

Influence of the steel disk on the NVH behavior of industrial wet disk clutches

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Abstract

Purpose – Industrial drivetrains use wet disk clutches for safe and reliable shifting. Advances over the past decades regarding the formulation of lubricants and the composition of friction materials have led to reliable clutch systems. In this context, the friction behavior is crucial for the correct operation of the clutch. Nevertheless, the friction behavior and its influencing factors are still the object of modern research. The purpose of this study is to investigate how the choice of the steel disk influences the noise vibration and harshness (NVH) behavior of wet industrial clutches.

Design/methodology/approach – To investigate the influence of the steel disk on the friction and NVH behavior of industrial wet disk clutches, experimental investigations with relevant friction systems are conducted. These tests are performed at two optimized test rigs, guaranteeing transferable insights. The surface topography of the steel disk and the friction lining are measured for one friction system to identify possible relations between the surface topography and the friction behavior.

Findings – The steel disk can influence the friction behavior of wet disk clutches. Using a different steel disk surface finish, corresponding results can show differences in the shudder tendency, leading to a nonfavorable NVH behavior – different gradients of the coefficient of friction over sliding velocity cause this phenomenon.

Originality/value – This work gives novel insights into the friction and NVH behavior of industrial wet disk clutches. It supports engineers in the optimization of modern friction systems.

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Keywords Boundary friction, Wet clutch, NVH behavior, Sinter friction lining, Steel disk, Surface topography

Paper type Research paper

1. Introduction

Wet disk clutches are critical components of, e.g. modern industrial, maritime and heavy-duty drivetrains. In these applications, friction systems with sinter-metallic friction linings are standard due to their high robustness and their comparatively low production costs. Due to their characteristic friction behavior, these friction systems tend to be more critical regarding noise vibration and harshness (NVH) issues. An increase in the coefficient of friction (CoF) toward lower sliding velocities is the reason for this phenomenon (Watts *et al.*, 1997) and is based on the physics of stick-slip friction. Also, the torsional stiffness and moment of inertia can influence the NVH behavior of drivelines (Tentarelli *et al.*, 2022). In literature, vibrations of friction systems are known as shudder (e.g. Watts *et al.*, 1997), judder (e.g. Chen *et al.*, 2022), chatter (e.g. Albers and Herbst, 1998), squawk (e.g. Ohtani *et al.*, 1994; Lam *et al.*, 2006), squeal (e.g. Dai and Lim, 2008), howl (e.g. Dai and Lim, 2008) or moan/groan (e.g. Dai and Lim, 2008). These vibrations are mostly self-excited but can also be forced externally. For dry automotive brakes, brake noise can be classified based on frequency range and excitation source

(Dai and Lim, 2008). In this classification, judder is classified as forced vibration with frequencies lower than 100 Hz, whereas moan/groan is self-excited with frequencies between 100 and 500 Hz. Vibrations with higher frequencies are howl, squeal and wire brush. In this publication, all these vibrations are called shudder and are generally nonfavorable.

Additives, particularly friction modifiers, can positively influence the shudder tendency (Hopkins and Anderson, 1983). For some common friction modifiers, a linear rise of the CoF over the logarithmic sliding velocity for very low sliding velocities can be observed and can be caused by the sliding of ordered close-packed monolayers (Campen *et al.*, 2012).

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From automotive applications, it is known that contamination of the lubricant with water or iron particles (Wirkner *et al.*, 2023) or the aging of the lubricant (Berglund *et al.*, 2010) can negatively affect friction and NVH behavior of wet clutches. The investigation of influencing factors of self-excited vibrations is still the object of modern research (Yang L.-K. *et al.*, 2016).

The friction material also influences the friction behavior of wet disk clutches. Besides the influence of the type (e.g. organic or sinter-metallic) of the friction material (e.g. Meingassner, 2017), studies concentrate on the identification of specific characteristics of the friction lining. For one sinter-metallic friction system for automotive applications, the influence of the surface topography of the friction material is investigated (Nyman *et al.*, 2006). This study shows that surface characterization with the help of surface parameters, in particular S_{sc} and S_{dq} , allows the prediction of the remaining lifetime of this friction system. Furthermore, it is shown that the friction material and its change over operation time influence the friction behavior and, therefore, the anti-shudder performance (Nyman *et al.*, 2006). Besides the topography, the friction material's permeability is essential for the friction behavior of systems with paper-based (Pauschitz and Franek, 2016) and sinter-metallic (Marklund *et al.*, 2008) friction lining. Investigations of the surface topography and ToF-SIMS for both steel and friction material of automotive wet disk clutches are performed (Zhao *et al.*, 2012; Stockinger *et al.*, 2019), but rather focus on the influence of the lubricant than on the friction surface.

Some studies investigated the influence of the different steel disk variants. Early studies show an influence of the steel surface on the static friction between lapped and polished surfaces (Pokorny, 1960). In this context, differences in the surface roughness of the steel surfaces are considered to influence the different static friction behavior. For sinter-metallic friction material, the hardening and finishing of the steel disk are crucial influences on damaging behavior (Pfleger, 1998). To describe the relationship between surface topography and friction, the parameter $sR_k + sR_{pk}$ (Geier, 2003) is found to be suitable for both steel and friction-lining disks in investigations with wet clutches and synchronizers with metallic and organic friction materials. Besides the investigations on the component level, tribometer tests with automotive paper-based friction systems show a lower shudder tendency for additionally nitrided steel disks, but the roughness of the steel disk does not affect the NVH behavior (Sittig, 2007). Newer studies focus on the influence of the choice of steel disk finishing on the friction behavior of automotive wet disk clutches (e.g. Baese, 2016; Baese *et al.*, 2016; Bukharov *et al.*, 2019) or the influence of the steel disk on the run-in behavior (e.g. Voelkel, 2020; Voelkel *et al.*, 2019). In this context, the steel surface is characterized using the arithmetic average roughness R_a , and the influence of different surface roughness of the steel surface on the friction and wear behavior with different organic friction linings is analyzed (Bukharov *et al.*, 2019).

Besides the friction systems themselves, the influence of the operating conditions on the friction behavior is essential (Strobl *et al.*, 2022). Investigations with friction systems with sinter friction lining (Marklund and Larsson, 2008) show a strong influence of operational parameters like the sliding velocity and the interface temperature obtained from both pin-on-disk tribometers and clutch test rigs. Tests show no load dependence on the friction behavior, which is reasonable due to their direct consideration of the interface temperature (Marklund and Larsson, 2008). Other studies (e.g. Geier, 2003) show a light

influence of the surface pressure on the friction behavior, which could be explained by its indirect influence on the interface temperature (Maeki, 2005). In genuine clutch systems, the interface temperature is hard to measure.

Because the friction behavior cannot yet be simulated, an accurate experimental measurement is required. The application-relevant testing of industrial wet disk clutches is crucial for determining relevant effects on the dynamic friction behavior of the clutch systems. Therefore, component test rigs are preferred. Load or brake shift operation is crucial for most clutches due to the high-energy input during shifting. Industrial clutches show a wide variety of applications in which they are engaged in slip-control mode. Test rig KLP-260 (Strobl *et al.*, 2024; Meingassner *et al.*, 2015) allows the testing of both modes. Previous studies (e.g. Meingassner, 2017; Voelkel *et al.*, 2021) show good transferability of the friction behavior over those operational modes. During the shifting process, the clutch heats up. Higher system temperatures influence the friction behavior at low sliding speeds, which can be reduced by the operation in steady slip mode (Meingassner, 2017).

In particular, the determination of the static and micro-slip friction measurement depends on the operational mode. Studies (Lloyd *et al.*, 1994) show differences between three ways to determine static friction. To determine the friction behavior at very low sliding velocities, a combined method is proposed using a specialized test rig (Voelkel *et al.*, 2021). Contrary to Lloyd, the measurement is not speed-controlled but uses torque as an input.

Previous investigations show an influence of the steel disk on the friction and damaging behavior of wet disk clutches. Most of these studies concentrate on automotive friction systems with organic friction linings. Previous research lacks investigations of the influence of the steel disk variant on friction behavior in a broad range of sliding velocities with different friction systems. The occurrence of shudder or the shudder tendency is generally not discussed.

This investigation analyzes industrial clutch systems' friction and NVH behavior at application-relevant operational modes through experimental studies with three steel disk variants combined with genuine friction linings and lubricants. The investigations show differences regarding the occurrence of shudder depending on the tested steel disk variant. The surface roughness of the steel and friction disks is determined for one friction system to better understand the observed behavior. These measurements show different reductions in the surface roughness of steel and friction disks depending on the underlying friction system. The methods are adapted from investigations with paper-based friction systems (Strobl *et al.*, 2023) for the application-relevant needs of industrial clutches. The publication gives novel insights into the friction of wet clutches from industrial applications with different steel disk variants that consider different operational parameters and modes. Due to the usage of genuine parts, the investigation guarantees a higher application relevance compared to, e.g. tribometer tests. The results give insights regarding the influence of steel disk finishing, which is essential for clutch disk manufacturers for both cost-efficient and functionally relevant surface finishing of the steel disks.

2. Materials and methods

Experimental investigations on well-proven test rigs KLP-260 (e.g. Strobl *et al.*, 2024) and LK-3 (e.g. Voelkel *et al.*, 2021) are

performed to investigate the underlying friction behavior. Among other operating modes, KLP-260 allows the experimental testing of wet disk clutches in brake shift and steady slip mode. These modes are both relevant for industrial clutches and brakes. LK-3 allows the measurement of the friction behavior at the transition from static to dynamic friction (Voelkel *et al.*, 2021). These tests allow the evaluation of the static torque transmitting capacity of the clutch. To characterize the surface of the steel disk and the friction disk, optical measurements of the 3D surface topography by focus variation are performed.

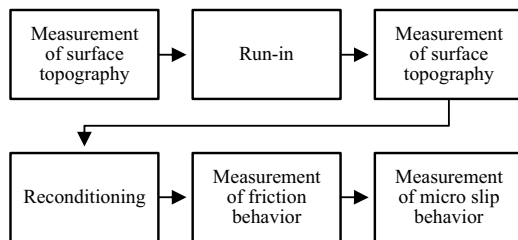
Figure 1 shows the overall test procedure for each clutch pack. The investigations are performed to analyze the influence of the steel disk variant on the friction behavior. The investigated friction materials are described in the following steps before the individual steps of this test procedure are explained in detail.

For the investigations, steel disks with three surface finishes are used [belt-ground (bg), belt-ground and nitrocarburized (bg+nc) and cross-ground (cg)]; see Figure 2.

The steel disks (see one in Figure 3) are paired with two types of genuine sinter-metallic friction disk materials, MS-A and MS-B, with waffle groove patterns. Two exemplary friction disks are shown in Figure 4. The mean diameter is $\varnothing d_m = 162$ mm.

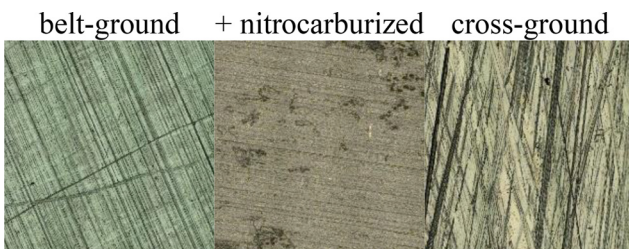
Six friction surfaces ($z = 6$) are considered for the investigations. Therefore, four friction and three steel disks are mounted in corresponding carriers. For the calculation of the specific surface pressure p , the nominal friction area A is used, which allows an application-relevant evaluation of the torque transmitting capacity in the design space. The technical data of the clutch disks is given in Table 1.

Figure 1 Test procedure



Source: Created by authors

Figure 2 Investigated steel disk surfaces before run-in ($750 \times 750 \mu\text{m}$)



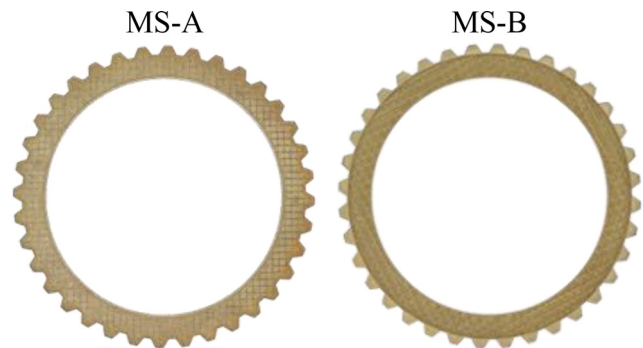
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Figure 3 Steel disk ($\varnothing d_m = 162$ mm)



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Figure 4 Friction disks ($\varnothing d_m = 162$ mm)



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The investigations were performed with two lubricants. L-301 is a typical industrial lubricant, and L-302 is a bio-degradable lubricant used in maritime applications. Both lubricants contain different additive components of phosphorus ($L-301 < L-302$) and sulfur ($L-301 \gg L-302$). The technical data of the lubricants are given in Table 2.

Three friction systems are tested as combinations of the friction material and lubricant. The tested systems are MS-A, L-301; MS-B, L-301; MS-A and L-302. Although parts and lubricants are gathered from genuine systems, the friction systems in their combinations were not optimized for a specific application.

The dynamic investigations are performed on the component test rig KLP-260. For this, the clutch disks are mounted in adapted carriers. The outer carrier is standing still (brake

Table 1 Technical data of the clutch disks

Parameter	Unit	Value
Outer friction diameter d_a	mm	176
Inner friction diameter d_i	mm	147
Mean radius r_m	mm	81
Nominal friction area A	mm ²	7,357

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Table 2 Technical data of the lubricants

Parameter	Unit	L-301	L-302
Density ρ at 20 °C	kg·m ⁻³	891	930
Kin. viscosity ν at 40 °C	mm ² ·s ⁻¹	99.3	101.2
Kin. viscosity ν at 100 °C	mm ² ·s ⁻¹	11.0	16.4

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operation). The inner carrier is connected with the driven shaft and flywheels, which allow the adjustment of the shifting inertia $\mathcal{J}$. Depending on the operational mode, the shaft is driven by a high-torque drive (forced slip operation) or a high-speed drive (brake shift operation). In forced slip operation, the speed n is measured via an incremental encoder. In brake shift operation, the speed is measured via a tachometer. This allows the operation in the most relevant ranges of sliding velocities. The friction torque T_f is measured via a load cell connected to the rotatably mounted outer carrier. The clutch is actuated by a force-controlled hydraulic piston that measures the axial force F_a . To determine the specific surface pressure p , the axial force F_a is divided by the nominal friction area A . With the measured information, the CoF is determined according to equation (1):

$$CoF = \frac{T_f}{p \cdot A \cdot r_m \cdot z} \quad (1)$$

The test rig's measurement accuracy is described in previous studies (Baumgartner, 2020; Groetsch *et al.*, 2021). As a result, the measurement of the CoF has an uncertainty of around $\pm 1.3\%$, and the measurement of the speed of around $\pm 0.2\%$ (forced slip) or $\pm 0.9\%$ (brake shift) at a confidence level of 95% (Baumgartner, 2020; Groetsch *et al.*, 2021). Further information on the experimental setup and the working principle of KLP-260 can be found in previous publications (e.g. Meingassner *et al.*, 2015; Strobl *et al.*, 2024).

The clutches undergo a run-in under harsh test conditions. The run-in procedure was chosen according to previous studies (Meingassner, 2017) and adapted to higher specific surface pressures. During run-in, the clutch is lubricated evenly with oil from the inside and the top with a specific oil flow rate $\dot{v}_{oil} = 0.8 \text{ mm}^3 \cdot \text{mm}^{-2} \cdot \text{s}^{-1}$ at an oil inlet temperature $\vartheta_{oil} = 80^\circ\text{C}$ and a connected inertia of around $\mathcal{J} = 1.1 \text{ kg} \cdot \text{m}^2$. The cycle time is set to $t_c = 15 \text{ s}$ with a closed phase of $t_{cl} = 5 \text{ s}$. In brake shift operation, the clutch is accelerated to a defined differential speed (or maximum sliding velocity $v_{s, \max}$). Then, the drive is disconnected from the inner shaft before the clutch is actuated and decelerated to a standstill. The run-in procedure contains five load stages (R1–R5) with increasing successively run loads. An overview of the run-in load stages is shown in Table 3.

Before and after the run-in, the surface topography of the middle steel disk and one inner friction disk are measured with the Alicona Infinite Focus G4. The measurement settings for the steel disk surface are chosen according to Table 4. This paper shows only topography measurements of friction system MS-A, L-302 due to a necessary change of measurement settings during the measurements of the other friction systems.

Due to the high roughness of the sinter-metallic friction material, the measurement settings for these surfaces were

Table 3 Run-in load stages

Name	$p / \text{N} \cdot \text{mm}^{-2}$	$v_{s, \max} / \text{m} \cdot \text{s}^{-1}$	Cycles
R1	0.5	5	100
R2	0.5	10	100
R3	1.0	10	100
R4	1.0	15	200
R5	2.0	15	2,500

Source: Created by authors

Table 4 Settings for the measurement of surface topography of the steel disk

Parameter	Setting
Magnification	50 x
Vertical resolution	50 nm
Lateral resolution	2 μm
Measuring field	$0.75 \times 0.75 \text{ mm}$
L-filter	250 μm
Polarization	Inactive

Source: Created by authors

adopted. Here, the L-filter is chosen to be large compared to the size of the measuring field, leading to a consideration of the surface as a whole. The measurement of a more extensive measuring field was not possible due to the waffle groove pattern and the demand for repositioning measurements. The measurement settings are shown in Table 5.

Due to a necessary cleaning process of the friction and steel disks for the optical surface measurements, the friction disks undergo 30 cycles according to the loads defined by run-in load stage R5 before testing continues.

Then, the clutches are tested in the main investigation at KLP-260 to measure the friction behavior at brake shift and steady low-speed slip operation. The tests are grouped in three blocks of oil inlet temperature ($\vartheta_{oil} = 80^\circ\text{C} - 110^\circ\text{C} - 40^\circ\text{C}$). During the dynamic tests, the clutch is lubricated evenly with oil from the inside and top with a specific oil flow rate $\dot{v}_{oil} = 0.8 \text{ mm}^3 \cdot \text{mm}^{-2} \cdot \text{s}^{-1}$ at an oil inlet temperature $\vartheta_{oil} = 80^\circ\text{C}$ and a connected inertia of around $\mathcal{J} = 1.1 \text{ kg} \cdot \text{m}^2$ (in brake shift mode). In contrast to the run-in, the cycle time in the main investigations is increased to $t_C = 30 \text{ s}$ to reduce the influence of preceding shifts. The brake shift load stages are run one after the other, according to Table 6.

Table 5 Settings for the measurement of surface topography of the friction disk

Parameter	Setting
Magnification	10 x
Vertical resolution	200 nm
Lateral resolution	4 μm
Measuring field	$2.5 \times 2.5 \text{ mm}$
L-filter	8,000 μm
Polarization	Active (90°)

Source: Created by authors

Table 6 Brake shift load stages

Name	$p / \text{N}\cdot\text{mm}^{-2}$	$v_{s, \max} / \text{m}\cdot\text{s}^{-1}$	Cycles
B1	0.5	10	10
B2	1.0	10	10
B3	2.0	10	10

Source: Created by authors

After each brake shift block, the clutches cyclically undergo 18 load stages (see Table 7) of steady slip in order of ascending friction energy (calculated with constant CoF). In steady slip operation, the clutch is actuated first before the drive accelerates the inner shaft to a maximum sliding velocity $v_{s, \max}$. This differential speed is kept constant for $t_s = 30$ s, and the steady slip phase CoF is determined.

After the dynamic investigations, the clutch is mounted in the LK-3 test rig for static and micro-slip friction tests. The test rig operates in brake operation with a fixed outer carrier. At first, the clutch is actuated with the desired axial force F_a or specific surface pressure p . After that, the inner shaft, connected to a lever with the length l , is loaded with a defined weight to apply torque on the clutch. If the static friction coefficient (SFC) is exceeded, the clutch creeps, and an incremental encoder measures the change in the angle of the lever ε , allowing the evaluation of the sliding velocity v_s the clutch. On the contrary, the weight m is converted into a friction coefficient UFC according to equation (2) (Voelkel et al., 2021):

$$UFC = \frac{m \cdot g \cdot l \cdot \cos(\varepsilon)}{p \cdot A \cdot r_m \cdot z} \quad (2)$$

Further information on the experimental setup and the working principle of LK-3 can be found in previous publications

Table 7 Steady slip load stages

Name	$p / \text{N}\cdot\text{mm}^{-2}$	$v_{s, \max} / \text{m}\cdot\text{s}^{-1}$	Cycles
S1	0.5	0.005	10
S2	0.5	0.01	10
S3	1.0	0.005	10
S4	1.0	0.01	10
S5	2.0	0.005	10
S6	0.5	0.025	10
S7	2.0	0.01	10
S8	0.5	0.05	10
S9	1.0	0.025	10
S10	1.0	0.05	10
S11	0.5	0.1	10
S12	2.0	0.025	10
S13	2.0	0.05	10
S14	0.5	0.2	10
S15	1.0	0.1	10
S16	1.0	0.2	10
S17	2.0	0.1	10
S18	2.0	0.2	10

Source: Created by authors

(Meingassner et al., 2016; Strobl et al., 2023; Voelkel et al., 2021).

In LK-3, the clutch is lubricated from the top at the desired oil inlet temperature ϑ_{oil} . Due to the high manual effort for the investigations, only oil inlet temperatures $\vartheta_{oil} = 110^\circ\text{C}$ and 40°C are tested sequentially. Inside these blocks, two specific surface pressures, $p = 2 \text{ N}\cdot\text{mm}^{-2}$ and $0.5 \text{ N}\cdot\text{mm}^{-2}$, are tested sequentially. Before each test, the clutch is subjected to an axial force of 1,000 N and forced to slide several times within an angular range of at least $\pm 15^\circ$ for a suitable preconditioning of the friction surfaces just before the test.

Every measurement consists of one creep rate at the tested value of the UFC . The measurements are repeated with different values of the UFC until no relevant creep rate can be measured within around 10 to 15 min measuring time. In this case, the SFC is defined. All measurements showing a creep rate > 0 are used for a curve fitting of type $f(v_s) = a \cdot x^b + c$ with $(0 < b < 1)$, according to (Voelkel et al., 2021).

The measurement uncertainty of test rig LK-3 is described in detail in the literature (Meingassner, 2017; Hieber, 2022). For the shown measurements, the expanded relative uncertainty for the UFC is between 1.1 and 2.0% (confidence level 95%) (Hieber, 2022).

After the measurements, the relation between the steady CoF and surface topography parameters is analyzed. Therefore, the median of the steady CoF of cycles six to ten is determined for each clutch at every load stage. The CoF is compared to the median from the 16 (steel disk) or 8 (friction disk) individual measurements of 21 3D surface topography parameters (see Table 8) after run-in. The strength of the linear relation is measured via the coefficient of determination R^2 for every load stage separately. This approach is based on the methodology in unsteady slip operation published in previous publications (Strobl et al., 2023).

3. Results and discussion

The friction behavior is shown for three operational modes of the clutch after run-in. First, the friction behavior at brake shift operation, then the friction behavior at steady slip, and then the behavior at the transition from static to dynamic friction is described and discussed. After that, the change of the friction surfaces is presented. Finally, the relation between surface topography and the CoF at steady slip is discussed for one friction system.

3.1 Brake shift operation

Brake shifting is a prevalent operational mode covering both high sliding velocities at the beginning and very low sliding velocities down to a standstill at the end of the shifting. Due to

Table 8 Investigated surface parameters

Sa	Sq	Sp	Sv	Sz	$S10z$	Ssk
Sku	Sdq	Sdr	Sk	Spk	Svk	$Sk + Spk$
$Smr1$	$Smr2$	Vmp	Vmc	Vvc	Vvv	Vvc/Vmc

Source: Created by authors

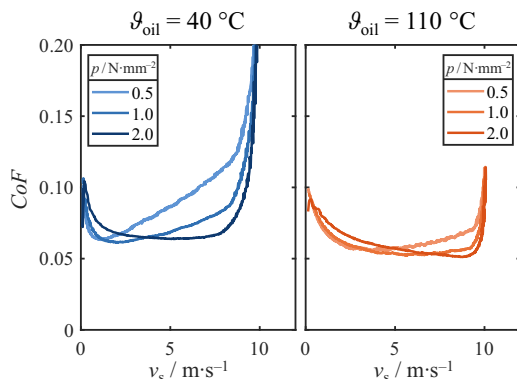
the great importance of this mode for the application, the investigation of the friction behavior in this operational mode is widespread in clutch testing.

First, the influence of the operational parameters is discussed. Figure 5 exemplarily shows the course of the CoF over the sliding velocity for brake shifts of friction system MS-B, L-301, cg for three distinctive specific surface pressures p and two oil inlet temperatures ϑ_{oil} . All friction curves show high CoF values at high sliding velocities during the build-up of the axial force. After that, the friction behavior shows differences in the gradient of the CoF over sliding velocity depending on the specific surface pressure and the oil inlet temperature. All friction curves show a steep rise of the CoF toward lower sliding velocities. This so-called “rooster tail” is typical for the friction behavior of sinter-metallic friction systems. The rooster tail is particularly strong for low oil inlet temperatures and signals a high shudder tendency. Conversely, the rooster tail is less steep for higher oil inlet temperatures. A reason for this could be the higher activity of the surface-active additives at higher surface temperatures.

To identify differences in the NVH behavior of friction systems in combination with different steel disk variants, Figure 6 shows the friction behavior of two friction systems (MS-B, L-301 and MS-A, L-302) with different steel disks for operational parameters with high shudder tendency and intensity ($\vartheta_{oil} = 40^\circ\text{C}$; $p = 2 \text{ N}\cdot\text{mm}^{-2}$). The investigations with belt-ground and cross-ground steel disks were repeated with another clutch using the same manufacturing process once. In contrast, the test with the additional nitrocarburized variant was not repeated.

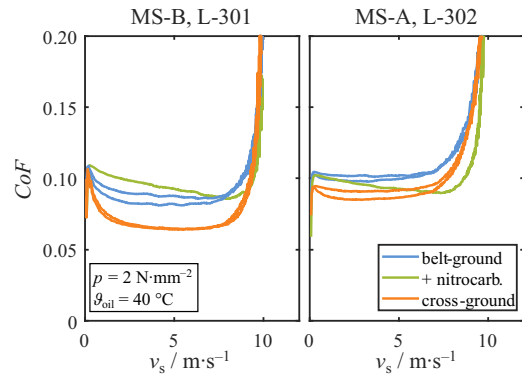
For the friction system MS-B, L-301 cross-ground steel disks lead to the most dominant rooster tail, showing the highest tendency to shudder. The dynamic CoF is comparatively low compared to the CoF at slow sliding velocities. This leads to the assumption that hydrodynamic influences are possibly present. These influences are typical if, e.g. the porosity of the friction material is low. On the other side, the CoF at low sliding velocities at the peak of the rooster tail is approximately equal for all steel disk variants. For the friction system MS-A, L-302, the influence of the steel disk on the friction and NVH behavior

Figure 5 CoF over sliding velocity v_s of brake shifts under variation of the specific surface pressure for two oil inlet temperatures (MS-B, L-301, cg)



Source: Created by authors

Figure 6 CoF over sliding velocity v_s of brake shift with different steel disks at $\vartheta_{oil} = 40^\circ\text{C}$ and $p = 2 \text{ N}\cdot\text{mm}^{-2}$ (bg; bg+nc; cg) for the friction system MS-B, L-301 (left) and MS-A, L-302 (right)



Source: Created by authors

is different. Here, cross-ground and belt-ground variants show similar characteristic behavior, but the CoF level is still lower for the cross-ground variants. In both friction systems, the additionally nitrocarburized variants show a different friction behavior with an equal rise of the CoF toward lower sliding velocities.

For the friction system MS-A, L-301, the influence of the steel disk on the friction behavior at brake shift operation was negligible, which emphasizes a strong dependency on the friction system regarding the underlying mechanisms and corresponding influences on the friction behavior.

Therefore, the influence of the steel disk variant on the friction behavior in brake shift operation should be considered separately for the friction systems.

3.2 Steady slip operation

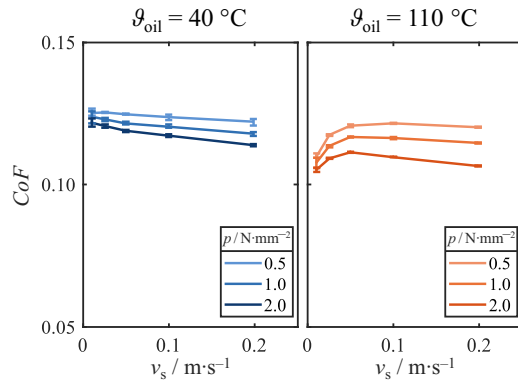
The friction behavior at low sliding velocities in brake shift operation is strongly influenced by the heat introduced by the brake shift itself. Therefore, the friction behavior at low sliding velocities is tested separately in steady slip operation. Here, the clutch is tested under forced, steady-state conditions, allowing the CoF to be evaluated at a defined sliding velocity. The median CoF values for the investigated operating points are connected with lines in the following evaluations to better understand the friction behavior.

First, the influence of the oil inlet temperature ϑ_{oil} and the specific surface pressure p on the CoF at low sliding velocities is depicted in Figure 7.

For low oil temperatures, the slope of CoF over sliding velocity remains negative until the lowest investigated sliding velocities are reached. On the contrary, the CoF decreases toward lower sliding velocities for higher oil inlet temperatures ϑ_{oil} after a stable or even increasing friction behavior at higher sliding velocities v_s .

The decrease in the CoF toward lower sliding velocities at higher oil inlet temperatures indicates the presence of corresponding friction modifier additives. In the context of the NVH behavior of the clutch system, this decrease mitigates the significant increase in the CoF toward low

Figure 7 CoF over sliding velocity v_s at steady slip for different surface pressures p at $\vartheta_{oil} = 40^\circ\text{C}$ of friction system MS-B, L-301 with steel disk variant cg



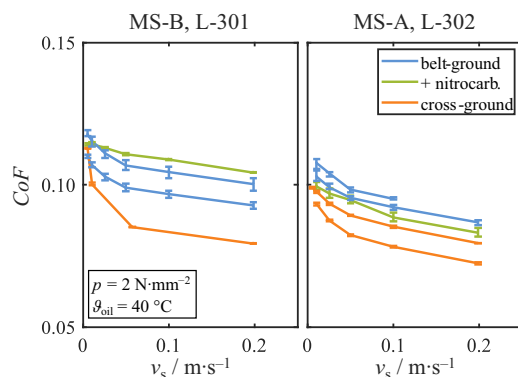
Source: Created by authors

sliding velocities and is therefore considered advantageous. Lower oil inlet temperatures lead to a higher tendency to shudder, which shows a strong connection between the system's temperature and its tendency to shudder. Because of this, the following analyses concentrate on the friction behavior at an oil injection temperature of $\vartheta_{oil} = 40^\circ\text{C}$. Due to the test rig's stiff construction, shudder typically does not occur. The higher the friction torque, the higher the possibility of the occurrence of shudder, which is why the subsequent evaluations are presented at the highest surface pressure in the test.

The influence of the steel disk finishing on the friction is shown in Figure 8. Here, the conclusions of the brake shift operation can be confirmed.

For friction system MS-A, L-302, belt-ground steel disks lead to a slightly higher level of CoF . All variants show similar friction behavior concerning the occurring gradients. Nevertheless, missing points (e.g. MS-A, L-302, bg, $v_s = 0.2\text{ m}\cdot\text{s}^{-1}$) are caused by the notable occurrence of shudder leading to problems concerning the evaluation of the steady

Figure 8 CoF over sliding velocity v_s at steady slip operation with different steel disks at $\vartheta_{oil} = 40^\circ\text{C}$ and $p = 2\text{ N}\cdot\text{mm}^{-2}$ (bg; bg+nc; cg) for the friction system MS-B, L-301 (left) and MS-A, L-302 (right)



Source: Created by authors

CoF of corresponding operating points. An additional nitrocarburization leads to slightly lower values of the CoF .

For friction system MS-B, L-301, the cross-ground steel disks show a steep rise of the CoF for lower sliding velocities. This leads to a vital occurrence of shudder compared to the other friction systems. Therefore, not all operating conditions could be evaluated, and, e.g. the repetition of the cross-ground variant was not evaluated. In this friction system, the additional nitrocarburization led to higher CoF values and a less pronounced rise of the CoF over sliding velocity.

Like in brake shift operation, the friction behavior of friction system MS-A, L-301 is less sensitive to different steel disk variants. Here, the friction behavior shows a lower shudder tendency than the shown friction systems at a comparatively high CoF level.

Shudder occurrence behaves differently for the tested sliding velocities in friction system MS-B, L-301. Figure 9 shows the friction behavior over time at the up-ramp and in the first seconds of steady slip operation at five different sliding velocities in the last collective run with a specific surface pressure $p = 2\text{ N}\cdot\text{mm}^{-2}$ and an oil inlet temperature $\vartheta_{oil} = 40^\circ\text{C}$ for the friction system MS-B, L-301 with cross-ground and with belt-ground steel disks.

For the sliding velocity $v_s = 0.005\text{ m}\cdot\text{s}^{-1}$, no shudder occurs for both steel disk variants. Only minor changes in the CoF during the up-ramp are visible. The level of the CoF is approximately equivalent for both variants. For the sliding velocity $v_s = 0.01\text{ m}\cdot\text{s}^{-1}$, strong shudder occurs in combination with the cross-ground variant but not with the belt-ground variant.

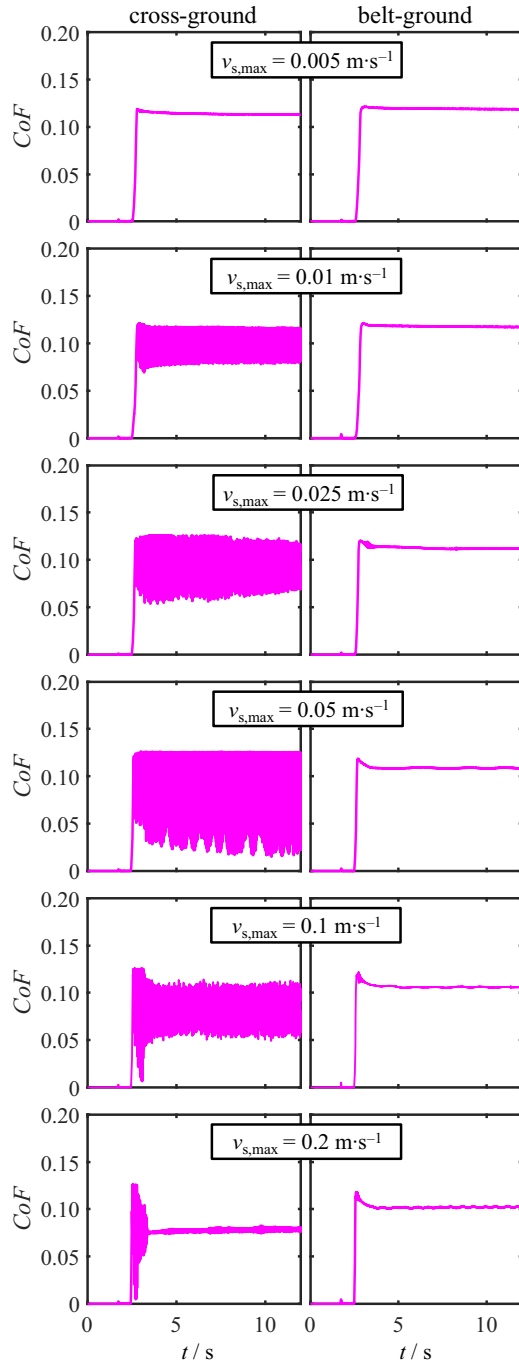
For sliding velocity $v_s = 0.025\text{ m}\cdot\text{s}^{-1}$, the vibration's amplitude with the cross-ground variant is increasing. It reaches the load cell's limits, meaning higher torques cannot be determined. This leads to a straight cutoff visible during the CoF and should also be considered for the upcoming sliding velocities. On the contrary, the belt-ground variant shows light shudder with low intensity at the up-ramp. Shudder occurs also in the seconds that are not shown here. The intensity of the occurring shudder gets lower with higher sliding velocities in the steady phase for the cross-ground steel disks. However, shudder in the up-ramp remains strong, reaching the limits of the load cell. The combination with belt-ground steel disks does not show shudder in the steady phase for higher sliding velocities and only light shudder in the up-ramp. Nevertheless, both variants show lower CoF values for higher sliding velocities, as shown in Figure 8.

Like in brake shift operation, the influence of the steel disk variant on the CoF in steady slip operation should be separately considered for different friction systems. Here, the systems showed differences in the influence of the steel disk on friction and NVH behavior dependent on the underlying friction system. Nevertheless, system MS-B, L-301 shows a dependency between its shudder tendency and the steel disk.

3.3 Operation at the transition from static to sliding friction

At the transition from static to sliding friction, the friction behavior shows a characteristic rise of the CoF over the sliding velocities, which is typical for sinter-metallic (Voelkel et al., 2021; Meingassner, 2017) and organic friction materials

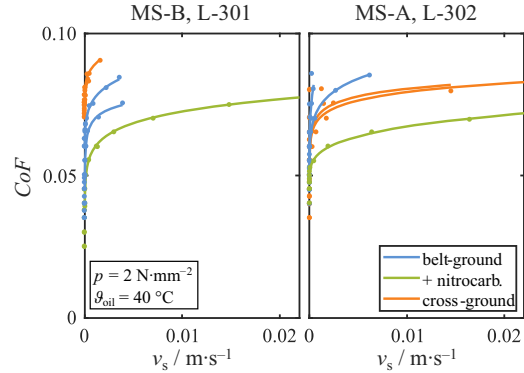
Figure 9 Occurrence of shudder for different sliding velocities v_s with steel disk variants cg (left) and bg (right) with friction system MS-B, L-301



Source: Created by authors

(Meingassner, 2017; Voelkel *et al.*, 2021; Strobl *et al.*, 2023). All *SFC* values are significantly lower than the *CoF* values at low sliding velocities in steady slip operation. This signals a steep increase of the *CoF* in microslip. The micro-slip behavior of the steel disk variants for two friction systems is shown in Figure 10.

Figure 10 *CoF* over sliding velocity v_s at micro-slip with different steel disks at $\vartheta_{oil} = 40^\circ\text{C}$ and $p = 2 \text{ N}\cdot\text{mm}^2$ (bg; bg+nc; cg) for the friction system MS-B, L-301 (left) and MS-A, L-302 (right)



Source: Created by authors

For friction system MS-B, L-301, the system with cross-ground steel disks shows the highest *CoF* level in micro-slip. The lowest *CoF* level was determined for the system with nitrocarburized steel disks. This influence of the nitrocarburization could also be confirmed for friction system MS-A, L-302. Here, the combinations with belt-ground steel disks show the highest *CoF* level in micro-slip.

The friction system MS-A, L-301 shows microslip behavior similar to that of MS-B, L-301. Cross-ground steel disks also lead to the highest *CoF* in microslip operation. In addition, nitrocarburized steel disks show the lowest values of the *CoF* in microslip operation.

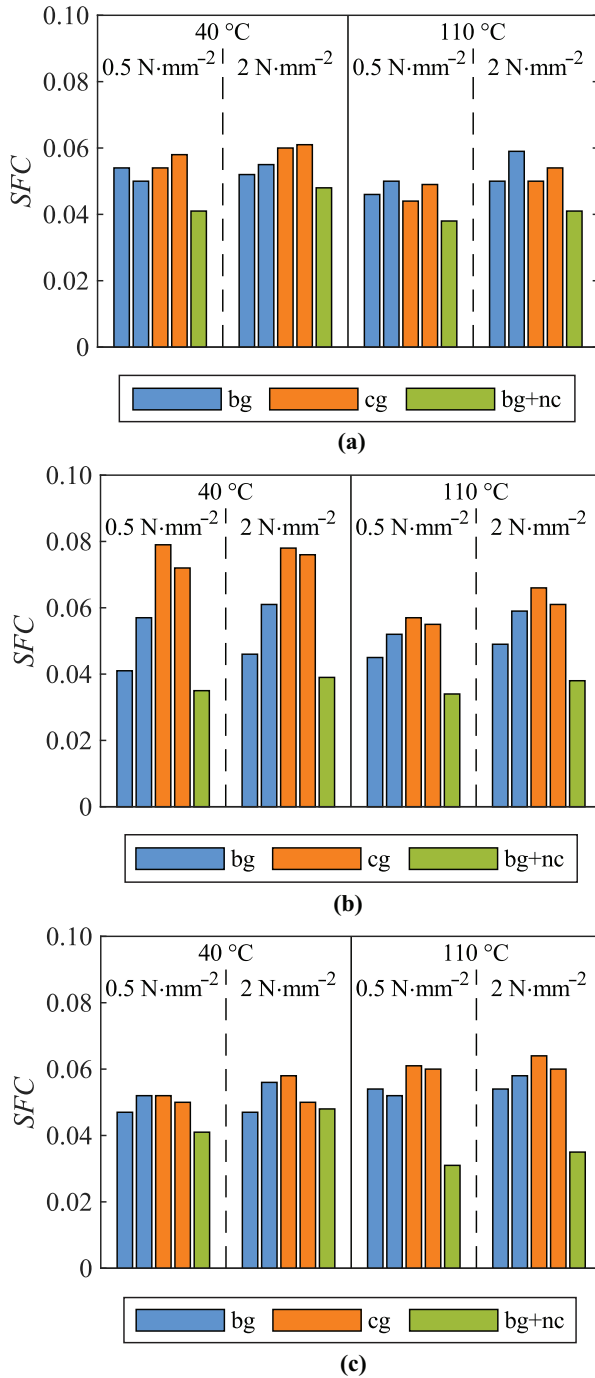
Besides the microslip behavior, these investigations also allow the determination of the *SFC* defined at the transition from static to sliding friction. The corresponding results for all friction systems and all tested operational parameters are shown in Figure 11.

For all investigated variants and operational parameters, friction systems, in combination with additional nitrocarburized steel disks, show the lowest *SFC* values. For many operational parameters, friction systems, in combination with cross-ground steel disks, show the highest values of *SFC*. Nevertheless, the highest values of *SFC* are recorded in combination with friction system MS-B, L-301, cg, which is the friction system with the strongest occurrence of shudder. Here, the differences in the *SFC* could be relevant, although the *SFC* is lower than the maximum of the *CoF* in steady slip operation.

The results of the microslip operation show a slight difference from the previous evaluations. The nitrocarburized steel disks show the lowest *SFC* values, significantly reducing those variants' static torque transferability. On the other side, the cross-ground steel disks show the highest values of the *SFC* for most operating points. Higher values of the *SFC* could imply a deterioration concerning the NVH behavior, although the values are significantly lower than *CoF* values in steady slip operation.

3.4 Investigation of the surface topography

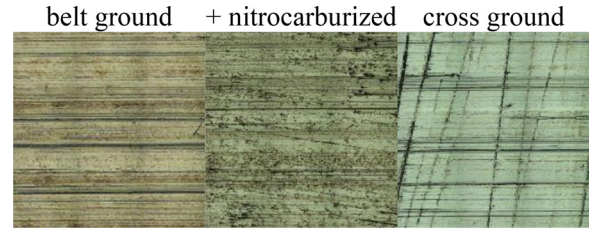
After discussing the friction behavior, the surface topography of the steel and friction disks is discussed.

Figure 11 SFC for all friction systems at all tested operating conditions (ϑ_{oil} and p) with different steel disks (bg; bg+nc; cg)

Notes: (a) MS-A, L-301; (b) MS-B, L-301; (c) MS-A, L-302

Source: Created by authors

According to Figure 12, the steel disks show significant smoothing due to the run-in procedure compared to the new surfaces shown in Figure 2. After the run-in, all variants showed a directional surface structure in the sliding direction (horizontally). In the case of the belt-ground surface, almost no recognizable structure of the initial surface texture is left. The

Figure 12 Investigated steel disk surfaces after run-in (750 × 750 μm; MS-A, L-302)

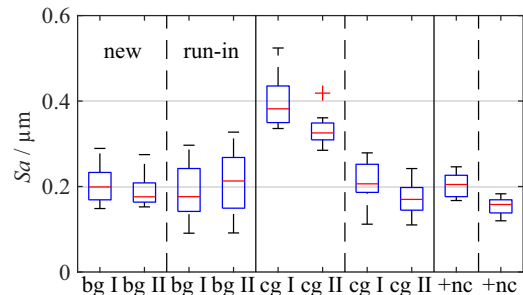
Source: Created by authors

initial surface structure is still identifiable after run-in for additional nitrocarburized steel surfaces. In contrast, the cross-ground surfaces only show deep surface characteristics from the initial texture.

Nevertheless, the surface of the disks of the combinations of friction system MS-A, L-302 was investigated using multiple surface topography parameters. Figure 13 shows the change of the arithmetical mean height S_a from the new to the run-in state of the steel disks. Belt-ground steel disks do not show any change in the median arithmetical mean height, but the scattering of this value increases after run-in. Cross-ground steel disks show an intense smoothing concerning the roughness parameter. On the other hand, the nitrocarburized steel surface shows only a slight change in its surface roughness.

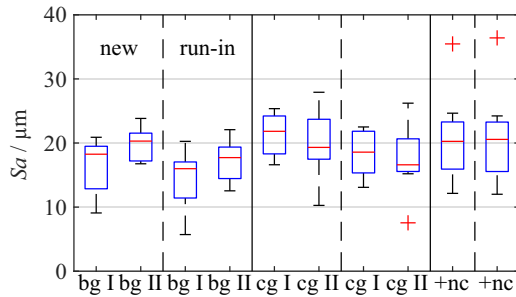
Figure 14 shows the change of the arithmetical mean height S_a from the new to run-in state of the friction lining with different paired steel disks. The friction lining, in combination with belt-ground and cross-ground steel disks, only shows a slight smoothing. In combination with nitrocarburized steel disks, the arithmetical mean height remains almost constant, indicating no significant smoothing of the surface.

Also, other surface topography parameters indicate differences in the smoothing of the surface depending on the steel disk variant. Therefore, the change in the surface topography of steel and friction lining depends on the steel disk used in the clutch system. Random samples of surfaces from the other friction systems confirm this and lead to the assumption that the choice of the steel disk is relevant not

Figure 13 Comparison of the 3D arithmetical mean height S_a of the steel disk in the new state and after run-in for friction system MS-A, L-302

Source: Created by authors

Figure 14 Comparison of the 3D arithmetical mean height S_a of the friction lining in new state and after run-in for friction system MS-A, L-302 paired with different steel disks



Source: Created by authors

only for changes in the friction and NVH behavior, as shown before, but also for the smoothing of the friction lining surfaces.

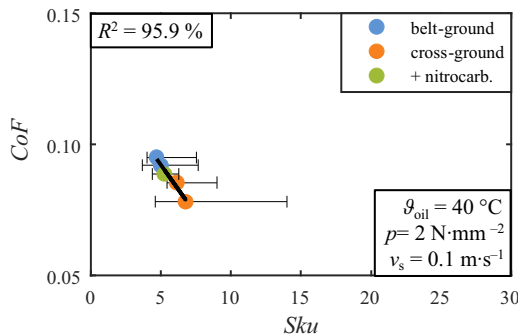
Nevertheless, systematic investigations with other friction systems should be obtained to support this hypothesis. Here, different results could be expected when differently formulated lubricants or very different friction materials are used. A separate investigation on only one component (e.g. the steel disk) could provide misleading assumptions.

3.5 Relation between the surface topography and the friction behavior

To find a relation between the surface topography and the friction behavior, linear correlations for the steady CoF of each load stage (five sliding velocities v_s , three specific surface pressures p and three oil inlet temperatures ϑ_{oil}) and 21 surface topography parameters were analyzed for both, steel and friction lining surface. Due to changes in the steel disk's surface measurement procedure, only measurements for the friction system MS-A, L-302 are discussed.

Figure 15 exemplarily shows one of 1890 (945 for the steel's surface and 945 for the friction lining surface) investigated relations between the surface topography and the CoF under steady slip conditions. This example shows a

Figure 15 Exemplarily relation of the steady CoF and the surface topography parameter Sku of the steel disk for one load stage



Source: Created by authors

combination with a high coefficient of determination R^2 . Due to the high number of investigated relations, only the most critical findings are mentioned subsequently. The coefficient of determination for each linear correlation between the CoF and a specific surface parameter can be found in the Appendix for the steel disk's surface (Figure A1) and the friction lining's surface (Figure A2).

For the friction system MS-A, L-302, the following findings can be obtained for the relation between the steel surface topography and the CoF in steady slip:

- No characteristic surface value of the steel disk shows a good correlation for all operational parameters.
- Some characteristic values show a different behavior for higher/lower sliding velocities and some for higher/lower oil inlet temperatures.
- The value Sku of the steel disk shows a good correlation for most of the operational parameters – but the relation gets weaker for higher oil inlet temperatures in combination with lower sliding velocities.
- The values of Sdq and Sdr of the steel disk show a good correlation for operational parameters with higher oil inlet temperatures. However, the relation becomes weaker for higher sliding velocities.

For the friction system MS-A, L-302, the following findings can be obtained for the relation of the friction lining surface topography and the CoF in steady slip:

- No characteristic surface value of the friction lining disk shows a good correlation for all operational parameters.
- Some characteristic values show a different behavior for higher/lower sliding velocities and some for higher/lower oil inlet temperatures.
- The valley-describing parameters, especially Sv , but also Vvv and Svk show good correlation for many operating conditions. Only conditions with very low sliding velocities do not show any relation – especially for higher oil inlet temperatures.
- The peak-describing parameters Sz , $S10z$, and slightly Sp and Spk also show good relation for similar conditions described above.

Nevertheless, random sample tests with the other friction systems show different surface topography parameters that could be relevant for describing the investigated relation to friction. Thus, some observations can be confirmed, like the different behavior of the relations for different operational conditions. This leads to the assumption that different underlying chemical and/or physical fundamentals should be assumed for different load stages.

In general, good drainage and large surface contact areas may be beneficial for high CoF values in boundary and mixed friction. The valley describing parameters may correlate with this effect. This leads to the hypothesis that hydrodynamic effects are relevant in this friction system. In this case, the peak describing parameters could describe the contact area under these circumstances. On the contrary, the kurtosis of the steel disk could indicate the ability of the steel surface to modify the friction lining surface. Nevertheless, this theoretical discussion of possible hypotheses should consider different friction

systems and be only a first step for further work in this area of research.

The findings of friction system MS-A, L-302 could also show that different surface topography parameters, combinations of them or even different approaches to describe the surface should be considered to optimize the overall reliability of an application-relevant description of the functional behavior. Additional evaluations on the dependency of the gradient of the friction behavior from the surface topography could be performed to investigate the NVH behavior. In addition, artificial intelligence could also be used to evaluate the relation between surface topography and the friction and NVH behavior of industrial wet disk clutches due to an additional clustering of the measured data to categorize operating conditions and their underlying chemical and/or physical fundamentals.

4. Summary

In this study, the steel disks in three industrial-relevant wet clutch friction systems are systematically varied. Low oil inlet temperature and specific surface pressure lead to a higher shudder tendency for all variants. The influence on the choice of the steel disk is stronger or weaker depending on the underlying friction system. Nevertheless, for one friction system, the occurrence of shudder could be mitigated using a different steel disk.

The investigations were performed under different operating conditions. In brake shift and steady slip operation, the influences of the steel disk variant were similar regarding friction and NVH behavior. Although both operating conditions showed similar *CoF* levels for the different steel disks at very low sliding speeds, microslip friction behavior depends on the steel disk used. For all variants, nitrocarburized variants showed a lower torque transfer capability than the variants without nitrocarburization. All variants show lower *SFC* values than the *CoF* level in steady slip operation.

The smoothing of the steel and friction disk surface due to the run-in depends on the steel disk variant. Initially, rough steel disks with cross-ground surface finishing show an intense smoothing. Nevertheless, the measurements should be systematically performed for multiple friction systems to confirm the findings.

The relation between the surface topography and the *CoF* in steady slip operation is investigated for one friction system. Potential surface parameters for describing the application-relevant relation between surface and friction were identified. Nevertheless, these parameters are not valid for all operating conditions and should be confirmed with other friction systems. Random samples with other friction systems question the usage of only one determined surface topography parameter. Further research should be performed to identify application-relevant combinations of surface parameters.

5. Conclusions

The choice of the steel disk can affect the NVH behavior of wet disk clutches. In this context, shudder could be mitigated using different steel disks. The steel friction surface and the friction lining surface are analyzed to characterize potential surface topography influences.

The investigation of the friction behavior should include a wide range of relevant operation modes. This study shows differences in the observed friction behavior in different operating modes. This also allows a more detailed evaluation of friction behavior.

In the design process of wet disk clutches, the choice of the steel disk typically has low priority. In the case of the unwanted occurrence of shudder, this publication shows that not only changing the friction material or the lubricant can optimize the shudder tendency, but also the choice of steel disk variant.

This study supports a better understanding of the influence of the surface finishing and hardening of steel disks on the friction behavior of industrial wet disk clutches. Thus, the choice of surface finishing should not only be decided regarding the frictional behavior but should also consider, e.g. the wear and damaging behavior. A combined investigation of these factors for future investigations is proposed.

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Appendix

Tables 9 and 10 show the coefficient of determination R^2 for each linear correlation between the surface topography of the steel surface or friction lining and the CoF for different operating conditions. The color coding indicates correlations:

dark green: $R^2 \geq 80\%$

light green: $80\% > R^2 \geq 60\%$

yellow: $60\% > R^2 \geq 40\%$

Empty cells indicate operating points with strong shudder where it was not possible to evaluate a stable value for the steady CoF .

Table A1 Coefficient of determination R^2 of the linear regression for every investigated combination of surface parameters of the steel surface and the CoF (high values of R^2 highlighted)

p		$v_s / m \cdot s^{-1}$		$\vartheta_{oil} = 40^\circ C$																$R^2 / \%$	
0.01	5	32	33	36	16	0	11	64	64	66	37	19	0	0	2	8	7	40	0	31	
	0.025	8	0	0	80	73	29	70	99	10	12	32	0	40	17	49	3	10	37	38	
	0.05	8	0	0	78	70	25	62	98	12	14	28	0	34	18	40	3	5	10	32	
	0.1	6	2	1	70	62	18	55	97	16	18	28	0	26	14	34	2	4	13	26	
	0.2	2	9	9	57	44	7	34	88	26	29	24	1	11	9	17	0	2	18	12	
0.01	0	14	11	54	38	4	33	88	37	40	35	5	8	4	17	1	0	28	7	10	
	0.025	10	0	0	79	74	29	69	99	9	11	29	1	40	19	48	4	6	9	38	
	0.05	11	0	2	80	77	34	78	98	7	8	32	1	47	19	58	5	6	8	44	
	0.1	5	1	1	76	65	21	58	98	17	19	32	0	28	14	37	1	3	14	27	
	0.2	4	3	2	71	59	16	52	97	19	22	31	0	23	12	31	1	2	16	22	
0.01	2	8	9	58	45	8	33	87	25	27	22	1	11	10	16	0	2	16	12	9	
	0.025	7	2	2	72	61	18	48	94	16	18	23	0	24	17	27	2	5	11	24	
	0.05	10	0	0	83	75	31	67	98	10	11	29	1	39	20	45	4	5	8	37	
	0.1	5	1	1	79	64	21	53	96	18	20	29	0	26	16	31	2	2	12	24	
	0.2																				
$\vartheta_{oil} = 80^\circ C$																					
0.01	90	71	52	12	36	60	21	1	73	70	27	88	59	66	20	81	93	57	69	22	
	0.025	21	58	35	12	3	8	7	52	78	81	57	44	2	9	3	30	18	73	3	
	0.05	3	34	25	29	17	0	17	72	54	57	40	17	0	7	8	2	45	0	4	
	0.1	1	27	22	35	22	0	20	76	48	51	35	12	1	0	8	4	1	38	1	
	0.2	1	24	18	41	26	1	24	80	46	49	38	10	3	1	11	3	1	36	3	
0.01	86	68	33	4	17	44	3	1	81	79	57	93	33	66	2	87	92	81	44	3	
	0.025	10	40	19	21	11	1	20	69	61	65	61	29	0	3	13	19	8	64	0	
	0.05	1	22	12	36	27	1	30	83	41	44	42	10	4	0	18	4	0	39	4	
	0.1	0	16	8	44	34	3	37	89	36	39	43	7	8	1	23	2	0	35	7	
	0.2	0	12	7	52	40	5	41	92	33	36	40	5	11	3	25	1	0	30	10	
0.01	45	47	9	1	0	13	11	29	67	68	93	65	2	37	12	59	46	94	5	11	
	0.025	2	19	4	29	25	0	42	82	38	41	62	14	7	0	33	8	2	51	5	
	0.05	1	6	0	46	45	8	58	92	18	21	43	2	21	3	45	0	1	27	19	
	0.1	1	5	0	49	47	9	59	94	19	21	43	2	22	4	44	0	1	26	20	
	0.2	2	3	0	55	53	12	63	96	17	19	41	1	26	6	47	0	2	23	24	
$\vartheta_{oil} = 110^\circ C$																					
0.01	52	91	87	1	11	46	11	5	88	88	20	68	44	30	17	53	42	58	46	22	
	0.025	34	79	65	2	0	24	0	26	89	91	37	57	17	17	2	41	28	69	18	
	0.05	19	60	50	13	2	8	2	45	82	85	42	41	5	6	0	26	18	61	6	
	0.1	8	38	36	30	12	1	9	61	69	72	40	24	0	0	2	11	9	46	0	
	0.2	1	22	24	48	25	1	17	73	53	56	35	11	1	1	5	3	3	32	1	
0.01	48	89	86	0	8	41	9	7	89	89	21	65	40	26	15	50	41	58	41	20	
	0.025	40	78	63	3	1	23	0	25	96	97	44	63	18	20	2	46	38	72	21	
	0.05	21	54	38	16	4	6	6	52	84	86	57	43	2	7	2	28	22	69	4	
	0.1	3	24	20	45	24	1	22	77	57	60	47	16	2	0	9	6	5	41	1	
	0.2	0	9	9	67	43	8	33	85	37	40	37	4	10	6	16	0	0	23	8	
0.01	45	88	82	0	5	37	5	11	91	91	25	64	34	24	11	49	38	61	36	15	
	0.025	29	67	54	8	0	13	1	37	91	92	47	52	9	12	0	35	28	69	11	
	0.05	8	31	16	33	18	0	25	75	93	66	65	25	1	1	15	14	10	59	0	
	0.1	0	12	7	57	39	5	40	90	42	44	52	8	10	2	24	2	1	35	8	
	0.2	3	1	0	76	63	20	61	98	19	22	41	0	29	10	41	0	1	18	25	
Sa Sq Sp Sv Sz S10z Ssk Sku Sda Sdr Sk Ssk Ssk Ssmr1 Ssmr2 Vmp Vmc Vvc V																					

Table A2 Coefficient of determination R^2 of the linear regression for every investigated combination of surface parameters of the friction lining and the CoF (high values of R^2 highlighted)

$p \quad v_s / \text{m} \cdot \text{s}^{-1}$		$\vartheta_{\text{oil}} = 40^\circ \text{C}$																				$R^2 / \%$	
0.01	40	50	60	92	88	90	44	57	10	11	34	63	82	28	6	26	41	34	62	52	46		
	2	7	53	72	68	72	0	3	10	10	7	75	42	40	2	28	2	0	44	22	17		
	0.025	6	12	63	78	74	79	1	6	7	6	13	83	52	49	1	36	4	2	54	28	25	
	0.1	8	17	71	84	81	85	2	10	4	3	16	88	60	55	1	38	6	3	63	30	29	
	0.2	21	34	84	93	91	95	11	25	0	0	30	93	79	62	0	46	17	12	79	41	44	
0.05	20	32	76	97	94	97	16	29	0	0	25	84	77	49	0	35	18	13	72	39	39		
	3	8	57	72	69	73	0	3	11	10	9	79	44	45	2	32	2	0	48	23	19		
	0.025	0	3	47	65	61	65	0	1	17	16	4	70	34	37	4	24	0	0	38	16	12	
	0.1	7	15	62	82	77	82	3	9	4	4	14	81	55	45	0	34	6	3	54	31	26	
	0.2	10	19	70	86	83	87	4	13	2	2	17	86	62	51	0	37	8	5	62	33	30	
0.1	22	36	85	92	90	94	11	25	0	0	32	94	80	64	0	49	18	14	79	44	47		
	13	23	74	84	81	86	4	13	2	2	23	92	66	58	0	46	10	7	66	38	37		
	0.025	4	9	54	72	67	72	0	4	9	8	10	77	45	42	1	33	3	1	45	26	21	
	0.1	11	18	60	81	76	81	4	12	2	2	18	82	58	43	0	37	9	6	53	38	31	
	0.2																						
		$\vartheta_{\text{oil}} = 80^\circ \text{C}$																				$R^2 / \%$	
0.01	15	11	1	6	5	4	63	47	48	50	1	4	4	16	5	15	25	22	0	1	0		
	21	33	56	88	90	86	39	52	8	9	14	45	66	23	0	6	22	16	62	19	21		
	0.025	24	39	79	99	100	99	26	43	3	3	24	73	81	47	0	24	22	16	81	31	36	
	0.1	26	41	82	100	99	100	24	41	2	3	28	80	84	51	0	31	23	17	82	36	41	
	0.2	23	37	79	100	99	100	22	37	1	2	26	80	81	49	0	30	21	16	79	36	39	
0.05	5	4	0	15	15	12	44	34	21	23	1	1	5	11	0	19	11	8	1	0	0		
	11	22	63	93	95	92	21	34	1	1	9	55	62	31	3	8	11	7	66	15	17		
	0.025	14	27	80	98	99	98	13	28	0	0	17	78	73	52	1	24	12	7	79	24	28	
	0.1	11	23	77	97	96	97	10	23	0	0	15	80	69	51	2	25	10	6	74	24	27	
	0.2	12	23	76	96	94	96	9	21	0	0	16	82	69	50	1	28	10	6	72	28	29	
0.1	0	1	11	44	46	40	13	15	0	0	3	6	13	0	16	9	0	0	14	0	1		
	0.025	2	8	61	86	89	86	5	13	3	2	2	58	48	36	12	8	1	0	60	7	9	
	0.1	1	7	68	82	83	83	0	6	9	8	4	72	47	50	12	18	0	0	62	9	11	
	0.2	1	8	68	83	83	84	1	6	9	8	5	74	48	50	10	19	1	0	62	11	13	
	0.2	1	7	66	81	81	82	0	5	10	9	5	75	47	49	9	21	1	0	59	12	13	
		$\vartheta_{\text{oil}} = 110^\circ \text{C}$																				$R^2 / \%$	
0.01	51	57	25	41	43	39	86	90	58	60	27	11	51	4	10	2	57	51	35	23	28		
	40	51	47	71	73	69	65	78	29	31	25	31	68	16	2	6	43	35	57	25	30		
	0.025	40	51	56	86	86	84	56	70	19	21	28	47	77	22	3	13	42	34	63	35	37	
	0.1	40	50	60	92	89	90	46	60	12	13	32	60	81	27	5	22	41	33	63	47	43	
	0.2	36	45	62	92	87	91	34	47	6	6	34	70	80	32	6	31	36	29	61	54	47	
0.05	53	59	27	45	46	43	87	91	56	59	29	14	54	5	11	3	59	53	37	26	30		
	39	46	36	67	67	63	70	77	31	33	21	24	60	8	4	3	44	36	45	25	25		
	0.025	28	37	48	86	85	83	47	58	11	13	17	43	67	16	1	7	31	23	54	27	25	
	0.1	26	35	58	94	89	92	30	41	3	4	23	65	72	27	2	22	27	20	58	42	35	
	0.2	23	30	55	86	80	85	18	28	0	1	24	72	66	30	3	31	22	17	51	48	37	
0.1	51	58	32	52	53	49	83	89	50	52	29	17	59	7	9	4	56	49	42	26	31		
	38	47	45	78	77	75	62	72	23	25	23	35	68	14	3	7	42	34	52	30	30		
	0.025	14	22	52	93	91	90	23	34	1	1	10	54	60	22	0	9	15	9	54	22	19	
	0.1	12	20	56	91	87	89	13	23	0	0	13	68	60	29	0	19	12	8	53	30	24	
	0.2	5	10	51	78	73	78	2	8	5	5	9	71	47	32	1	24	4	2	44	26	19	
		$\vartheta_{\text{oil}} = 140^\circ \text{C}$																				$R^2 / \%$	
0.01	51	58	32	52	53	49	83	89	50	52	29	17	59	7	9	4	56	49	42	26	31		
	38	47	45	78	77	75	62	72	23	25	23	35	68	14	3	7	42	34	52	30	30		
	0.025	14	22	52	93	91	90	23	34	1	1	10	54	60	22	0	9	15	9	54	22	19	
	0.1	12	20	56	91	87	89	13	23	0	0	13	68	60	29	0	19	12	8	53	30	24	
	0.2	5	10	51	78	73	78	2	8	5	5	9	71	47	32	1	24	4	2	44	26	19	
		$\vartheta_{\text{oil}} = 170^\circ \text{C}$																				$R^2 / \%$	
0.01	51	58	32	52	53	49	83	89	50	52	29	17	59	7	9	4	56	49	42	26	31		
	38	47	45	78	77	75	62	72	23	25	23	35	68	14	3	7	42	34	52	30	30		
	0.025	14	22	52	93	91	90	23	34	1	1	10	54	60	22	0	9	15	9	54	22	19	
	0.1	12	20	56	91	87	89	13	23	0	0	13	68	60	29	0	19	12	8	53	30	24	
	0.2	5	10	51	78	73	78	2	8	5	5	9	71	47	32	1	24	4	2	44	26	19	
		$\vartheta_{\text{oil}} = 200^\circ \text{C}$																				$R^2 / \%$	
0.01	51	58	32	52	53	49	83	89	50	52	29	17	59	7	9	4	56	49	42	26	31		
	38	47	45	78	77	75	62	72	23	25	23	35	68	14	3	7	42	34	52	30	30		
	0.025	14	22	52	93	91	90	23	34	1	1	10	54	60	22	0	9	15	9	54	22	19	
	0.1	12	20	56	91	87	89	13	23	0	0	13	68	60	29	0	19	12	8	53	30	24	
	0.2	5	10	51	78	73	78	2	8	5	5	9	71	47	32	1	24	4	2	44	26	19	
		$\vartheta_{\text{oil}} = 230^\circ \text{C}$																				$R^2 / \%$	
0.01	51	58	32	52	53	49	83	89	50	52	29	17	59	7	9	4	56	49	42	26	31		
	38	47	45	78	77	75	62	72	23	25	23	35	68	14	3	7	42	34	52	30	30		
	0.025	14	22	52	93	91	90	23	34	1	1	10	54	60	22	0	9	15	9	54	22	19	
	0.1	12	20	56	91	87	89	13	23	0	0	13	68	60	29	0	19	12	8	53	30	24	
	0.2	5	10	51	78	73	78	2	8	5	5	9	71	47	32	1	24	4	2	44	26	19	
		$\vartheta_{\text{oil}} = 260^\circ \text{C}$																				$R^2 / \%$	
0.01	51	58	32	52	53	49	83	89	50	52	29	17	59	7	9	4	56	49	42	26	31		
	38	47	45	78	77	75	62	72	23	25	23	35	68	14	3	7	42	34	52	30	30		
	0.025	14	22	52	93	91	90	23	34	1	1	10	54	60	22	0	9	15	9	54	22	19	
	0.1	12	20	56	91	87	89	13	23	0	0	13	68	60	29	0	19	12	8	53	30	24	
	0.2	5	10	51	78	73	78	2	8	5	5	9	71	47	32	1	24	4	2	44	26	19	
		$\vartheta_{\text{oil}} = 290^\circ \text{C}$																				$R^2 / \%$	
0.01	51	58	32	52	53	49	83	89	50	52	29	17	59	7	9	4	56	49	42	26	31		
	38	47	45	78	77	75	62	72	23	25	23	35	68	14	3	7	42	34	52	30	30		
	0.025	14	22	52	93	91	90	23	34	1	1	10	54	60	22	0	9	15	9	54	22	19	
	0.1	12	20	56	91	87	89	13	23	0	0	13	68	60	29	0	19	12	8	53	30	24	
	0.2	5	10	51	78	73	78	2	8	5	5	9	71	47	32	1	24	4	2	44	26	19	
		$\vartheta_{\text{oil}} = 320^\circ \text{C}$																				$R^2 / \%$	
0.01	51	58	32	52	53	49	83	89	50	52	29	17	59	7	9	4	56	49	42	26	31		
	38	47	45	78	77	75	62	72	23	25	23	35	68	14	3	7	42	34	52	30			