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LUBRICATION OF RAILWAY RAIL WHEEL-FLANGE CONTACTS

Streszczenie. W artykule przedstawiono rezultaty analizy numerycznej grubości filmu 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.

Резюме. В работе представлены результаты численного анализа толщины смазки и разпределения давления во время смазки гребней железнодорожных колёс. Проведён анализ толщины смазки в зависимости от геометрических параметров колеса, используя упругогидродинамическую теорию смазки.

Summary. The paper presents the results of an analysis of the film thickness and pressure distribution likely to be found in the heavily loaded, elliptical, lubricated point contacts between railway wheel flanges and curved track. The lubrication is seen to be in the elastohydrodynamic regime, and film thicknesses for a range of wheel transverse profile radii are presented.

1. INTRODUCTION

Lubrication of wheel flange - rail contacts as a means of reducing friction and wear on curved track has become a common practice in railway technology. Details of a typical lubricator design in use on Polish railways are given by Barylski and Koc [1], and reports of experimental work and field trials on track lubrication are common in the literature and have shown significant differences over dry track running. The change in wear rates on curved track is very significant. Fujinawa [2] reports a hundredfold reduction in rail gauge face/wheel flange wear whereas Czuba [3] 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% [2]. This is due to the reduced wear which is itself dependent on the level of lubrication. Allan and Reiff [4] report that for standard carbon rails there is a fivefold reduction in wear rate with a 'low' level of lubrication, but this is improved to an eightyfold reduction with a 'high' level of lubrication. 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 leads to lower energy consumption. During the course of the experiments reported in reference [4] the throttle settings required to maintain set speeds were significantly different between lubricated and unlubricated conditions, and it was deduced that fuel savings of the order of 30% were possible over the curved experimental track used.

2. CONTACT CONDITIONS

The lubricated contact between rail and flange has a geometry that is modified by the state of wear of the wheel and track. For a new wheel and new track the contact is between a convex rail and a concave section of the wheel having nominally equal radii of curvature (13 mm). This results in a line contact whose radius is of the order 0.5 m, with the extent of the contact being determined by the transverse arc lengths of the contacting components. As the wheel wears, its effective radius of curvature increases over the part of the flange in contact with the rail, and flange wear can result in a considerable increase in the ammount of material that must be removed when the wheel is reprofiled.

The change in wheel flange profile radius causes a change in the contact conditions between it and the rail. The initial line contact becomes a point contact with an elliptical contact area. With low wear the contact ellipse has its minor axis in the direction of lubricant entrainment, but as the wear increases the orientation of the contact ellipse changes so that it has its major axis in the entrainment direction. This latter arrangement is less able to generate effective films due to the increased lubricant side leakage.

3. LUBRICATION REGIME

The loads carried by wheel flange contacts are sufficiently high, and the contacts sufficiently concentrated, for the contacts to operate in the Elastohydrodynamic (ehl) regime. To establish the film thickness a solution to both the Reynolds equation for the lubricant and the elasticity equations for the contacting bodies is required. A description of the numerical methods used to obtain the results presented in this paper are given by Evans and Snidle [5] and Kweh et al [6] and will not be rehearsed here. They have been developed to solve the isothermal point contact ehl problem, and are particularly appropriate for the wheel flange - rail problem due to the high stress level found in these contacts.

4. RESULTS

The effective radius of curvature along the rail, R_x, is given by the contact diameter of the wheel and the angle the tangent plane at the point of contact makes with the wheel axle. In this work we have taken R_x to be 0.483 m. The load considered was fixed at 100 kN, as being representative of the load likely to be carried by the contact. The lubricant corresponds to that used currently for this purpose on Polish railways, and the values assumed for its viscosity and pressure viscosity coefficient are 0.018 Pas and 15 GPa^{-1} respectively. The entraining velocity was fixed at 22.2 m/s corresponding to a vehicle speed of 80 km/h.

For this study the rail profile radius, R_{yr}, is taken to be 13 mm, and that of the wheel in contact with it, R_{yw}, has been varied from 13 mm to 23 mm. The effective transverse radius of curvature, R_y is given by

$$1/R_y = 1/R_{yr} + 1/R_{yw} \qquad (1)$$

and the range of R_{yw} values considered above leads to R_y in the range 30 mm to 1.5 m so that R_x/R_y varies from 0.3 to 16. (Note that R_{yw} is negative).

Results were obtained for eight different contacts and Figure 1 shows the film thickness contours for the case $R_y = 50$ mm, giving $R_x/R_y = 9.7$. This is an elongated contact with the major axis along the rail, whose semi major axis, a, is 9.2 mm and semi minor axis, b, is 2.1 mm. We see that the minimum film thickness, h_m, occurs in the contact's side constructions close to the transverse centre line. This is a feature of heavily loaded contacts and the pressure distribution is close to Hertzian with little pressure generated outside the dry contact area. The value of h_c, the film thickness at the centre of the contact, is 1.9 μm which is reasonably thick in ehl terms. The value of h_m however is an order of magnitude smaller at 0.16 μm.

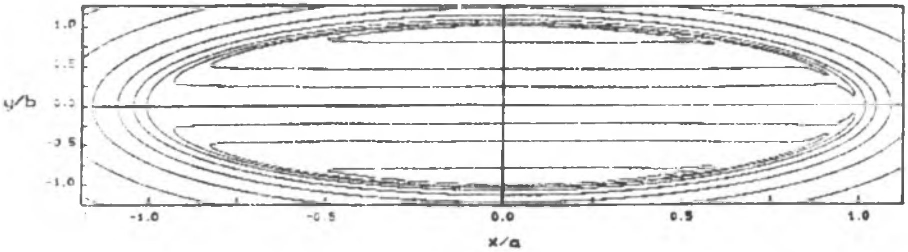

Fig. 1. Film thickness contours for the case R_y = 50 mm
(Contours shown are 0.5, 1, 1.5, 2.5, 5, 10, 20, 40 and 80 μm)

Rys. 1. Warstwice grubości filmu dla przypadku R_y = 50 mm
(Warstwice pokazują 0.5, 1, 1.5, 2.5, 5, 10, 20, 40 i 80 μm)

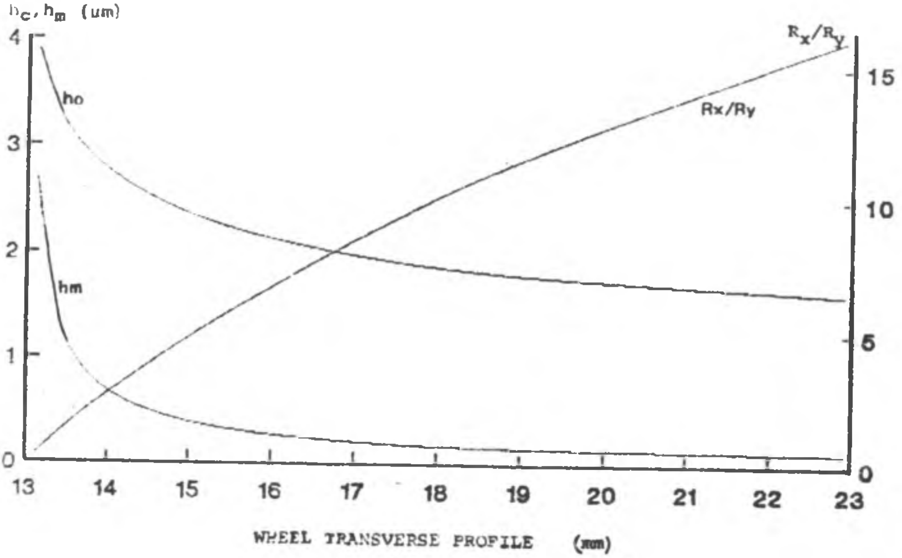

Fig.2. Variation of h_c, h_m and R_x/R_y with R_{yw}

Rys.2. Zmiany h_c, h_m oraz R_x/R_y w zależności od R_{yw}

The results for film thickness are summarised in Figure 2 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 2-4 μm over the whole of the range of R_{yw}. However, the minimum film thickness is very sensitive to R_{yw} and falls from a healthy value of approximately 2 μm for almost circular contacts 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 from 13 mm to 15 mm.

5. CONCLUSIONS

Lubrication of wheel flange - rail contact on curves is a common practice on railways today. It is effective in reducing wear and energy requirements. The contacts operate in the elastohydrodynamic regime and there may be considerable variation in film thickness over the contact in cases where the contact 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 theory applied in this paper presumes isothermal conditions and smooth surfaces. The possible effect of shear heating and the effect of surface roughness in such contacts needs to be assessed.

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SMAROWANIE OBRZEŻY KÓŁ KOLEJOWYCH ZESTAWÓW KOŁOWYCH

W artykule 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 koło-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.1 i rys.2, 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 kolejowych, jak również sposobem zmniejszenia zużycia energii.