduced, and how under starved conditions this results in an increase in required torque. It is believed that the factor of 'reflow time' (derived from articulation angle and reciprocal frequency between each cycle) is critical in defining the lubricating conditions in reciprocating journal bearings such as this.

Further work is required to investigate the time it takes for grease to reflow into the contact patch and further deduce a relationship which can be used for future aircraft landing gear and journal bearing design.

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