Methods for Clutch Dimensioning

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Abstract In early phases of the product development of clutch or brake systems not every component, especially the facing is not available in its finish shape. Therefore the friction behavior is investigated by using a generic shape of the desired facing material. In order to predict the behavior of the friction material in full ring form for the future application simulative and experimental methods are needed. This chapter presents a method to describe the dynamic behavior of a drivetrain with a dry running clutch, whose facing is not available in a full ring form, but in segment or pellet form. The chapter deals on the one hand with the opportunities of the transfer of the tribology behavior from low test level (segment or pellet shape) to high test level on the example of a dry running clutch and on the other hand it is also an objective to connect the results (behavior of the friction coefficient) of the experimental investigations with the simulation model of the future drivetrain to predict the dynamic behavior. In order to achieve the research objective the first step is to identify the influence of the facing shape on the friction coefficient. In several experimental investigations on a clutch test bench, which represents the drive train stiffness and inertia, the influence of the shape form on the friction coefficient was identified. In addition to that the influences on the tribological behavior of the test bench (rotational stiffness, thermal mass, providing of the axial force) are also important and need to be known. Furthermore the mounting of the friction pairs on the test bench has a huge influence on the temperatures during the synchronizations, the wear and the friction coefficient and is also analyzed. Beside the experimental investigations a detailed simulation

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model of the test bench was build up, which considers the mechanical (stiffness, inertia, damping) characteristics and the electric engines of test bench. After having identified the influence of the shape form of the facing on its tribological behavior it is possible to predict the dynamic behavior of a full ring facing on the test bench by using the results of a e.g. segmented facing, the transfer function and the simulation model.

Keywords Dry running clutch · Simulation · Validation

Nomenclature

ω	Angular speed output machine (rad/s)
r_m	Calculated friction radius (mm)
q_A	Specific friction work (J/mm^2)
q	Specific friction power (W/mm^2)
p	Contact pressure (Mpa)
v_0	Sliding speed at the beginning (m/s)
μ	Coefficient of friction
$T_{surface}$	Temperature 0.2 mm under the friction surface (°C)
μ_{mean}	Mean friction coefficient out of 100 cycles

1 Introduction

The dimensioning and calculation of dry and wet clutch systems is still one of the biggest challenges of a power train development. The main function of the clutch system is to transmit the torque of the engine to the powertrain and to interrupt the power flow to enable shifting. Beside that the comfort is also very important. Self-excited or forced-excited shudder causes rotational vibrations in the drive train, which are percept as longitudinal oscillations by the driver. The effect of self-excited shudder on the drive train is especially hard to predict, because there are a lot of interactions and typically the specific components are not available at a certain time in necessary maturity. Due to this the system borders or clutch dimension and investigations are difficult to define. Depending on the system borders a lot of experimental investigations with huge parameter field are necessary to make sure that the clutch is working in the intended drive train. A further challenge is to validate such subsystems or components adequately. In order to achieve this goal new suitable methods must be applied. Actual dimensioning don't take all of these aspect in account, e.g. VDI 2241 [1]. The XiL (see Fig. 1) allows developing and validating virtual and real systems in consideration of the systems driver and environment. This allows taking all relevant aspects for the considered phenomena or maneuvering into account.

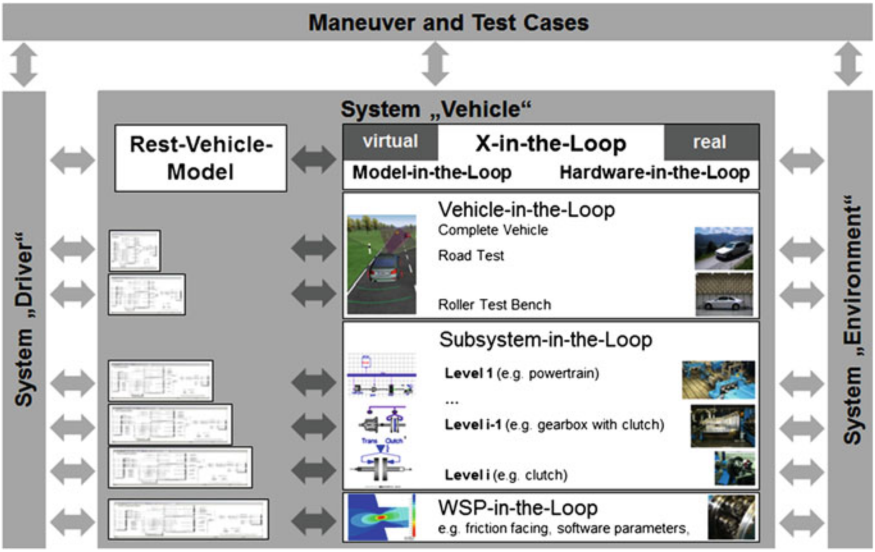

Fig. 1 X-in-the-loop-validation environment [2, 3]

The XiL takes the complete system as well as the environment and the driver into account. "X" represent the unit under test. The unit under test can be a single component, a vehicle of software code. For each unit under test the systems driver an environment will be simulated (e.g. torque oscillations by a dynamic electric engine): During product development usually no complete products but only partitions are available at a certain time. Therefore the validation must be feasible on a partitioned level as well. In the XiL approach this is realized by simulating the required remaining system virtually or physically. At the same time the unit under test might also be only available as a virtual model.

This chapter shows an approach for a clutch dimension method for dry running clutch systems. Experimental and simulation investigations are conducted and connected in appropriate method. The chapter starts with an extract of experimental investigations regarding the geometry (variation of inner and outer diameter) of the friction lining. Beside the experimental methods also simulative test environments were developed. The multi-body simulation model of the intended is used to calculate the dynamic effect of the coefficient of friction (cof) on the drive train. The model represents in addition to the stiffness, the damping, and the inertias also the dynamic of the clutch actuation system. A finite-element model was built up to calculate the temperature and the deformation during a coupling procedure.

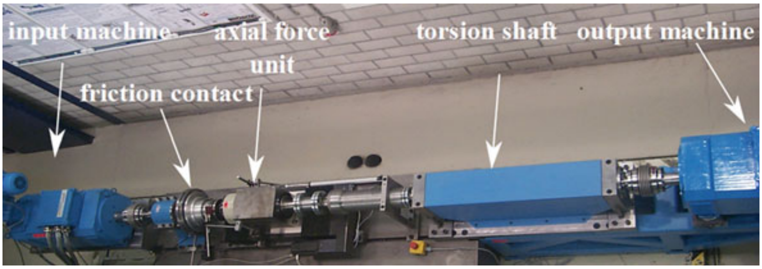

Fig. 2 Dry running clutch test bench of the IPEK

2 Experimental Investigation Methods

The experimental investigations are conducted on the dry running clutch test bench of the IPEK. The test bench consists of two high dynamical electric engines, a torsion shaft and an axial force unit to apply the force on the friction pairings. The input engine represents the power machine. The output machine is the work machine, which simulates the rest system (vehicle). Hence test under realistic conditions are possible. In addition the friction pairing is imbedded in a climate chamber, which defines the ambient temperature and humidity (Figs. 2, 3).

The axial force unit actuates the clutch and applies the contact pressure on the friction contact. During the test the speed of the engines, the friction torque and axial force close to the test probe are measured. In addition the speed of the test probe is measured by a laser, to identify and evaluate the torsional oscillations caused by a negative friction gradient (Fig. 4).

In order to identify the geometric influence three pairings (A, B, C) were investigated. Table 1 shows the geometric dimensions. Pairing A has the largest friction surface and ratio of the inner and outer diameter as opposed to pairing C. The specific values pressure, friction work, friction energy are constant for all variants. The calculated friction radius r_m is also constant. Figure 5 shows the schematic test procedure. The input machine is mechanical blocked. At the beginning of one cycle the test probe is open and the output machine is accelerated to specific speed (blue dashed line). After the speed is reached, the axial force unit applies the contact pressure (red line). The green dotted line shows the temperature behavior close to the friction surface. There are five temperature positions overall which are distributed in radial direction. All temperatures are measured 0.2 mm under the surface.

Figure 6 shows the influence of the contact pressure ($p = 0{,}35$ and $0{,}7$ Mpa) and the sliding speed v_0. The temperature of the surface at the beginning of each cycle was $T_{surface} = 75$ °C. The friction work was the same for all variants and test programs. The sliding speed v_0 was 7 and 14 m/s (in relation to r_m). The friction coefficient is the mean value (μ_{mean}) out of 100 measured cycles.

With the objective to minimize the experimental volume procedure a multiple regression model is built up. The multiple regression analysis allows predicting a

Fig. 3 Test bench with climate chamber

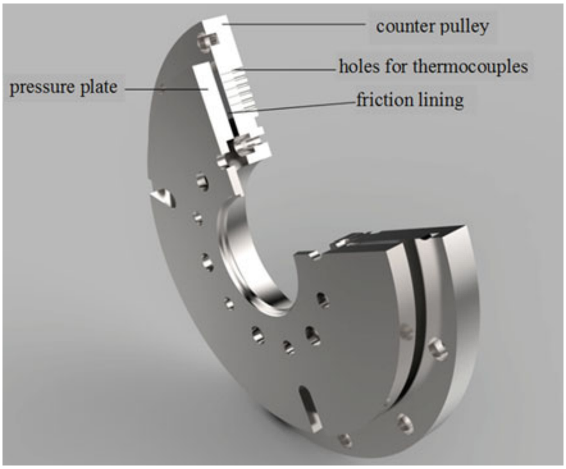

Fig. 4 Test probe with friction lining [4, 5]

target variable y_i in dependence of several variables x_i, which correlates with the target variable. The variables x_i are weighted with the regression coefficients β_i. The regression coefficients are calculated by using the criteria of the robust least square method. Within this method there is an iterative procedure to minimize the effect of outliner on the target variable β_i.

$$y_i = \beta_o + \beta_1 \cdot x_{1i}^m + \beta_2 \cdot x_{2i}^n + \ldots \tag{1}$$

Equation 1: Basic equation for multiple regression [6]

The variables x_i in Eq. 1 are in this chapter the pressure p, the initial velocity v_0 and the product of them (see Eq. 2). In order to consider the non-linear influence an exponent n is introduced for all variables. The equation was modified for this chapter to the following form:

Table 1 Investigated friction pairings (r_m is constant)

Pairing	Shape	Friction surface (mm^2)	da/di (mm)
A		12475,07	188/139,5
B		10304,42	184/144
C		8010,28	180/149

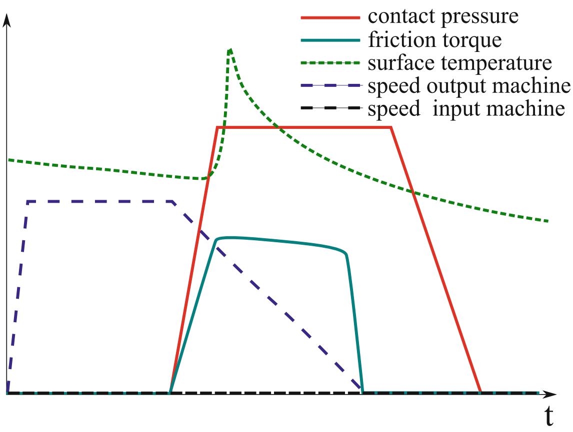

Fig. 5 Schematic progress a cycle (breaking operation)

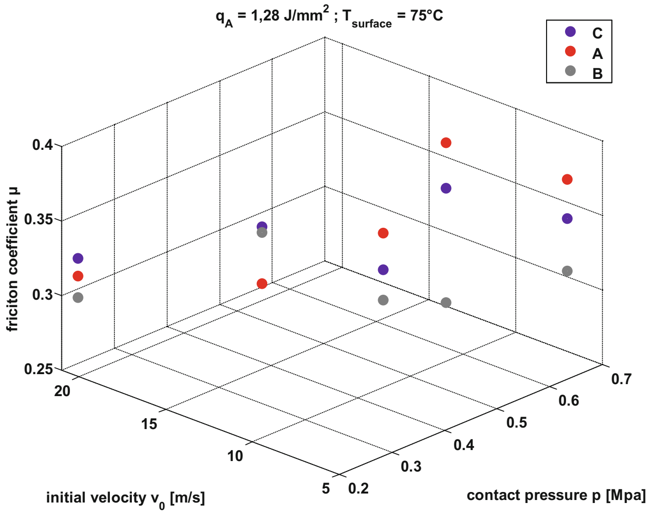

Fig. 6 Experimental results of different friction pairings (μ_{mean} out of 100 cycles)

Table 2 Calculated parameters for each friction pairing

	$ß_0$	$ß_1$	$ß_2$	$ß_3$	n	R^2
A	3.65	−3.04	−2.63	2.43	0.1	0.87226
B	−1.41	−1.4085	−3.0e-015	−6.2e-012	10	0.97949
C	0.36	−0.21	−2.0e-011	−9.3e-010	7	0.99998

$$\mu_i = \beta_o + \beta_1 \cdot p_{1i}^n + \beta_2 \cdot v_{2i}^n + \beta_3 \cdot (p \cdot v)_{3i}^n \tag{2}$$

Equation 2: Approach for multiple regression analysis

After the experimental investigations the regression coefficients and the exponent n for every friction pairing were calculated. Equation 2 shows the parameters β_i, the optimized exponent n and the coefficient of determination (R^2). The closer the coefficient of determination is to the value 1 the better fits the model the measured data. The influence of the pressure p is dominating in all models (high $ß_1$), in contrary to the parameter v_0 (low $ß_2$). The initial speed has only a significant influence in pairing A. It is observable that there is a low influence of v_0 on the pairings B and C. The regression model of pairing C shows the best coefficient of determination (Table 2).

Figure 7 illustrates the regression model for pairing C. The measured and calculated friction coefficients are very similar. Furthermore the model gives an approximation about the level of the friction coefficient. At higher v_0 and contact pressure p the model predicts a significant declining of the friction level. This behaviour can be explained thus the specific friction work increases, which leads usually to a collapse of the friction coefficient of organic linings.

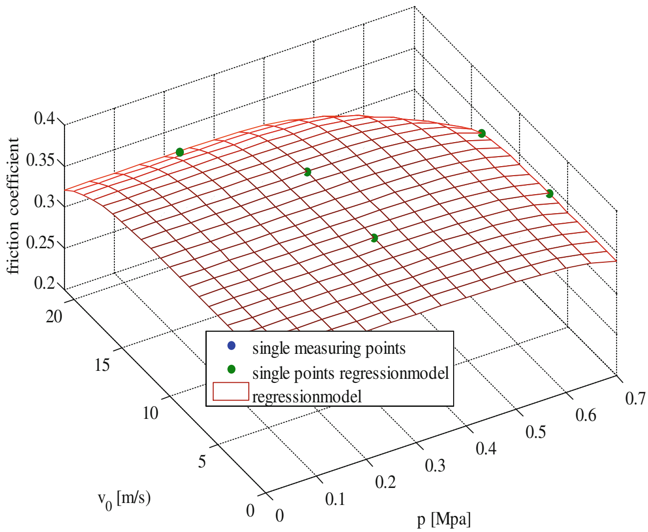

Fig. 7 Measured and calculated friction coefficient of pairing C

The light declining of the cof at higher v_0 and pressure is represented by the model. In the range of $v_0 = 0$ to 15 m/s and $p < 0.6$ Mpa a nearly constant cof is predicted. This means for the possible application that in this range a stable behaviour of the system is probable. It is not only a stable or in certain ranges predictable friction coefficient important for the future application, but also the dynamic behavior of the whole powertrain.

3 Simulative Investigation

The simulative investigation methods have the target to one the one hand to reduce the experimental effort and on the other hand to conduct simulation on a complete powertrain with experimental results (e.g. friction coefficient, friction gradient).

The powertrain is represented by the dry running test bench. Figure 8 shows the developed simulation model. During the model buildup the focus was not only on the inertia, stiffness and damping, but also on the dynamic (transfer function) of the axial force unit and the electric inertia simulation.

The control structure and the controller itself are considered as well. This makes it possible to carry out detailed simulations and to investigate influences on the dynamic of the drive train.

The simulation model in Fig. 8 does not take the temperature development into account. One solution is a FE-Simulation to calculate the temperature on the surface. The comparison of the simulated and measured temperature shows Fig. 9. The high temperatures are simulated at the inner radius ($\sim$75 mm); this fact is

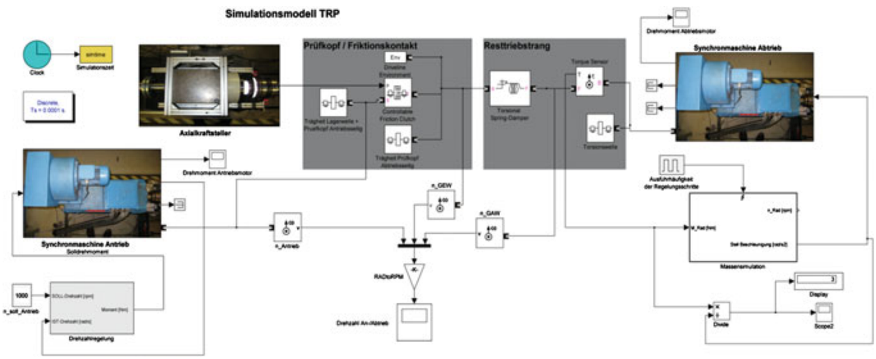

Fig. 8 Simulation model of the dry running test bench

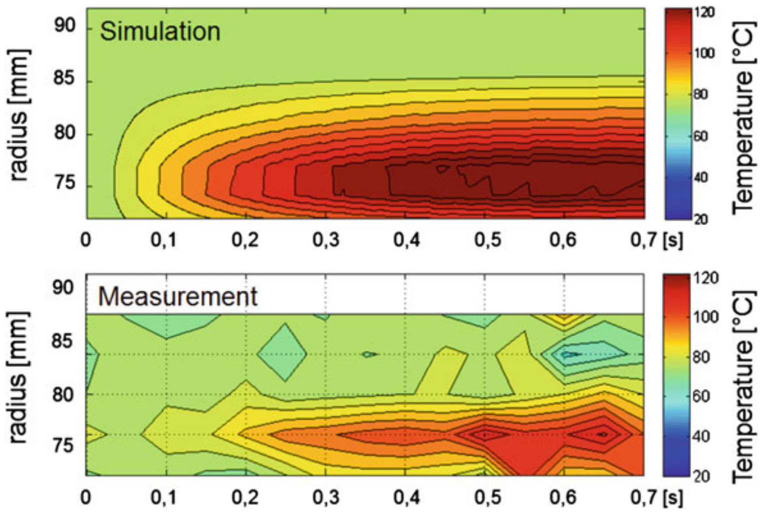

Fig. 9 Simulated and measured temperature distribution during one cycle (see Fig. 5)

confirmed by the measurements (Fig. 9, top). This makes it possible to determine also the local pressure, which is important for the local friction coefficient. Future research works will deal with the task how the procedures and the results of these different domains can be connected in an appropriate way.

4 Summary and Outlook

The present chapter gives a short view on the opportunity's to integrate experimental results in a simulative environment. At first the shape geometry of the friction pairing on the friction coefficient was investigated and a mathematical model was created by using the multiple regression analysis. A detailed simulation model of the dry running test bench (represents the power train), which was used to conduct the experimental test, was built up. Now the experimental results, the regression analyses and the simulative model can used to make simulation to investigate the dynamic behavior of the future powertrain.

In prospective research work the temperature development during the synchronization needs to be considered and to implement into the simulation environment. In addition the local contact pressure is unequally distributed over the friction lining, which results in different local friction coefficients.

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