Heat loading of hoist brakes by example of drum brakes

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Abstract. Due to the technological development in drive technology, drives controlled by frequency inverters in hoists of cranes are almost standard. Since these drives offer the possibility of electric braking, the operation of the mechanical brakes changes as a result. As a result, the mechanical brakes are used more rarely and, if so, more likely in critical operating conditions. In this paper, an analysis of the changes that occur in the structure under the influence of thermal load is presented.

1. Introduction
Technological development in the field of drive technology does not stop only on crane systems. Asynchronous motors controlled by means of frequency inverters, for example, offer possibilities for control, which has so far been unavailable for the vehicle fleet. Although this technology has not yet completely penetrated the market, corresponding drives are already widely used where sensibility, particular speed in the partial load range and energy efficiency are required.

A special feature of these drives is that electrical braking is possible in addition to the electric drive. Thus loads can be moved with hoists without the need to remove energy from the hoist system via a mechanical brake. As a result, an operating brake, usually located on the axis of the motor shaft, is modified to the holding brake. Instead of regular braking, the brake is used to hold an already broken load. The work done in this case is small or even equal to zero. The brake must work only if the electric braking system fails.

2. Consequences
The described altered operation of the brakes results in various consequences for the planning, operation and testing of corresponding systems.

If the brake is no longer operated in normal crane operation, the friction partners are no longer exposed to frictional contact. Corrosion can occur in the friction surfaces. Furthermore, components of the atmosphere such as moisture, oil and dust can precipitate. When using the brake, at least a significant change in the friction conditions and therefore the friction coefficient should be expected. Against this background, a design and/or operation of the brake is recommended, which regularly ensures the desired and necessary frictional conditions.

If a brake is not used or hardly used in normal operation, it is unclear in the course of time, whether the full performance of the brake is still present. Availability of full performance can not necessarily be derived from the presence of partial performance. For this reason, procedures for regularly
checking the performance of the brake must be defined and be executed.

The changed operating mode also affects the thermal load of the brake.

In case of a service brake, braking operations with a specific frequency are to be considered. Braking must occur with a sufficiently small period that the brake could not cool completely from braking to the braking. In this case, the steady-state peak temperature which is established after a certain time has to be proofed.

A holding brake acting as an emergency stop brake, on the other hand, rarely has to provide braking with maximum braking energy. After braking, the brake can cool down and the next braking process starts again from the fully cooled state. In this respect, a holding brake is to be designed for the maximum temperature occurring after a single braking. The peak temperature after a singular braking (functionality as a holding brake) is lower than the stationary peak temperature during a pulsed braking (functionality as a service brake) of one and the same lifting mechanism.

3. Reference system

Basis of further investigation is the reference hoist described in [1]. Considered is the brake located on the axis of the motor shaft, described as service brake or holding brake, depending on the operation mode. A drum brake according to DIN 15435 with a drum diameter of $d_T=630\,\text{mm}$ (Fig. 1) is used for this brake.

| Table 1. Parameters of the drum brake on the basis of manufacturers’ information |
|-----------------------------------|-------------------------------|
| Geometry drum                     | Geometry lining               |
| Drum diameter $d_T=630\,\text{mm}$ | Lining thickness $t_B=12\,\text{mm}$ |
| Drum thickness 1 $t_T=30\,\text{mm}$ | Lining width $b_B=225\,\text{mm}$ |
| Drum width $b_T=236\,\text{mm}$ | Lining length $l_B=400\,\text{mm}$ |
| Drum thickness 2 $t_S=30\,\text{mm}$ | Lining angle $\alpha_B=70^\circ$ |
| Drum surface $A_T=0.467\,\text{m}^2$ | Lining surface (total) $A_{B,\text{ges}}=0.18\,\text{m}^2$ |

| Material properties drum           | Material properties lining    |
| Density drum $\rho_T=7850\,\text{kg/m}^3$ | Density lining $\rho_B=2000\,\text{kg/m}^3$ |
| Heat conductivity drum $\lambda_T=50\,\text{W/(mK)}$ | Heat conductivity lining $\lambda_B=0.9\,\text{W/(mK)}$ |
| Heat capacity drum $c_T=465\,\text{J/(kgK)}$ | Heat capacity lining $c_B=1000\,\text{J/(kgK)}$ |

Figure 1. Drum brake according to DIN 15435 [2]
The rated braking torque of the brake corresponds to the static torque of the load \( M_s = 2901.8 \) Nm and the service brake ratio \( BF_{BB} = 2.0 \) set to \( M_{BB} = 5803.6 \) Nm. With the assumption of a coefficient of friction \( \mu = 0.4 \), the linings are pressed onto the drum with a surface pressure of:

\[
p = \frac{2 \cdot M_{BB}}{A_{\text{ges}} \cdot \mu \cdot d_T} = 0.256
\]

A braking process with an initial drum speed of \( n_1 = 2000.9 \) min\(^{-1}\) is considered. This is the speed occurring in full-load-lowering-operation after switching off the motor and the brake dead time of \( \Delta t_{BB} = 0.4 \) s. The braking takes place over a braking time of \( t_B = 1.6 \) s. During this time, the drum speed decreases linearly down to zero due to the constant deceleration.

During braking, this results in a maximum specific and maximum absolute friction performance at the beginning of the braking process [2].

The maximum frictional speed is:

\[
v_{\text{max}} = \omega \cdot \frac{d_T}{2} = 2 \cdot \pi \cdot n_1 \cdot \frac{d_T}{2} = 66.0
\]

The maximum specific frictional performance is:

\[
P_{\text{spec., max}} = p \cdot \mu \cdot v_{\text{max}} = 6.8 \cdot 10^6
\]

The maximum absolute frictional performance is:

\[
P_{\text{max}} = P_{\text{spec., max}} \cdot A_{\text{ges}} = 1216.6
\]

With this the mean braking power of \( P_{\text{medium}} = 608.3 \) kW is clearly above the nominal motor power of \( P_{\text{nom}} = 456.2 \) kW.

Qualitative analysis of structures of cast samples showed that there was no significant difference in the amount of eutectoid at different concentrations of the modifier [3]. By means of the metallographic studies it has been found that the change in the eutectoid content is not so significant as it was during the influence of different cooling rates on the same bronze. The maximal difference (a decrease from 15 % to 10 %) was detected in case of the sample, which contained 0.75 % of the modifier. At that, the samples with 0.75 and 1.5 % addition of the modifier possessed such vastly branching morphology of eutectoid that it did not allow determining a sphericity coefficient and an average size for them.

For the calculation, the drum-lining-contact was modeled as a three-dimensional FEM model under the described conditions. The transient thermal behavior of the model was determined during the simulation [4]. Due to the short-term nature of the processes, an adiabatic system was used. Since in reality, a heat outflow is to be expected, this assumption leads to a conservative view, i.e. higher temperatures than in reality [5].

4. Results
For braking in drum and lining, the temperature field shown in Fig. 2 is to be expected after the braking time \( t = t_B/2 = 0.8 \) s. This is the point of time at which the maximum temperatures occur in the lining approximately. In the lining, a maximum temperature of \( T_{B_{\text{max}}} = 577.9 \) K (heating by \( \Delta T_B = 284.8 \) K) occurs at the inlet edge of the drum. The zone affected by this temperature level is very thin. The drum experiences a maximum temperature of \( T_{T_{\text{max}}} = 352.5 \) K (heating by \( \Delta T_T = 59.3 \) K) on its surface.
Fig. 3 shows the time dependent temperatures in lining and drum in points close to the zone, where the drum is running onto the lining.

![Figure 2. Temperature field in drum and lining after half of braking time](image)

**Figure 2.** Temperature field in drum and lining after half of braking time

Obviously, significantly higher temperatures occur rather in the lining than in the drum. This seems to be understandable since the linings are continuously subjected to an inflowing heat flow. For the drum, this heat flow is interrupted by the rotation of the drum [6].

Maximum temperatures in the lining and drum do not occur at the end of the braking process, but before it. In the lining after approximately half the braking time. This is caused by the braking power steadily declining during the braking process and the given heat dissipation from the friction surface by heat conduction into the brake lining and in particular into the brake drum. Hasselgruber demonstrated this relationship for the one-dimensional case already [7].

The temperature in the lining develops relatively continuously. The likewise relatively continuous temperature development in the drum is superimposed by a temperature fluctuation (Fig. 4). This results from the permanent change between heating under the linings and cooling in phases without contact with the linings [8].

![Figure 3. Temperature vs. time in lining and drum for single braking process](image)

**Figure 3.** Temperature vs. time in lining and drum for single braking process

![Figure 4. Short term temperature fluctuation in drum](image)

**Figure 4.** Short term temperature fluctuation in drum
Fig. 5 shows the course of the temperature in the drum over its width. In the middle, the highest temperatures occur. The temperatures drop off at the edges of the drum. The drop to the side of the drum assembly side is more pronounced [9]. This is due to the increased heat dissipation into the accumulation of material given there.

Figure 5. Temperature across drum width

Fig. 6 shows the temperature curves in the lining and the drum for a multiple brake process. The cooling of the lining and drum surface after braking is clearly visible. However, it is also clear that, in the event of further braking, higher temperatures are reached than those after the first braking, if this further braking is only carried out sufficiently close in time.

Figure 6. Temperature vs. time in lining and drum for multiple braking process

Figures 7 a and b show the temperature trends in drum and lining over the component width and the time.

Figure 7. Temperature vs. time for multiple braking process: a - in drum, b - in lining.
It should be pointed out, that the processes occurring in the frictional zone are quite complex. In fact, surface pressure, temperature and, ultimately, the chemical conditions may vary significantly at a particular location [9]. In this respect, the maximum temperatures determined here are certainly "average maximum temperatures".

5. Conclusions
In the case of comparable braking processes, lower maximum temperatures in the drum and lining are achieved as a result of a single braking than in the case of multiple braking. As a result, the detection of the steady-state temperature, which is mostly relevant for mechanical braking, is no longer of central importance. Rather, the maximum temperature due to a one-time braking is placed in the centre of the consideration for electrically braked lifting gear [10].

The expected maximum temperatures can be determined on the basis of FEM models [11]. However, for a practical application it would be desirable to create simple calculation methods for a sufficient estimation of the expected temperatures.

On the part of the components, the maximum temperatures tolerable for a short time get into the focus. These should be higher than the permanently tolerable maximum temperatures. This could result in the fact that brakes on a previous design basis now have thermally higher safety during operation with usually electric braking. For new systems, this could result in a reduction of size for brakes, which function as holding brakes and emergency stop brakes in addition to operation with electric braking.

As a result of the modified drive technology, modified requirements for hoist brakes result. As a result of the technology of the frequency inverters, which is established in the wide range, operating brakes arranged on the axis of the motor shaft mutate into holding brakes. Instead of performing braking at a certain frequency, braking with a corresponding energy conversion occurs rather only in the exceptional case. This reduces the maximum temperatures to be expected in drum and lining. Furthermore, the maximum temperatures occur only briefly. These developments result in the potential of greater thermal safety or a smaller size of the brake. In order to exploit this potential, sufficiently accurate data on the expected and permissible states must be known.

References
[1] Vöth S 2015 Hoists with safety brakes Vol 1: Loads of components, Lifting conveyors 3 150-152
[2] Ishak M R, Abu Bakar A R, Belhocine A , Taib J M, Omar, W Z W 2016 Measurement 94 487-497
[3] Severin, Lührsen: Change of temperatures and coefficient of friction in the friction surfaces of industrial brakes under thermal load, DHF, 5/6 143-152
[4] Aleksendric D, Barton D C 2006 Tribology international 42(7) 1074-1080
[5] Day A J, Harding P R J, Newcomb T P 1984 Proc. Inst. Mech. Eng., Part J: J. Eng. Tribol. 198D(15) 287-294
[6] Hwang P, Wu X, Jeon, Y B 2009 Proc. Inst. Mech. Eng., Part J: J. Eng. Tribol. 223(J7) 1041-1048
[7] Bao J, Zhu Z, Tong M 2012 Industrial lubrication and tribology 2012 64(4) 230-236
[8] Vöth S, Vasilyeva M A 2016 Journal of Mining Institute 2016 219 455-458
[9] Cai P; Wang Y; Wang T 2015 Tribology international 2015 87 1-10
[10] Cantiziano A; Carnicero A; Zavarise G 2002 Computational materials science 25(1-2) 54-60
[11] Gui L, Zhang F, Fan Z 2014 Advances in materials and materials processing iv, pts 1 and 2, 887-888 (886)