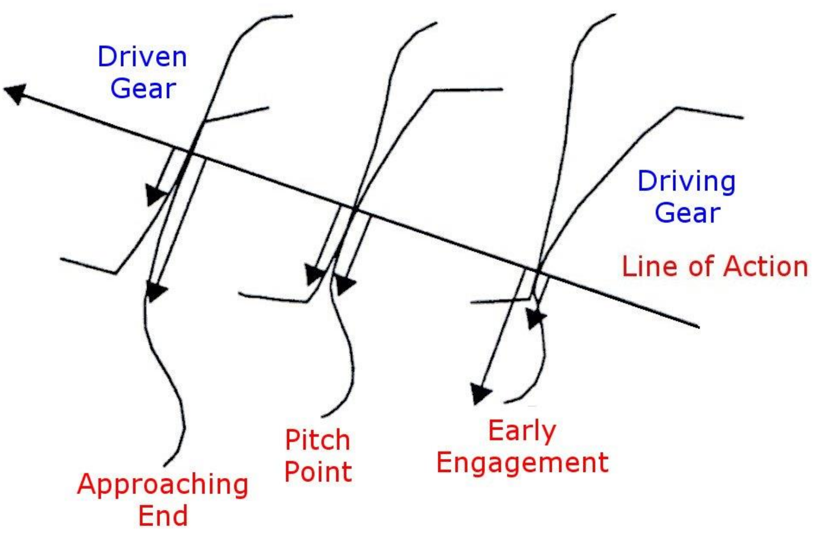
Guidance - Modelling Involute Gears with a Two Roller Machine

Background

One of the most commonly occurring misapprehensions is to assume, wrongly, that the pitch-line velocity of actual gears to be modelled, is the same as the surface sliding speeds of the rollers in a two-roller machine test model. The pitch-line velocity determines the contact time for gear tooth pairs. However, the rolling and sliding velocities between gear tooth pairs depend on the gear tooth profile plus the contact time. It is these velocities that should be modelled in any two roller experiment.

With involute gears we have two contacting surfaces with variable curvature, moving together with a complex combination of rolling and sliding. An added complication is that, away from the pitch point, there is load sharing between overlapping pairs of teeth, adding the uncertainty of dynamic loading, to an already complex system.



Involute gears of 20° pressure angle

Shortly after engagement, the surface of the driving gear is moving with a small velocity relative to the point of contact, whereas the driven gear has a much higher velocity. These are defined as the rolling velocities of the two surfaces.

The sliding velocity of the driven tooth across the surface of the driving gear is in the same direction as the rolling velocities, and is conventionally described as a negative sliding velocity.

At the pitch point, the rolling velocities are equal and there is no sliding in the contact.

As the point of contact nears the end of the contact path, the driving gear is moving faster than the driven gear, relative to the point of contact.

The sliding velocity of the driven tooth, across the surface of the driving tooth is in the opposite direction to the rolling velocities, and is conventionally described as positive.

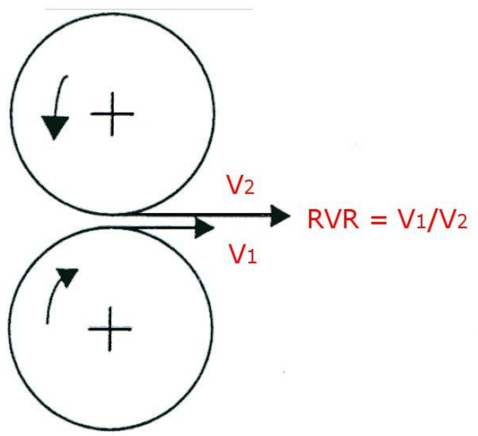
Note that in the case of the driving gear, sliding is always away from the pitch point. This imposes a tension in the surface layers and is the reason for the observed greater tendency of the driving gear to pit in the region of the pitch point.

Conditions for the driven gear are the mirror image of those for the driving gear, with sliding always towards the pitch point, imposing compressive forces to the surface layers, thus discouraging pitting.

Definitions

Rolling and Sliding Velocity Ratios

With gears, the relationship between rolling and sliding velocity requires careful definition. Merritt defines the rolling velocity ratio (RVR) as the ratio of the smaller to the larger velocity of the two surfaces relative to the point of contact, taking algebraic sign into account.



For pure rolling (without sliding):

$$RVR = 1$$

For pure sliding:

$$RVR = 0$$

The sliding velocity ratio (SVR) is defined as follows:

$$SVR = (V_1 - V_2) / (V_1 + V_2)$$

For pure rolling (without sliding):

$$SVR = 0$$

For pure sliding:

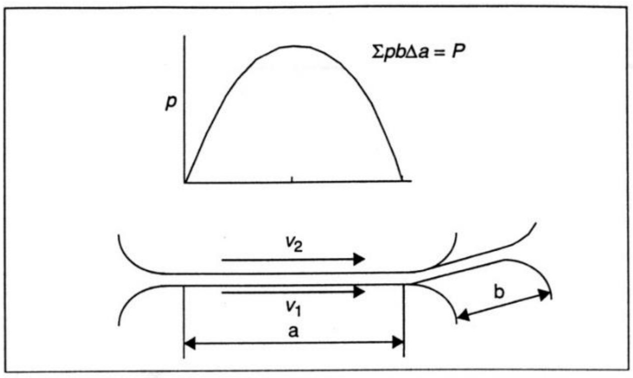
$$SVR = 1$$

Energy Pulse

The Energy Pulse is the product of the Friction Power Intensity (FPI) and the contact transit time. The EP therefore takes into account the length of time during which the material is subjected to energy input during its transit of the contact zone, where t_t is the transit time in seconds.

Energy Pulse: $\mathbf{EP = \mu\, P\, V_s\, t_t / A}$ $\quad Jmm^{-2}$

where	μ	=	friction coefficient	
	P	=	applied load	N
	V_s	=	relative sliding velocity	ms^{-1}
	A	=	area of contact	mm^2
	t_t	=	transit time	s



Generic sliding/rolling Hertzian line contact

The transit times for the contact are:

Upper body: $t_t = a / v_2$

Lower body: $t_t = a / v_1$

The Energy Pulse is analogous to the Archard Wear Law, however, the Energy Pulse equation uses the friction force rather than the applied load. This is perhaps more logical as it takes into account the work done in the contact.

Archard Wear Law: $\mathbf{\Delta V = k\ P\ V_s\ t_t / A}$ mm^3

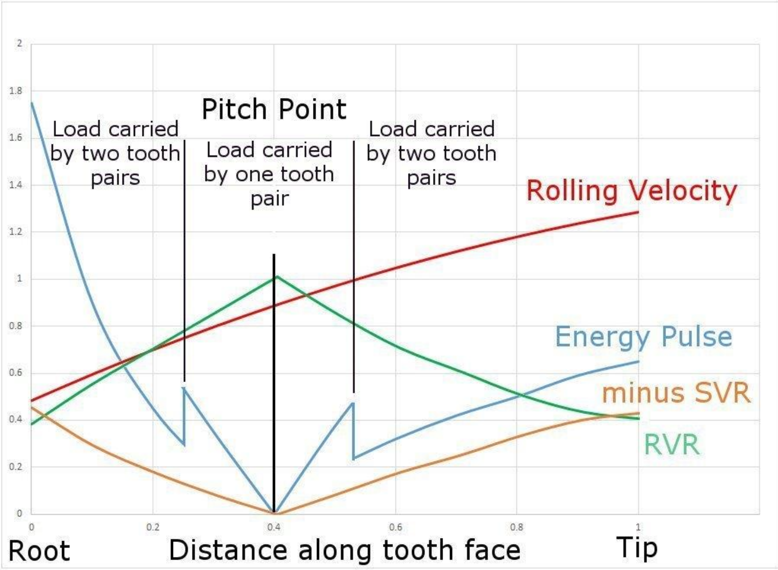
Each Energy Pulse can be regarded as an incremental contribution to wear or surface damage in the contact. The sum of the Energy Pulses can be used as a measure of the total wear.

Correct analysis of the EP in the real contact and subsequent modelling in the experimental design significantly enhances the chances of achieving a satisfactory emulation of sliding and combined sliding and rolling contacts.

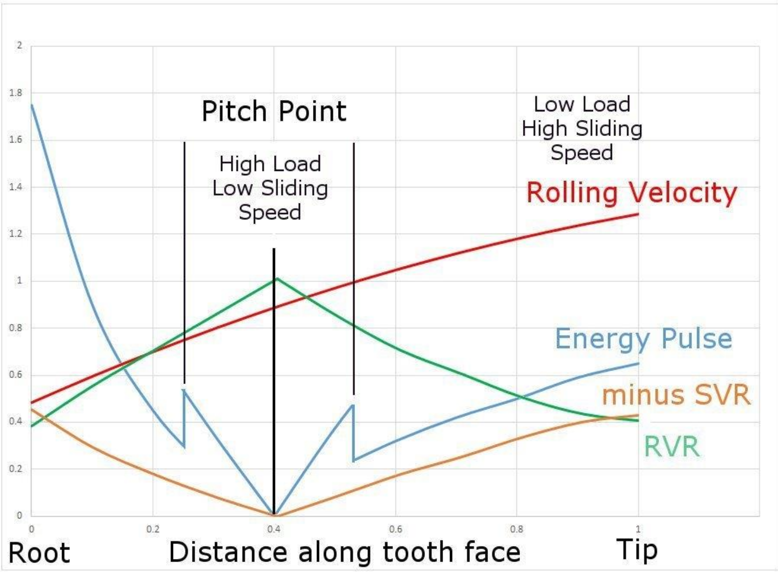
It is important to note that in many machine components there can be very high FPIs but, because the contact durations are short, the EP is low and hence the incremental damage is low.

Parameter Variation along Gear Tooth

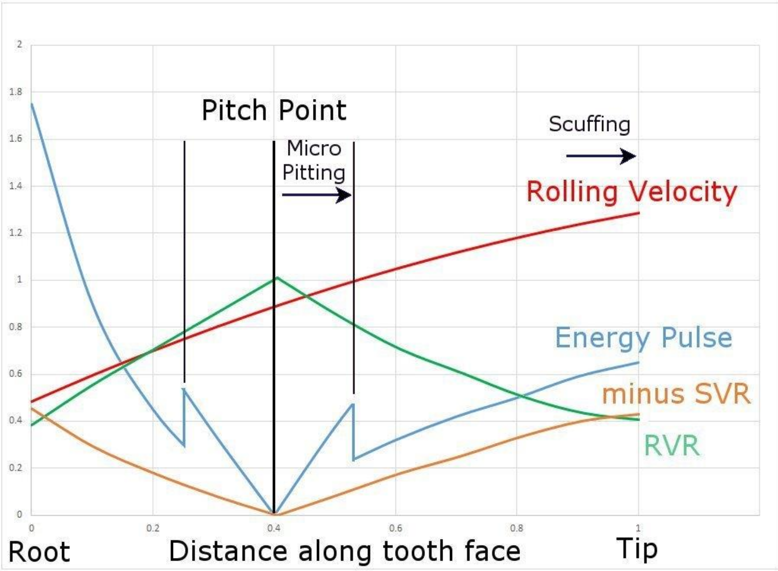
The following schematics are based on two 30 tooth gears with a 20° pressure angle. Note that SVR, which has a negative value, is here plotted as positive, so as to appear above the x-axis.



As the EP is a function of load, it is clear that the EP for a single tooth contact will not only vary with relative sliding velocity, but also as a result of dynamic loading and the sharing of load between successive pairs of teeth.



The EP is zero at the pitch point, despite the load potentially being at a maximum, because the SVR is zero. The EP increases in the direction of the root and the tip, as the SVR increases.



Zero EP at the pitch point provides the machinism for generating micro-pitting. High EP at the tooth tip produces conditions conducive to scuffing.

Modelling Gear Tooth Contact with a Two Roller Machine

It will be apparent that there is a significant difference between a pair of dynamically loaded involute gears and a conventional two roller machine test geometry. These can be summarised as follows:

	Gear Tooth Pair	Twin Rollers
Contact Geometry	Curvature varying with position on tooth	Fixed by disc diameters
Rolling Velocity Ratio	Varying with position on tooth	Fixed with steady state motor speed set-points
Sliding Velocity Ratio	Varying with position on tooth	Fixed with steady state motor speed set-points
Load	Varying with position on tooth	Fixed with steady state load set-points
Contact Pressure	Varying with position on tooth	Fixed with steady state load set-points
Energy Pulse	Varying with position on tooth	Fixed with steady state load and speed set-points

There is, however, one parameter that, for gears, is fully deterministic, but for two roller machines is less certain. With gears, the points of contact between pairs of teeth plus which pairs of teeth engage, each cycle, is determined by the design of the gears and the number of teeth on each gear. With a two roller machine, especially where surface speeds are independently controlled, the relationship between corresponding points on each roller is continuously variable.

Parameters affecting performance of both gear and two roller contacts are as follows:

- Contact pressure
- Lubricant film thickness
- Frequency of encounter
- Friction power intensity (FPI)
- Energy pulse (EP)

All these parameters are easy to define for the two roller contact, but less so for the gear tooth contact. However, it is clearly necessary that if we wish to model the complex operating conditions in a gear contact, with a simplified, steady state model, in a two roller machine, we must start by evaluating the conditions in the former. Because of the general complexity and uncertainty, significant assumptions are inevitable.

Contact Pressure

Whereas the contact pressure is relatively straightforward to calculate at the gear pitch point, the uncertainty caused by load sharing, combined with varying tooth curvature, renders simple calculation of contact pressure impossible.

Lubricant Film Thickness

The lubricant film thickness at the pitch point can readily be estimated using any of the established elasto-hydrodynamic film thickness calculations, for example, the Dowson and Higginson equation. There is, however, a significant caveat: the calculations all assume a fully flooded inlet to the contact and minimal side leakage. In practice, most gears run under conditions of starved lubrication.

Estimating the lubricant film thickness away from the pitch point clearly requires the calculation to be performed taking into account local contact pressure and local entrainment conditions.

Frequency of Encounter

Gear tooth contacts are intermittent, in other words, a given pair of gear teeth is subjected to a brief period of engagement, followed by longer period rotating out of contact, before once again coming into contact. The period out of contact allows time for dissipation of frictional heat and for lubricant additive chemistry to react. It is well known that in gears running at very high speeds, the frequency of encounter can be too short to allow the chemistry to work, resulting in scuffing.

FPI and EP

The FPI and EP clearly have no meaning at the pitch point and thus need to be calculated taking into account local contact pressure, hence local load and sliding velocity. This is, of course, by no means easy, hence a simple bench-mark estimate is to use the hertzian contact pressure at the pitch point and the mean sliding speed across the contact. This should provide basic order of magnitude values.

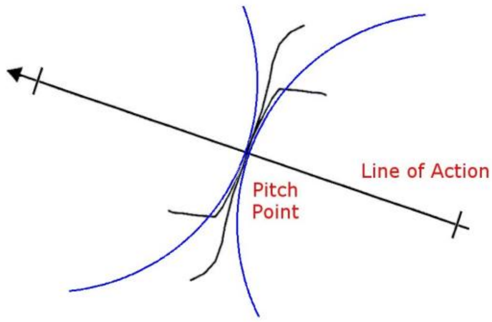
Practical Choices for Two Roller Experiments

It is important to state that there are no established and proven test configurations or test parameters for modelling gear contacts, hence there is no proven right or wrong answer. However, experiments based on sensible estimates and rational choices are likely to be more meaningful than randomly chosen test parameters.

Experimental Design

Roller Sizes

The contact between involute gears at the pitch point can be modelled as cylinders of the same local contact radius as the gears.



Using rollers of equal radius to the gear radii at the pitch point is a choice made by numerous experimenters. It would seem a rational choice, if one wished to perform experiments modelling conditions at the pitch point. It would seem a somewhat arbitrary choice, if one were intent on modelling conditions away from the pitch point. In practice, roller contacts are essentially scalable, so choosing roller diameters that conveniently provide contact radii somewhere within the range of radii of the gear teeth profiles would seem acceptable.

Machine Capacity

To determine the required two roller machine capacity we need to perform the following calculations, to match the machine capacity to the gear tooth contact speeds and pressures, having chosen suitable sized test rollers.

- Speed/RPM Calculations
- Load/Contact Pressure Calculations
- Machine Torque/Power Calculations

Typical machine torque and power calculations are as shown below, for 70 mm diameter by 10 mm wide contacts at 2000 MPa and 1000 MPa maximum contact pressure,with moderately realistic surface velocities:

INPUT DATA				2000	MPa
Roller 1 - Diameter	70	mm	Friction	1891	N
Roller 2 - Diameter	70	mm	Torque 1	66.185	Nm
Load	18.91	kN	Torque 2	66.185	Nm
Traction Coefficient	0.1		Roller 1 - Power	6.93	kW
			Roller 2 - Power	17.33	kW
Roller 1 - Speed	1000	rpm	Surface Speed 1	3.67	m/s
Roller 2 - Speed	2500	rpm	Surface Speed 2	9.16	m/s
			Sliding Velocity	5.50	m/s
			Friction Power	10.40	kW
			Rolling Velocity	6.41	m/s
			Slide-Roll Ratio	85.70	%

INPUT DATA				1000	MPa
Roller 1 - Diameter	70	mm	Friction	473	N
Roller 2 - Diameter	70	mm	Torque 1	16.555	Nm
Load	4.73	kN	Torque 2	16.555	Nm
Traction Coefficient	0.1		Roller 1 - Power	1.73	kW
			Roller 2 - Power	4.33	kW
Roller 1 - Speed	1000	rpm	Surface Speed 1	3.67	m/s
Roller 2 - Speed	2500	rpm	Surface Speed 2	9.16	m/s
			Sliding Velocity	5.50	m/s
			Friction Power	2.60	kW
			Rolling Velocity	6.41	m/s
			Slide-Roll Ratio	85.70	%

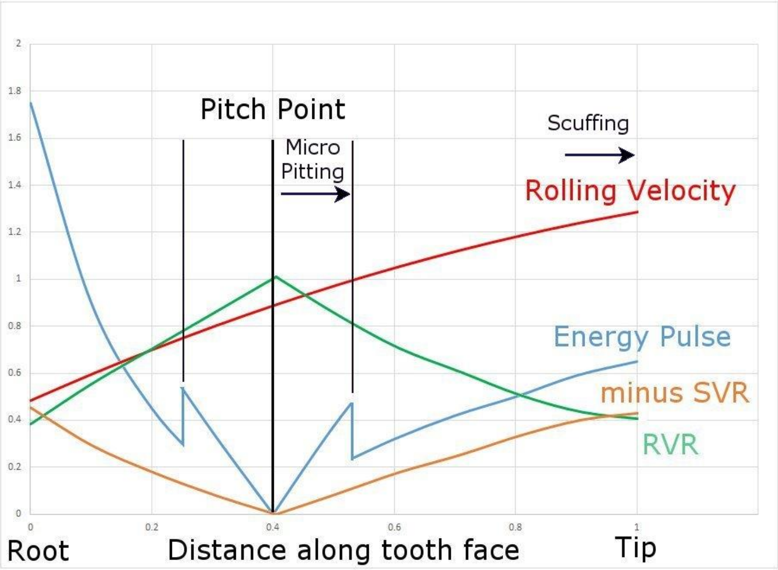
Having calculated the potential machine capacities, it is sensible to:

- Review FPI to confirm that it is sensible
- Calculate the nominal lubricant film thickness

Lubrication

The normal practice with two roller tests is to jet test lubricant into the in-running side of the roller contacts. To model starved lubrication, jetting lubricant against the out-running side of the contact may be worth considering.

Test Procedures



Micro-pitting Tests

Micro-pitting tests should be run at high contact pressures equivalent to those at or near the gear pitch point, but with low sliding velocities, hence low frictional energy input. Note that two roller tests have shown that negative sliding is more conducive to pitting than sliding in a positive direction. The level of asperity engagement can be varied by:

1. Varying the lubricant entrainment velocity
2. Varying the lubricant inlet temperature, hence viscosity

Scuffing Tests

Scuffing tests should be run at lower contact pressures equivalent to those at or near the gear tip, but with higher sliding velocities, hence high frictional energy input. Typical sliding speeds are between 5 and 20 ms^{-1}.

As scuffing is a wear transition (the onset of adhesive wear), tests sensibly involve increasing the severity of conditions within the contact, with the aim of precipitating the transition, but preferably not causing catastrophic failure.

There are various mechanisms for precipitating scuffing in a two roller machine:

Progressively increasing the load:

> As EHD film thickness is only weakly dependent on load, the main effect of increasing load is thus to increase the frictional energy input, hence contact temperature.

Progressively reducing the lubricant film thickness:

> There are two methods for achieving this. Firstly, by increasing the lubricant inlet temperature, hence reducing the lubricant viscosity. Secondly, by reducing the lubricant entrainment velocity.

Progressively increasing the frictional energy input:

> This is best achieved by increasing the sliding velocity in the contact, while limiting the entrainment velocity.

Running-in

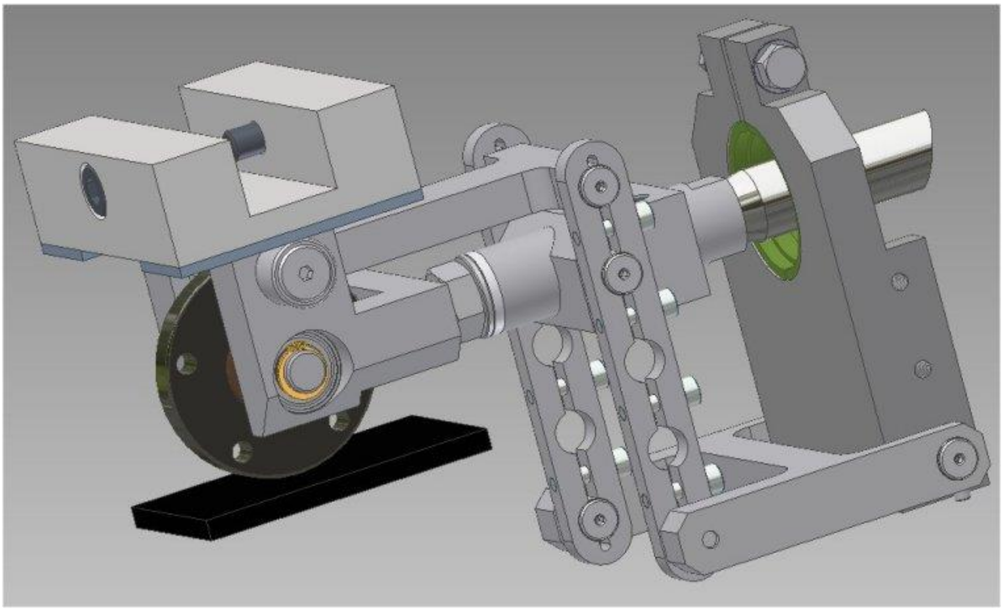
The need for satisfactory running-in of gears is well understood. There is a similar requirement to run-in test rollers. This is best performed at modest loads and low sliding velocities. Running-in performs two functions, firstly, generating plastic shakedown, which is the process of initial plastic deformation of the sub-surface, and secondly, flattening the peaks of the surface asperities. Shakedown imparts residual stresses to the sub-surface material, after which the contact should be elastic. This is analogous to a controlled work hardening process. The tips of surface asperities are flattened by a combination of plastic deformation and mild wear. Increasing the sliding velocity during running-in alters the shakedown behaviour and increases the risk of scuffing at the asperity tips.

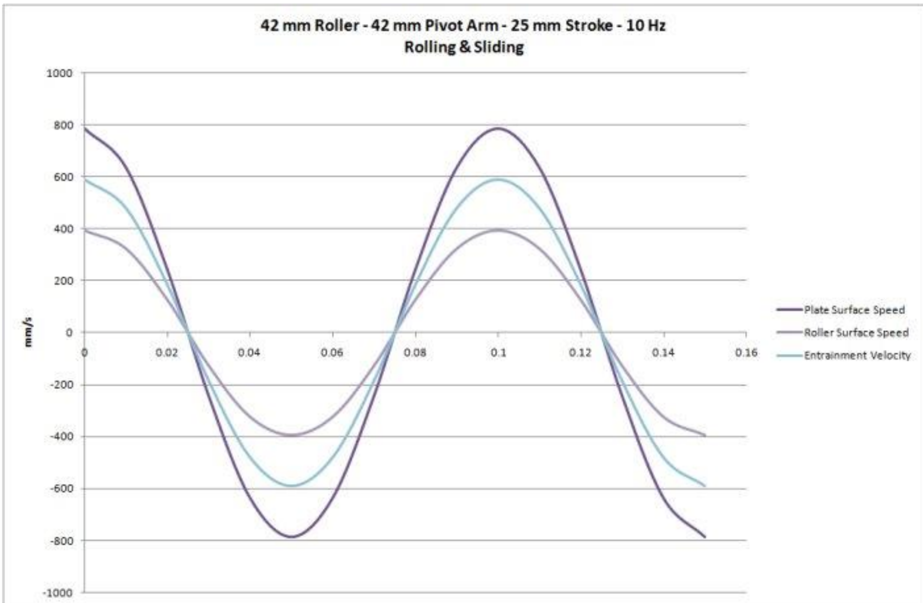
Sliding/Rolling Reciprocating Adapter

An alternative to the steady state slide-roll ratio behaviour achievable with a two-roller machine, a variable slide-roll ratio contact can be generated by imposing a degree of rotation on a test roller, as it is reciprocated against a flat plate.

In this device, a crowned or flat roller is reciprocated, with a linkage connected to the side of the roller opposite the tribo-contact. As the roller is reciprocated, the RVR changes with stroke position, with RVR = 1 (pure rolling) at mid-stroke, and RVR <1 (rolling and sliding) away from the mid-stroke position. Hence, the point of contact moves on both the surface of the roller and the surface of the plate, with a motion similar to a pair of gear teeth rolling backwards and forwards, on either side of the pitch point.

By changing the linkage position, RVR range can be changed.



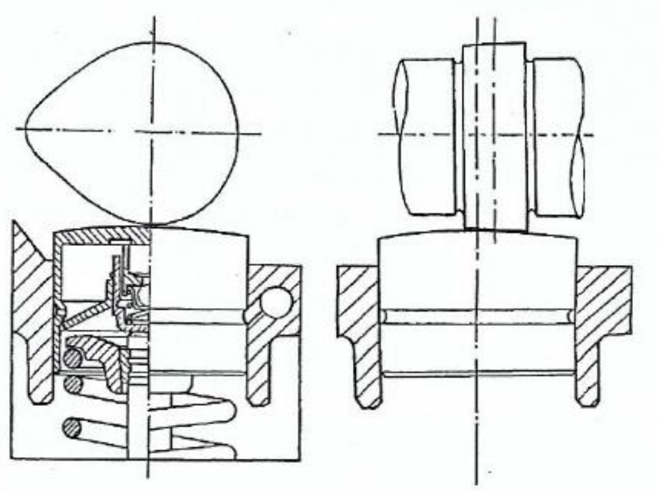


The device has not been extensively used for micro-pitting tests, but has been successfully used as a scuffing screening test, for gear oils. In this case, scuffing is precipitated in a controlled way, not by increasing the frictional energy input, but by increasing the contact temperature by electrically heating the plate specimen.

Cam-Follower Contact

I have covered the simplified analysis of a cam-follower contact in my lecture on lubricated friction measurement. The key wear issue with cams and followers is the contact at the cam nose, where a combination of poor entrainment conditions and high peak loads can give rise to scuffing, in other words, the onset of adhesive wear. Under these conditions, additive protection is essential.

Example - Ford Zetec Engine Cam and Tappet



Engine cam and tappet

This is quite a complicated “wedge on sphere” geometry, designed to promote rotation on the bucket follower, thus producing a circular wear track. For our bench mark calculations, we use a simplified geometry, comprising a “cylinder on flat” with a 10 mm wide contact width:

Nose Radius:	5	mm
Nose Load:	540	N
Contact Width:	10	mm
Lift:	9	mm
E*:	115 x 10^9	Pa
Speed:	1500	rpm

Calculate Peak Hertz Pressure:

$$P_0 = (WE^*/\pi \mathrm{LR})^{1/2}$$

$$P_{0-Nose} = 0.63\ GPa$$

Lubricant Film Thickness (Ertel-Grubin Equation):

$$\frac{\bar{h}}{R} = 1.37\left(\frac{\eta_0 \alpha 2\bar{U}}{R}\right)^{\frac{3}{4}}\left(\frac{E^* RL}{W}\right)^{\frac{1}{8}}$$

Assuming:

$$\eta_0 = 0.01\ Pa\ s$$

$\alpha = 2 \times 10^{-8} \, Pa^{-1}$

$$h_{Nose} = 0.186 \, \mu m$$

Calculate Lambda Value:

Assuming:

$R_a = 0.3 \, \mu m$

$\lambda_{Nose} = 0.62$ Boundary Regime

Only by doing the appropriate analysis can we establish the lubrication regime under which our component is running, hence determine test conditions, in an appropriate model system, in which to generate relevant and meaningful wear or failure data.