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Experimental and numerical investigation of the heat transfer inside a hollow piston rod

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ABSTRACT

The heat transfer in the system piston - piston rod - crosshead of reciprocating compressors can be influenced by a fluid enclosed in an internal cavity. Thus, heat is transferred by the fluid flow inside the hollow parts utilizing the reciprocating motion. Subsequently, the high working frequency of such compressors can lead to an improved cooling of the high temperature compressor parts in comparison with conventional solid material designs.

For the experimental examination of this cooling concept a test rig has been set up at the Technische Universität Dresden which comprises a vertically oscillating hollow rod that is heated at its upper end and cooled at its lower end. Among other factors the heat transfer of this cooling concept is affected by the internal diameter of the piston rod. In order to quantify the influence of the internal diameter on the axial heat transfer three different hollow rods with varying internal diameters were investigated by means of temperature measurements.

The experimental results are presented and the influence of the size of the cavity is discussed. Furthermore, the measured temperatures are used for numerical simulations of the test rig in order to derive the axial heat flux of the fluid inside the cavity. Based on the resulting values the heat transfer capability of the investigated cooling concept can be examined with respect to the internal diameter and the benefit for the cooling of different compressor components can be estimated.

1. INTRODUCTION

When there is a demand for oil-free compression typically dry-running reciprocating piston compressors are used. The applied sealing elements are usually made of polymer composite materials which do not require an additional lubricant, offering economic and environmental benefits. In this field not only a growing interest in research but also an increasing utilization can be observed over the last decades (Kriegel, 1977; Tomschi, 1995; Feistel *et al.*, 2003; Thomas, 2013).

Especially for higher capacities the advantages of crosshead-based compressor designs are typically preferred, requiring a sealing at the piston rod – the so-called packing – in order to minimize the leakage to the environment. Due to the lack of lubrication the mechanical and thermal stress of the sealing elements are significantly higher compared to oil-lubricated designs. Hence, on the one hand friction increases and on the other hand the omitted lubricant reduces the heat rejection. This leads to higher temperatures and to an increased wear of the sealing elements. As a result the service life of the sealing elements is reduced and the operating costs of the compressor increase.

According to Kriegel (1977) the temperature in the contact zone between the non-lubricated sealing and the counter surface is crucial to the wear rate of the polymer sealing elements. Their attrition increases with increasing temperature. Hence, their cooling is expedient.

Up to the present a coolant circuit provided in the housing of the packing is often applied for the rejection of heat (Feistel *et al.*, 2003). However, such an external packing cooling shows several drawbacks in terms of additional constructional and operating means giving rise to higher investment and operating costs. In addition, the risk of a blocking of the coolant ducts as well as leakage of coolant is well-known.

Based on these circumstances a different cooling approach was created for which a coolant is filled in an internal cavity in the piston rod. For the sake of simplicity this cavity can be designed as a cylindrical bore. The hereafter presented investigation and related results show the influence of the internal diameter on the heat transfer through a reciprocating cooling cavity within a reasonable range.

2. INTERNAL COOLING CONCEPT FOR RECIPROCATING COMPRESSORS

The idea of an internal cooling of the piston rod for reciprocating piston compressors was firstly delineated and patented by Quack and Nickl (2009). The following section gives an overview of the principle of operation and the latest state of the internal cooling concept as well as the consequential motivation for this publication.

2.1 Principle of Operation

The aim of the internal cooling concept is to extract heat from components that need to be cooled, *e.g.* the packing and the cylinder area. For this purpose, a coolant filled into a cavity inside the piston rod and/or the piston is used to absorb heat from the warmer regions, to transfer it through the piston rod and to reject it to the oil-lubricated crosshead (see figure 1).

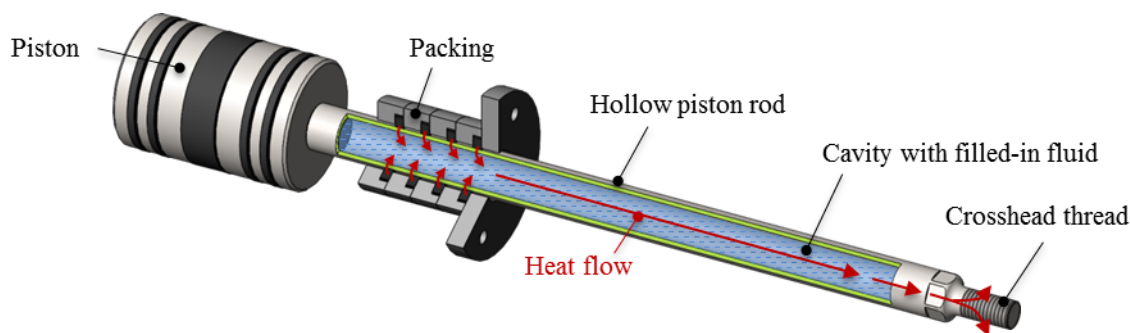


Figure 1: Design and principle of operation of the internal piston rod cooling concept

Although the thermodynamic drive is given by the temperature gradient, the main idea and benefit of this cooling technology is to exploit the reciprocating motion of the piston rod to cause an enhanced fluid flow inside the piston rod's cavity. In this way, the fluid absorbs heat by flowing through warmer areas and transfers it to colder areas where it is finally rejected. The typical high working frequency of such crosshead-type compressors (up to 1800 min^{-1}) enables an enhanced heat transfer through the piston rod and an increased cooling of high temperature compressor parts compared to conventional solid piston rods.

2.2 Previous Work and Results

Since the first reference of the described idea several experimental and theoretical studies were conducted focusing on its feasibility and general applicability. Hammer *et al.* (2012) measured a significant cooling effect at a test rig with a vertical oriented oscillating hollow rod induced by the internal cooling of the rod. In addition, results were presented indicating the possibility to model the heat transfer processes inside the rod's cavity by means of a Finite Element (FE) simulation. Thomas (2013) presented extensive results from both experimental and theoretical investigations which show the effectiveness of the internal cooling at three different test rigs, including a single-stage double-acting balanced-opposed compressor. Making use of the internal piston rod cooling the measured gas

and solid body temperatures inside the packing of this compressor could be reduced by up to 70 K. In addition, the sealing quality could be improved by reducing the temperatures in the sealing gap.

Comparable approaches in terms of an enclosed internal cooling cavity supported by a reciprocating motion can be found in the field of combustion engines, for instance the cooling of engine valves (Bush and London, 1965) and the heat transfer from the piston to the crankcase oil via heat pipes molded into the piston (Cao and Wand, 1995). Unfortunately, different scales impede the transfer of these results to internally cooled compressor piston rods.

2.3 Open Tasks

Since the feasibility and effectiveness of the internal piston rod cooling has been demonstrated, the next target is to optimize the heat transfer. A variety of parameters, influencing the heat transfer of the discussed cooling technology, allow more detailed investigations. The results of these investigations are the basis to enhance the heat transfer capability. These parameters are:

- the fluid itself and the charge,
- the cross section (shape, size),
- the kinematics of the rod (speed, stroke),
- the orientation relative to gravity,
- the temperature difference between the heat source and sink, and
- the heat flow to be transferred

as well as other thermal resistances.

One of the most significant impacts of the enumerated parameters constitutes the cross section of the cavity in axial direction. It is represented by the internal diameter in case of a cylindrical bore. So the question arises how the internal diameter affects the heat transfer through a hollow piston rod.

3. TEST RIG AND EXPERIMENTAL INVESTIGATION

At present the theoretical examination of the fluid flow, *e.g.* by CFD simulations, seems not expedient since the transient and turbulent processes in the shaken-up cavity are not educible with reasonable effort. In contrast, experimental investigations offer promising results (Thomas, 2013). Hence, the general construction of the used test rig and the obtained results are shown in this section.

3.1 Test Rig Setup

For the experimental investigation of the heat transfer inside a reciprocating hollow rod a test rig is available at the Technische Universität Dresden (see figure 2). It comprises a vertically reciprocating hollow rod actuated by a crank drive that is driven by an electric motor. The stroke of the crank drive is 100 mm. The rod is mounted onto a temperature-controlled crosshead. It comprises a cooling duct connected to an external cooling circuit providing the entire heat to be completely rejected at the rod's lower end, *i.e.* the temperature at this location can be kept constant for all operation points. At the upper end of the rod a cylindrically shaped electrical heater is installed to provide a defined heat input and the rod is closed by an end cap. All relevant components – except the crosshead – are insulated against the ambient. By this, most of the heat input is transferred through the rod and the filled-in fluid. In order to connect all electrical signals to the measurement system and to transfer the coolant fluid from the cooling unit to the moved system an energy chain is attached to the rod's cap.

To accurately record the temperature field of the relevant components with reasonable extent 38 type K thermocouples (TCs) are installed whose signals are sampled by 1 Hz. Their positions are depicted in figure 2 (c). Five TCs positioned inside the cavity (FT1 ... FT5) are distributed along the rod's axis measuring the fluid temperature. Twelve temperature sensors (WT1 ... WT12) are located 2 mm below the rod's exterior radial surface capturing the wall temperature. They are distributed along the rod's length similar to the fluid TCs. For the striven thermal simulation of the experiments and therefore required boundary conditions two TCs are installed at the rod's cap and 3 TCs at the lateral surface of the cap bracket as well as 2 TCs inside the cooling duct of the crosshead.

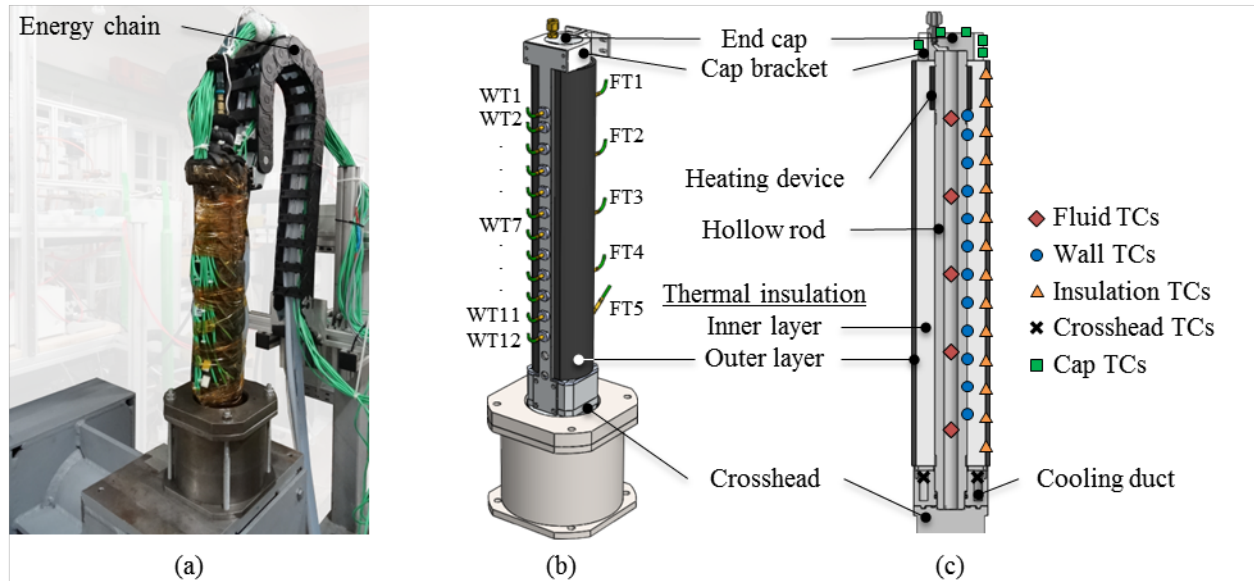


Figure 2: Setup of the test rig (a), CAD model (b), and sectional view with thermocouple positions (c)

The rod is equipped with two insulation layers. In between 14 TCs are placed measuring the temperature along the rod's axis. A former setup of the facility exhibits only one layer and the boundary condition at its outer surface was a convective heat transfer which is rather difficult to specify for a FE simulation (Hammer *et al.*, 2012). The TCs at the insulation are now used to provide the temperatures as boundary condition for the numerical simulation which is more accurate than the previous setup.

3.2 Measurement Procedure

At first the fluid to be investigated is filled into the cavity and the hollow rod is closed. This yields a coolant-air mixture at ambient conditions. Then the rotational speed (in this study only 600 min^{-1} are presented) is adjusted and the flow through the crosshead's cooling circuit as well as the power supply of the heater is switched on. After a transient period in which the temperatures are converging towards a steady state the following interval is used to record the final temperatures which are averaged over 5 minutes. In this way temporary fluctuations and phenomena are excluded. All measurements are carried out under laboratory conditions ensuring the ambient temperature to be sufficiently constant over time.

3.3 Measurement Series

For the experimental results shown here, 3 different hollow rods are investigated that differ only by their internal diameters (see test cases A, B, and C in table 1). For all measurements the same coolant-air-mixture was used as the filling. The power input of the heating device and the temperature of the crosshead coolant were constant for all measurements, thus the boundary conditions are consistent in terms of the heat sink and source. The cross sectional flow areas are referenced to the rod's value of test case A. Table 1 summarizes the relevant conditions.

Table 1: Relevant information on the test series

Parameter	Symbol	Unit	Test case		
			A	B	C
Rotational speed	N	$[\text{min}^{-1}]$	600	600	600
Heat input	P	$[\text{W}]$	70	70	70
Temperature of crosshead coolant	t_c	$[\text{°C}]$	10.3	10.3	10.3
Cross sectional flow area ratio	$A_{\text{flow}}/A_{\text{flow,A}}$	$[-]$	1.00	1.78	2.78

Since this paper aims for the determination of the heat transfer capability of the fluid, only the temperatures over time inside the cavity of the measurement's last six minutes are shown (see figure 3).

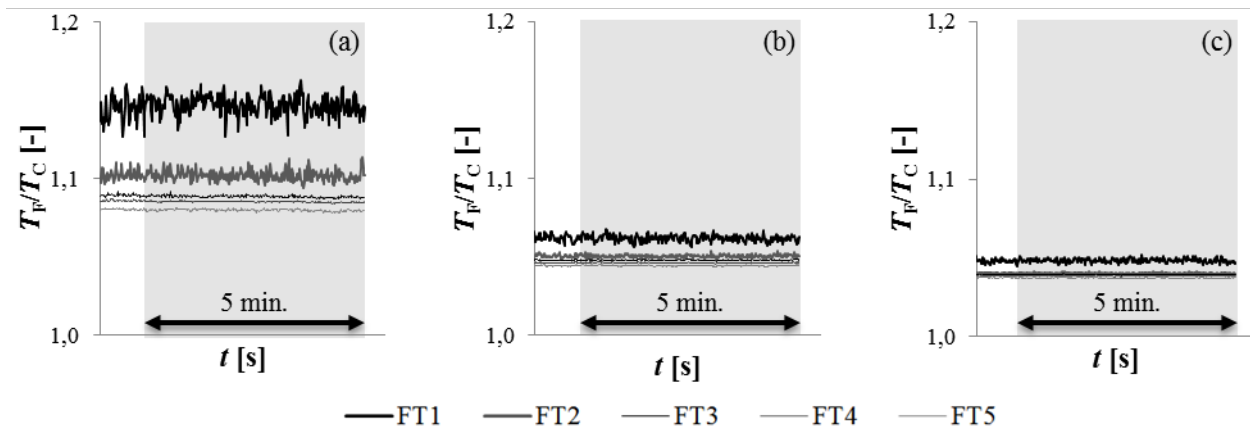


Figure 3: Fluid temperatures from the evaluation period (highlighted gray) of the test cases A (a), B (b), and C (c)

It can be seen that all fluid temperatures decrease with an increasing cross sectional flow area. The larger temperature reduction occurs between test case A and B. In comparison, the difference between test case B and C is less significant. Therefore it can be stated that increasing the internal diameter from test case A to B is considerably more effective than to increase it from test case B to C in terms of the axial heat transfer. Furthermore, the upper temperatures (FT1 and FT2) appear less steady for smaller internal diameters. This indicates a more fluctuating local heat transfer. Optical measurements of the fluid motion could help verifying how this is caused by or interrelated to the fluid's turbulent behavior. The temperatures exponentially increase with increasing height. Since the TCs are equally spaced along the rod the influence of the gravity cannot be neglected for the heat transfer in a vertical oriented internally cooled reciprocating piston rod even at 600 min^{-1} . The resulting values of the evaluation at the end of each test case measurement are summarized in table 2. The fluid temperatures are referenced to the temperature of the crosshead's coolant.

Table 2: Summary of measured steady-state temperatures of the test cases A, B, and C

Sensor	Quantity	Symbol	Unit	Test case		
				A	B	C
FT1	Mean referenced temperature	$\overline{T_{FT1}/T_C}$	[-]	1.147	1.062	1.049
	Relative deviation	$\sigma_{t_{FT1}}/t_{FT1}$	[%]	3.3	2.1	1.5
FT2	Mean referenced temperature	$\overline{T_{FT2}/T_C}$	[-]	1.102	1.051	1.040
	Relative deviation	$\sigma_{t_{FT2}}/t_{FT2}$	[%]	2.4	0.9	0.5
FT3	Mean referenced temperature	$\overline{T_{FT3}/T_C}$	[-]	1.089	1.048	1.040
	Relative deviation	$\sigma_{t_{FT3}}/t_{FT3}$	[%]	0.5	0.4	0.3
FT4	Mean referenced temperature	$\overline{T_{FT4}/T_C}$	[-]	1.085	1.047	1.039
	Relative deviation	$\sigma_{t_{FT4}}/t_{FT4}$	[%]	0.3	0.3	0.3
FT5	Mean referenced temperature	$\overline{T_{FT5}/T_C}$	[-]	1.080	1.045	1.038
	Relative deviation	$\sigma_{t_{FT5}}/t_{FT5}$	[%]	0.5	0.3	0.3

The measured values of both the fluid and the wall temperatures can be averaged over the steady-state period (highlighted gray in figure 3). Thereby, the thermal gradient along the rod represents the heat transfer capability according to the one-dimensional Fourier's law.

$$\hat{q}_z = -\lambda \frac{dt}{dz} \quad (1)$$

The mean fluid and wall temperatures for the three test cases are presented in figure 4.

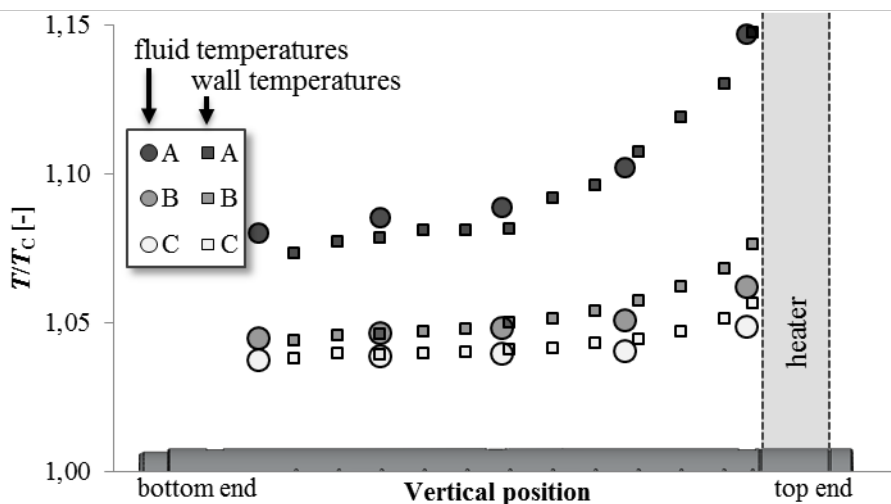


Figure 4: Fluid and wall temperatures from the evaluation period of the test cases A, B, and C

Besides some small differences it can be seen that the fluid and the wall temperatures show a similar distribution for each test case, respectively. As it is indicated in figure 3 larger internal diameters enhance the heat transfer and therefore lead to an improved cooling. Additionally, the larger differences between fluid and wall temperatures, *e.g.* at the top end of the rod for test case B and C, indicate a lowered radial heat transfer at this height. Those temperature differences can be seen at the bottom end of the rod for test case A as well. This is important to consider when it comes to optimize the heat transfer by minimizing the individual thermal resistance factors.

4. SIMULATION OF THE TEST RIG TEMPERATURE FIELD

The experimental results presented in the previous sections allow a qualitative evaluation of the axial heat transfer capability of reciprocating hollow rods depending on their internal diameter. To estimate the axial heat flux inside the cavity the recorded temperatures can be exploited to simulate the temperature distribution of the relevant parts. Therefore, the following section introduces the developed simulation model and the related assumptions as well as two derived quantities that are describing the axial heat transfer.

For calculating the thermal distribution of the test rig a steady-state thermal simulation model is used which is based on the finite element method. This simulation model consists only of solid elements, thus the fluid's thermal behavior (and not its flow characteristics) is represented by an equivalent thermal conductivity.

Since most of the test rig components are nearly rotationally symmetric the three-dimensional construction is transferred into a two-dimensional model (see figure 5). Moreover, the domain to be calculated is demarcated to an essential extension. This means that the outer layer of the rod's insulation and the insulation of the end cap are not adopted to the simulation model.

As shown in figure 5 (b) the area of the filled cavity (depicted white) is divided into several sections taking into account different dominant heat transfer mechanisms. According to different temperature gradients and corresponding to the design of the test rig this area is separated in the direction of the z axis. Additionally, these sub-areas are segmented into one part close to the rod's wall and into a second internal part which yields 12 areas representing the filling.

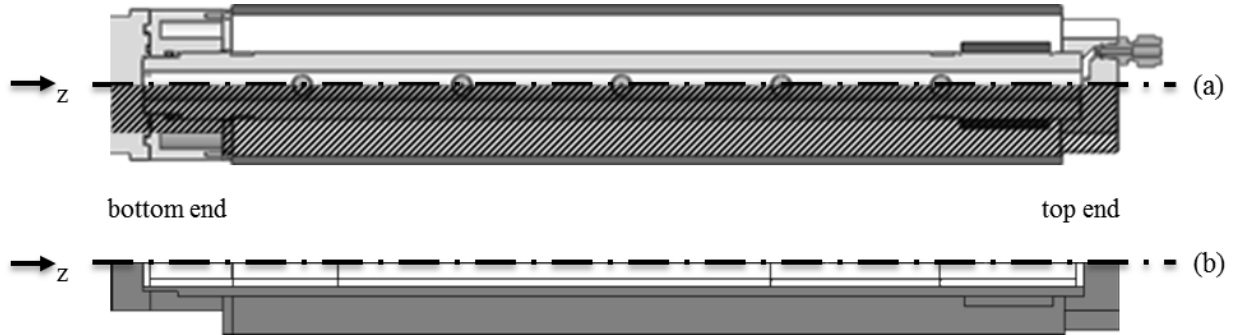


Figure 5: Cross sectional view of the CAD model (a) with areas adopted to the FE model (highlighted striped) and derived simplifying FE model (b) with white areas representing the cavity

After meshing the model with reasonable accuracy (Klotsche, 2012) all boundary conditions (temperature specifications and rotational symmetry) provided by the temperature measurements (see figure 2 (c)) and the thermal conductivity of all parts of the model except the fluid's one were predefined. The temperatures of the insulation are chosen to be piecewise linear functions between the thermocouple positions. Additionally, the heat input is specified as the heat generation rate for the cross sectional area of the heating device.

Then, the only unknown influence is the thermal behavior, *i.e.* the thermal conductivities of the filling areas. The idea here is to find a thermal behavior of the modelled filling that corresponds to the fluid's one in the experiments. This is done by an iterative procedure. In order to check this correspondence the calculated temperature field is compared to the measured fluid and wall temperatures. The result of such a comparison for test case A is presented in figure 6.

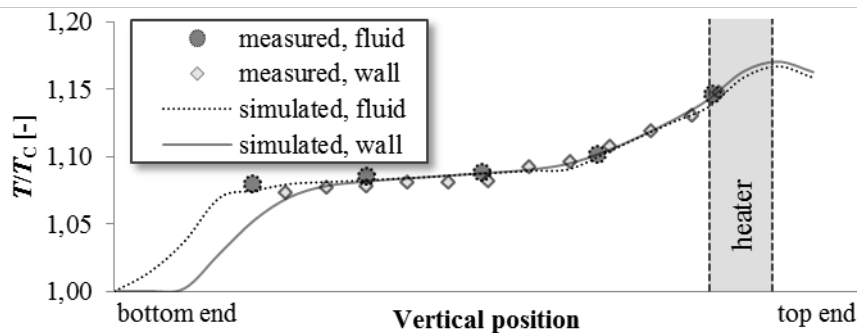


Figure 6: Measured and simulated steady-state temperatures of test case A

A sufficiently good correspondence between the measured and simulated fluid and wall temperatures can be seen in figure 6. Hence, the assumed equivalent thermal conductivities of the simulated filling areas accurately represent the thermal behavior of the fluid in the experiment. With this result the temperature field is exploited to deduce how much heat is actually transported by the fluid in axial direction. On the one hand, the power input of 70 W is not entirely transferred from the top end to the crosshead. On the other hand, some heat will be conducted by the rod's wall as well as by the insulation. To achieve an appropriate comparison between different measurements the axial heat flux at only one significant height is used. This position is chosen to be at the middle of the rod's length as it seems to be the most suitable representation of the axial heat transfer for the largest part of the rod.

To determine this parameter the axial heat flow through all finite elements at this height associated with the fluid are obtained at first. Then, each of these values is divided by the associated cross sectional area. This yields the axial heat flux for every circular ring of the fluid cross section. Finally, all values are added which gives the axial heat flux for the entire fluid cross section. The procedure of adjusting the simulation model to the experiment and ascertaining the axial heat flux was done for all measurements resulting in the values summarized in table 3.

Table 3: Axial heat flux and piston rod heat transfer capability at the middle of the rod's length of the test cases A, B, and C

Parameter	Symbol	Unit	Test case		
			A	B	C
Axial heat flux	$ \hat{q}_z $	W/mm ²	0.285	0.191	0.109
Piston rod heat transfer capability	C	W/K	302.1	548.2	715.1

The axial heat flux is a useful indicator for the heat transfer inside of an internally cooled piston rod. However, for the designing process not only the transferred heat in relation to the cross sectional area is of importance but also the temperature level occurring at the heated end. Besides the heat flux also the temperature gradient over the relevant axial length have to be taken into account. Consequently, a novel parameter – the piston rod heat transfer capability C – is introduced covering most of the influences that affect the axial heat transfer:

$$C = \dot{Q}_z * (\Delta T)^{-1} * L * (d_i)^{-1} \quad (2)$$

where \dot{Q}_z is the transferred heat in z direction, ΔT is the temperature difference between both ends of the piston rod, L is the length and d_i is the internal diameter of the piston rod. For the calculation of C (see table 3) the temperature difference and the distance between the thermocouples FT4 and FT3 is used. The cross section used for obtaining the axial heat flux shown in table 3 is located in-between those thermocouples.

When comparing both of the different parameters of table 3 two contrary behaviors can be identified. The test case A with the smallest internal diameter leads to the highest axial heat flux. In contrast, the piston rod heat transfer capability increases with an increasing internal diameter. This can be explained with the help of the simulation results. They show that nearly the same amount of heat is axially transferred by the fluid at the middle of the rod within all 3 test cases. If the heat flow is referenced to the cross sectional flow area the smallest internal diameter of test case A leads to the highest heat flux. The thermal gradient that causes the heat flow is not taken into account for the determination of the heat flux. Therefore, only the higher temperature difference between the ends of the rod for test case A enables a similar heat flow through the fluid compared to the other test cases B and C with larger cross sectional flow areas.

This illustrates the benefit of the self-defined piston rod heat transfer capability. This parameter allows the designing of an internally cooled piston rod based on the thermodynamic requirements. In particular, the necessary internal diameter can be calculated according to the heat flow that is to be transferred driven by a certain temperature gradient over a certain length. In contrast to the axial heat flux, the piston rod heat transfer capability of test case C is superior to the other test cases mainly because of the smallest temperature gradient.

5. CONCLUSIONS

Experimental and theoretical investigations have been carried out in order to verify the impact of different internal diameters in a vertically oriented internally cooled piston rod with regard to the axial heat transfer. The comparison of the resulting temperature distribution and a subsequent thermal FE simulation yield the axial heat flux inside the fluid for all test cases. Based on these heat fluxes, a new parameter is introduced – the piston rod heat transfer capability. This parameter characterizes the transferable heat flow as a function of the internal diameter and of the thermal gradient over the length of the piston rod as they are the most important influences.

The results constitute the basis for the selection of the internal diameter for future configurations and lead to an enhanced performance of the internal piston rod cooling. In a scientific perspective the attained results allow an improved insight into the thermal mechanisms of an internally cooled reciprocating hollow rod.

NOMENCLATURE

A	Area	(m ²)
C	Piston rod heat transfer capability	(W/K)
d	Diameter	(m)
L	Piston rod length	(m)
N	Rotational speed	(min ⁻¹)
P	Power input of heater	(W)
\dot{Q}	Heat flow	(W)
\hat{q}	Heat flux	(W/mm ²)
t	Temperature	(°C)
T	Temperature	(K)
z	z axis	(m)
λ	Thermal conductivity	(W/m/K)
σ	Standard deviation	

Subscript

A	test case A
C	Crosshead
F	Fluid
FT1 ... FT5	Fluid temperature 1 ... fluid temperature 5
flow	flow
i	internal
t	Temperature
W	Wall
z	z axis

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