Guidance - Two Roller Machine Test Design

Slide-Roll Ratio

The Slide-Roll Ratio is the preferred parameter when modelling liquid lubricated contacts as it takes into account the lubricant entrainment velocity. It is defined and calculated as:

Slide-Roll Ratio = Sliding Velocity / Rolling Velocity

Where:

Sliding Velocity = |U1 - U2|

Rolling Velocity = ½ (U1 + U2)

For lubricated tests the **Rolling Velocity** is the lubricant **Entrainment Velocity**.

Hence:

Slide/Roll Ratio% = $200 \times |U_1 - U_2| / (U_1 + U_2)$

This definition of Slide/Roll Ratio means that for "pure sliding", in other words, for $U_2 = 0$, the **Slide/Roll Ratio = 200%**.

Slip Ratio

The Slip Ratio is a term that is used for nominally dry contacts, mostly in the automotive industry, in particular with regard to traction control and anti-lock braking systems. It is also used extensively by the rail industry.

Slip Ratio% is usually defined as:

Slip Ratio% = 100 x (Vehicle Speed - Wheel Speed) / Vehicle Speed

Slip Ratio% = $100 \times (U_1 - U_2)/ U_1$

Hence, for "pure sliding", in other words, for $U_2 = 0$, the **Slip Ratio = 100%**.

For unequal sized rollers, the convention is to define Slip Ratio as:

Slip Ratio = Sliding Velocity / Velocity of Larger Roller

For equal sized rollers, one roller is chosen to be U_1.

Test Applications

Constant Slip% against variable Speed

$(U_1 - U_2)/ U_1$	=	Constant
Plot:		CoT against U_1

U_1: Discrete set-point values incremented in steps

U_2: Single error correction loop at highest resolution with Slip% as conditional step

Example:

Set-point:	U_1	=	400 rpm
Loop:	U_2	=	400 rpm +0.05 rpm
Until:	Slip	=	1% (Sample CoT)
Set-point:	U_1	=	401 rpm
Loop:	U_2	=	404 rpm +0.05 rpm
Etc:			

If required, repeat at different loads.

Variable Slip% against constant Speed

U_1	=	Constant
U_2	=	Variable
Plot:		CoT against U_1

Example:

Set-point:	U_1	=	400 rpm
Ramp:	U_2	=	400 rpm +0.05 rpm
Until:	Slip	=	Required Maximum

If required, repeat at different loads.

Constant Slide-Roll ratio and varying Entrainment Velocity

Running experiments at constant slide-roll ratio, load and lubricant viscosity, with varying entrainment velocity, will generate the equivalent of a Stribeck curve, however, the speed of both test rollers, must be varied for each data-point.

$(U_1 - U_2) / (U_1 + U_2)$	=	Constant
$(U_1 + U_2)$	=	Variable
Plot:		CoT against Log $[½(U_1 + U_2)]$

A spreadsheet calculation table providing discrete speed set-point values is the simplest means of generating the relevant values, pre-test, rather than using real-time calculations within the test sequence, as such calculations may become unstable and not converge.

Example:

Set-point:	U_1	=	0.00 rpm
	U_2	=	0.00 rpm
Ramp:	U_1	=	38.19 rpm
	U_2	=	37.44 rpm
Etc:			

Slide-Roll Ratio	2	%	Constant
Roller 1 - Diameter	50	mm	Constant
Roller 2 - Diameter	50	mm	Constant

Entrainment Velocity	U1+U2	U1-U2	U1	N1	U2	N2
m/s	m/s	m/s	m/s	rpm	m/s	rpm
0	0	0	0.000	0.00	0.000	0.00
0.1	0.2	0.002	0.100	38.19	0.098	37.44
0.2	0.4	0.004	0.200	76.38	0.196	74.87
0.3	0.6	0.006	0.300	114.58	0.294	112.31

0.4	0.8	0.008	0.400	152.77	0.392	149.74
0.5	1	0.01	0.500	190.96	0.490	187.18
0.6	1.2	0.012	0.600	229.15	0.588	224.62
0.7	1.4	0.014	0.700	267.35	0.686	262.05
0.8	1.6	0.016	0.800	305.54	0.784	299.49
0.9	1.8	0.018	0.900	343.73	0.882	336.92
1	2	0.02	1.000	381.92	0.980	374.36
1.1	2.2	0.022	1.100	420.11	1.078	411.80
1.2	2.4	0.024	1.200	458.31	1.176	449.23
1.3	2.6	0.026	1.300	496.50	1.274	486.67
1.4	2.8	0.028	1.400	534.69	1.372	524.10
1.5	3	0.03	1.500	572.88	1.470	561.54
1.6	3.2	0.032	1.600	611.08	1.568	598.98
1.7	3.4	0.034	1.700	649.27	1.666	636.41
1.8	3.6	0.036	1.800	687.46	1.764	673.85
1.9	3.8	0.038	1.900	725.65	1.862	711.28
2	4	0.04	2.000	763.84	1.960	748.72
2.1	4.2	0.042	2.100	802.04	2.058	786.15
2.2	4.4	0.044	2.200	840.23	2.156	823.59
2.3	4.6	0.046	2.300	878.42	2.254	861.03
2.4	4.8	0.048	2.400	916.61	2.352	898.46
2.5	5	0.05	2.500	954.81	2.450	935.90

Zero Slide/Roll Condition

The zero slide/roll condition depends not only on the relative rates of rotation of the rollers but also on the absolute diameters of the two samples. These diameters may vary as a result of manufacture or wear. Depending on the precision required, the user may choose simply to enter the measured diameters of the test rollers, alternatively, the zero slide/roll condition can be established by searching for the zero traction condition. This may be done either under Manual or Automatic control.

At the start of a test, the measured diameters of the roller samples should be entered in the respective screen boxes.

It should be noted that as these values will vary with wear, a period of running-in is advised, before adjustments are made for the zero slide/roll condition.

Manual Control

First choose and set rotational speeds within the proposed range for the experiments, close to the zero slide/roll condition. Adjust the load to the required value.

Next reduce the rotational speed of the U_2 roller incrementally, by the smallest step achievable, using the on-screen toggles, allowing the friction force to settle between each adjustment.

Continue adjusting the speed until the friction signal switches sign, passing through zero. Adjust the U_2 disc speed up and down to allow the friction signal to switch from positive to negative and vice versa.

Finally, set the U_2 speed to give as near zero friction signal as possible.

If the roller diameters are correct, for pure rolling (the zero friction condition):

$N_1D_1 = N_2D_2$

If this is not true, the values for D_1 and D_2 are incorrect and:

Slide-Roll Ratio $\neq$ 0

Now adjust the disc diameters by small increments until the Slide/Roll Ratio falls to zero or as near zero as possible, thus corresponding with the zero friction condition.

If:

Slide-Roll Ratio = +ve

Increase D_2 and Decrease D_1 incrementally

If:

Slide-Roll Ratio = -ve

Decrease D_2 and Increase D_1 incrementally

A number of iterations may be required to achieve a condition where:

Friction = 0

Slide-Roll Ratio = 0

Automatic Control

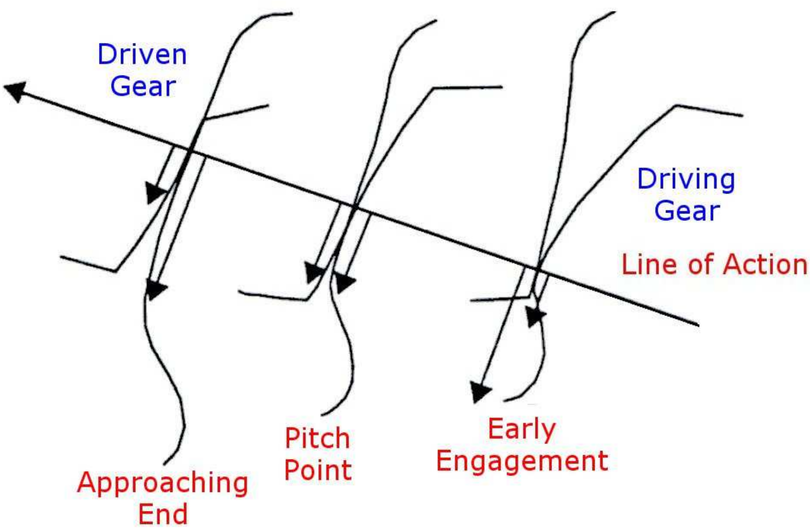
The above procedure can be run automatically under test sequence control using the installed programme ZeroSlideRoll.tsf. This will automatically search for the zero friction condition and then adjust the roller diameters incrementally to register the zero slide/roll ratio condition.

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.

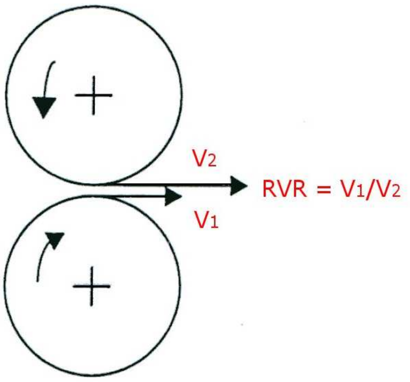
Note that in the case of the driving gear, sliding is always away from the pitch point. This imposes a tension in the surface layers and is the reason for the observed greater tendency of the driving gear to pit in the region of the pitch point.

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):

$$SVR = 0$$

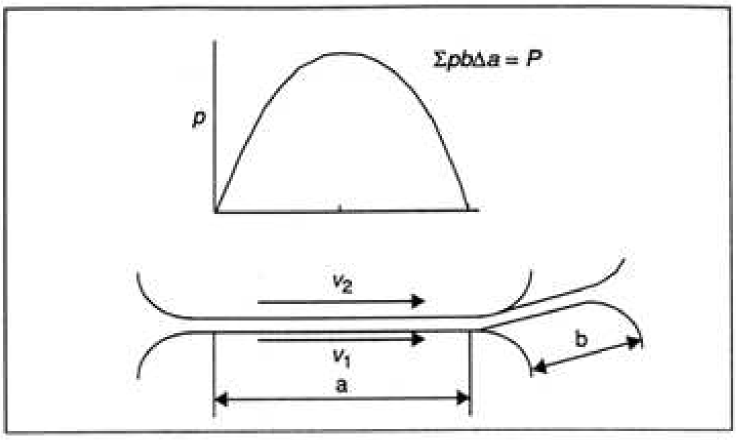
For pure sliding:

$$SVR = 1$$

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:		EP	=	$\mu P V_s t_t / A$	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: $\Delta V = k P V_s t_t / A$ mm^3

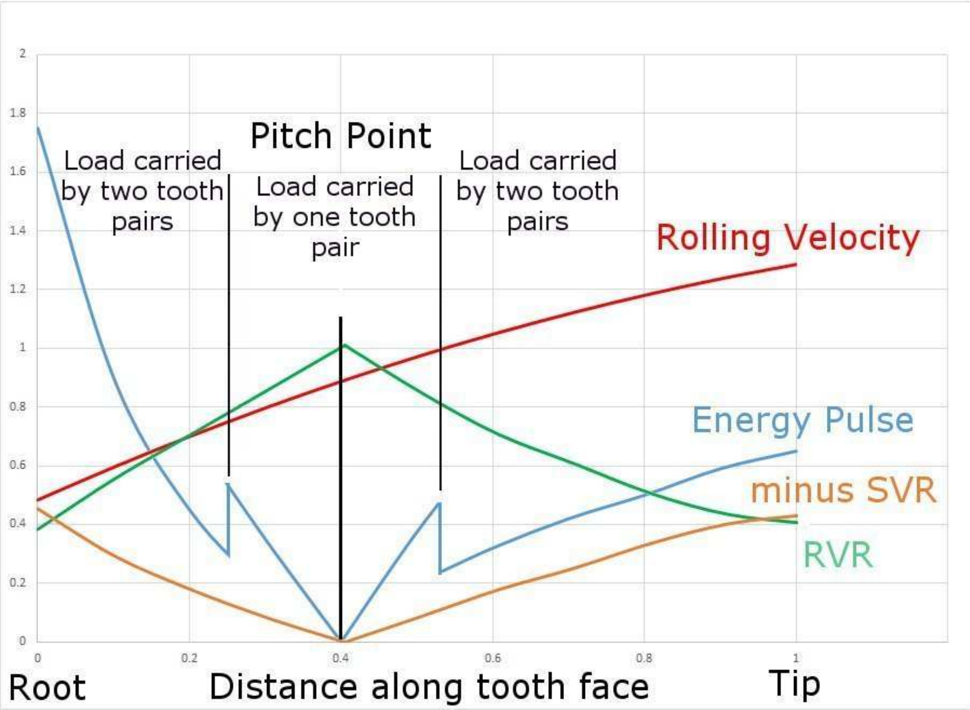
Each Energy Pulse can be regarded as an incremental contribution to wear or surface damage in the contact. The sum of the Energy Pulses can be used as a measure of the total wear.

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.

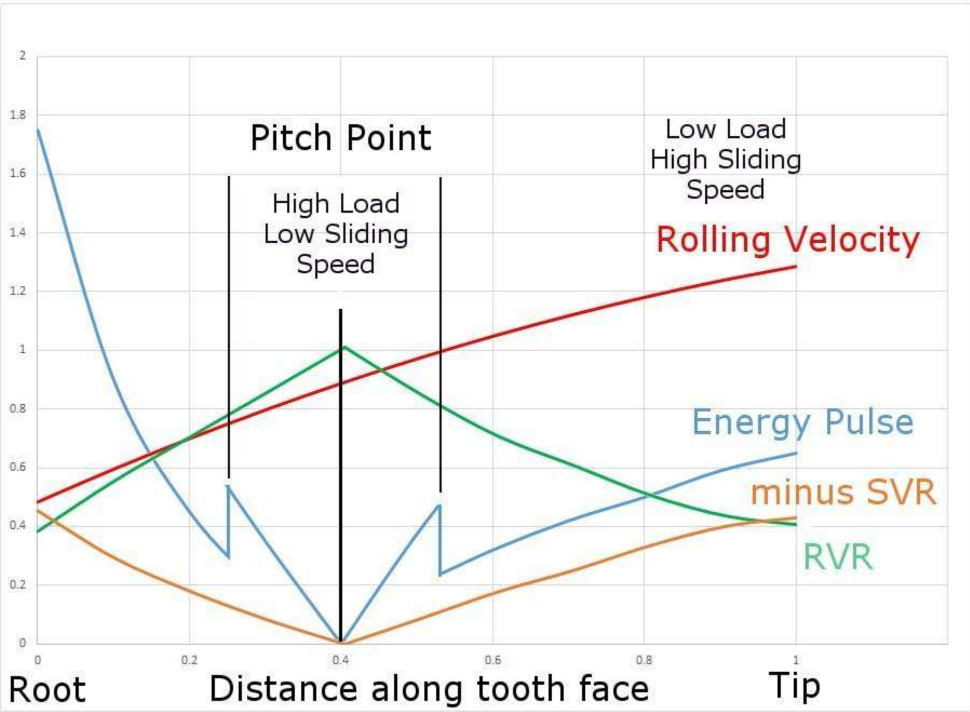
It is important to note that in many machine components there can be very high FPIs but, because the contact durations are short, the EP is low and hence the incremental damage is low.

Parameter Variation along Gear Tooth

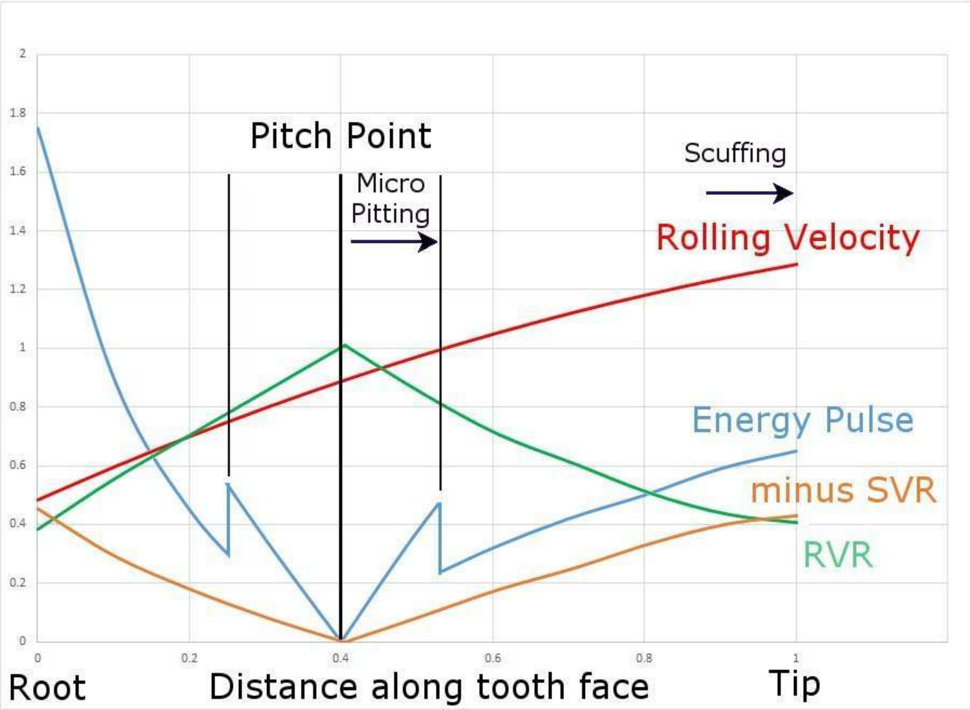
The following schematics are based on two 30 tooth gears with a 20° pressure angle. Note that SVR, which has a negative value, is here plotted as positive, so as to appear above the x-axis.



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 mechanism 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

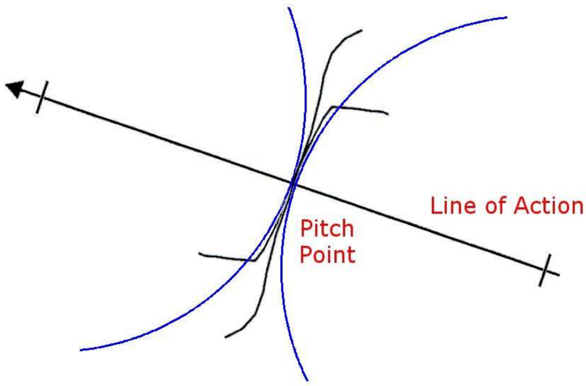
The FPI and EP clearly have no meaning at the pitch point and thus need to be calculated taking into account local contact pressure, hence local load and sliding velocity. This is, of course, by no means easy, hence a simple bench-mark estimate is to use the hertzian contact pressure at the pitch point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

Practical Choices

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.

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.

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.

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.