ZESZYTY NAUKOWE POLITECHNIKI ŚLĄSKIEJ 1994

Seria: TRANSPORT z. 22 Nr kol. 1210

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APPLICATION OF ELASTOHYDRODYNAMIC THEORIES OF LUBRICATION TO RAIL/WHEEL SYSTEMS WITH CURVED TRACKS

Summary. The paper reviews the problem of rial/wheel flange wear and its control by a variety of means including lubrication of the contact. The results of an analysis of the lubricant film thicknesses to be expected in such contacts are presented. These indicate theat the degree of profile mismatch between the contacting rail and wheel is an important factor in determining the effectiveness of the lubricating film.

ZASTOSOWANIE ELASTOHYDRODYNAMICZNEJ TEORII SMAROWANIA SYSTEMU KOŁO-SZYNA W ŁUKACH O MAŁYCH PROMIENIACH

Streszczenie. W pracy rozpatrywano teoretycznie i eksperymentalnie zagadnienia związane ze smarowaniem obrzeży kół kolejowych w łukach przy dużych obciążeniach zewnętrznych.

Przedstawiono rezultat analizy numerycznej grubości filtru olejowego i rozkładu ciśnienia podczas smarowania obrzeży kół kolejowych w łukach przy dużych obciążeniach zewnętrznych. Przeprowadzono analizę grubości filmu olejowego w zależności od parametrów geometrycznych koła wykorzystując elastohydrodynamiczną teorię smarowania.

ПРИМЕНИЕНИЕ УПРУГИГИДРОДИНАМИЧЕСКОЙ ТЕОРИ СМАЗКИ СИСТЕМЫ КОЛЕСО-РЕЛЬС В КРИВЫХ

Резюме. В работе рассматривается теоретически и экспериментально проблемы смазки гребней железнодорожных колес при больших нагрузках. Представлены результаты численного анализа толщины смазки и разпределения давления во время смазки гребней железнодорожных колёс. Проведен анализ толщины смазки в зависимости от геометрических параметров колеса, используя упругогидродинамическую теорию смазки.

1. INTRODUCTION

The problem of wheel flange/rail contact wear is discussed together with methods for its control. Lubrication of the contact is considered and the resulting elastohydrodynamic point contact is analysed for a range of transverse wheel radius profiles using a typical lubricant with a representative load and speed. The minimum film thicknnes occurs at the sides of the contact and is found to be sensitive to the effective transverse radius of the wheel/rail contact varying between 2 μm and 0.1 μm for the range of radii considered with assumed representative operating conditions of load, speed and lubricant viscosity.

2. RAIL AND WHEEL WEAR IN CURVES

2.1. Performance characteristics of the rail/wheel tribosystem

In recent years a number of new railway systems have been introduced, some of which have encountered high rates of wheel and rail wear. The continuing trend of increasing loads and speeds, particularly for goods trains, causes increasingly higher wheel-rail stresses and correspondingly greater wheel and rail maintenance problems. The wheel/rail tribosystem consists of the wheelset, the rails and the lubricant which may be present. The most important parameters influencing the performance of the tribosystem are axle load, vehicle speed, track design and traffic characteristics. The combination of these parameters in curves determines the nature of the external load applied to the rial, which will thus be different for each individual railway.

Axle load determines the vertical load acting on the wheel/rail contact areas and current typical values fall in the range 180 to 220 kN/axle [1]. The corresponding stress levels will vary with contact geometry and may be taken to be in the range 500 to 2000 MPa [2]. The radial, centripetal force varies between 10% and 40% of the vertical load. On sharp curves the wear process typically takes place on the rail/wheel flange. The coefficient of friction between wheels and rails can vary between 0.1 and 0.5, and has a great influence on the wear process. Much information concerning axle load, vehicle speed, traffic track design and the relationship with wear is available in the literature, for example references [3] and [4].

A number of factors make studies of wheel and rail wear difficult [5] namely: the lack of an applicable quantitative wear theory; several wear mechanisms contribute to the wear; plastic flow is a factor; the problem is not easy to scale down to a laboratory test level; it is difficult to cally out metallurgical and metallographic samplin of in service rails; load and creep histories of specimens are not known.

The areas of friction, contact stresses in rolling contact adhesion, creep and vehicle dynamics are covered in considerable detail in references [5], [6], [7], [8].

2.2. Wheel and rail wear in curves

Rail wear on curved track is considerably higher than that for straight tangent track and is illustrated in Figure 1 where typical wear patterns are shown.

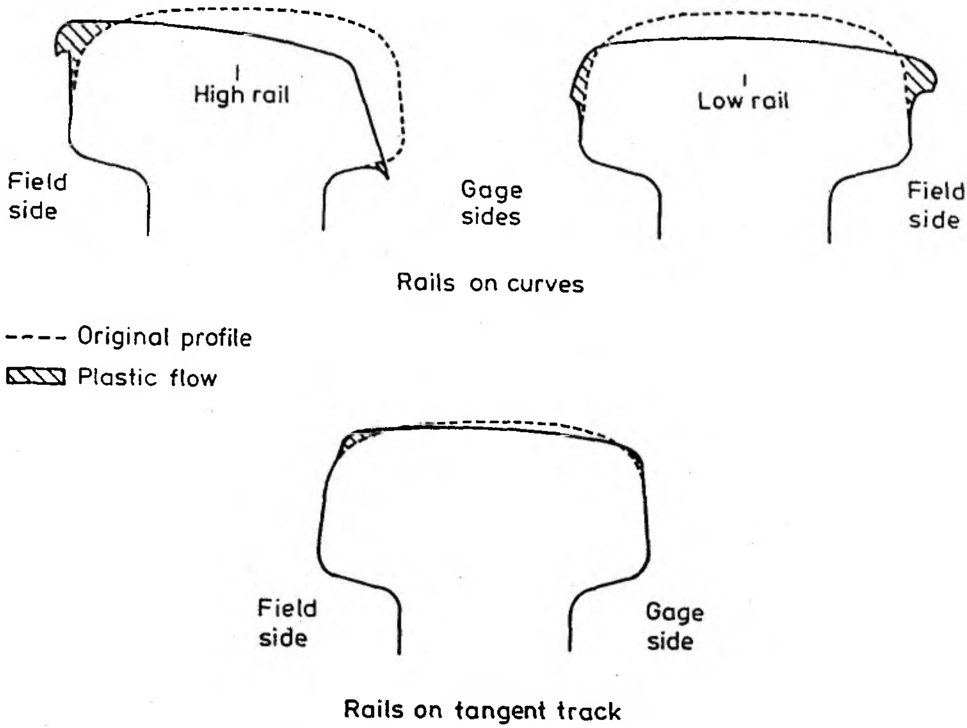

Fig. 1. Rail wear pattterns for curved and tanget track

Rys. 1. Zużycie szyn na prostych odcinkach i łukach

Figure 2a shows a typical wheel wear profile for tangent track running and Figure 2b shows a wear profile produced by predominantly curved track running. It is seen that although the amount of material lost in the wear process is approximately the same for both of the cases shown in Figure 2, the additional flange wear in the case of Figure 2b is a significant factor in reprofiling the wheels. The amount of material that must be removed in order to reprofile·the wheel with severe flange wear can be as uch as three times that for the wheel worn by running on tangent track, as illustrated in Figure 3. As a consequence, as few as two of three reprofiling cycles may be possible berfore the wheelset must be scrapped. For the case of the wear pattern shown in Figure 2a the loss of 2 to 3 mm of material corresponds to running in excess of 100 000 km. The technical and economic aspects of this have been discussed by the author [9,10].

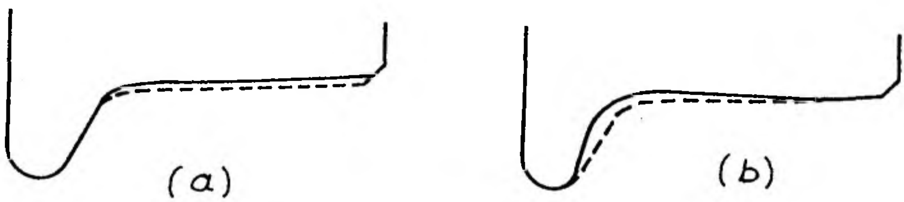

Fig. 2. Wheel wear patterns for (a) predominantly tangent track running, and (b) predominantly curved track running
--------- Original Profile, ―――――― Worn Profile
Rys. 2. Zużycie kół z przewagą odcinków
a) prostych, b) z łukami --------- profil orginalny, ―――――― profil zużyty

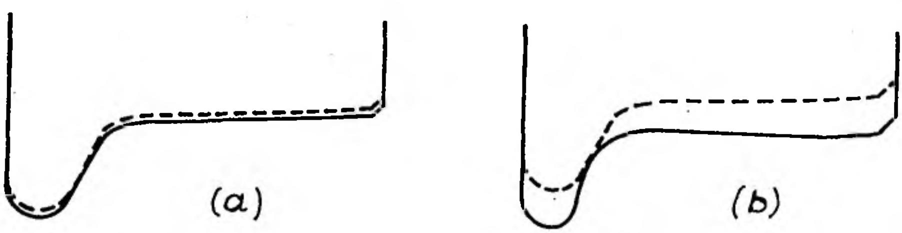

Fig. 3. Schematic of material removal necessary for wheel reprofiling for wear patterns of figure 2
―――――― Worn Profile, -------- Refinished Profile,
Rys. 3. Schematyczne przedstawienie niewykorzystanego materiału koła podczas regeneracji dla przykładów pokazanych na rys. 2
―――――― profil zużyty, ---------- profil po regeneracji

2.3. Methods of increasing the life of wheels and rails in curves

Theoretical and experimental studies of the problem of excessive wear of the wheel/rail system for curves lead to the conclusion that the wear is influenced by aspects of metallurgy (steel, heat treatment), construction (limiting wheel forces) and lubrication. The effects of heat treatment and lubrication are additive. At high axle loads both heat treatment and lubrication can individually have the wear rate (with lubrication being marginally) more effective). However when both factors are present the wear rate can be reduced by a factor of five [11]. An evaluation of rail type behaviour by Hargrave et al [12] indicates that the effect of lubrication is greater with standard rail types than with higher quality steel and that lubrication-metallurgy interactions can be an influence. The problem of rail wear in curves has received much attention, for example references [13] and [14].

One of the current authors has presented methods of increasing the life of the elements of wheel/rail systems [15]. The factors considered were metallurgical treatments of the railhead and the influence of geometrical parameters on the wear. Wear experiments were carried out using 40 mm diameter discs finished to representative wheel and rail profiles. The discs were loaded together with a force of 700 N and rotated for 50.000 revolutions with 2% slip. Figure 4 shows the influence of heat treatment on the disc wear which is presumed to be representative of a wheel rail system. The equation for selection of optimal parameters for the surface layers in a wheel/rail system have been shown to be

$$
\begin{gathered}
Z_w + Z_r = Z_{min} \\
\frac{F_w}{F_r} = A_{opt}
\end{gathered}
\qquad (1)
$$

where Z_w is the wheel wear and Z_r is the rail wear. The factors F are properties such as hardness, yield stress metallurgical composition, etc and the optimum value, A_{opt}, for each ratio nis dependent on economic considerations which vary with the application. For example, for the hardness ratio in an underground application a small value of A results in most of the wear taking place in the wheel which can be coped with more easily and economically.

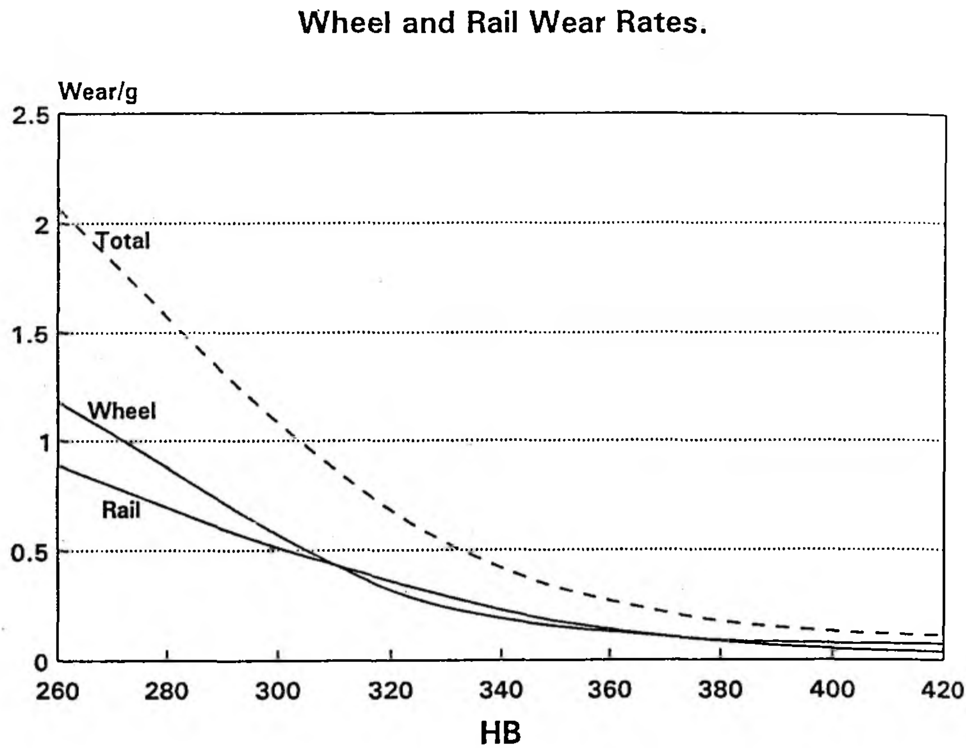

Fig. 4. Wear rates of the wheel profile disc and the rail profile disc expressed in terms of Brinnell Hardness for disc machine wear experiments

Rys. 4. Zależność zużycia badanych kół i szyn w zależności od twardości (HB)

The potential for reducing wear by adopting different constructional factors have been investigated by many authors, for example references [7] and [16]. In reference [7] an interesting experiment comparing the radial force on curves for cylindrical and tapered wheels with standard stiff suspension and experimental soft suspension systems is reported. Tapered wheels exhibit smaller forces with softer suspenion contributing a further beneficial effect.

Lubrication as a means of reducing friction and wear between steel contacting surfaces and at the same time reducing corrosion is a long established practice. The lubrication of railway lines is probably also quite a long standing practice, although this was at first achieved inadvertently by the use of journal bearings with consequent leakage of lubricant onto the track [14].

Systematic track lubrication is a relatively recent development in the history of railways, although its benefits have been reported in the literature since the 1940 s. In recent years several experimental track lubrication programs have been reported showing significant differences in comparison with dry track running. The change in wear rates on curved track is very significant.

Fujinawa [18] reports a hundredfold reductin in rail gauge face/wheel flange wear whereas Czuba [19] reports a more modest but nevertheless substantial improvement by a factor of between 5 and 7 in a revenue service experiment using trackside lubrication. The increase in rail life due to lubrication is typically 50 to 100% [20]. This is due to the reduced wear which is itself dependent on the level of lubrication. For standard carbon rails a fivefold reduction in wear rate is reported with a 'low' level of lubrication, but this is improved to an eighty fold reduction with a 'high' level of lubrication [20]. In addition to reducing the wear rate, lubrication has been shown to reduce rail end batter, and the rate of corrugation growth.

Together with the clear benefit of reduced wear (and consequent reduced maintenance) lubrication gives rise to a reduction in friction which in turn leads to reduced energy consumption. During the course of the experiments reported in reference [20] it was noticed that the throttle settings required to maintain set speeds were significanty different between lubricated and unlbricated conditions, and it was deduced that fuel savings of the order of 30% were possible over the curved experimental track. A further experiment reported [20] measured th energy required to pull unit coal trains over a 320 km route. Tests were conducted with no lubrication, trackside lubricators, onboard lubrication and both types of lubrication combined. The results are presented in the form of gross ton miles per kilowatt-hour with the improvement over the unlubricated state being 17% for onboard lubricators, 24% for trackside lubricators and 43% when both lubrication methods were used. Trackside lubricators, however, introduce a further maintenance requirement and seasonal variations in temprature may require different lubricants to be used in winter and summer.

Lubrication Equations

The way in which a pressurised oil film is developed between lubircated contacting surfaces is described by the Reynolds Equation

$$\frac{\partial}{\partial x}\left(\rho\,\frac{h^3}{\eta}\,\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial y}\left(\rho\,\frac{h^3}{\eta}\,\frac{\partial p}{\partial y}\right) = 12u\,\frac{\partial(\rho h)}{\partial x} \tag{2}$$

where p is the lubricant pressure, ρ its density, η its viscosity, h is the separaion of the surfaces and u is the mean velocity of the contactin surfaces relative to the point of contact. The x and y axes are in the common tangent plane with the origin at the point of contact and x in the direction of u.

If the pressures ar high the density and viscosity cannot be regarded as constant and are assumed to vary according to

$$\rho = \rho_o \left(\frac{1 + \lambda p}{1 + \gamma p}\right) \tag{3}$$

$$\eta = \eta_o l^{\alpha p} \tag{4}$$

where λ, γ and α are constants, and ρ_o and η_o are the values of density and viscosity t ambient pressure.

The regime of lubrication for the rail wheel contacts considered is elastohydrodynamic, with the contacting surfaces being significantly deformed by the pressure developed in the lubricant. The deformation is taken to be that occurring in semi-infinite bodies,

$$d(x,y) = \left\{\frac{1-\gamma_1^2}{\pi E_1} + \frac{1-\gamma_2^2}{\pi E_2}\right\} \iint \frac{p(x_1,y_1)}{\sqrt{(x-x_1)^2+(y-y_1)^2}}\,dx_1\,dy_1 \tag{5}$$

which consequently gives a film thickness that depends on the pressure according to

$$h(x,y) = \frac{x^2}{2R_x} + \frac{y^2}{2R_y} + d(x,y) + s \tag{6}$$

where R_x and R_y are the effective radii of curvature in the x and y directions and s is an arbitrary constant chosen to obtain the required load, w,

$$w = \iint p(x,y)dy \tag{7}$$

The integrations in equations (5) and (7) are carried out over the area in which fluid pressure is generated. The equations are solved numerically using an iterative technique as described in [22] and [23]. The contacts to be considered are specified by the load w, entrainment velocity u, lubrication parameters η_o, α, λ, γ together with the radii of curvature R_x and R_y.

Results

The results presented in detail in this paper have been reported in summary in [24]. The load considered for the work was fixed at 100 kN, as being representative of the load likley to be carried by the contact. The entraining velocity was fixed at 22.2 m/s corresponding to a vehicle speed of 80 km/hr. The lubricant corresponds to that used currently for this purpose on Polish railways and the parameter values assumed are η_o = 0.018 Pas and $\alpha = 15\text{x}10^{-9}$ Pa^{-1}, although these parameters canbe expected to vary with temperature. The effective radius of curvature in the x (rolling) direction is given by the contact diameter of the wheel and the angle the wheel axle makes with the tengent plane at the contact. In this work we have taken R_x to be 0.483 m.

Figure 5 shows the transverse profiles of a wheel and rail, and contact is possible between a number of different sections of the two profiles, as outlined in Table 1.

During this study the contacts considered have been assumed to take place on the 13 mm radius arc at the top corner of the railhead, i.e. section 1' in Figure 5. Section 1' of the rail normally makes contact with section 2 of the wheel. The wheel transverse radius in this area is nominally 13 mm which will give prefect conformity and a contact that extends over the whole of the 13 mm radiused section. However, as the wheel becomes worn the profile radius will increase and a range of wheel transverse radii from 13.1 m to 23 mm has been considered.

The effective transverse radius of curvature, R_y, is given by

$$\frac{1}{R_y} = \frac{1}{R_{yr}} + \frac{1}{R_{yw}} \tag{8}$$

and with R_{yr} = 13 mm the range of R_{yw} values specified above leads to R_y in the range 30 mm to 1.5 m. (Note that R_{yw} is negative as the surface is concave).

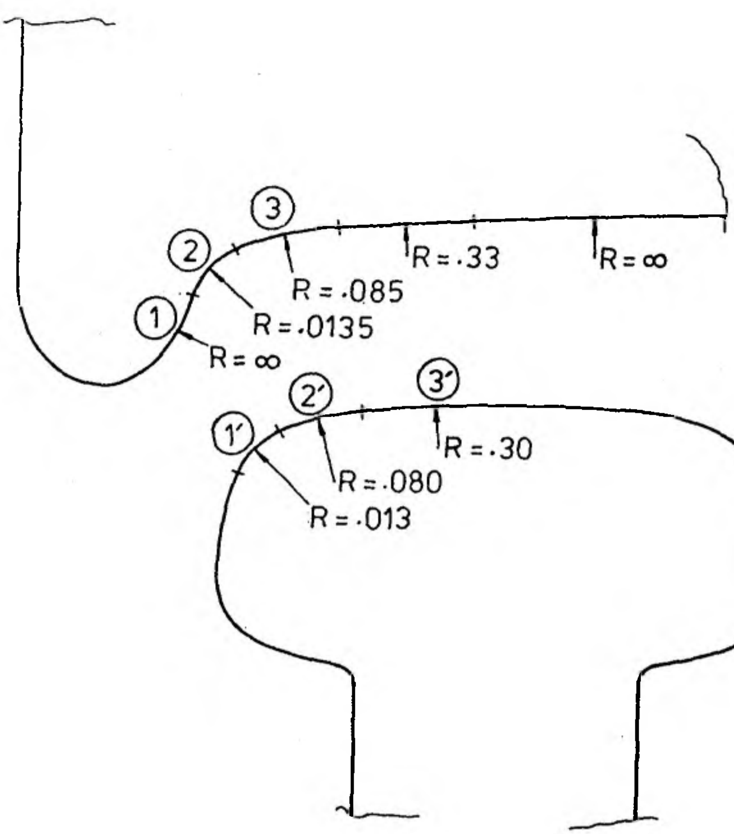

Fig. 5. Schematic of wheel and rail transverse profiles showing the radii of curvature of the different sections

Rys. 5. Schematyczne przedstawienie profili koła i szyn ich promieni w określonych przedziałach

Table 1

Possible contacts (+) between wheel and rail

Rail \ Wheel	1	2	3	4	5
1'	+	+	+	+	-
2'	-	-	+	+	+
3'	-	-	-	+	+

Results were obtained for eight different values of R_y, with R_x/R_y varying from 0.33 to 16. The lower value of this ratio is the case where wheel and rail have a profile radius mismatch of about 1% and represents a relatively unworn wheel on a new rail. The higher value of the ratio corresponds to a badly worn wheel whose profile radius at the contact has increased to 23 mm. Figure 6 shows the results for the case $R_x/R_y = 0.33$, which results in an elliptical area of contact whose major axis is 22 mm long. Such a contact would extend for more than the available arclength of the 13 mm profile and would be modified by end effects as the radii of curvature of the surfaces change within the loaded area of the contact. This effect is not taken into account in this analysis. The figure shows longitudinal and transverse sections of the pressure and film thickness, and also the contours of film thickness over the contact area. The pressure distribution is close to Hertzian showing little pressure generation outside the Hertzian region and a vestigial pressure spike in advance of the film constriction at the rear of the contact. The film formed over the dry contact area has an effectively constant value of 3.9 μm with the exception of the constriction which extends to the sides of the contact where side lobes form in which the minimum film thickness h_m of 2.7 μm occurs. These features are characteristic of relatively heavily loaded EHL contacts.

Figure 7 shows the corresponding results for the almost circular contact obtained when the radius ratio $R_x/R_y = 1.41$. The longitudional sections again show an almost Hertzian pressure variation with a parallel film whose central value h_c is 3.1 μm. The variation in film thickness in the transverse direction is now more pronounced and we see that the side lobes are considerably deeper than those shown in figure 6, with a minimum film thickness value of 1.1 μm. Figure 8 shows an isoparametric projection of half of the pressure distribution viewed from the exit side of the contact and illustrates the close to Hertzian nature of the pressure distribution.

The results for the most extreme radius radio considered, $R_x/R_y = 16$, are shown in Figure 9. This case is very heavily loaded in EHL terms with the pressure spike absent except for a change in curvature of the pressure curve at the location of the film constriction. The transverse film thinning in this case is severe with the central film thickness of 1.6 μm reduced by 90% to 0.13 μm in the side lobes.

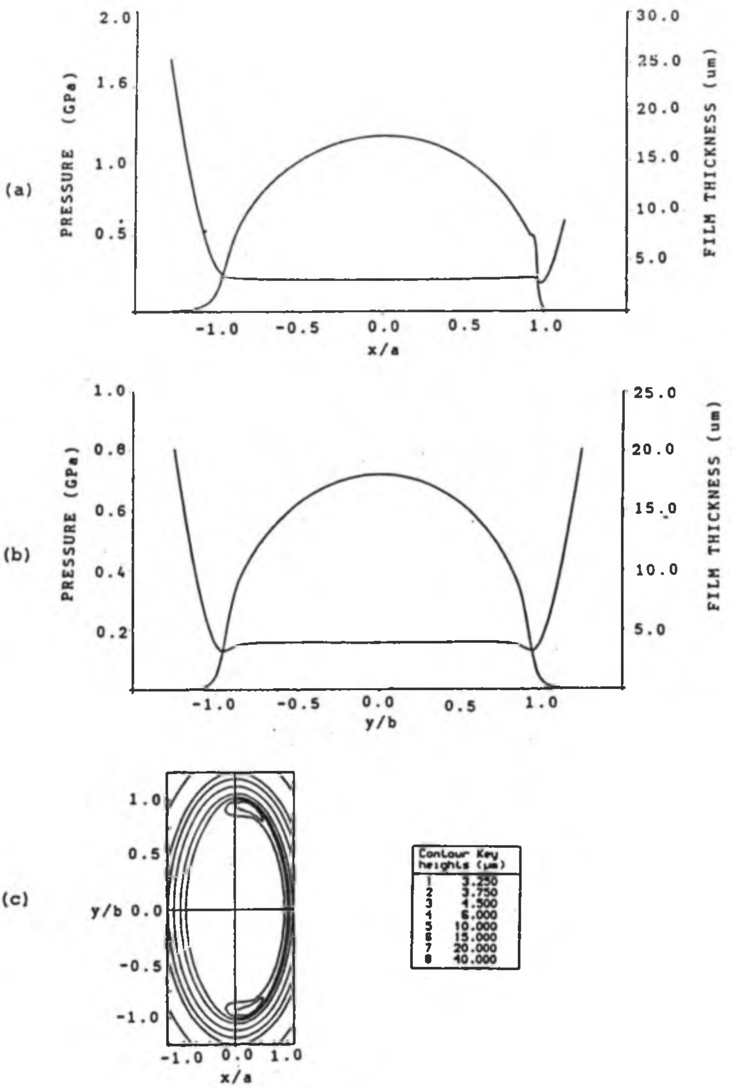

Fig. 6. Results for the case having $R_x = 0.48$ m, $R_y = 1.45$ m
a) Sections of pressure and film thickness on line y = 0,
b) Sections of pressure and film thickness on line x = 0,
c) Film thickness contours; $h_c = 3.9\ \mu m$, $h_m = 2.7\ \mu m$

Rys. 6. Rezultaty dla przypadku $R_x = 0.48$ m, $R_y = 0.34$ m
a) Ciśnienie i grubość warstwy oleju na prostej y = 0,
b) Ciśnienie i grubość warstwy oleju na prostej x = 0,
c) warstwice grubości filmu dla $h_c = 3,9\ \mu m$, $h_m = 2,7\ \mu m$

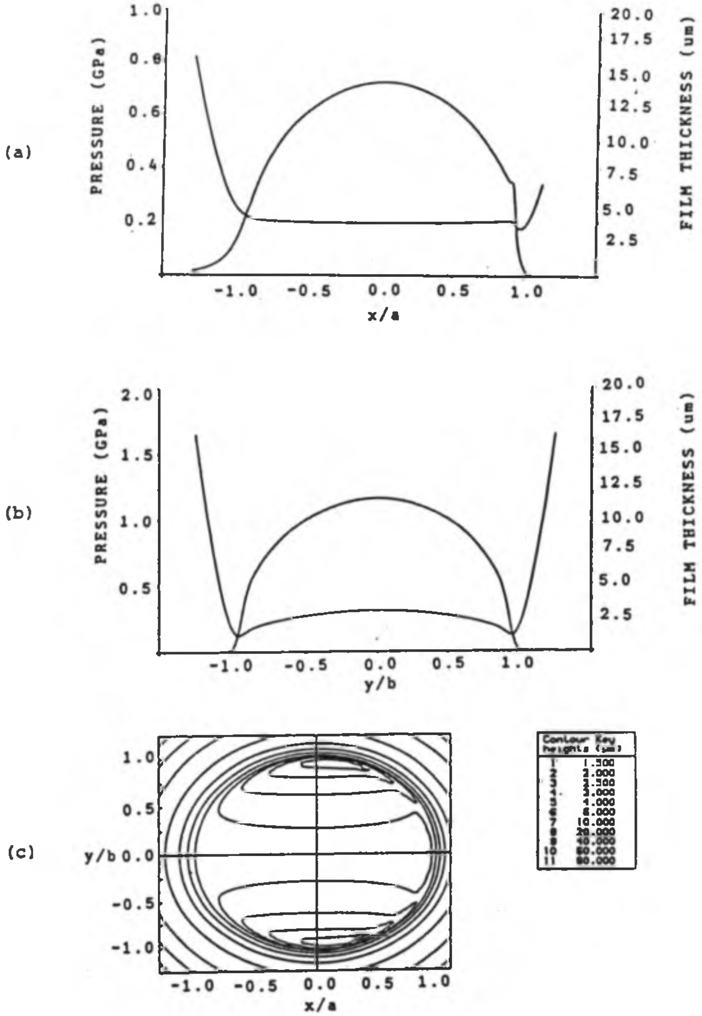

Fig. 7. Results for the case having R_x = 0.48 m, R_y = 0,34 m
a) Sections of pressure and film thickness on line y = 0,
b) Sections of pressure and film thickness on line x = 0,
c) Film thickness contours: h_c = 3.1 μm, h_m6= 1.1 μm

Rys. 7. Rezultaty dla przypadku R_x = 0.48 m, R_y = 0.34 m
a) Ciśnienie i grubość warstwy oleju na prostej y = 0,
b) Ciśnienie i grubość warstwy oleju na prostej x = 0,
c) warstwice grubości filmu dla h_c = 3,9 μm, h_m = 2,7 μm

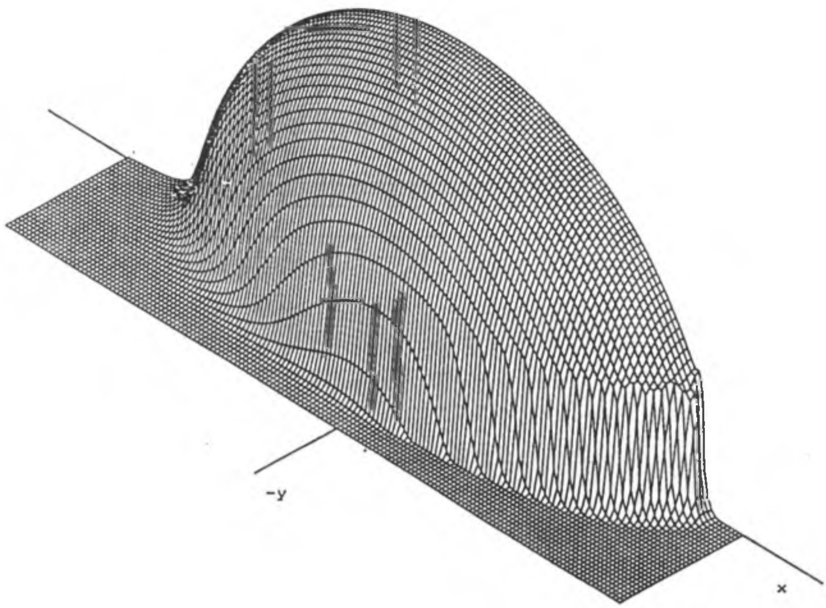

Fig. 8. Isoparametric projection of the pressure distribution for the case having R_x = 0.48 m, R_y = 0.34 m. Maximum pressure = 1.16 GPa

Rys. 8. Izoparametryczny rozkład dla przypadku R_x = 0,48 m, R_y = 0,34 m. Maksymalne ciśnienie P = 1,16 MPa

The results for film thickness are summarised in Figure 10 where the variation of h_c and h_m with the wheel transverse radius R_{yw} are shown. It can be seen that the central plateau film thickness remains high at 2 to 4 μm over the whole of the range of R_y. However, the minimum film thickness is very sensitive to R_y and falls from a healthy value of approximatly 2 μm for the circular contact to an asymptotic level of about 0.1 μm as the wheel's transverse radius is increased (leading to reduced R_y). The most rapid change in minimum film thickness takes place as the wheel radius increases to 15 mm.

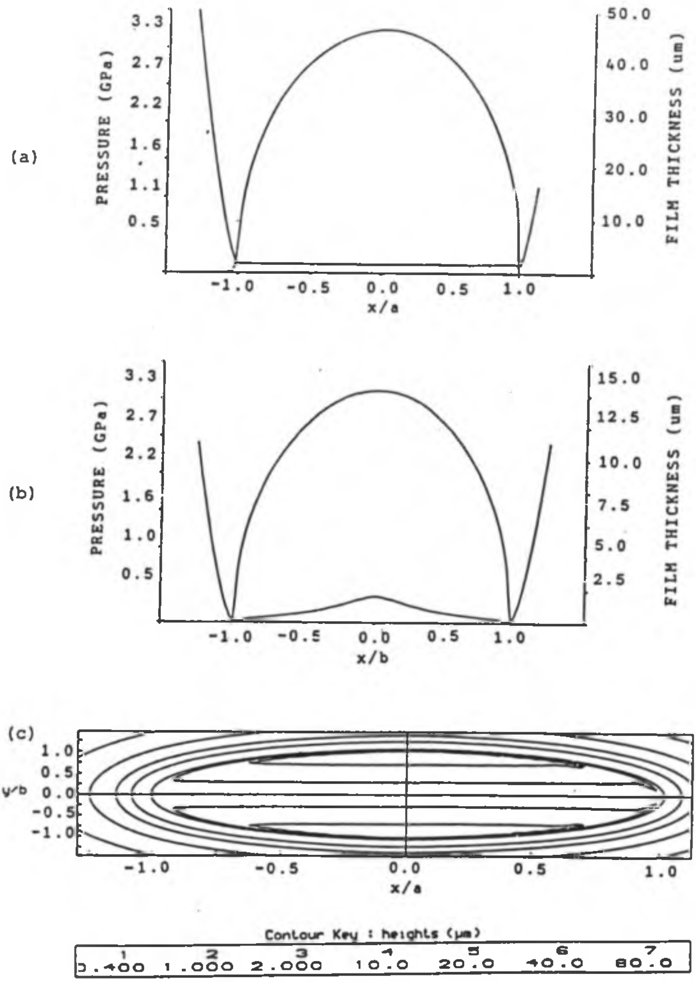

Fig. 9. Results for the case having R_x = 0.48 m, R_y = 0.03 m
a) Sections of pressure and film thickness on line y = 0,
b) Sections of pressure and film thickness on line x = 0,
c) Film thickness contours; h_c = 1.6 μm, h_m = 0.13 μm

Rys. 9. Rezultaty dla przypadku R_x = 0.48 m, R_y = 0.03 m
a) Ciśnienie i grubość warstwy oleju na prostej y = 0,
b) Ciśnienie i grubość warstwy oleju na prostej x = 0,
c) warstwice grubości filmu dla h_c = 1,6 μm, h_m = 0,13 μm

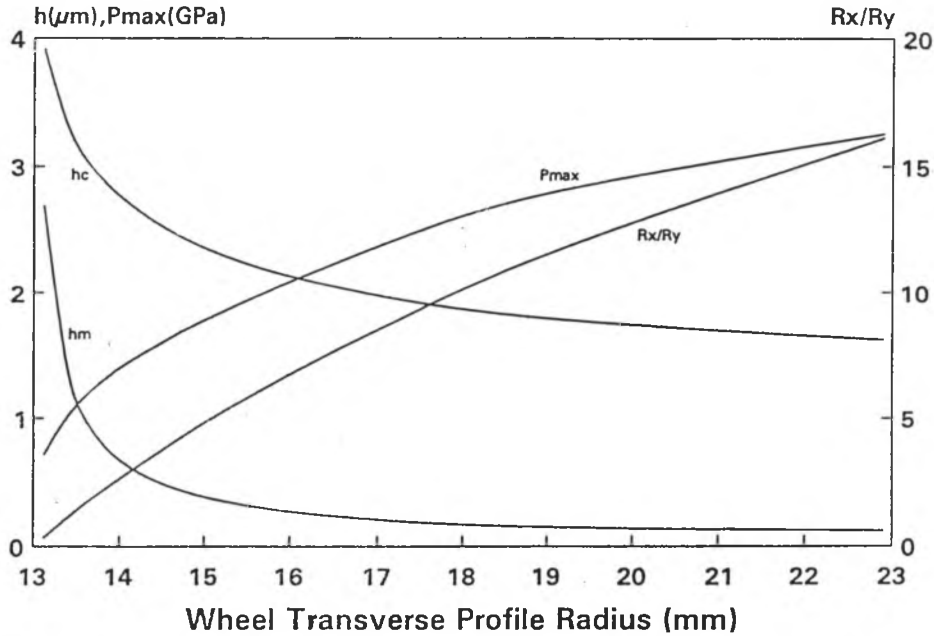

Fig. 10. Variation in h_c and h_m with wheel transverse profile radius at the contact. Also shown are the maximum contact pressure, P_{max} and the radius ratio R_x/R_y

Rys. 10. Zmiany h_c i h_m w zależności od promienia względnego koła w miejscu styku. Pokazano również zmiany maksymalengo ciśnienia P_{max} i zależności R_x/R_y

Conclusions

Lubrication of wheel flange/rail contact on curves is a common pracrtice on railways today. It is effective in reducing wear and energy requirements. The contacts operate in the elastohydrodynamic regime and would appear to be heavily loaded in EHL terms for the representative operating conditions considered in this paper. There may be considerable variation in film thickness over the contact in cases where the cotact aspect ratio is large. The way in which the geometric properties and lubricant type affect the wear rate in elliptical point contacts requires experimental evaluation. The teory applied in this paper presumes smooth surfaces and the effect of surface roughness needs to be assessed.

ACKNOWLEDGEMENT

The authors acknowledge that the financial assistance of the E.C. TEMPUS scheme provided the opportunity or the study reported in this paper to take place.

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Wpłynęło do Redakcji 22.01.1991

STRESZCZENIE

W pracy rozpatrywano teoretyczne i eksperymentalne zagadnienia związane ze smarowaniem kół kolejowych w łukach przy dużych obciążeniach zewnętrznych.

Przedstawiono rezultaty analizy numerycznej grubości filmu olejowego i rozkładu ciśnienia podczas smarowania obrzeży kół kolejowych w łukach. W literaturze dotyczącej smarowania systemu koło-szyna brak było dotychczas opracowań teoretycznych dotyczących zagadnień optymalizacji grubości filmu olejowego i rozkładu ciśnienia w strefie ich współpracy. Eksploatacyjne badania [2,3] potwierdzają znaczne korzyści techniczne i ekonomiczne, jakie można osiągnąć stosując smarowanie systemu koła-szyna w łukach. I tak na przykład stosując smarowanie tego systemu w łukach można od 5 do 7 razy zmniejszyć zużycie szyn i kół kolejowych oraz o 30% zmniejszyć zużycie paliwa.

W praktyce największe zużycie powierzchni tocznej kół kolejowych występuje w miejscu najmniejszego promienia $R = 13$ mm. Dlatego w pracy przeprowadzono analizę grubości filmu olejowego w zależności od parametrów geometrycznych koła, wykorzystując elastohydrodynamiczną teorię smarowania. Parametry oleju odpowiadały parametrom olei stosowanych aktualnie w Polsce do smarowania obrzeży kół w łukach. Przyjęto prędkość zestawu kołowego 22,2 m/s, co odpowiada 80 km/h oraz obciążenie 100 kN. Rezultaty obliczeń numerycznych przedstawiono na rys. 6,7,8,9, z których wynika, że największe zmiany grubości filmu olejowego występują w przedziale promieni $R = (13 - 15)$ mm.

Podsumowując należy stwierdzić, że smarowanie systemu koło-szyna w łukach jest obecnie ważnym praktycznym problemem. Jest efektywnym sposobem zmniejszenia zużycia materiału kół i szyn kolejowyh, jak również sposobem zmniejszenia zużycia energii.