

Lubricated Bearing Testing

Introduction

The kinds of wear and failure mechanisms are of interest:

Adhesive Wear:

This occurs under boundary lubrication, in other words, during stop/start cycles. Journal bearing tests, involving stop/start cycles, thus make sense.

Scuffing Resistance:

This is essentially failure, following the onset of adhesive wear. It is caused by thermal or mechanical overload during stop/start cycles.

Conformability:

This is a measure of the ability of a bearing shell to conform to the housing and shaft.

Embeddability:

This is a measure of the ability of the bearing to resist damage from entrained particles in the lubricant. These particles can either be those generated within the system (from ingested hard particles, wear debris etc) or, more seriously, residual particles generated during the manufacturing process.

Fatigue Strength:

This is a measure of the de-lamination life of the soft metal coating, caused by cyclic loading. This is sometimes referred to as "Babbitt Fatigue".

Tribo-corrosion:

This is the degradation of materials caused by simultaneous mechanical and chemical interactions between surfaces in relative motion.

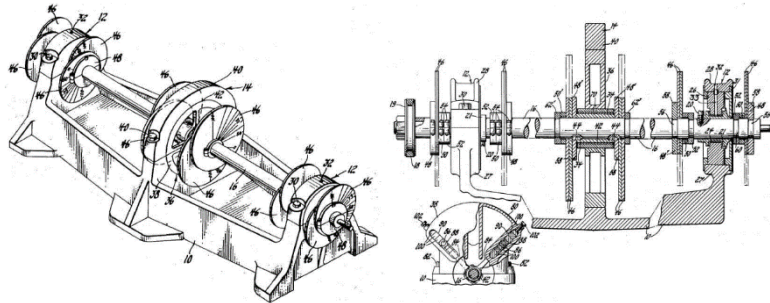
Cyclic Load Test Rigs – Fatigue Strength

The best way to describe this is to consider what has happened to date. A brief history of dynamic bearing test rigs follows. Most such rigs have been designed to model crankshaft bearings. These rigs can be divided into two basic categories:

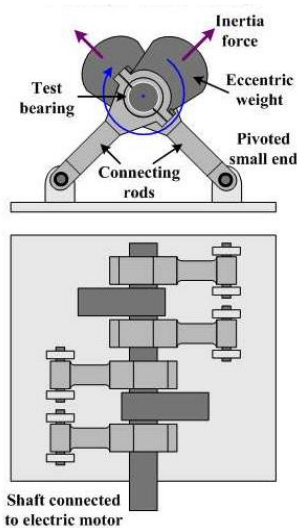
- Machines Applying Pulsating Alternating Loads - Full Wave Load Cycle

- Machines Applying Pulsating Loads Between Zero and Maximum - Half Wave Load Cycle

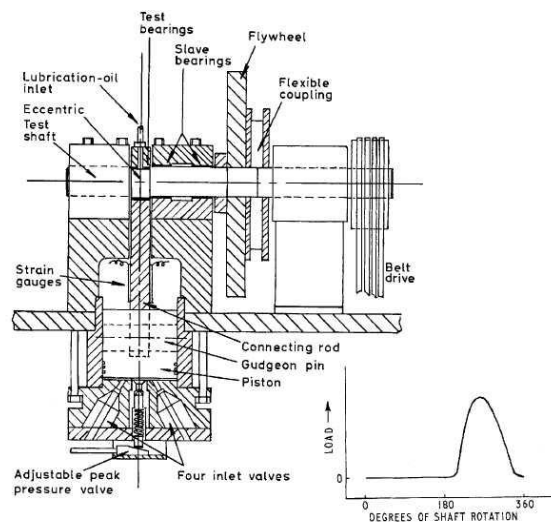
1945 - Original Underwood Machine Patent:



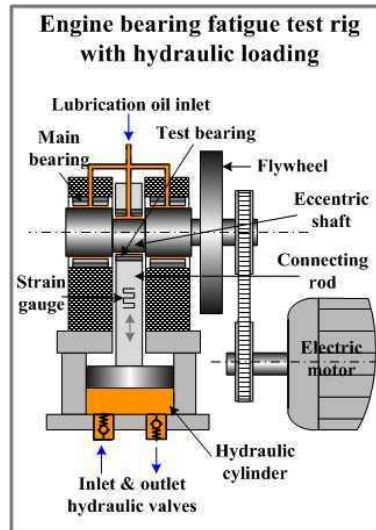
This machine, which is still used in various guises, involves out-of-balance masses generating a cyclic pulse on a test bearing. The bearing is loaded in both directions.



1958 - "Sapphire" Machine (Glacier Metals - now Mahle):



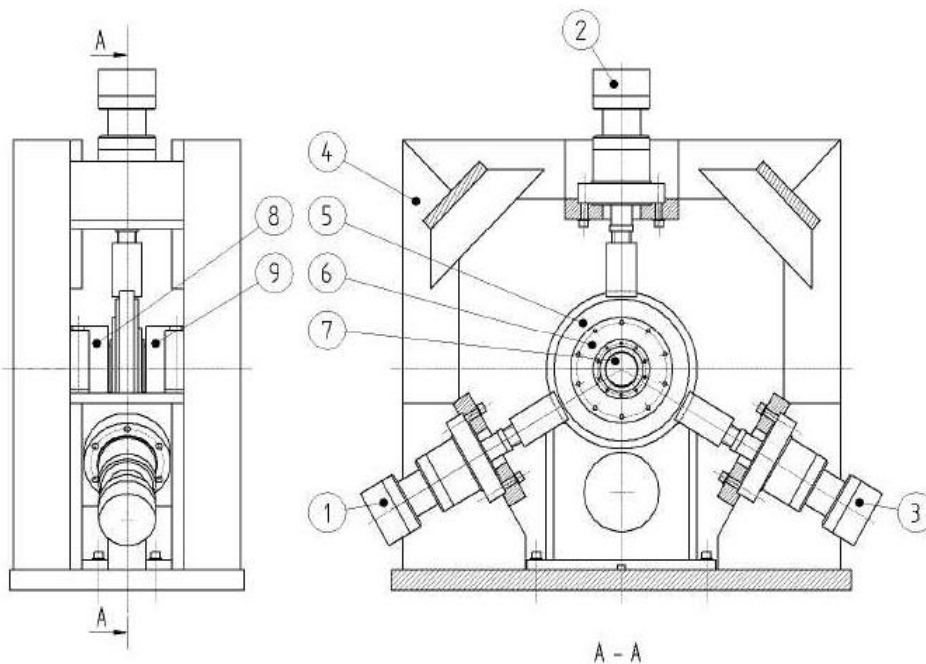
This is a half-wave actuator machine. A journal bearing sample is mounted on an eccentric shaft with a connecting rod attached to the equivalent of a dashpot, making the equivalent of a badly pulsing pump. This is however an ideal and simple method of generating a pulsating load.



Tests on the Sapphire machine are typically run at 2,500 rpm with a 50 mm diameter x 28 mm wide bearing and a load of 87 Nmm^{-2} , which equates to a dynamic load of approximately 122 kN.

Sapphire machines remain in service to this day with Mahle.

2000+ - Servo Dynamic Bearing Test Rigs:

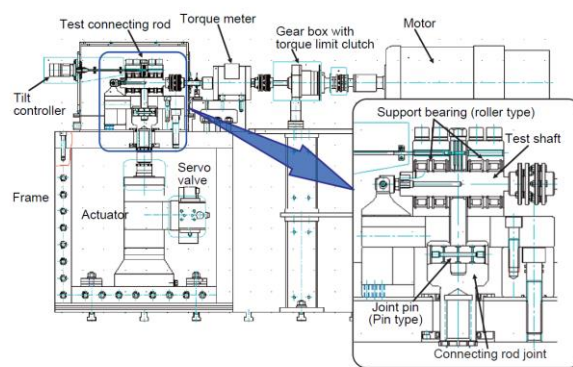


With either one or three actuators, depending on budget, these machines use high dynamic force servo actuators, usually fitted with multiple super high response Moog servo valves. In addition to high capital costs, the requirement for large numbers of Moog valves etc has a significant impact on servicing costs. The requirement for high capacity hydraulic power packs has a negative impact on operating costs.

To understand the basic issues and costs, consider the price of a high dynamic force servo hydraulic test machine from Instron or MTS, then build it into a bearing test rig.

Various examples of these more complicated and expensive servo hydraulic dynamic test machines are in existence including units developed by Daimler and, more recently, by Honda.

Proposal of New Bearing Simulation Rig Tester for High Efficiency Engine Bearing Development



Structure of new bearing rig tester

Table 1 Performance of new bearing rig tester

Item	Performance
Bearing slide conditions: Similar to actual engine	<ul style="list-style-type: none"> • Use of various entity connecting rods • Synchronization of X, Y direction loads • Generation of various load waves • Frequency: 300 Hz Load: ± 10 kN • Synchronization of bearing load with shaft angle
Tester protection from seizure	<ul style="list-style-type: none"> • Construction that can separate torque meter from motor torque
Repeatability of test results	<ul style="list-style-type: none"> • Adjusting mechanism of bearing-shaft tilt • Roller-type support bearing • Multi-point measurement of bearing temperature

The Honda bearing rig, which is rated to ± 100 kN quasi static, actually achieves approximately ± 20 kN at 2500 rpm

All these rigs use full size bearing shells and as a consequence require dynamic loads of 100 to 120 kN. As a result, the supporting test assembly has to be massive and very stiff.

Adhesive Wear and Scuffing - Potentially Starved Contacts

The following arrangements can be used for:

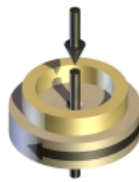
Friction measurement under boundary and potentially mixed lubrication

Adhesive wear tests

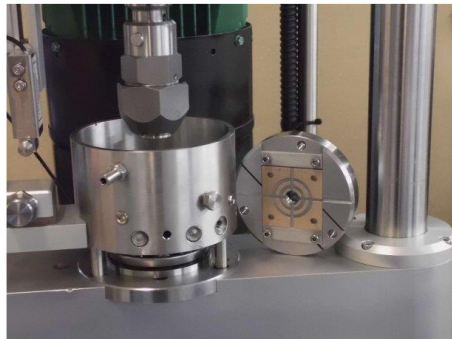
Scuffing tests

There may be the potential to run tests under hydrodynamic lubrication, but only with specially designed specimens and with adequate entrainment conditions.

Thrust Washer

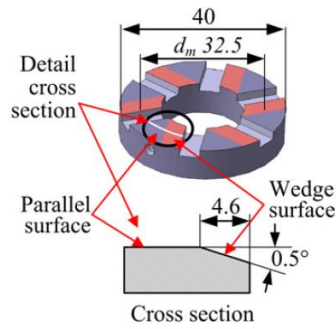


The thrust washer geometry is effectively a sliding face seal, in other words, a system designed to prevent lubricant from getting from one side of the contact to the other. For perfectly flat surfaces, lubricant entrainment is impossible, hence such a contact can only operate dry or under boundary lubrication.



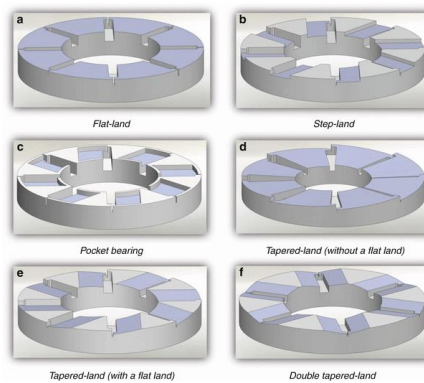
To facilitate lubricant entrainment, radial grooves must be machined in one surface, as in various designs of plain thrust bearing and in the JASO Suzuki test geometry. This arrangement will still only operate, at low speeds, under boundary lubrication and, at high speeds, under a mixed regime.

To produce more predictable lubricant entrainment, grooves with a converging wedge profile are necessary. With this type of geometry, and sufficient entrainment velocity, it should be possible to generate mixed and hydrodynamic lubrication regimes.

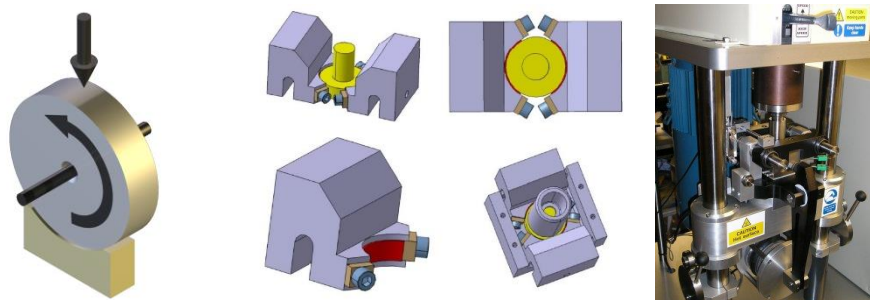


Tribological performance of thin overlays for journal bearings
 F Grün, I Godor, W Gärtner, W Eichlseder
 Tribology International 44(2011)1271–1280

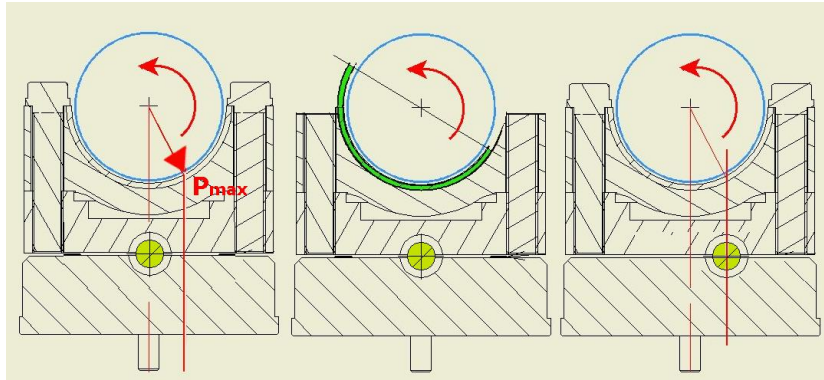
This arrangement is essentially a “tapered land/flat land” thrust bearing arrangement. Other geometries could be considered, for example, “flat land” with radial flow restriction and central oil feed:



Partial Journal Bearing



This sort of arrangement (University of Leoben) looks promising for lubricated friction tests, however, it only works satisfactorily for tests requiring starved lubrication.



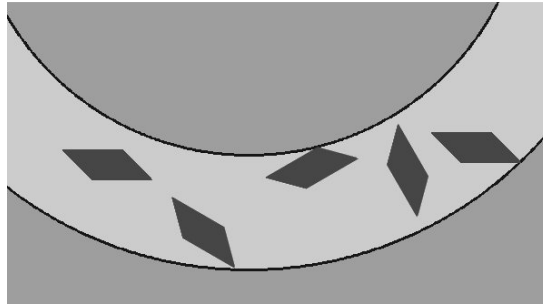
In a journal bearing, the point of peak pressure is not on the centre line. With a half journal bearing contact configuration (which includes conforming block on ring), this results in the inlet being closed, preventing lubricant entering the bearing contact. Designers of partial journal bearings address this problem by designing bearings with the required “pre-load” and “off-set”. Pre-loading is achieved by adjusting the bearing radius relative to the journal, to ensure converging wedge entrainment conditions. It will be noted that the greater the pre-load, the closer the contact becomes to a line contact.

Embeddability/Abrasion Tests

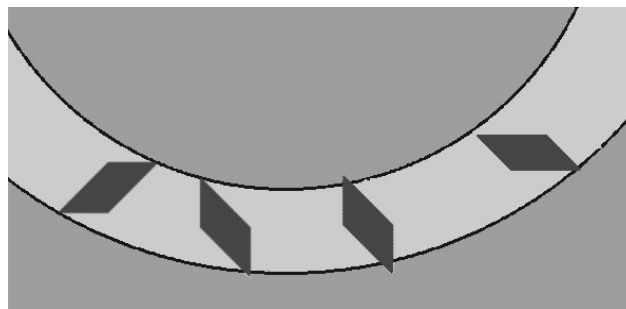
Journal bearings, except during start up, are essentially designed to operate with comparatively thick lubricant films between the loaded surfaces. Transit of particulate contaminants through the bearing gap can give rise to different wear mechanisms, each producing different wear rates.

When the size/gap ratio is small, worn surfaces may consist of a large number of small pits and indentations, usually with no obvious orientation in the direction of relative sliding, indicating either free movement of the particles through the fluid film and subsequent impact with the surfaces or the rolling of the lightly loaded particles through the contact. In both cases, the actual load on the bearing is of no relevance other than as a mechanism for setting the bearing gap. This mechanism, perhaps similar in nature to conventional polishing wear, has been termed “tumbling” wear.

With polishing wear, we would expect the free particles to roll through the tribo-contact in continuous contact with both sides. The term tumbling is used to describe the situation in which particles are not in continuous contact with both surfaces, but are free to tumble through the bearing gap.



Below a certain size/gap ratio, the particles are no longer free to roll through the contact, instead being dragged through the contact, generating grooving wear. As with the pitting wear mechanism, the actual load on the bearing is of no relevance other than as a mechanism for setting the bearing gap. It will be apparent that the load on a particle will be a function of the size/gap ratio, the relative hardness of the particle and the bearing surfaces and the number of particles sharing the load. It is not a function of the load on the bearing itself.



For surfaces of similar hardness, grooving wear may occur on both surfaces of the tribo-contact. For surfaces of different hardness, there are two possible mechanisms that may not be mutually exclusive. If the surface roughness of the harder surface is sufficiently large, particles may become trapped by asperities and be dragged through the contact producing grooving or micro-machining wear of the softer surface. However, increasing the hardness ratio between the two surfaces may cause hard particle to become embedded in the softer surface, resulting in more severe grooving wear on the hard surface.

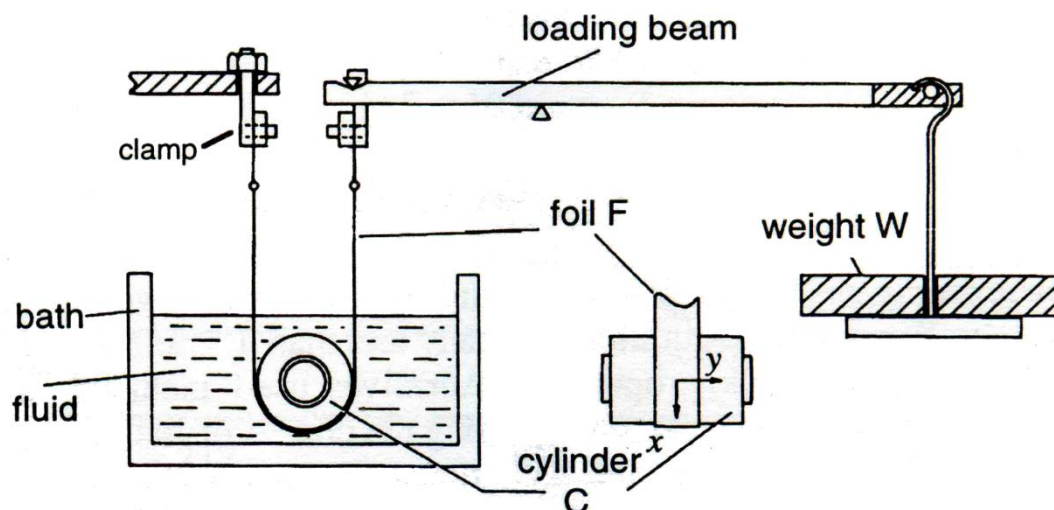
The critical parameters for an adequate test model are therefore:

1. A test configuration that allows precise control of the bearing gap.
2. A means of introducing abrasant particles of carefully controlled shape and size into the contact.

It will be noted that load (either static or dynamic) is not considered of importance except in as much as it may provide a mechanism for setting the bearing gap.

There are numerous models that could be considered, including bearing rigs with either static or dynamic loading, or with a well-defined mechanically set bearing gap. Examples of the latter would be simple half-journal bearing rigs in which a converging wedge is generated by running a journal of significantly smaller diameter than the corresponding bearing shell, in other words, a bearing with a significant "pre-load".

Of the more innovative solutions, the simple foil bearing test device, used at Cambridge University, is possibly the most interesting and potentially productive. A key feature of a foil bearing is that the lack of circumferential (bending) stiffness of the foil results in a more or less uniform lubricant film thickness, away from the immediate entry and exit zones. The entry zone provides a converging wedge geometry. The exit zone, at the point where the pressure is falling, produces a classic hydrodynamic "nip", with the film thickness falling to approximately 70% of the mean film thickness.



The foil bearing arrangement provides easy control of the lubricant film thickness and a ready mechanism for introducing abrasant into the lubricant.

As the ratio of particle size to film thickness is varied the different wear mechanisms characteristic of actual bearing wear can be generated, with corresponding variations in wear rate.

A major advantage of this arrangement, other than its simplicity, is that the film thickness, a key parameter in the experiment, is not affected by wear during the experiment or, unlike all other types of journal bearing rig, minor changes in diameter of the journal sample as manufactured.

By standardizing the film thickness and abrasive conditions, the rig can be used to compare abrasion resistance of different tribological pairs under both "tumbling" and "grooving" wear processes, or a combination of both.

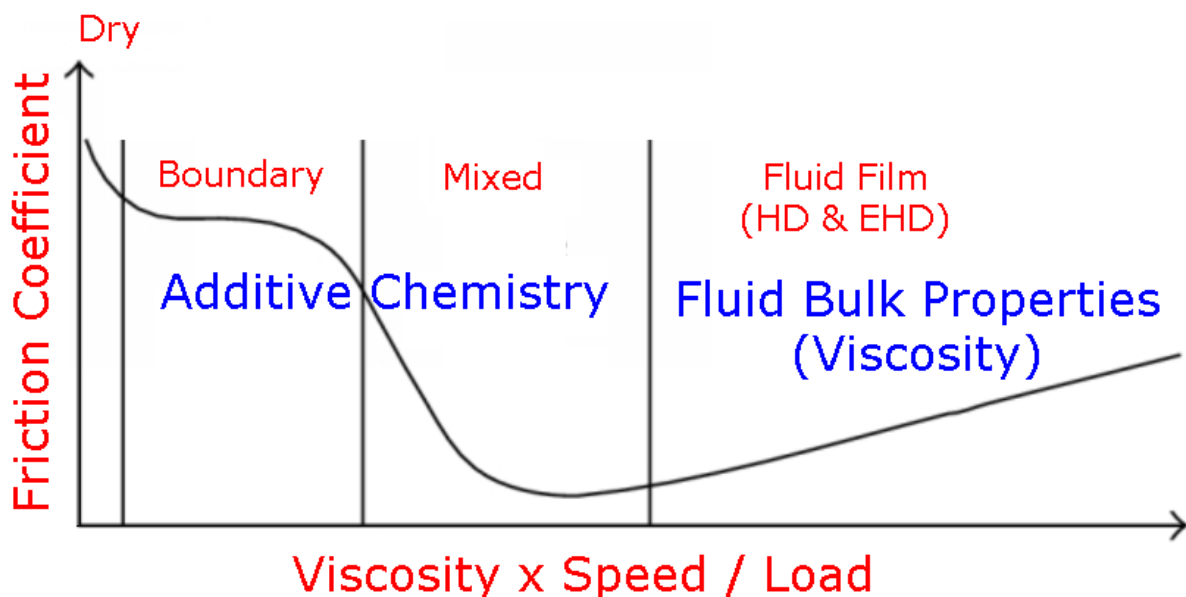
By standardizing tribological pairs, the rig can be used to rank the abrasiveness of different of different particle contaminants.

Typical foil samples are made of steel foil usually no more than 50 microns thick and 10 mm wide, wrapped around a 25 mm journal. This could of course be scaled up with ease to match more closely the diameter of actual bearings, but the effect of this increase in size is likely to be purely cosmetic.

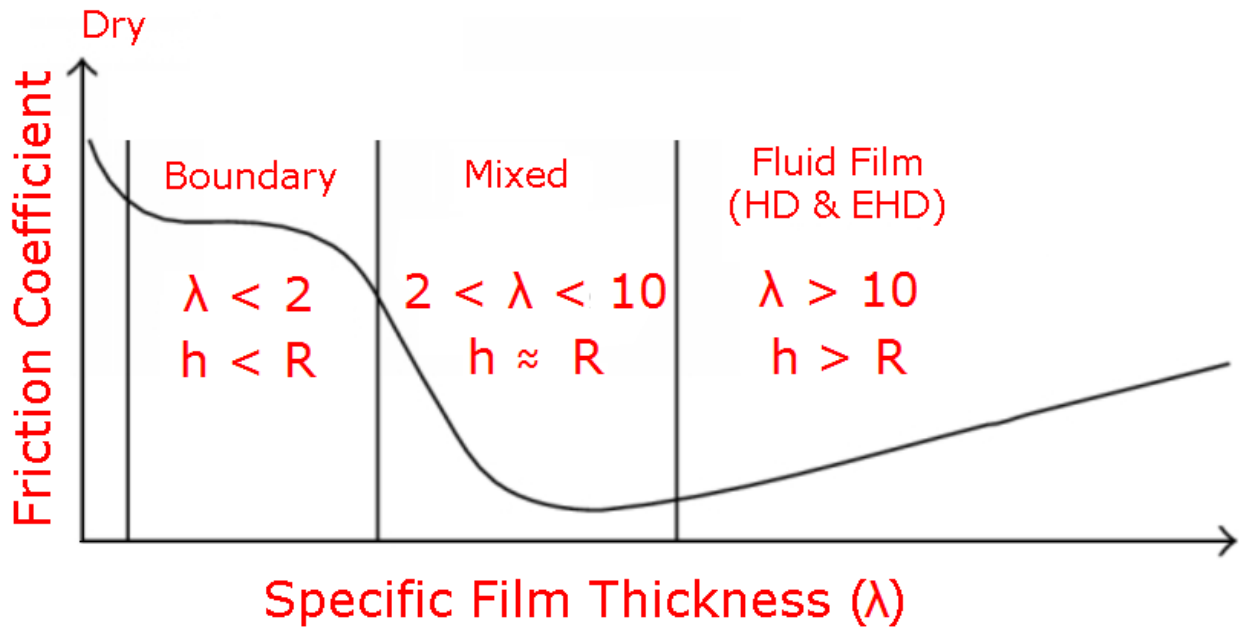
This is of course not a classical journal bearing rig, in the engineering sense, but is a tribologist's model system.

Tribo-corrosion Tests – Speculative Proposal

One of the key requirements of any lubricated tribological test is to know under which lubrication regime or regimes the contact is running, in essence, the operating point on the Stribeck curve. This is because under different regimes, different lubricant properties are at play. For example: viscosity index improvers, which affect response under fluid film lubrication, have little or no effect under boundary lubrication. Similarly, surface active friction modifiers and other additives, which are effective under boundary lubrication, have little or no effect under fluid film lubrication.



It follows that being able to control the lubricant film thickness, especially with reference to surface roughness, could be beneficial experimentally.



With this in mind and considering that a foil bearing produces a more or less uniform film thickness, with $h \propto U^{2/3}$, tests could be run with fully defined area contact, anywhere on the Stribeck curve.

One might consider running foil bearing tests with a copper or beryllium copper strip as the foil, to investigate issues associated with tribo-corrosion. Increasing lubricant temperature would obviously reduce viscosity, potentially moving the operating point down the Stribeck curve. Increasing speed could then be used to restore the operating point, to the required position on the Stribeck curve.