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Simulation method for the floating of hydrodynamic guides

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Abstract

The flowing lubricant in the oil wedge between the slide and guide rail creates hydrodynamic pressure that leads to a displacement of the carriage, which depends on the sliding surface geometry and the slides velocity. This paper describes an approach to calculate the resulting complex displacement. Therefore, two methods were combined. The simulation of the floating by using the finite difference method was integrated in the numerical calculation of the balance of the whole carriage. Considering a real sliding surface geometry of a test bench, the experiment verification of the approach for velocities up to 40m/min showed a good conformity.

Keywords: oil wedge; hydrodynamic guides; simulation

1. Introduction

A survey [1] from Hirsch in 2011 of machine tool manufacturers showed, that hydrodynamic guides are mostly used beacause of their excellent damping behavior. According to this survey, one of their biggest disadvantages is the stick-slip-effect and their maximum feed velocity u. When reaching a specific sliding velocity, slide and rail are completely separated by hydrodynamic pressure and form a lubrication wedge as shown in Fig. 1a. In this state wear and friction decrease to a minimum. Fuller uses in his standard work onlubrication [2] Reynold's hydrodynamic equation [3] to describe the state of full fluid friction analytically. In full fluid friction, slide and rail form a lubrication wedge ABCD to generate hydrodynamic pressure. This wedge geometry determines that the distance BD is bigger than AC. By putting these two distances in relation to the slides length, the floating angle α can be calculated.

Experiments with long, slender, plastic coated and non-scraped sliding parts (500x50mm²) showed an unusual behavior. The friction increased, compared to scraped ones (Fig. 2) and the tilting direction caused by floating inverted (AC > BD).

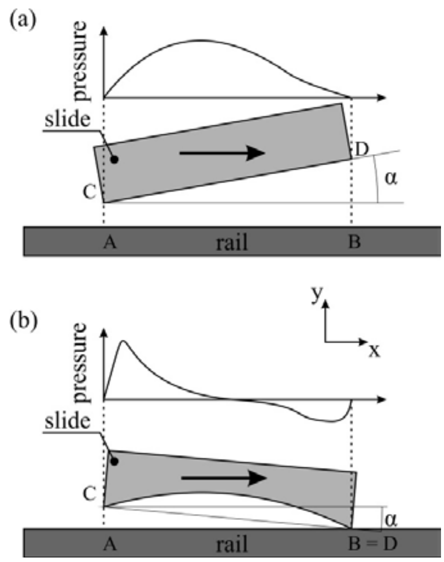

Fig. 1: Schematic pressure distribution and resulting floating behavior for plain (a) and concave (b) wedge geometry

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A topographical measurement showed, that the sliding surfaces of both slides were not plain as assumed by Reynolds. Their basic shapes were concave, as shown in Fig. 1b. It is assumed, that shrinking processes after moulding the plastic coatings stochastically generate form derivations of approximately 0.03 mm. This form derivation can explain the inverted tilting by it causing an altered hydrodynamic pressure distribution on the sliding surface.

This concave shape forces a lubrication wedge at the end of the slide AC, where hydrodynamic pressure occurs. On the slides other end, near BD, the inverted wedge creates an underdraft (as sketched in Fig. 1b). This pressure distribution causes a momentum which forces contact between rail and slide in BD. The experiments results in Fig. 2 show an up to 60% increased friction coefficient for concave slide geometries, compares to plain ones.

To evaluate that effect, it is necessary to develop a numeric model to describe the pressure distribution in concave wedge geometries. The described model in this paper calculates the pressure distribution and the resulting floating angle.

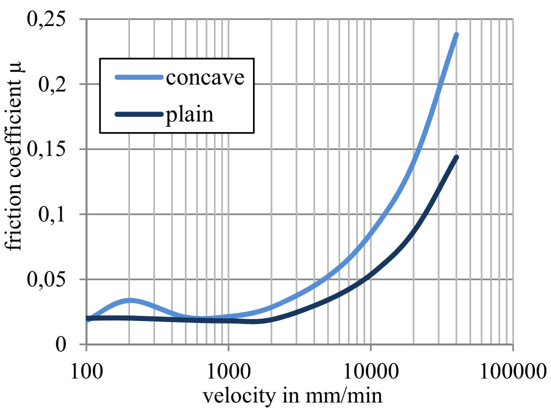

Fig. 2: Friction coefficient for concave and plain wedge geometry

2. Numeric calculation of floating behavior

2.1. Overall calculation method

Laurien presents in [4] a general method for solving simulation tasks, which is used to simulate the floating behavior of hydrodynamic guides.

The first step is to describe the problem to be solved mathematically. Therefore it is supposed that there is an equilibrium of force at the regarded system of the guided slide. Fig. 3 shows the active forces in the system, consisting of the slides weight F_g, friction force F_{fr} with the opposing feed force F_{fe}, hydrodynamic pressure p and acceleration forces. The simulation model is designed for constant feed, so acceleration forces are negligible. A moment concerning point B puts the remaining forces in relation. The test bench design

Nomenclature

F_{fe}	Feed force [N]
F_{fr}	Friction force between slide and rail [N]
F_g	Gewichtskraft des Schlittens [N]
h	Grid point distance [mm]
i	Grid index, parallel to x []
j	Grid index, parallel to z []
m	Index for iteration loop determining α
M_g	Momentum resulting from F_g to reference axis B [N·m]
M_{max}	Abort condition for iteration loop determining α
M_p	Momentum resulting fromp_{wedge} to reference axis B [N·m]
M_{res}	Resulting Momentum from M_g and M_p; auxiliary variable [N·m]
n	Index for iteration loop determining p
p	Hydrodynamic pressure [Pa]
u	feed [m/min]
x	Dimension parallel to feed
x_B	Distance of a discrete areas center to reference axis B [mm]
x_S	Distance of slides center of gravity to reference axis B [mm]
y	System geometry
y_0	Slide geometry = System geometry, for u = 0
z	Dimension parallel to sliding surface
α	Floating angle [°]
Δp_{max}	Abort condition for iteration loop determining p
η	Dynamic viscosity of lubricant [Pa·s]

ensures that the screw-nut-drive and the carriage with slides are in the same plane and linked by a spherical joint. Therefore F_{fe} and F_{fr} are in the same plane. On this plane lies the oil inlet B. Therefore no moment can be generated by F_{fr} and F_{fe}. They are negligible as well. What remains, are F_g and p and their resulting moment to B:

$$0 = M_g - M_p \tag{1}$$

The simulation program needs to find a floating angle α that, given the boundary condition of feed u, the sliding surfaces geometry y_0 and the lubricants dynamic viscosity η, solves (1).

The second step describes the wedge geometry at the start of the simulation program. The slides geometry y_0 describes the system for $u = 0$. Therefore an starting angle α_0 is set to start the simulation with the wedge geometry y

$$y = f(y_0, \alpha) \tag{2}$$

For a system with several guides it is supposed that the mechanical links between them are stiff. Therefore one angle α is valid for the whole simulated system.