LUBRICATION AND SURFACE DISTRESS

OF

LOADED TOOTH FLANK OF GEARS

by

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1 . State of Lubrication of Loaded Tooth Flank and Surface Distress

1 .1 Contact* of Smooth Surfaces

The lubrication state between contacting bodies with rolling and sliding under loaded condition is generally understood by the conception shown in Figure 1.1. When the lubricating oil film formation between facing bodies is good enough to separate these bodies by the hydrodynamic pressure, this state is called by the expression of "hydrodynamic lubrication". The thickness of oil film is so large that the lubricating oil between facing bodies behaves as fluid and metal-to-metal contact between surface roughness asperities on facing bodies does not occur. When the oil film thickness becomes thinner or when the surface roughness height becomes larger, top of surface roughness asperities on facing bodies reaches very near to each other and there the oil or absorbed oil molecules on the surface of facing bodies behave no more as fluid. Partly metal-to-metal contact of surface roughness asperities occurs. Such lubrication state is called by the expression "mixed lubrication". When the oil film thickness becomes more thinner or surface roughness height becomes larger, metal-to-metal contact or contact via absorbed oil molecules dominate at most of the part in contact zone. Such state is called by the expression "boundary lubrication". Schematic representation of these three regimes of lubrication is shown in Figure 1.1.

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* The wording "contact" is employed here in the chapters concerning to gear lubrication to express the situation, that two bodies are pressed and face to each other under loaded condition. This state should be rather expressed by "conjunction", which include metal-to-metal contact, boundary-, mixed- and hydrodynamic lubricated state. But gear engineer never call e.g. "tooth conjunction" instead of "tooth contact" or "line of conjunction"

instead of "line of action" on tooth flank.

To investigate the lubrication state of gear teeth, contact between meshing tooth pair is substituted by the model of contacting cylinders which have the same radii of curvature as that of tooth form, see Figure 1.2. This modelling is not proper for the state at the beginning or ending of tooth meshing. The result of investigation by use of this model is therefore not valid especially for the tooth contact at the beginning of meshing. Lubricating oil film formation at the beginning of tooth meshing becomes generally poorer than expected and severe metal-to-metal contact occurs rather easily which can give damage at the root of teeth.

Tooth flanks of power transmission gears contact to each other with very large load and tooth flank locally deforms by contacting pressure. The lubrication state under such condition is called Elasto Hydrodynamic Lubrication (EHL or EHD) and many investigations have been worked out in this field /1/2/3/. Figure 3.3 shows one example of analyzed pressure distribution and shape of oil film thickness on the contact zone between two rollers under the supposition of the state of hydrodynamic lubrication and, that the contacting bodies have smooth surface /3/. In order to induce the hydrodynamic pressure, oil film shape or gap shape between contacting cylinders has the narrow part near the outlet of contact zone. The hydrodynamic pressure induced differs from the shape of Hertzian contact pressure at the inlet position and at near the outlet position. The minimum film thickness is approximately calculated by following equation /4/

$$H_{min} = 2.65\ G^{0.54}\ U^{0.7}\ W^{-0.13} \qquad (1.1)$$

where

$$H_{min} = h_{min}/R \qquad (1.2)$$

$$G = \alpha E' \quad (1.3)$$

$$U = \eta_0 u/(E'R) \quad (1.4)$$

$$W = w/(E'R) \quad (1.5)$$

R is the effective radius of the roller pair (mm)

E' the effective elastic modulus of the materials of rollers (kgf/mm^2)

η_0 viscousity of oil in atomospheric pressure ($kg\ s/mm^2$)

α pressure-viscousity index of lubricant (mm^2/kgf)

w applied load per unit width (kgf/mm)

u rolling velocity (mm/s)

h_{min} minimum oil film thickness

When relative sliding exists between contacting bodies under EHL contact, heat is generated in the oil film in contact zone. Figure 1.4 shows schematically the EHL condition with consideration of thermal effect on it. Suppose the lubrication state is hydrodynamic and, that the contacting bodies have smooth surface and the surface velocity of contacting bodies be denoted by V_1 and V_2 respectively. The hydrodynamic pressure between the bodies makes the contacting bodies deformed elastically and make the gap between the bodies. The oil flow in this gap is laminar and all mass of oil come in the gap is expelled from the out-let of the gap. When V_1 is not equal to V_2, frictional heat is generated in the oil film due to shearing of oil. This heat is transfered with oil mass in the flow direction, but some amount of heat is conducted across the oil film and transfered to the contacting surface of the bodies. This heat is conducted into the bodies. Figure 1.5 and figure 1.6 show one example of pressure distribution and shape of oil film over the contacting zone (Hertzian contact width) between two cylinders in steady state respectively. The abscissa takes the nondimensional contact width: -1.0 to 1.0 corresponds to the Hertzian contact width. On the ordinate nondimensional pressure and film thickness are taken, which are

$P = p(4\,\phi^{0.5})/E'$

$H = h\,\phi\,/R$

$X = x/b_H$

$b_H = (8\,w\,R\,/(\pi E'))^{0.5}$

$\phi = (R/b)^2 = 0.125\,\pi\,/W$

p pressure distribution

h oil film thickness

x position in contact width direction

b_H a half of Hertzian contact width

The shape of film thickness takes the minimum point near the out-let of contact zone and that shape changes slightly when the slip ratio $S=(V_1-V_2)/V_1$ changes under constant rolling velocity $u=(V_1+V_2)/2$. The shape of pressure distribution changes at the height of its spike near the outlet. Figure 1.7 shows one example of temperature distribution in longitudinal direction (along the oil film) when the inlet oil temperature is kept constant 50 deg.C. The oil temperature on the ordinate is the average temperature of oil film in thickness direction. The temperature rises strongly with rise of slip velocity between contacting bodies. The maximum temperature appears at the place near the pressure spike. It is remarkable, that the oil temperature after outlet of Hertzian contact width becomes very lower than the maximum temperature in EHL contact zone. It means, that most of the heat generated in the oil film transfers into the contacting bodies and raise the surface temperature of the body. Figure 1.8 shows the temperature distribution in oil film. In the direction of oil film thickness the maximum oil temperature appears at the middle of thickness. The surface temperature of contacting body rises gradually from the inlet position and takes the maximum value near the front of the pressure spike. Because the surface of contacting bodies move, this temperature on a definite point on the surface is instantaneous. This instantaneous surface temperature under Hertzian contact is often called by the expression "flash temperature".

The concept of flash temperature corresponds sometimes to the increment of instantaneous temperature over the inlet temperature, but sometimes to the sum of surface temperature of contacting body at inlet and the increment of instantaneous temperature. Here the former definition is incorporated for "flash temperature".

Historically the analysis of flash temperature was worked out with considering only the one dimensional heat conduction into the contacting body: The existence of lubricating oil film, heat conduction in oil film and heat transfer from oil film to the contacting body have not been considered. Under the supposition that the shape of heat source moving of the contacting body surface be after Hertzian pressure distribution and the instantaneous temperature at every corresponding point on contacting high speed body surface and low speed body surface be equal, Blok /5/ showed following equation for the maximum value of flash temperature:

$$T_f = \frac{1.11\, \mu\, w\, [V_1 - V_2]}{(\sqrt{V_1} + \sqrt{V_2}) \cdot \beta \cdot 2a} \qquad (1.6)$$

where $\beta = \sqrt{\lambda \rho_s c}$

for average kind of steel takes a value between 0.9 and 2.3 (kgf/(mm s deg))

λ = thermal conductivity of surface material

ρ_s = density of gear material

C = specific heat of gear material.

When the existence of lubricating oil film, heat conduction in oil film and heat transfer from oil film to the contacting body is considered, the instantaneous surface temperature in Hertzian contact zone becomes somewhat different from the Blok's flash temperature. Figure 1.9 shows one example of comparison between these instantaneous temperatures. The absolute value of these three kinds of instantaneous temperatures do not coincide well. The calculation of flash temperature under EHL condition is possible, but the oil viscousity is not well known under the condition of very high pressure, existence of shearing action of oil film and on-off of very high pressure in a very short time. The property of such viscousity has although very large influence on the calculated value of flash temperature in EHL condition. These facts mean, when the Blok's flash temperature or any other kinds of flash temperature is used for estimation of scuffing resistance of gears, the calculated value of flash temperature should not be understood as the actual physical quantity, but as one reference value or guiding value for the instantaneous surface temperature of EHL contacting bodies. The reliability of absolute value of the calculated flash temperature is of some question.

1.2 Effect of Surface Roughness

The surface of actual contacting body is not smooth and it has surface roughness. When the wave length of surface roughness, i.e. pitch between surface roughness asperities is enough shorter than the Hertzian contact width, the minimum oil film thickness calculated for the model of smooth surface is almost valid, where the oil film thickness is defined as the distance between average lines of surface roughness height of both contacting bodies. In practical case, wave length of roughness on contacting surface is sometimes comparative or larger than the Hertzian contact width. For such case, no simple method has not been known for the estimation of minimum oil film thickness. The equation (1.1) is although also used for such case to obtain a reference value for the minimum oil film thickness. It must be mentioned, that the oil film thickness calculated e.g. by equation (1.1) cannot show the actual value of the minimum oil film thickness for such case. The degree of reliability of the calculated value for the minimum oil film thickness between rough surfaces is different therefore as function of the ratio between wave length of surface roughness and Hertzian contact width /6/, cf. Sec.2.1.

The effect of surface roughness on the flash temperature is not investigated and not well known until now. But a rough explanation is as follows. When oil film thickness is enough larger than the surface roughness height and no metal-to-metal contact occurs, the situation concerning to flash temperature is thought to be almost equal to that in the case of smooth surface. For the contact of rough surfaces and/or for the contact via small oil film thickness at which asperities of surface roughness can make metal-to-metal contact, very local frictional coefficient at that point must be quite different from the value at the place of oil film existence. This partial difference of local coefficient of friction changes the magnitude of heat source in contacting zone due to sliding. The absolute value of flash temperature must have the influence of it. The degree of reliability of the calculated flash temperature using e.g. equation (1.6) is therefore different for the case of hydrodynamic lubricated condition, for mixed lubricated condition and for boundary lubricated condition, although it is understood as a reference or nominal value for the instantaneous temperature on the surface under contact.

1.3 Surface Distress

The driving condition of todays power transmission gearing becomes severer than those of old time: running speed becomes higher and specific tooth load becomes heavier and easy maintenance is although required by users, for instance frequent change of gear oil is not welcome. These conditions induce often failure of gears due to lubrication problem.

1.3.1 Wear

Figure 1.10 shows schematically the state of the cross section of contacting tooth flanks and typical failure which can occur there. When surface roughness asperities of contacting surfaces make directly or via contaminant metal-to-metal contact, that part wears due to shearing force acting at surface roughness asperities in the forms of so called "Adhesive Wear" or "Delamination" or "Abrasive Wear". In practice, so called wear becomes of problem, when almost no EHL oil film exist over whole part of contact width. Wear problem becomes of course more serious for soft material.

1.3.2 Micropitting and Macropitting

Under mixed lubricated condition in which partial metal-to-metal contact between surface roughness asperities occurs, shearing stress inside the surface roughness asperities due to asperity contact induces the fatigue of material there and micropitting appears usually on hard surface such as on case carburized gear teeth, because the hard surface will not "wear" in the sense mentioned above. On soft surface the micropitting can occur, when the contacting surface does not wear in mixed lubricated condition. The observation of micropitting looks like frosted area and it is sometimes called as gray staining, cf.Fig.1.11. Clear form of pit is usually not found. The depth of micropitting is about 20 μm and many micro cracks are found in its area. Micropitting can increase vibration and dynamic loading of gears and can develop to severe pitting or to scoring/scuffing or to tooth breakage. In some cases, the micropitting can appear first after 10^{10} loading cycle. When the state of contact between surface asperities becomes more severe, high subsurface shear stress in roughness asperities induces macro pitting. Macro pitting of this kind breaks out therefore on projected part of surface roughness, such as at ridge of ground marks on tooth surface. Figures 1.12 and 1.13 show such example of macro pitting on Maag 15° ground and Maag 0° ground gear tooth respectively. Macro pitting of this kind is usually small and can break out both on soft and hard gear tooth. When loading condition is not very severe, macro pitting can stay in the state of so called initial pitting and does not develop to destructive pitting.

Under high Hertzian pressure, subsurface shear stress in the contacting body, not in surface roughness asperity, becomes higher. Under the existence of frictional force on contacting surface, the position of the maximum shear stress approaches to the surface. At very near the surface, there can be stress concentration due to notch effect of surface roughness and of macropitting, or there additional stress can be induced due to asperity-to-asperity contact. Consequently the total stress level at or very near the contacting surface exceeds the endurance limit of material and initial crack is induced on the surface and the crack propagates into subsurface and makes a pit of considerable size. The depth of the pit has a relation with a material property and the position of the maximum shear stress induced /6/: For the material with large fracture toughness, the crack propagate upto the maximum shear stress position and secondary crack which initiate from there to the surface make pits. For the material with small fracture toughness, the crack propagate deeper than the position of the maximum shear stress due to the stress concentration at the tip of the crack and deep and large pit breaks out. The size of pit becomes in general larger, when the loading condition becomes severer or the module of gears becomes larger.

1.3.3 Spalling and Case Crushing

Figure 1.14 illustrates the difference of the state of crack between pitting, spalling and case crushing. On the initiation of pitting, it is believed, that the friction on the contacting surface has a large relation as mentioned before.

When the loading becomes very large, the ratio of the increment of contact stress due to surface roughness to the maximum Hertzian pressure for the contact of smooth bodies becomes relatively small, because the stress increment due to asperity contact is approximately independent from the loading value, while the Hertzian stress for contact of e.g. smooth cylinders increases proportionally to the square root of loading value. The subsurface stress state for such very heavily loaded case can be approximately discussed therefore with the model of contact of smooth bodies. The black curve in the middle right figure in Fig.3.14 shows the state of shearing stress in subsurface of contacting body. The shear stress takes the maximal value at a certain depth from the surface. At the surface there is another maximal of stress due to the friction on the surface. If the durability of material is stronger than these value of induced shear stress distribution, no fatigue failure breaks out. According to the condition of applied load and sliding on the contacting surface, the distribution of subsurface shear stress changes. When the material is soft, plastic deformation occurs at high shear stress zone, and surface failures due to plastic deformation such as rippling, ridging and so on occur. But if the material is not soft enough to break out plastic deformation and the durability of case hardened material takes, for instance, the pattern shown with the broken curve in the same figure, the fatigue crack will initiate at the position at which the material durability becomes smaller than the induced shear stress level (at the area indicated by small circle in the figure). The crack propagates in this region of stress-durability relationship, mostly parallel to the surface. Some secondary cracks propagate from there to the surface and makes very large pit of undefinable shape. Such surface failure, the initiation of which crack lies in subsurface, is defined commonly by the expression of "Spalling". When the hardness of material is constant, the durability of material is almost constant in the depth direction from the surface and crack can initiate at two different depthes corresponding to the positions of the maximum shear stress τ_{xy} and of τ_{45°. Case crushing is a special case of spalling: when the hardness distribution in depth direction shows very sharp change as seen ,for instance, in the right low figure of Fig.1.14, the condition of spalling crack initiation is fulfilled at this position and only the hardened case is peeled off. In the case of pitting, each pit appears independently and the number of pits increases, but the area of already appeared pit will hardly grow. On contrary in the case of case crushing, case crushed area spreads continuously, as if the case of the surface is continuously peeled off.

As it is clear from the explanation about spalling and case crushing, lubrication of contacting surface has little influence on the durability of surface failure of this kind.

1.3.4 Scuffing

Scoring or scuffing failure is tooth flank failure of gears in relation with gear lubrication, which occurs at the beginning of operation or suddenly during the operation of gear drives, often without any signs of break out, and stops the further operation of the unit, because of the increase of vibration and noise and also because of the hazard to induce the total break down of the facilities which include the gear drive in question.

Scoring or scuffing failure is a thermal damage of tooth flank: local welding of contacting surface breaks out suddenly and develops itself explosively and the contacting surfaces are damaged in a very short time. The specialist in the field of Tribology distinguishes the definition of scoring and scuffing failure, but gear engineer commonly mixed up both the expression. German gear researchers use sometimes the expression of "hot scuffing/scoring" and "cold scuffing/scoring", which correspond almost to "scuffing" and "scoring" in the sense of tribologist respectively. Here the wording "Scuffing" is used including all the sense of this kind of failure for gear engineer.

The physical state of scuffing can be roughly defined as follows:

(1) Two lubricated surfaces slide to each other with/without rolling under loading.

(2) Atoms constituting both surfaces approach near enough to bond each

other.

(3) Enough energy for bonding atoms of both surface materials is delivered by the shearing action between the surfaces.

(4) The bonding and transfering action of atoms between both surfaces and abrading away of surface material make the state to produce more energy for this reaction. And this state of positive feed back accelerate itself explosively.

Scuffing can initiate even the lubrication state is complete hydrodynamic or in mixed or boundary lubricated condition. The scuffing failure which initiates under the condition of very high contact pressure and high sliding velocity is so called "hot scuffing". The damaged surface looks not so rough and have an appearance of melted surface, cf. Figure 1.15. The cause of it is believed pure thermal: shear energy dissipated in the contacting area raises the temperature of oil film and of absorbed layer on contacting surface and vapourize them off to realize metal-to-metal contact and local welding of the surfaces. The surface is then abrading away. The real procedure of the initiation and development of hot scuffing is not yet clear, but shear energy rate on contacting surfaces, maximum value or average value of flash temperature, and bulk temperature of the contacting surface are considered to have big correlation with the condition of initiation of hot scuffing /7/, cf. sections 4 and 5. Thermal stability of EHL oil film /8/ or of surface temperature /9/ is also considered to have some relation on the initiation of hot scuffing.

Under very high loading but very slow speed condition, a kind of scuffing failure can occur. This kind of damage is often called as "cold scuffing". Cold scuffing initiate under boundary lubricated condition or under the condition of very poor existence of lubricant between contacting surfaces under which surface roughness asperities contact each other metal-to-metal via absorbed surface layer. When relative sliding is given under such codition, absorbed layer is teared off by shearing force or vapoured off due to generation of very local frictional heat and metal-to-metal contact is realized. The metal-to-metal contacting part is teared off and abraded away with relative sliding of contacting surfaces, and there more frictional heat is generated to enlarge the state of local welding of surface roughness asperities. At the phenomena of cold scuffing, the hot area is limited to very local, only to the contacting surface roughness asperities. Other part in the region of contact zone stays cold. This is the biggest difference of the cold scuffing from the hot scuffing at which not only surface roughness asperities but also the surface bulk under the region of whole contact width becomes considerably hot. An example of damaged surface due to cold scuffing is shown in Figure 3.16 : Deep scratch is observed and the material near the surface usually shows plastic flow. Cold scuffing is often found together with pitting, spalling and plastic deformation of tooth flank.

The phenomenon of scuffing is very complicated and there are too many factors which give influences on the condition of its break out. Many results of research have been published, but almost all of them deal with hot scuffing and cold scuffing is not well investigated. In spite of huge amount of research on hot scuffing, no reliable method to predict the hazard of hot scuffing failure of gears is known until now.

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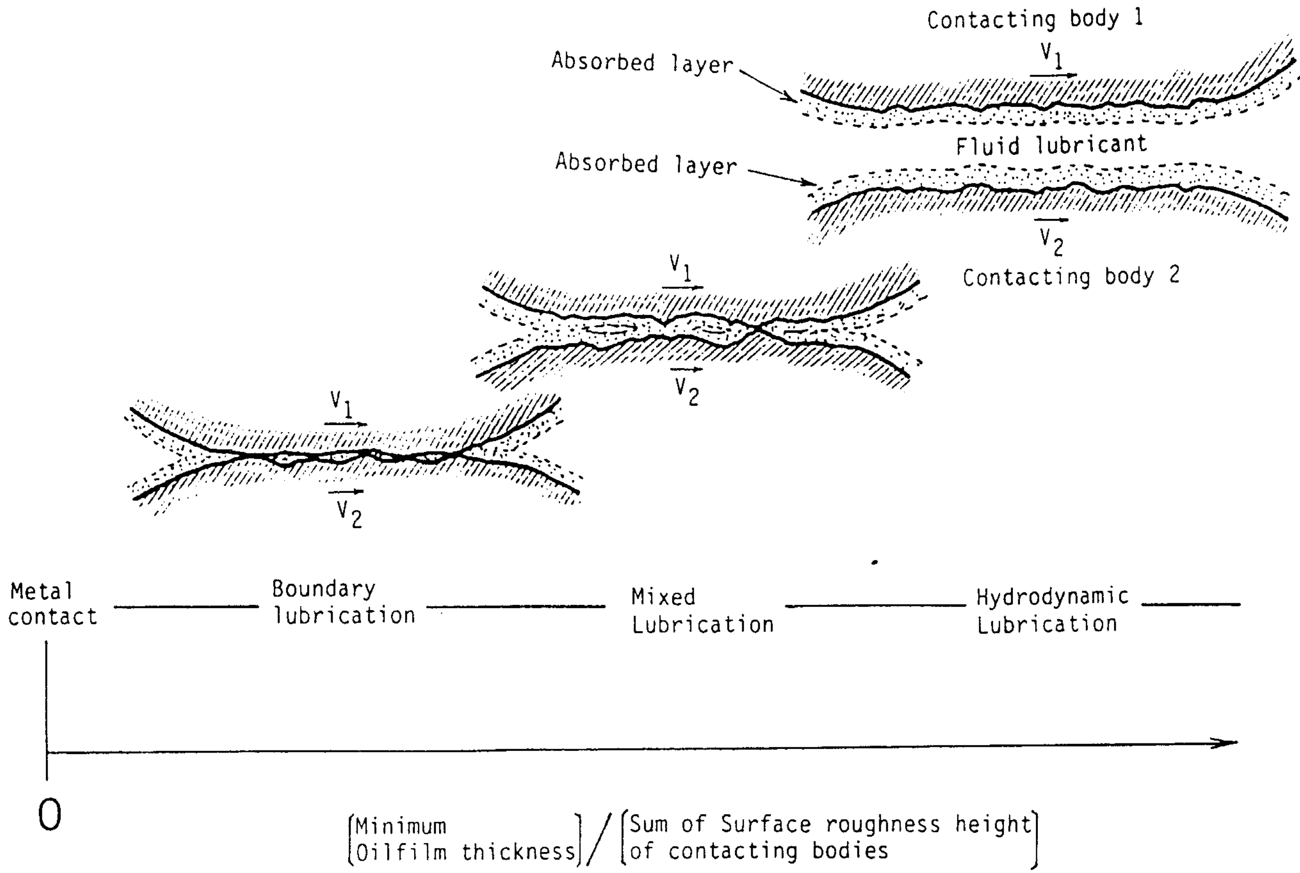


Fig. 1.1 Concepts of the regime of lubrication

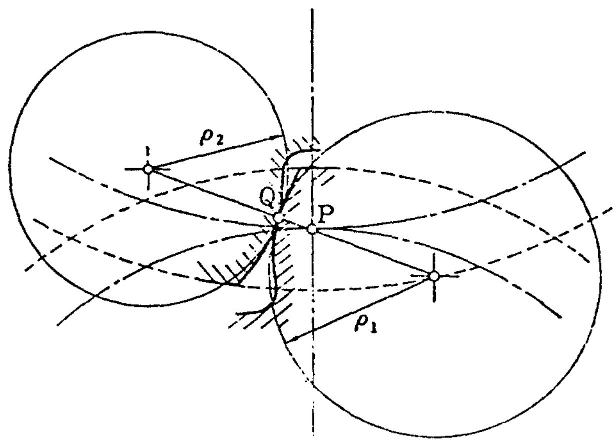
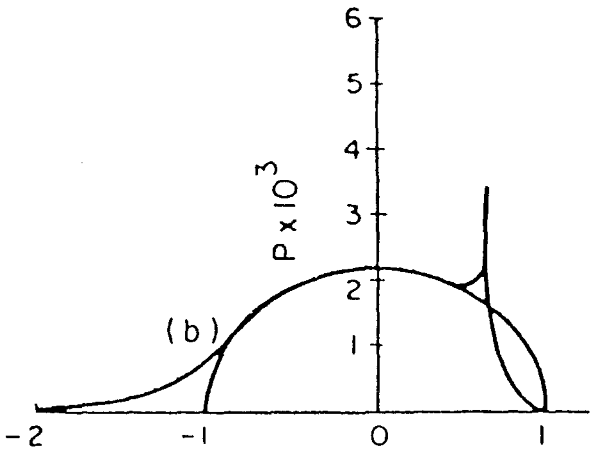
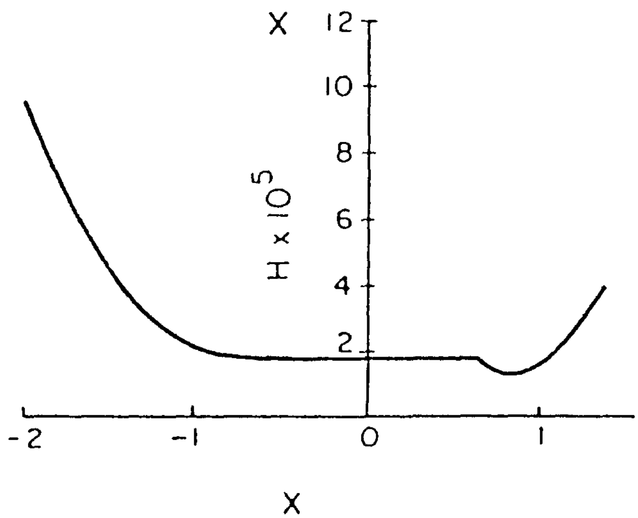


Fig. 1.2 Substitution of contacting tooth flanks with two rollers



(a) Pressure distribution



(b) Film shape

Fig.1.3 Example of calculated EHL oil film shape and pressure distribution (iso thermal model)

$W = 3 \times 10^{-5}$.

$U = 10^{-11}$. $G = 2500$.

[Reproduced from the *Journal of Mechanical Engineering Science*, Vol. 2, No. 3, pp. 188–194 (1960)

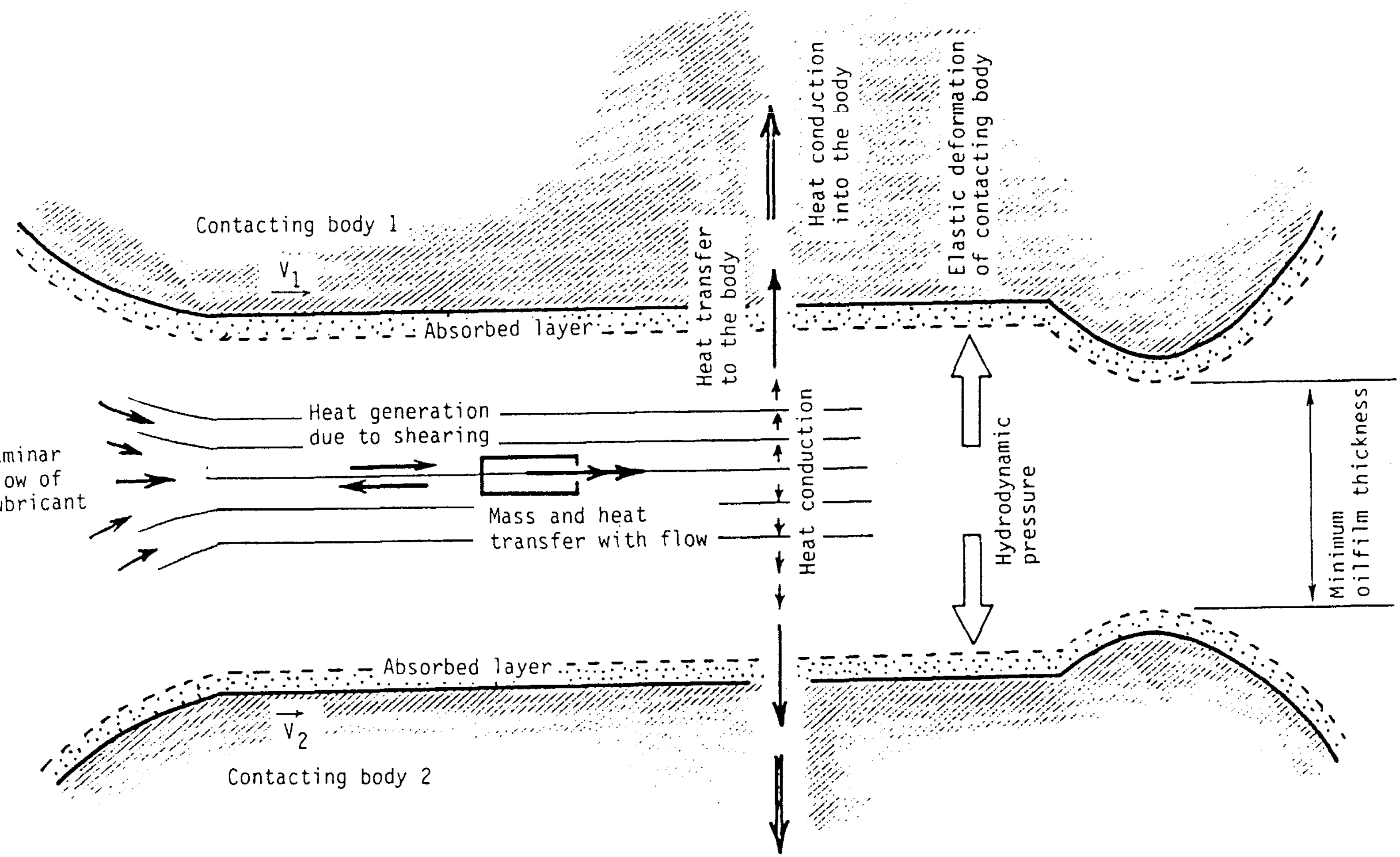


Fig. 1.4 Model of Elasto-Hydrodynamic Lubrication with considering the thermal effects (TEHL model)

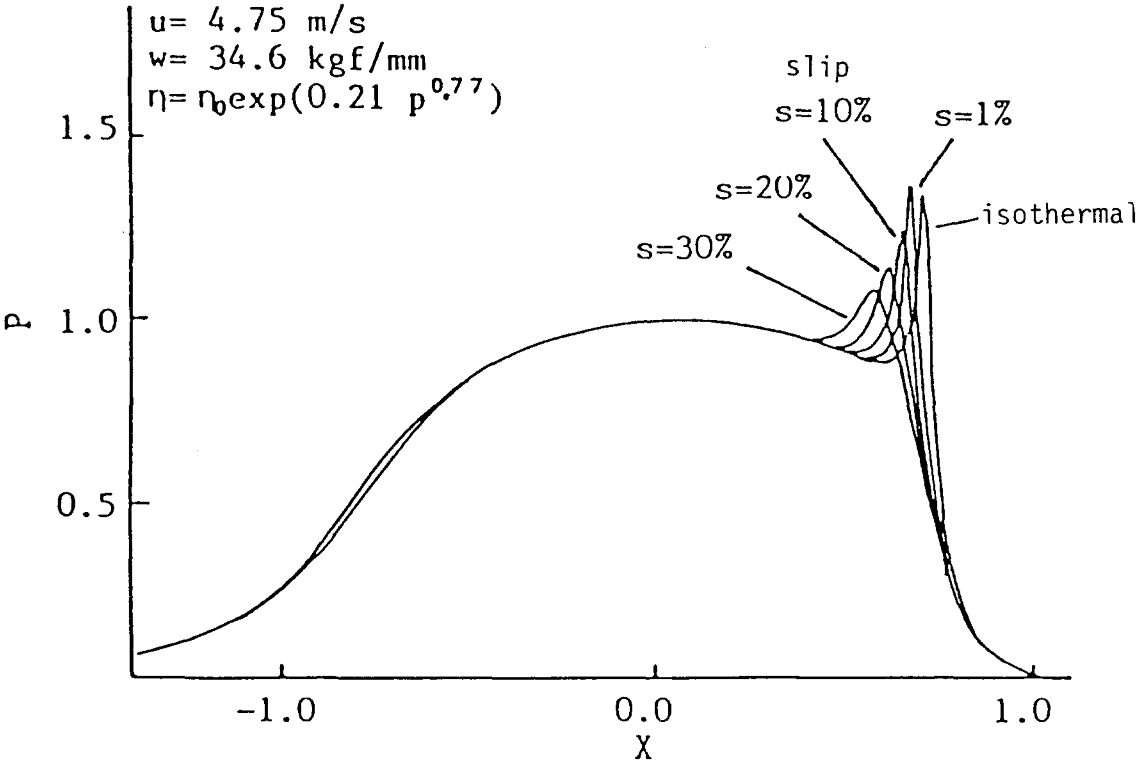


Fig. 1.5 Pressure distribution of TEHL model for showing the effect of slip ratio between contacting bodies

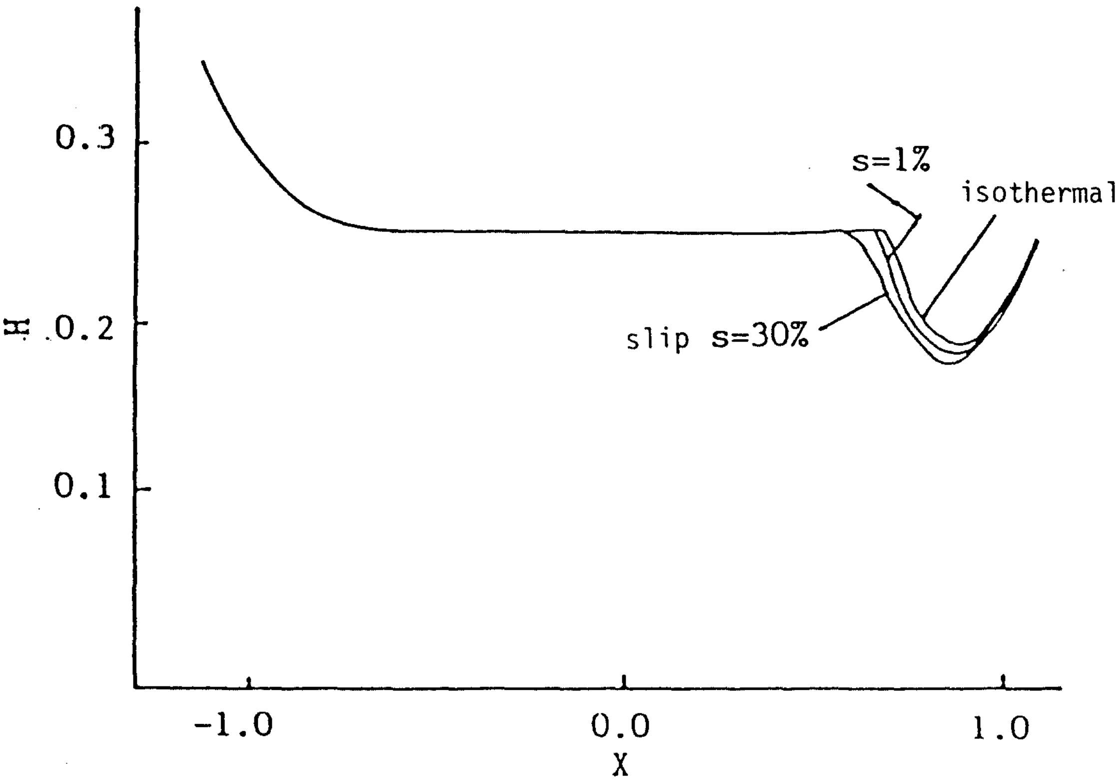


Fig. 1.6 Oil film shapes with TEHL model and the effect of slip ratio

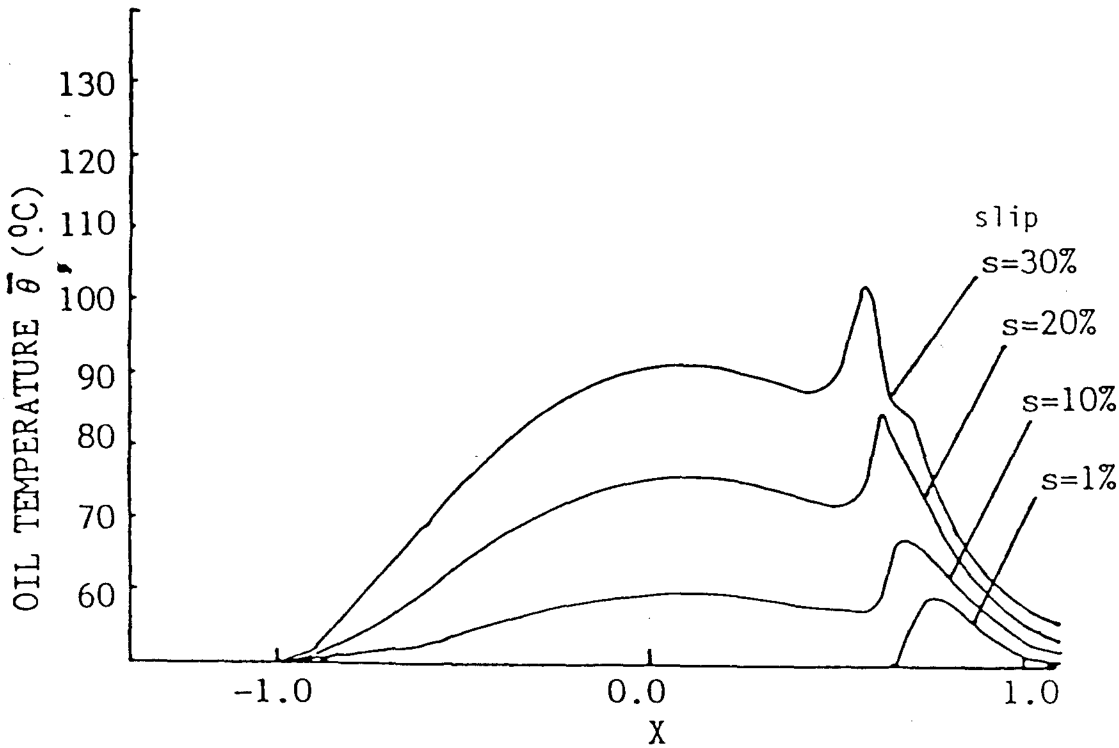


Fig. 1.7 Average temperature distribution in oil film under TEHL contact and the effect of slip ratio

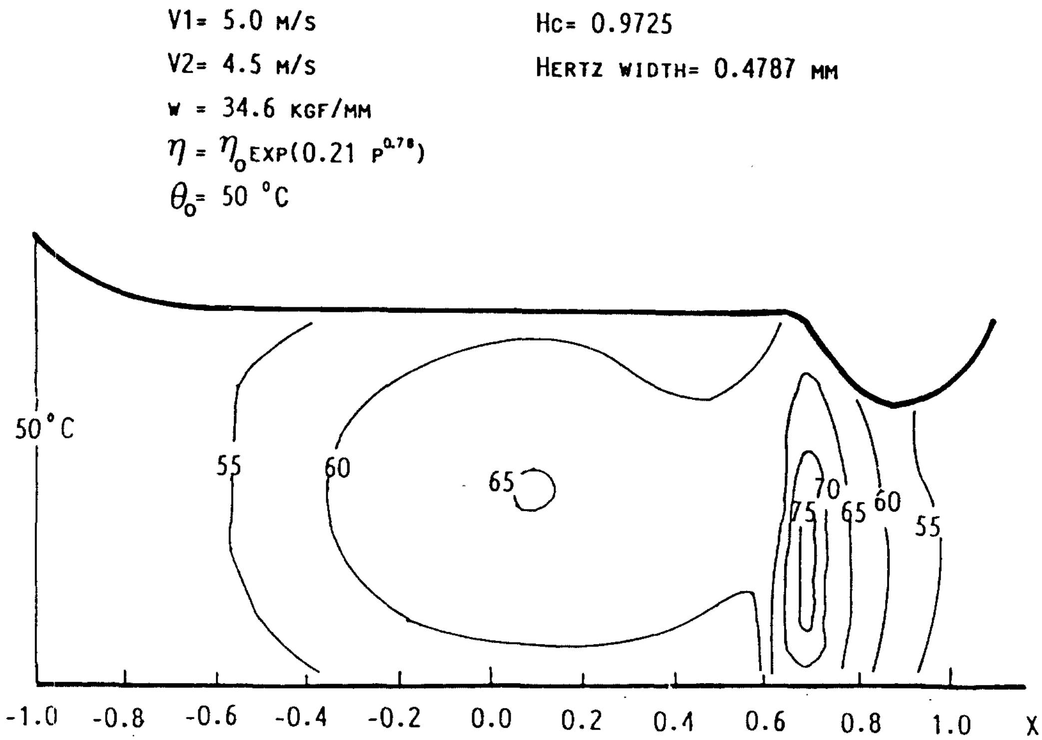


Fig. 1.8 Temperature distribution in oil film under TEHL contact

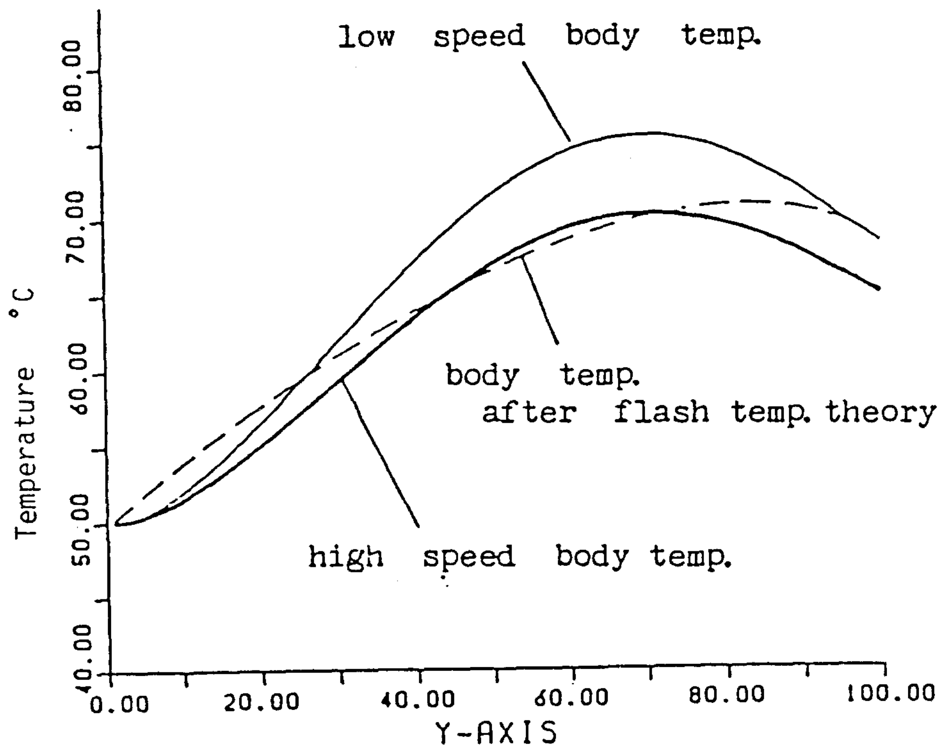


Fig. 1.9 Comparison between the classical flash temperature after Blok with that of the model considering heat generation and heat conduction in side the oil film and heat transfer to the contacting bodies

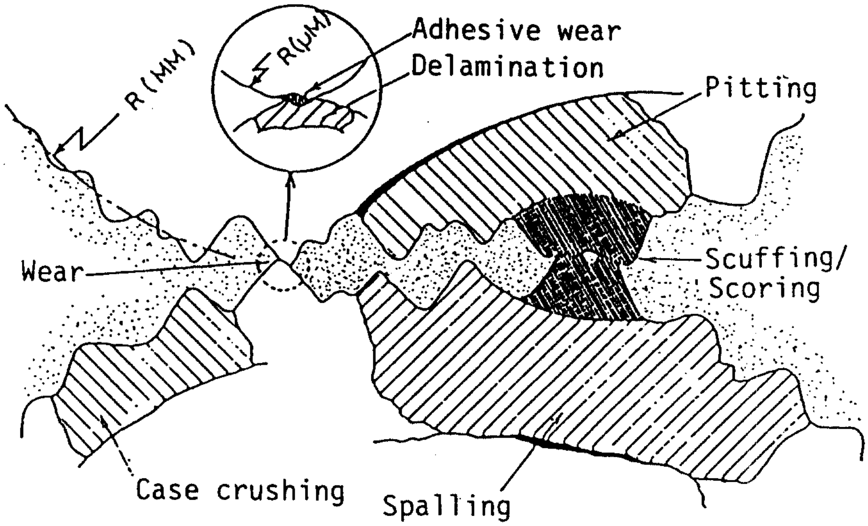


Fig. 1.10 State of distress of lubricated surfaces

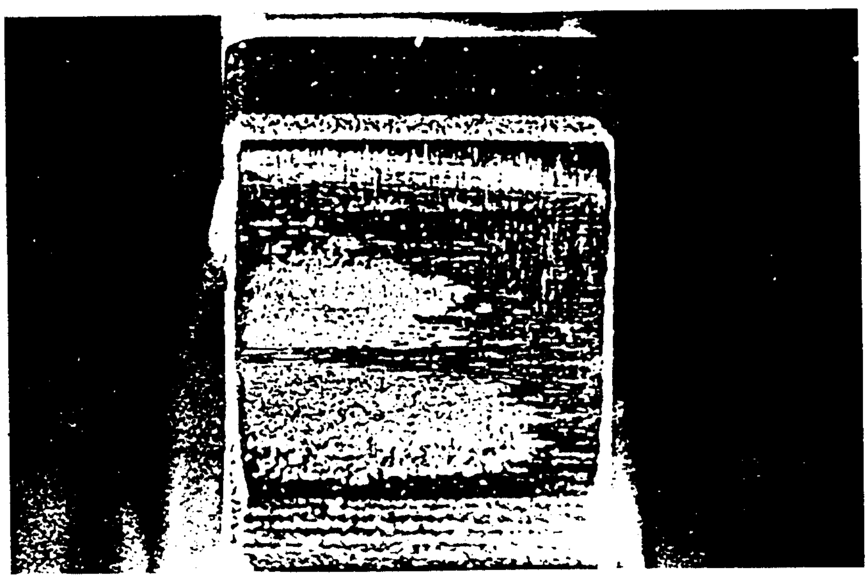


Fig. 1.11 Micro pitting of carburized tooth flank

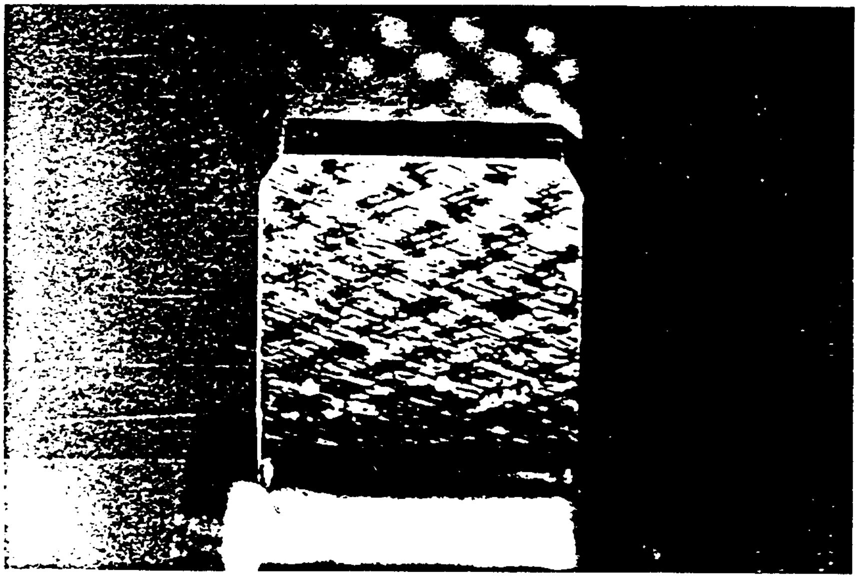


Fig. 1.12 Macro pitting on crisscross grinding mark

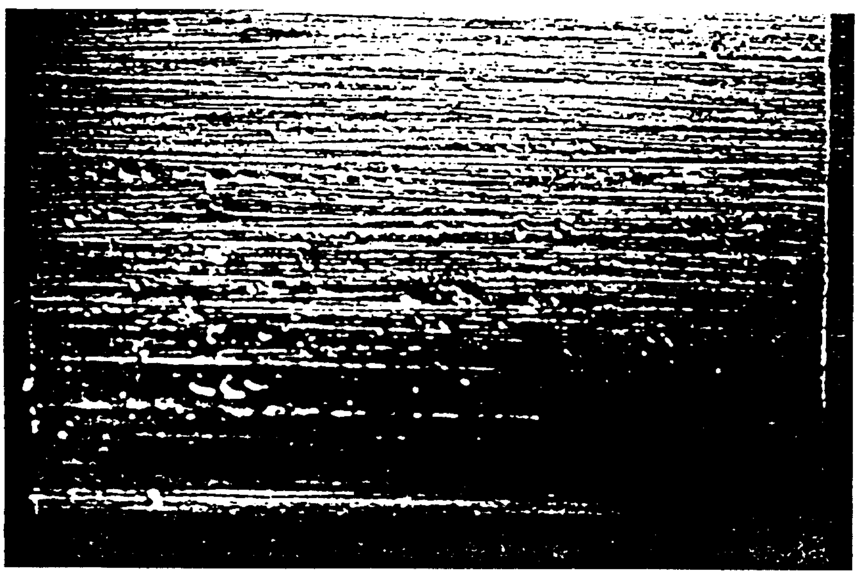


Fig. 1.13 Macro pitting on the ridge of Maag 0 deg.grinding mark

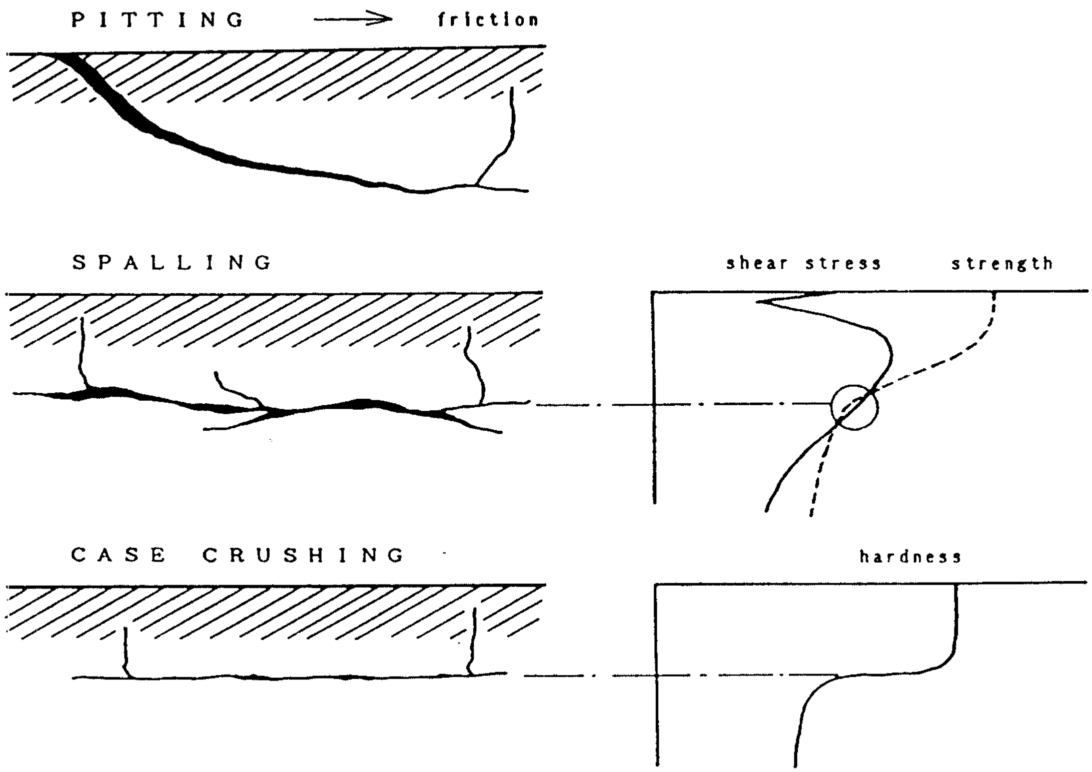


Fig. 1.14 Schematic explanation between pitting, spalling and case crushing

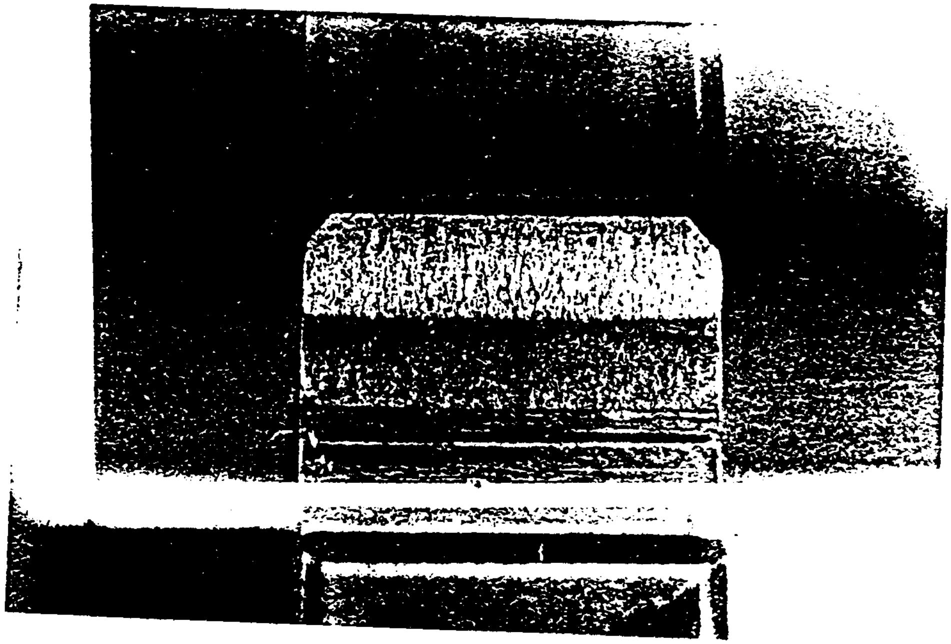


Fig. 1.15 Typical look of hot scuffing

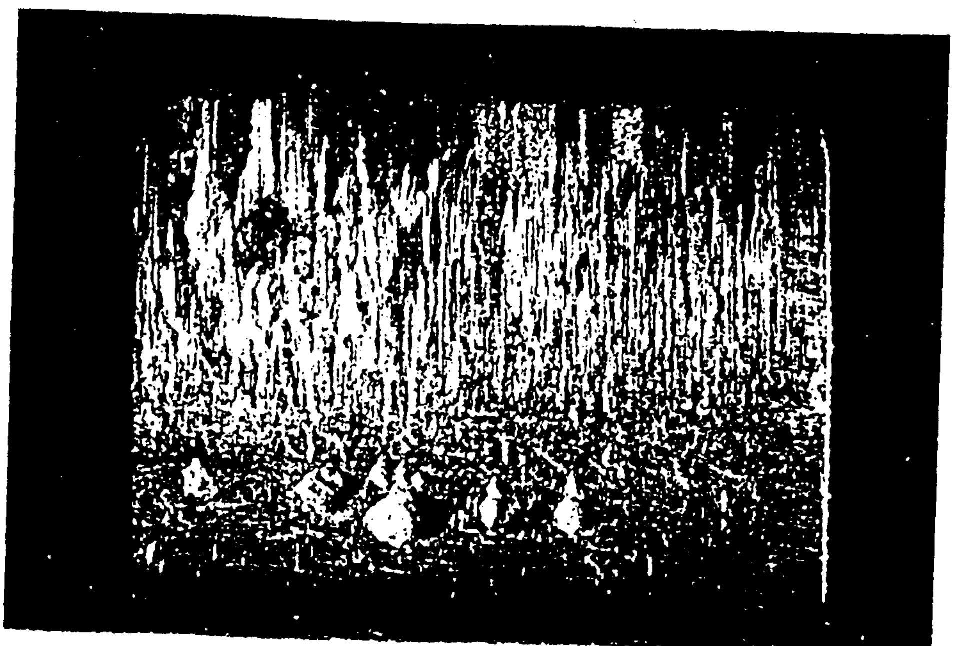


Fig. 1.16 Typical look of cold scuffing