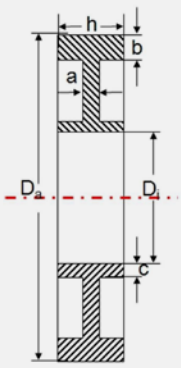
Measurement of a cooling process in an oil quench bath of a multi-purpose chamber furnace

by **Michael Taake, Thomas Lübben**

The analysis of the dimensional and deformation behaviour of metal parts during heat treatment is usually very costly, since the component distortion is affected by a number of factors. By using heat treatment simulation techniques, in principle many possible influencing variables can be numerically analysed in direct relationship to part distortion. However, due to the necessary characterisation of the material behaviour and the process, considerable costs are initially incurred. The cost of achieving such potential benefits needs therefore careful consideration. When deciding on the numerical approach, the question of heat transfer arises very quickly. The size of product is of critical importance, since in many cases the dimensional and shape changes are very sensitive to it. For simple geometric forms, measurement results can be found in literature. On the other hand, complex components such as gears require accurate monitoring and analysis of the cooling step especially in oil quench processes to determine actual physical distortion. In general, even if the oil quench process is accessible monitoring the quench cooling rate is very difficult, potentially dangerous and with a high risk of damage to the trailing thermocouples used. A few years ago, the development of a new thermal barrier which can go through oil quench baths created new opportunities. In the following, a collaborative project between PhoenixTM, Bad Oeynhausen, Germany, and Leibniz-IWT, Bremen, Germany, is reported.

The investigation of the dimensional and deformation behaviour of metal parts during heat treatment usually requires a large number of experiments that cause high costs. The reason for this is that the component distortion is usually subject to relatively large variation and is influenced by a large number of influencing variables. This is where the heat treatment simulation comes in. Although a considerable effort is required to characterise the material behaviour and the process, a validated model can be used to answer many questions, so that the question of costs is put into perspective.

In recent years, the heat treatment simulation has gone on to leave behind the calculation of simple samples (cylinders, disks, rings) and to approximate real components [1–9]. From the material side, this does not result in much greater characterisation effort. On the process side additional expense arises especially in the understanding of the heat transfer during quenching, which plays an important role in the heat treatment simulation. Here, an adequate description of the heat transfer coefficient must be provided for the entire surface of the component. For gas quenching, this model can also be obtained for complex geometries by using Computational Fluid Dynamics



Measure	Passenger car [mm]	Commercial vehicle [mm]
Outside radius r_a	57.5	115
Inside radius r_i	20	40
Height h	30	60
Hub thickness c	7.5	15
Web thickness a	7.5	15
Gear rim thickness b	7	14
Web location d	15	30
Radius r_c	4	8

Fig. 1: Examined geometry (left), dimensions of the reference variant (type A) for the size categories passenger cars and commercial vehicles (right)

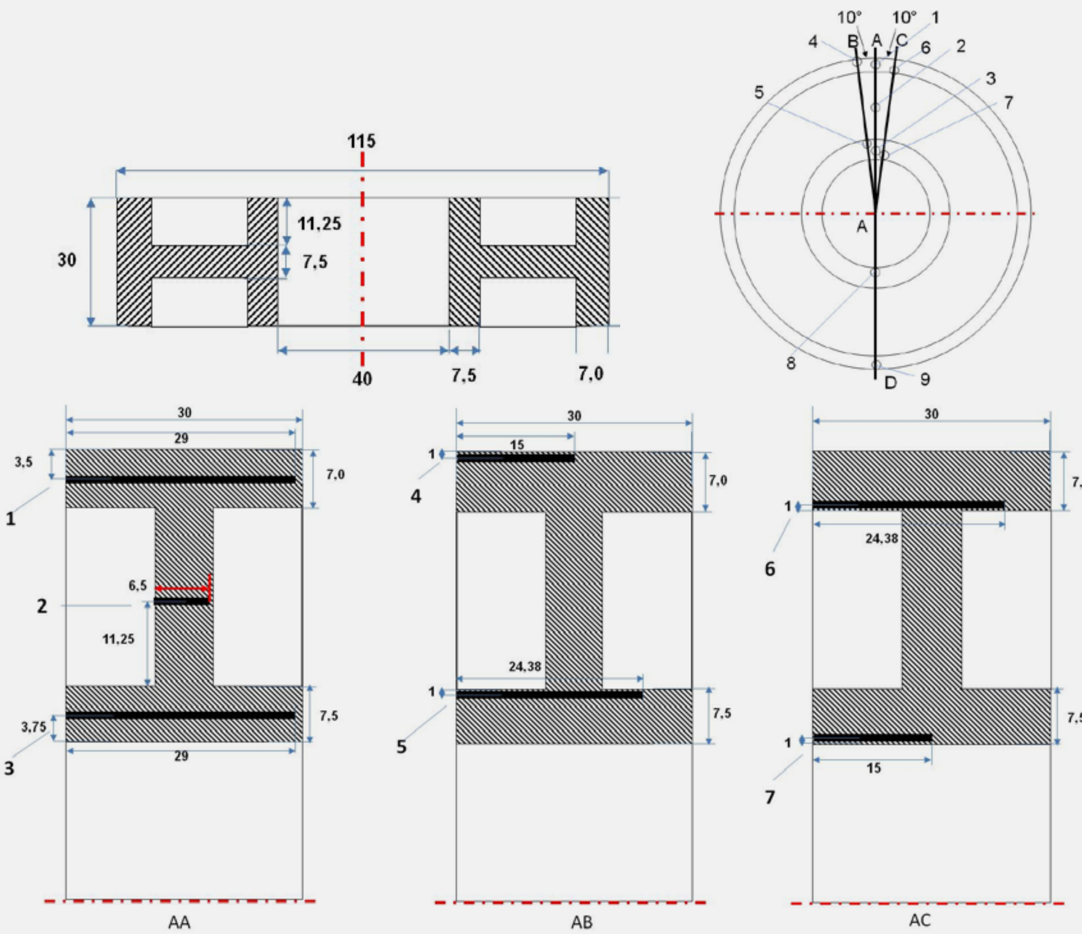


Fig. 2: Dimensions (top left) and schematic of thermocouple drilling for geometry type A and size category passenger car

(CFD) [10]. For evaporating liquids only the convective heat transfer in combination with gears is considered as too oversimplified. For simple geometries such as cylinders, first calculation approaches were tested and roughly describe the phenomena of evaporative cooling: steam skin phase, rewet, cooking phase and convection phase [11]. For more complex geometries, these approaches have so far not been used to the knowledge of the authors.

The bottom line is that the heat transfer coefficient of more complex geometries in evaporating media has so far only been determined experimentally. Until a few years ago, corresponding tests could only be carried out in laboratory systems with individual parts. For this purpose, the parts were fitted at defined positions with thermocouples. From the determined cooling progressions, the heat transfer coefficient can then be determined by solving the inverse heat conduction problem. This procedure was usually done for geometrically simple parts like cylinders [12–14].

Standard heat treatment facilities, especially those employing oil quench, are often impossible to monitor using tradition trailing thermocouple techniques, and even if possible, are difficult, unsafe and create high risk of damage to the trailing thermocouple.

Here a few years ago, the new development of a thermal barrier, which can go into the oil bath, created new opportunities. In the following, a joint project of PhoenixTM and Leibniz-IWT Bremen is reported.

OVERALL OBJECTIVE

In order to evaluate the influence of mass reductions on the distortion of gear parts, close-meshed experimental and numerical investigations were carried out as part of a joint industrial research project. The top level or fundamental objective was the identification of constructive measures to limit distortion in case of reduced mass and space optimisation [15–16]. For the numerical investigations, a description of the heat transfer coefficient on the entire component surface was necessary.

OBJECTIVE OF THE COOPERATION PROJECT

As part of the cooperation project, on the one hand, the concept for sealing a prototype of the newly developed heat protection container against oil ingress was to be tested and the maximum duration of the thermal protection determined. On the other hand, temperature measurements should be made during a quenching process in oil in a multi-purpose chamber furnace using prototyping. From the temperature curves simplified descriptions of the heat transfer coefficient (HTC) on the entire component surface for different parts should be derived.

EXPERIMENTAL BOUNDARY CONDITIONS

Component geometry and dimensions

This work was carried out on gear main bodies made of 20MnCr5 case hardening steel. The test pieces employed was used without formed gear teeth to reduce the calculation time in the experimental simulation of the heat treat process. However, this approach is expedient because the gear tooth correlated with the body distortion [17] and the assessment of the delay behaviour is therefore possible from the simplified model. The basic geometry is shown in **Fig. 1**. It corresponds to a disc with a hole, from the top and bottom of circumferential grooves were taken to reduce the mass. If needed to address space limitations then this material removal can also take place asymmetrically. The dimensions of the reference variants – here designated Type A – on which, inter alia, cooling curve measurements were carried out are also shown in Fig. 1. The two size classes have a mass of approx. 1 kg (size category passenger cars – cars) or 8 kg (size class commercial vehicles – commercial vehicles). The ratio of all dimensions of commercial vehicles and passenger cars is always two (Fig. 1).

Measuring points and temperature measurement

The following assumptions were made to describe the location dependency of the HTC:

- In the case of suspended charging of the components, a mirror symmetry of the HTC with respect to the centre plane is present (Fig. 1)
- The HTC is essentially independent of the circumferential angle axial symmetry
- The HTC is location-independent but temperature-dependent on sub-surfaces of the surface
- The temperature-dependent functions for each subsection or part-area can be estimated from a cooling curve measurement per subsection or part-area.

Taking into account the mirror symmetry, the surface of the component can be subdivided into seven partial areas,

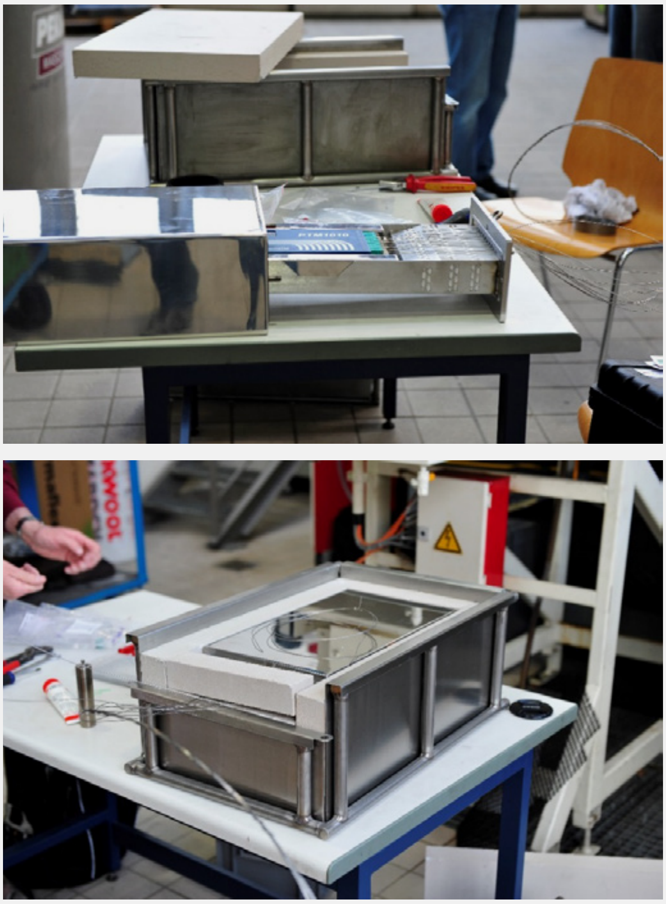


Fig. 3: Data logger in oil-tight protective housing (top), oil-tight protective housing with heat insulation and outer protective housing (below, here without cover)

regardless of the size class. The seven associated temperature measuring points each have a surface distance of 1 mm and are centred on the associated partial surfaces (**Fig. 2**). For the NFZ size class, the thermocouple holes were inserted at the equivalent positions.

All holes were made from one side of the disc, so that the heat transfer on the side of the disc to be measured is not disturbed by the thermocouple exit. All holes have a diameter of 1.1 mm and were fitted with 1.0 mm type K thermocouples.

To estimate the heat transfer coefficient, a cylindrical sample (diameter 28 mm × length 112 mm) made of austenitic steel (1.4301) was integrated into the batch in addition to the instrumented gear base body. This so-called Q-sample was equipped with two thermocouples, which detect the cooling process in the core of the sample or close to the edge.

Another thermocouple was used to measure the temperature on the surface of the data logger. In addition,

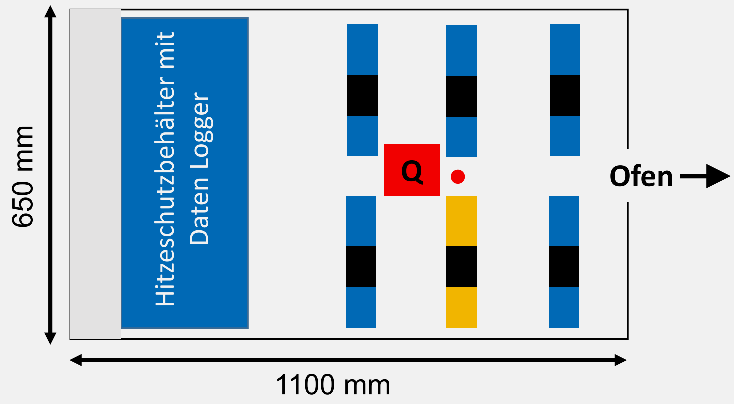


Fig. 4: Schematic representation of the batch structure: instrumented component (yellow), five dummy parts (blue), Q-sample (red)

the temperature at the reference junction inside the data logger was determined using a resistance thermometer. The temperature at all eleven measuring points was recorded every 0.5 s during the entire temperature-time cycle and stored on the data logger.[1]

Thermal barriers

All ten thermocouples were first connected to the data logger and led out of the inner thermal barrier case using oil-tight compression fittings (**Fig. 3**, top). This housing was then surrounded with a special insulation and installed in the outer protective housing (Fig. 3, bottom).

Batch layout

The thermal barrier was placed on the batch grate at a maximum distance from the load to minimise the influence of the flow. **Fig. 4** shows schematically the structure of this test batch. The thermocouples were inserted into corresponding holes and secured against movement by means of strain relief (**Fig. 5**). The real conditions of this batch build-up are shown in **Fig. 6**.

Furnace and temperature-time curves

The tests were carried out in a multi-purpose Aichelin type furnace with oil quenching and a batch volume of 1,100 × 600 × 650 mm^3 (max 650 kg). A total of four experiments were carried out with different component types and size classes. The processes were driven without a carburising atmosphere. The temperature-time profiles were, however, based on real carburising processes. On the one hand, it was crucial to be able to "touch" the maximum possible time in the oven for the data logger and, on the other hand, to thoroughly heat the component before quenching from a hardening temperature of 860 °C. **Table 1** shows the shortest (120 min) and **Table 2** the longest (256 min) of these processes.

RESULTS

Temperature development before quenching

Fig. 7 shows the measurement data of a test without the quenching segment. The dark lines characterise the temperature distribution in the component. Here, only very small temperature differences have occurred so that no appreciable stresses resulted in these partial steps of the process.

1 At the time of the experiments, telemetry real-time transmission of the temperature data from the oil-tight heat-protection tank was not yet available. In the meantime, this function is available. Furthermore, a measuring cycle of 0.2 s is now possible.

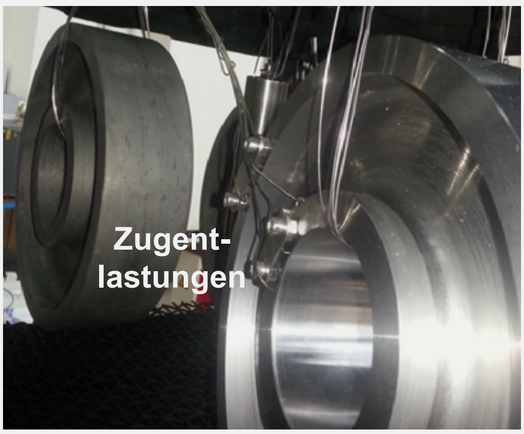


Fig. 5: Strain relief of the thermocouples on the instrumented gear main body

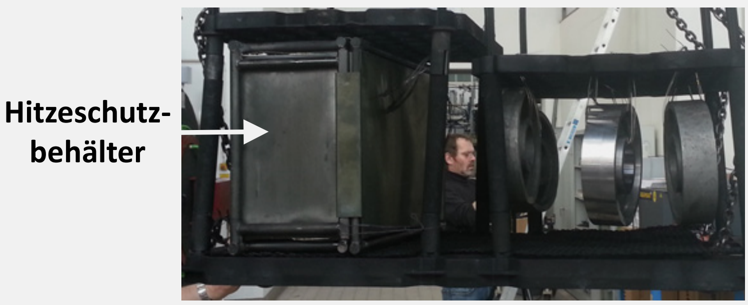


Fig. 6: Real batch build-up

Table 1: Temperature-time curves for testing the prototype: shortest process (120 min)

Description	Set point temperature [°C]	Duration [min]
Oven entry	860	45
Hold	860	45
Quench	60	30

Table 2: Temperature-time curves for testing the prototype: longest process (256 min)

Description	Set point temperature [°C]	Duration [min]
Oven entry	850	64
Hold	850	30
Preheating	>=945	45
Cooling	<=865	27
Hold	860	50
Quench	60	40

The maximum temperature within the wear insulation was 353 °C. Inside the data logger a maximum value of only 36.6 °C was reached. The values calculated during the development of the thermal barrier container are just above these measured values. In addition, the measurement in the data logger is still well below the allowable temperature of 85 °C and the phase change material used in the heat sink, with a melting temperature of 58 °C, remained un-melted at the end of the cycle time. It is therefore proposed that longer residence times or higher furnace temperatures would be also possible.

Cooling processes in the gear main bodies

Fig. 8 shows the resulting cooling curve for the geometry type A of the size category standard passenger car. The steam phase at the beginning of the process or cycle only takes a few seconds. But even in this early stage of cooling, the profile graph shows a clear separation of the individual temperature traces. This is amplified in the subsequent cooking phase. After falling below about 400 °C, the cooling rate is reduced significantly. In this phase (convection cooling), the differences between the individual measuring positions are most pronounced. The fastest cooling is therefore at the end faces of the sprocket (position 1) and hub (position 3). Cool the hole (position 7), the bridge (position 2) and the outer surface of the hub (position 5) much slower. These differences result partly from the geometric conditions at the measuring points. But also the locally different flow velocities lead to location dependencies of the HTC. This effect is clearly visible on the bridge.

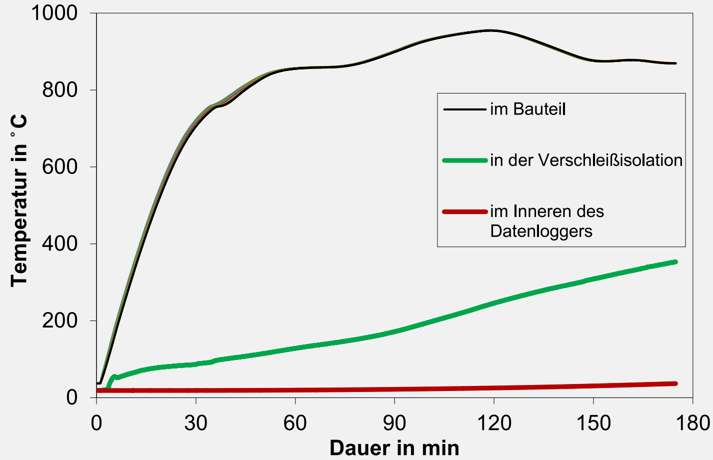


Fig. 7: Development of the temperature in the component, in the wear insulation (green curve) and inside the data logger at the temperature comparison point (red curve)

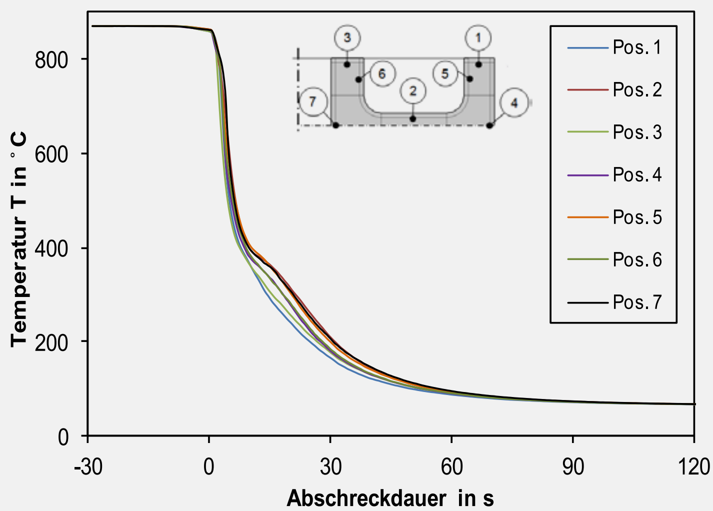


Fig. 8: Measured cooling curve on geometry type A, size class passenger cars

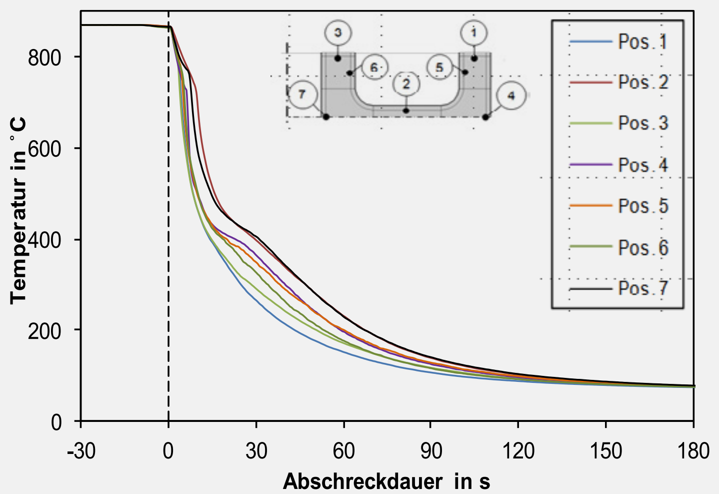


Fig. 9: Measured cooling curve on geometry type A, size class commercial vehicle

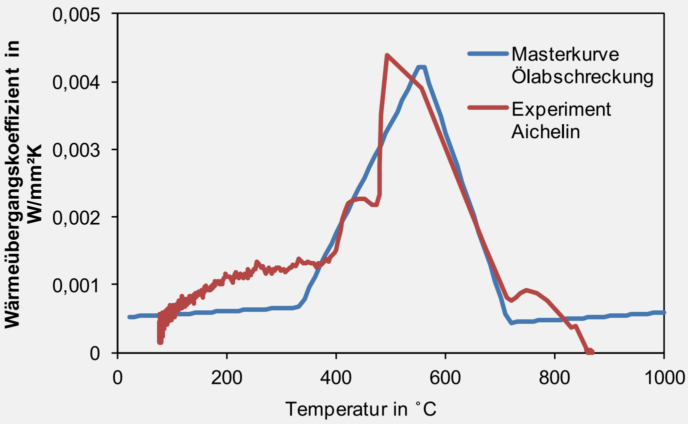


Fig. 10: Master curve "HTC" over surface temperature" for the adaptation of the simulated cooling curves to the measurements

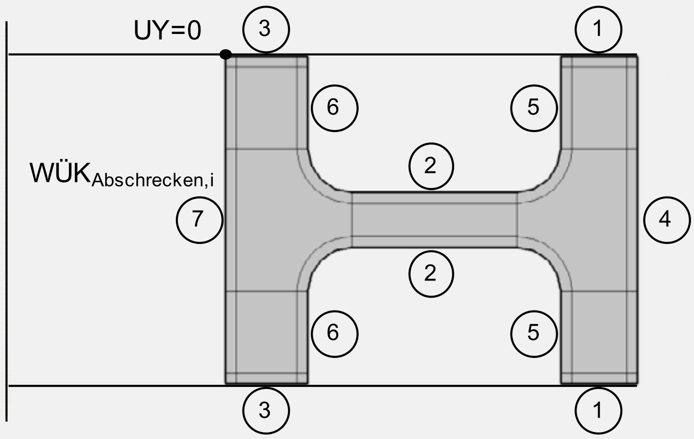


Fig. 11: Allocation of the location-dependent heat transfer coefficient to surface areas

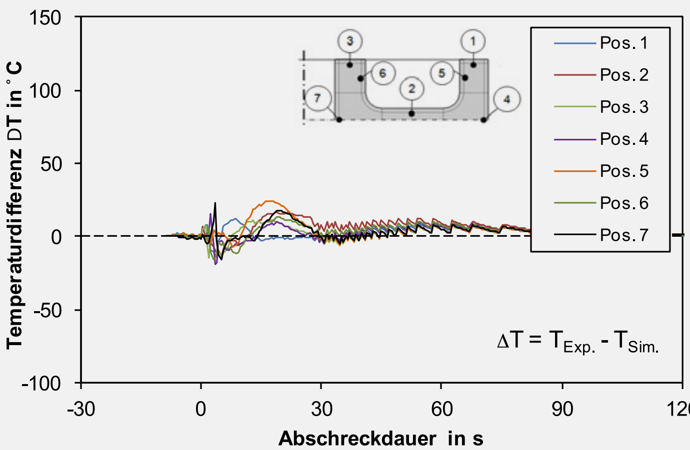


Fig. 12: Difference between measurement and simulation after optimization of the heat transfer coefficient (type A, size category PKW)

Fig. 9 shows the resulting cooling curve for the geometry type A of the size class commercial vehicle. Of course, with this eight times heavier component cooling is slower overall. The steam phase is more pronounced and shows a clear spatial dependence: At the web and in the hole (positions 2 and 7), the vapor film is much more stable than at the other positions. Accordingly, the cooling is slowest here. At positions 1 and 3 the fastest cooling takes place also for this component.

Determination of the heat transfer coefficient

In a first step, the heat transfer coefficient as a function of the surface temperature was determined from the cooling process of the Q-sample using a finite difference method [12]. The result of this evaluation is shown in **Fig. 10**. On one hand, this heat transfer coefficient characterises the quenching oil used (high-performance quenching oil W 72) and, on the other hand, the flow conditions at the Q-sample. Of course, the latter do not match the flow on the component. Therefore, this curve cannot be used directly for the simulation. But it can be used to generate a master curve for the derivation of the HTC.

Based on this, in the second step, a trial and error adjustment between measurement and simulation was carried out by variations in the HTC, which took place both as a function of the temperature and the position. In order to keep the effort for this procedure within acceptable limits, a simplification of the measurement curve was necessary. Fig. 10 shows the above-mentioned master curve, which is defined by only five points and roughly reproduces the original curve. These points of the master curve were varied for each of the seven surface segments according to **Fig. 11** using trial and error until a maximum difference between measurement and calculation of 20 K resulted at all measurement positions.

The result of this adaptation is shown in **Fig. 12** in the form of the difference between the respective temperature measurement and the associated simulation using the example of the geometry type A of the size category PKW. The upper limit for the difference between measurement and simulation is reached only at some times and is usually smaller.

CONCLUSION

The system used (**Fig. 13**) allows temperature measurements in the heating chamber and in the oil bath of a multi-purpose chamber furnace, which are difficult or impossible with other systems. The handling of the system in the preparation of measurements has been found to be very practicable.

The construction of the thermal barrier container combining complementary inner sealed case and outer insulative skin has proved to be reliable, oil-tight and thermally

Fig. 13: Components of the PhoenixTM measuring system used [18]

effective (also in subsequent experiments). The thermal barrier design prevented any damage or interference to the data logger by the process conditions, resulting in a very high quality of the measurement data.

As a result, the derivation of functions for describing the heat transfer coefficient of gear main bodies of different geometries and dimensions was possible. Thus, an essential prerequisite for the derivation of a method for the preparation of design guidelines for the low-warpage hardening of gear wheels under the aspects of mass reduction and space optimization was created [16].

ACKNOWLEDGEMENTS

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