Guidance – Test Fault Investigation

Contents

Introduction 2
Useful assumptions and calculations when assessing data 2
- Friction coefficient 2
- Machine torque and power capacity 3
- Drive Ratio 3
- Friction Power Calculation 3
- Temperature calculation 4
- Friction power intensity 5

Particular Issues 5
- Vibration 5
- Reciprocating Sliding 5
  - Specimen alignment 5
  - Fixed specimen not flat 6
- Thrust washer samples 6

Introduction

When responding to customer's concerns, it is often necessary to identify whether there is a problem with the machine or a problem with the experiment; the trigger for the many enquiries is usually some kind of unexpected result.

Except in the case where there is an obvious machine failure, when an apparent fault is reported on a machine, we need to establish, at an early stage, whether the problem is a genuine fault with the machine or the result of an experimental issue: do an inappropriate experiment and you get a stupid result! We can only do this if we are provided with the following information:

Type of experiment being performed

Sample materials and sample dimensions

Whether dry or lubricated

Low speed data file (original .tsv, not post processed data)

High speed data (original .tsv, not post processed data), if available

Useful assumptions and calculations when assessing data

When reviewing test data, it is often worthwhile doing a few simple back-of-the-envelope calculations. The following may be useful:

Friction coefficient

We can usually assume the following:

μ	=	0.01 to 0.05	Hydrodynamic lubrication to mixed lubrication
μ	=	0.05 to 0.1	Mixed lubrication to boundary lubrication
μ	=	0.1 to 0.3	Boundary lubrication to scuffing failure
μ	=	0.3 to 0.7	Dry friction to galling

The friction coefficient is simply calculated from the friction force divided by the load. In a rotary tribometer, we measure friction torque, so to convert this from to a friction force we must divide by the mean friction radius at the specimens, hence:

Friction coefficient = Friction torque (Nm) / [Load (N) x Mean friction radius (m)]

The mean friction radius for a pin on disc or thrust washer test is the mean wear track radius. For a more complicated geometry such as the four ball test, the track diameter is less than the diameter of the ball. The calculation with regard to what this is and how the load is shared by the different balls is covered in a separate Guidance Note.

Machine torque and power capacity

It will be apparent that if the friction generated in an experiment exceeds the available torque capacity of the machine, the motor will go into over-current or otherwise stall. The torque-speed characteristic of an a.c. vector motor is slightly more complicated than that for a d.c. motor, but for rule-of-thumb calculations, it is perhaps enough to use the latter, in other words, assume that the motor produces constant torque from 0 rpm to the motor base speed and constant power between the base speed and the maximum speed. As an approximation, we can assume that the base speed of a 4-pole motor is 1500 rpm (1440 to 1460 in practice) and a 2-pole motor is 3000 rpm. We mostly use 4-pole motors.

$$\text{Motor Power (W)} = 2 \times \pi \times N \times T / 60$$

Where:

$$N = \text{Rotational speed (rpm)}$$

$$T = \text{Torque (Nm)}$$

As an example, consider a 2 kW 4-pole motor:

$$2000 = 2 \times \pi \times 1500 \times T / 60$$

Hence torque from 0 to 1500 rpm:

$$T = 2000 \times 60 / [2 \times \pi \times 1500]$$

$$= 12.7 \text{ Nm}$$

Note that as the speed is reduced, at constant torque, the power reduces progressively to 0 W at 0 rpm.

If we assume that the power remains constant above the base speed, then the torque decreases as the speed increases, so, for example, at twice the base speed, we have half the torque:

$$T = 2000 \times 60 / [2 \times \pi \times 3000]$$

$$= 6.35 \text{ Nm}$$

Drive Ratio

If there is a gear-box or drive belt pulley ratio between the motor and the test machine spindle, we can assume that the power at the spindle is the same, but the available torque will have been altered by the drive ratio. So, if we have a motor producing 6.25 Nm at 1500 rpm connected via a 1:2 pulley ratio to the test spindle, the test spindle will be rotating at twice the speed but with half the available torque.

Friction Power Calculation

As well as generating resisting friction, which must be less than the available machine torque in order for the samples to remain in motion, the tribological contact must also be capable of absorbing the friction power: the tribological samples are nothing more than a brake applied to the driving system. It is useful to calculate the friction power; this can be done in two ways, both of which amount to the same thing, of course:

From friction:

Friction Power (W) = Friction force (N) x Sliding speed (ms^{-1})

Where:

Friction force = Friction torque / Mean friction radius

Sliding speed = 2 x π x N x Mean friction radius / 60

From friction torque:

Friction Power (W) = 2 x π x N x T / 60

Where:

N = Rotational speed (rpm)

T = Friction torque (Nm)

For reciprocating sliding, we can assume that sliding speed is the mean sliding speed:

Sliding speed = 4 x Amplitude (m) / Frequency (Hz)

Temperature calculation

It is useful to do some simple temperature calculations, especially if the apparent fault is uncontrolled specimen heating, plus bluing or melting of the sample surfaces. Once again a back-of-the envelope calculation may be worthwhile. Keep in mind a useful number:

Specific heat capacity of steel = 500 J kg^{-1} $°C^{-1}$

Let us assume that we have two 0.5 kg specimens (sample and immediate tooling) being rubbed against each other. Assume that we lose no heat to the surroundings through conduction or convection. Assume that the frictional energy dissipated is shared equally between the two halves of the contact. Assume that we have a friction power (frictional energy dissipation) of 1 kW. This means that each half of the contact is subjected to 500 W of friction power input.

500 (W) = 500 J s^{-1}

= Mass x Heat capacity x Rate of temperature rise

= 0.5 x 500 x Rate of temperature rise

Hence:

Rate of temperature rise = 2 °C s^{-1}

Or:

Rate of temperature rise = 120 °C min^{-1}

Pump 1 kW of friction power into two small specimens and they will get hot very quickly, with the heat source the point of contact between the specimens!

The only way to run experiments avoiding uncontrollable frictional heating is to run with low frictional energy input and this is usually best achieved by running with low sliding speeds, which is exactly what Bowden and Tabor did in the 1940s and 1950s and Matveevsky did in the 1960s.

Friction power intensity

It is also sometimes useful to calculate the friction power intensity. This is the watts dissipated per unit area in the contact. For example, the area of the small ASTM standard thrust washer sample is 129 mm^2; that means if we are pumping 1 kW of friction power into such a specimen, assuming it is shared equally between the two halves of the contact, we have 500 W dissipated across 129 mm^2, or about 3.9 W mm^2. This is about an order of magnitude greater than the watt density for the average electric cartridge heater!

Particular Issues

Vibration

In the absence on materials with no mass or infinite stiffness, it is impossible to design a machine that does not have a resonant frequency. We can alter the resonant frequency of the system by adjusting the stiffness or the inertia, but we cannot eliminate vibration. Vibration can of course be generated by out of balance mechanical forces in the machine; these can of course be minimised by dynamic balancing in the design and manufacture of the machine. Dynamically balancing rotating machines is of course easy. It is much more difficult to dynamically balance reciprocating machines.

In addition to machine generated vibration, the tribological test contact itself can generate what we term “friction induced” vibration. Numerous well-known examples exist, including brake squealing, stick-slip in machine tool slides and windscreen wipers, violin strings etc. If the tribological test contact generates vibration, this is not a machine fault, it is a “system”, in other words a “tribological” response.

It is important to note that a mass (in the case of a tribometer, one side of a tribological contact) attached to a force sensor, is the equivalent of an accelerometer, which will measure both the required force input (friction) and the unwanted noise (any vibration induced acceleration forces). The “friction” measurement is always subject to some “noise”; the issue is to choose operating conditions that, where possible, optimise the signal to noise ratio.

Reciprocating Sliding

Specimen alignment

In a lubricated reciprocating contact, we can usually tell whether there is a problem with specimen alignment by examining the electrical contact resistance signal. If there is a high signal in one direction, but not the other, the specimen alignment may be suspect. Note that this misalignment may not have any effect on the instantaneous friction signal, but it may have an effect on the eventual wear.

Fixed specimen not flat

If the instantaneous friction signal is highly asymmetric, this may indicate that the fixed specimen plate sample is not flat, in particular when running tests with an area contact specimen. It only takes a few microns out of flat to generate a micro hydrodynamic regime as opposed to a boundary or mixed regime. If this happens, the instantaneous friction signal may become more sign wave in shape as opposed to square wave and the r.m.s. friction signal may fall sharply. A similar effect can occur with area contact specimens if wear debris becomes entrained in the contact.

Plate specimens should be machined and ground on both surfaces, to ensure they remain flat.

Thrust washer samples

For hydrodynamic lubrication, there has to be a mechanism for lubricant entrainment (lubricant to get into the contact) plus sufficient entrainment velocity to generate hydrodynamic separation of the surfaces. A thrust washer configuration, without radial grooves in one sample, does not provide a mechanism for lubricant entrainment; it is in effect a face seal, so runs dry. Note that the ASTM standard thrust washer test is meant for tests on self-lubricating (polymer) samples, not lubricated samples.