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Comparing load conditions of plane and false-ellipse running surface contours of track links in running gears for construction equipment

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Abstract

To improve durability and fatigue strength, it is advisable to give running surfaces of track links in undercarriages for construction equipment a false-ellipse profile. Such a profile resembles the contour of a running surface of a railway rail. Kinematic conditions as they exist for the track run in the undercarriage of construction equipment can however not be compared with those of a railway. Moreover, construction equipment in field operation is in particular faced with problems like lateral inclinations and misalignments between running surfaces of track link and bottom roller. For all this it is necessary to have a look at the load conditions of track links. This article makes clear that, above all with a view to the fact that in the past only plane running surface profiles were used for construction equipment applications, designing track link running surfaces with a false-ellipse profile has considerable advantages compared to plane running surfaces even at a misalignment ratio of just 1/5 of the maximum width of track link running surface.

Keywords: Construction equipment; Running surface; Durability; Fatigue strength; Hertz compression

1. Introduction

As demonstrated in [1], running surfaces of track links should advisable be given the contour of a false ellipse in order to prevent and/or minimize so called link edge fractures – also known as *spalling* – on track link running surfaces (Fig. 1).

Fundamental theoretical considerations to be made on this point have, in particular, been discussed in [2]. Results of comparison have also been confirmed in [3] by numerical calculations.

Ever since the theoretical basis had been published the author was asked to compare and evaluate the stress of the different types of link running surface profiles (plain and curved) regarding the misalignment of link and bottom roller occurring in practice. Here, the main question is: at which ratio of this misalignment is the curved link running surface with false ellipse favorable to the plain one.

An answer to this will be given in the following.

2. Background

The link running surfaces are designed at present to give the best kinematic roll pair, i.e. cylinder on plane (similar Fig. 3c). The edges of the running surface are not rounded which would prevent the over stressing due to transition between plane strain in the middle portion of the rail profile and plane stress at the edge. This is a common characteristic known in roller bearing technology, too.

Therefore it could be advantageous to give the link surface an "edge rounding" contour.

Before changing the geometry of the link running surface it is necessary to evaluate the stress status under working conditions and determine the state of stress of the rail profile of the link under different loads. This was done using the Finite-Element-Method (FEM) in [4] and is illustrated and described also in [1].

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Nomenclature

a_L	half width of contact area	α	ratio D/d
b	width of running surface	μ	friction coefficient
b_K	width of contact area	φ	angle
D	diameter of bottom roller	τ_L^{max}	maximum shear stress-line contact without misalignment
d	diameter of track link running surface curvature	τ_{LK}^{max}	maximum shear stress-line contact with misalignment
p_{OL}	Hertz compression with line contact	τ_P^{max}	maximum shear stress-point contact
p_{OLK}	Hertz compression with line contact and misalignment	$\dot{\tau}_\varphi$	shear stress at angle φ
p_{OP}	Hertz compression with point contact	$\sigma_x, \sigma_Y, \sigma_z$	principal stress components
E	Young's modulus	σ_φ	normal stress at angle φ
F_R	wheel load	$\leftrightarrow$	symbolizes the interface between running surface of track link and running surface of bottom roller
r	radius of rail profile		
y_L^{max}	depth of max shear stress (line contact)		
y_{LK}^{max}	depth of max shear stress (line contact with misalignment)		

Moreover in [1] the conditions for the service durability could be shown using the example of a real case study of a link profile.

The service endurance investigations show that the edge rounding alone is not sufficient when the real service conditions of the track assembly on the machine are examined. The general working conditions of crawler machines are in sloped ground, with a continual front to back and sideways motion between the roller and rail which forces edge loading and leads to damage on a plane running surface.

For this reason, considerations were made to prevent the root problem of edge load by optimizing the geometry of the running surface, as mentioned above.

3. Advanced running surface geometry

The roll pairing of cylinder and plain is an optimum for kinematics but not for strength. Links in normal service show, after sufficient time, that the original plain takes on a false ellipse (3-center curve Fig. 2). This profile tends towards a certain relationship between the radii of curvature and the width of the running surface. As this process is proportionally faster relative to the load height, the principle of least resistance can be assumed. Therefore, the resulting curved running surface must depict the strength optimum for rail profiles. It is then a logical step to design and manufacture the links with this shape.

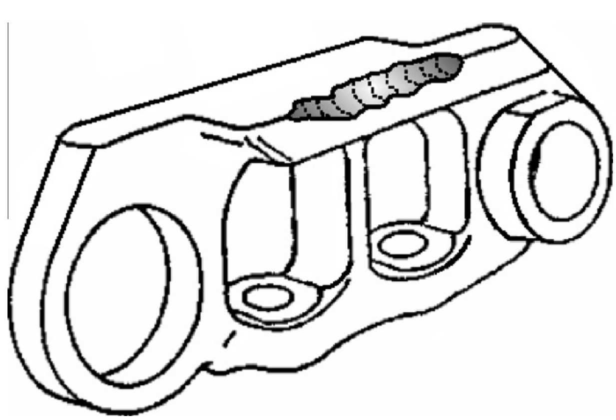

Fig. 1. Track link with edge fractures (so called *spalling*).

First and foremost in [1] the following considerations to estimate the radius of curvature could be shown, which is the starting point for experimental studies:

In the middle of the rail profile there is an almost ideal state of plane strain. This is less susceptible to yielding than the plane stress state at the edge (Fig. 4).

The lateral curvature of the running surface of the link tends towards the outside of the link due to plastic yielding of the material. The material on the running surface edge is then removed by abrasion from the roller flanges.

Fig. 5 shows the state of stress on the link edge. The shear forces must be in equilibrium with the lateral friction forces, i.e.:

$$\tau_\varphi = \mu\sigma_\varphi \tag{1}$$

τ_φ and σ_φ must also be in equilibrium with the edge stresses σ_y and $\sigma_x = 0$, resulting from Hertz compression. For the state of plane stress, with the principal stresses σ_x and σ_y (σ_z is neglected) applies:

$$\sigma_\varphi = \frac{\sigma_y + \sigma_x}{2} + \frac{\sigma_y - \sigma_x}{2}\cos 2\varphi$$

$$\tau_\varphi = \frac{\sigma_y - \sigma_x}{2}\sin 2\varphi$$

with $\sigma_x = 0$ and (1) follows:

$$\sin\varphi = \frac{\mu}{\sqrt{1+\mu^2}}$$

For $\mu = 0.15, \ldots, 0.2$, the sought relationship is found: $r/b \approx 3$.