

**XXII INTERNATIONAL CONFERENCE ON
"MATERIAL HANDLING, CONSTRUCTIONS AND LOGISTICS"**

4th - 6th October, 2017

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**Edited by
Nenad Zrnić, Srđan Bošnjak and Georg Kartnig**

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Faculty of Mechanical Engineering
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BELGRADE, SERBIA, 2017

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Determination of the longitudinal creep of a driven crane wheel on a crowned rail

The longitudinal creep on a crane wheel has considerable effects, both for the design and the control of the drive technology of a modern crane. The creep occurring during operation is, however, widely unknown. This article presents a calculation model for longitudinal creep of a driven crane wheel, which can be used for a fast analytical determination. Based on the contact area between the crane wheel and the rail an equivalent line contact is calculated instead of the complex real point contact situation. The approach was validated at the institute's wheel-rail test rig.

Keywords: Longitudinal creep, crane wheel, rail-wheel-contact, analytic method, running-in characteristic

1. INTRODUCTION

Currently applicable standards for crane design (DIN EN 13001 and sub-standards [1]) consider only rail geometries with flat heads at the contact between crane wheel and rail. Nowadays, crowned rails are used almost exclusively. As a result of the wear-specific run-in behavior, a change in the rail head profile occurs with increasing operating time (overrun cycles). This also results in a change of the initially ideal elliptical contact surface to an approximate line contact geometry, but not over the entire width of the railhead.

For this reason, a research project was launched in 2014 at the Department of Transport, Handling and Conveying Systems (KLFT) in cooperation with Hans Künz GmbH, which aims at transferring existing approaches of the static and structural design for the flat rail head to the general case of a cambered rail. The effects of the modified rail geometry on relevant system parameters, e.g. contact pressure, rolling friction, skewing forces, longitudinal creep, adhesion and wear of wheel and rail are to be investigated. Furthermore, the characteristics of the run-in behavior of the rail head due to plastic deformation are considered in the context of the project.

In 2016 the Excellence Center of Tribology (AC²T) was included in the project to achieve a better understanding of the tribological aspects within the contact surface.

In this publication, the longitudinal (tangential) slip between a driven or braked crane wheel and a cambered rail is to be described in more detail.

For various reasons it is necessary to know the occurring slip:

- The maximum transferable braking and driving force depends on the slip ratio (i.e. traction).
- The utilization of the coefficient of static friction is associated with higher slip ratios and higher wear.

- Different wheel loads on the individual wheels result in different slip ratios. Production tolerances on the wheel diameters also result in different circumferential forces on the wheels and thus deviating slip values. The crane clamps or distorts itself according to different wheel speeds.

However, actual slip values at the crane wheel during operation are widely unknown. Therefore an application-oriented and quick-to-calculate analytical approach for the longitudinal slip of current cranes is desirable.

When considering slippage a fundamental distinction between micro-slip (creep) and macro-slip (sliding) needs to be done. In the case of creep, the contact surface is subdivided into a stick zone with the same speed and a slip zone with a relative speed between the contact partners. The coefficient of static friction is the limiting factor for the tangential force that can be transmitted. If the slippage becomes larger, the stick zone disappears and the slip zone extends over the entire contact surface. From this point on there is pure sliding (macro-slip), and the coefficient of sliding friction is decisive (see Figure 1). On a driven or braked wheel, macro-slip corresponds to wheelspin, which in principle is to be avoided in crane construction. All slip curves considered in this work concern the micro-slip region.

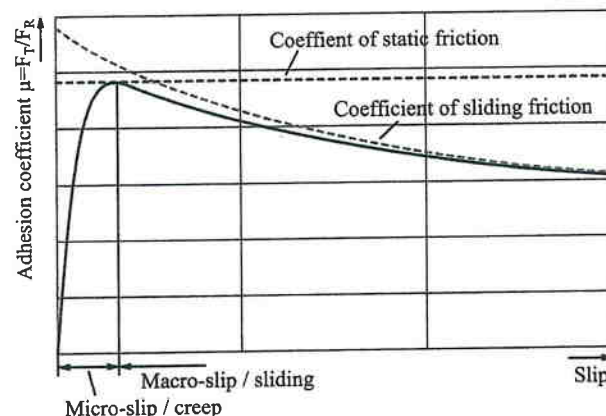


Figure 1. Traction-creep-relation (qualitative)

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2. EXISTING CALCULATION APPROACHES

The following calculation approaches are only described with respect to the tangential slip ratio. Axial slip and spin (rotation around the vertical axis) are not considered in this work, as far as the calculation methods are concerned. Furthermore, a contact geometry between a crane wheel and a crane rail according to DIN 536 has been given [2].

The tangential slip is generally defined as a related velocity difference between the circumferential speed of the wheel and the absolute velocity.

$$\xi_T = \frac{R\omega - V}{\max(R\omega, V)} \quad (1)$$

2.1 Calculation approach for line contact according to Carter

For the slip-bearing contact of a cylinder with a plane, relationships for longitudinal creep were derived by F.W. Carter [3] as early as 1926. These were again completely elaborated by G. Heinrich and K. Desoyer [4] and extended to incorporate lateral slip effects.

The following relationship for the longitudinal creep is obtained as a function of the contact force F_R , the circumferential force at the wheel F_T , the wheel radius R , the contact width in the contact b_K , the coefficient of static friction μ_0 and material constants G and ν . [4]

$$\xi_T = \sqrt{\frac{4(1-\nu)}{\pi} \frac{F_R}{G R b_K}} \mu_0 \left(1 - \sqrt{1 - \frac{F_T}{\mu_0 F_R}} \right) \quad (2)$$

Figure 2 shows the contact area as well as the shear stress distribution in the contact surface between the rolling cylinder and the plane under radial load and transmission of a torque.

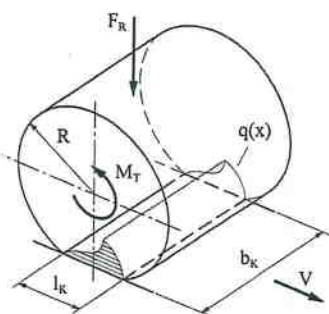


Figure 2. Contact between cylinder and plane

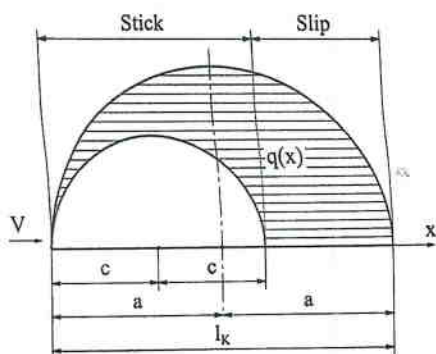


Figure 3. Distribution of shear stress and division in slip and stick zone in contact

Along the contact length l_K , the division into the slip and stick zone as well as the shear stress distribution as shown in Figure 3 are obtained.

The last term of Equation (2) corresponds to the proportion of the stick zone over the entire contact length and is a determining parameter for the creep.

$$a^* = \frac{c}{a} = \sqrt{1 - \frac{F_T}{\mu_0 F_R}} \quad (3)$$

The relationship between the contact length and the contact width for a line contact according to Hertz is included in Equation (2).

$$\frac{l_K}{2} = \sqrt{\frac{8(1-\nu)^2 F_R R}{\pi E b_K}} \quad (4)$$

Thus, the formula can be rewritten to use the length instead of the width of the contact surface. Along this length the division into stick and slip zone is also determined.

$$\xi_T = \frac{l_K}{2R} \mu_0 \left(1 - \sqrt{1 - \frac{F_T}{\mu_0 F_R}} \right) \quad (5)$$

The correctness of the approach was confirmed by J.J. Kalker amongst others using numerical methods [5].

For the contact between a crane wheel and a flat rail head, Equations (2) or (5) can be used right away. Certain deviations to the real contact situation between the wheel and the rail are to be expected since the approach does not take any edge effects at the boundary surfaces of the cylinder into account.

2.2 Calculation approaches for point contact

Approaches for calculating the slip conditions at point contact, as occurs with a cambered rail head, are not trivially solved. The transmitted tangential force is not constant over the width of the contact surface (here the long semi-axis of the contact ellipse). The distribution of the shear stress and the separation in the slip and stick zone of the contact surface is shown in Figure 4.

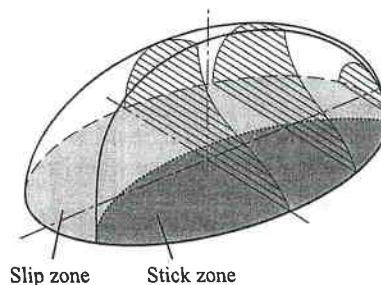


Figure 4. Shear stress distribution at point contact (qualitative)

Strip theory by Haines and Ollerton

The strip theory is a pure analytical calculation method, in which the elliptical contact surface is divided into thin strips parallel to the rolling direction and then integrated. Each individual strip is treated as a line

...but any influence of the strips on each other is neglected. The approach of B.J. Haines and E. Ollerton on tangential creep was elaborated by J.J. Kalker and developed further in order to be able to take into account creep and a small proportion of spin into account [6, 7, 8]. Figure 5 shows the discretization of the contact surface and the shear stress distribution in the

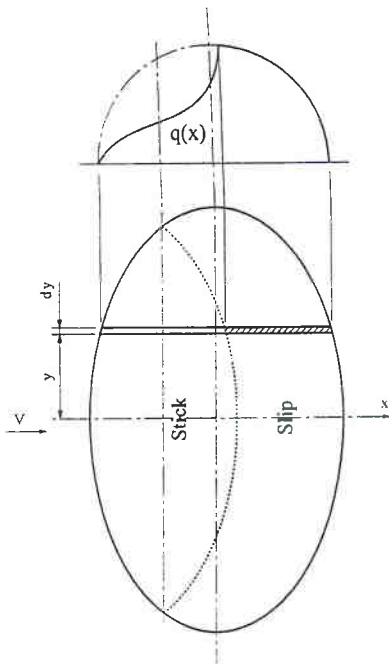


Figure 5. Strip theory according to Haines and Ollerton

The relationship between tangential force and creep defined by:

$$F_T = \mu_0 F_R \left\{ \frac{3}{2} \zeta \cos^{-1}(\zeta) + \left[1 - \left(1 + \frac{1}{2} \zeta^2 \right) \sqrt{1 - \zeta^2} \right] \right\} \quad (6)$$

with the factor ζ as a function of the slip ξ_T and the Hertzian pressure p_0 in the contact.

$$\zeta = \xi_T \frac{E}{2 \mu_0 (1 + \nu) p_0} \quad (7)$$

The results of this calculation method correspond very well with experimental results and numerical methods for slender contact areas (half-axis ratio $b/a \approx 0.2$). If, however, the contact surfaces deviate severely from this shape, the errors become large due to the lack of influence of the strips on each other [7].

The theory was not pursued further after the development of the simplified theory of Kalker in favor of the more exact numerical calculation.

Numerical methods according to Kalker

Calculation models of the exact and simplified theory developed by Kalker can only be solved numerically. They are implemented in the contact models of the programs CONTACT (exact) and FASTSIM (simplified). Both models divide the contact area into rectangular parts, which must be balanced in relation to the stress state over the entire contact surface. The exact

Kalker theory provides accurate results. At pure tangential stress on the contact the simplified theory deviates by up to 5% [9].

For more information on the numerical methods according to Kalker, see [5] and [10].

Linear method according to Kalker

This theory uses the numerically determined Kalker coefficients for the relation between slip and tangential stress. These are defined in tabular form as a function of the half-axis lengths of the contact ellipse and the Poisson ratio. Interpolations are necessary for intermediate values. The linear theory is applicable only for very small slip values since the existence of a slip zone is neglected in this approach. It represents the slope of the linear branch of the creep curve from the origin. Larger slip values caused by the influence of the increasing slip zone are not reproduced correctly. The deviation of the linear theory from the real creep curve is shown qualitatively in Figure 6.

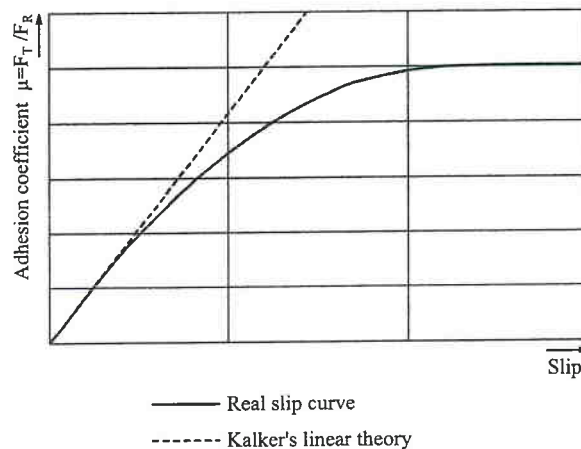


Figure 6. Discrepancy linear theory and real creep curve

According to Kalker's linear theory the dependence of the tangential force on the longitudinal creep is defined via the following relation:

$$F_T = G a b C_{11} \xi_T \quad (8)$$

For the purely tangential contact problems, only the Kalker coefficient C_{11} is required. (For values see [5].)

3. CALCULATION APPROACH FOR THE ACTUAL CONTACT GEOMETRY

The actual contact geometry between the crane wheel and the rail does not correspond to an ideal point contact after a short operating time of the crane system. Due to plastic deformation, the curvature of the rail head changes until the stresses inside the rail no longer exceed the yield point.

Imprints recorded using pressure measuring films already show run-in behavior on a new crane during the commissioning phase, which leads to a leveling of the rail head. Due to the limited measuring range of the Fujifilm Prescale films, the occurring contact pressures cannot be evaluated, but they provide very good information about the shape of the contact surface. Figure 7 shows the

measuring film on the crane rail after loading by the crane wheel. Figure 8 shows the resulting imprint on the film. In addition, the calculated contact ellipse of an ideal point contact is superimposed. Since the crane has been moved across the film, the exact contour of the contact area is not visible, but the contact width is significantly greater than the result according to Hertz's theory.

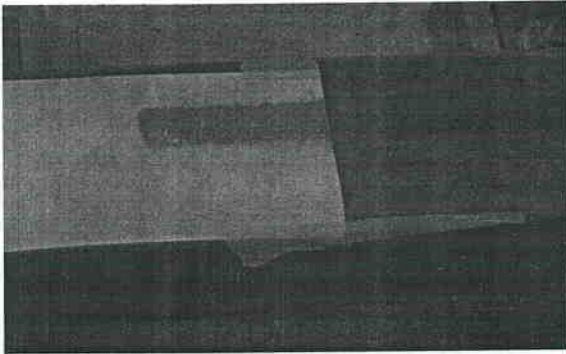


Figure 7. Fujifilm pressure measurement film on crane rail

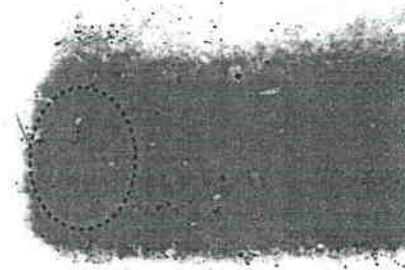


Figure 8. Imprint of the contact surface and theoretical contact ellipse

The contact width stabilizes after a certain time, so that even in the case of cranes that have been in operation for years, the contact surface does not extend over the full width of the rail. As an example of this a photograph of a crane rail after approximately seven years of operation is shown in Figure 9.



Figure 9. Run-in crane rail

Similar running-in behavior also takes place at the test rig at the Institute for Engineering Design and Logistics Engineering at the Vienna University of Technology described in more detail in Section 4. After approximately twenty operating hours with maximum wheel load the camber of the rail-wheel is flattened to a permanent geometry. Figure 10, taken at a load of 50 kN, compares a recorded imprint to the theoretical contact ellipse according to Hertz.

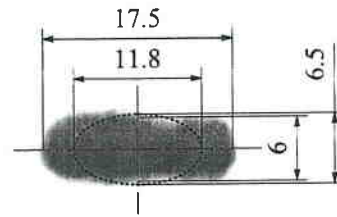


Figure 10. Run-in contact area (dimensions in mm)

For the calculation of the creep ratios for such real contact surfaces, the following approach uses the method for line contact according to Carter as well as Heinrich and Desoyer. This is adapted in such a way that a fictitious linear contact is calculated in which the average tangential stress in the contact surface coincides with the actual contact situation.

For pure tangential slippage the length in the direction of movement (l_{real}) has been identified as a determinant measure for the size of the contact surface. If the width of the fictitious contact surface is calculated according to the Hertzian theory for line contact as a function of the contact force and this contact length, the same surface areas as in the case of the real contact surfaces are obtained. Figure 11 shows two imprints, on the right a run-in rail, on the left in a new condition. The rectangle drawn corresponds to the fictitious contact surface used for slip calculation.

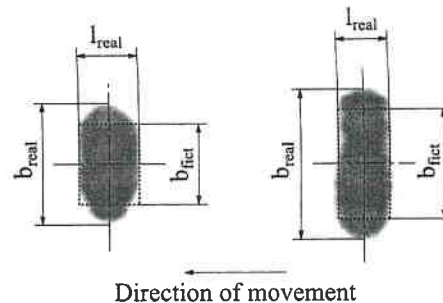


Figure 11. Dimensions of the contact areas (left: new, right: run-in)

This correspondence of the surface areas was only checked for the field of application and the geometries of crane wheels and rails. In the case of strongly divergent forms of the contact surfaces, the validity of this relationship would have to first be verified.

The essential procedure for determining the slip of a wheel according to this method is as follows:

When considering a new crane, or assuming that the wheel loads are sufficiently low to prevent plastic deformation, the required contact length l_{real} can be determined by the Hertzian theory. The half-axis length of the contact ellipse in the direction of motion can be used as a good approximation for the contact length.

If the slip on a run-in crane system is to be calculated, a determination of the real contact area is necessary. How and to which geometry a crane rail runs in is part of the investigations at the Vienna University of Technology, but the contact surface cannot yet be estimated after plastic deformation. A measurement by means of pressure measurement films represents a simple and favorable solution. The crane wheel to be evaluated is lifted by means of hydraulics and put back on the rail