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Internal Cooling of the Piston Rod in Non-Lubricated Piston Compressors

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ABSTRACT

In non-lubricated piston compressors the wear of the piston rod packing rings is still a problem that needs to be considered. Due to the demands of high gas pressure and leak-tight sealing the frictional heat of the rings is causing high temperatures in the area of the piston rod packing. High temperatures are – among other factors – one reason for an increase of wear and thus for a lifetime reduction of the packing. To increase this lifetime – and so to increase the time between two technical services – the idea of an internal cooling of the piston rod with a phase changing medium was created. To investigate this new cooling method for non-lubricated piston compressors a research project between the EFRC R&D Working Group (European Forum for Reciprocating Compressors) and the Technical University of Dresden was initiated. Within this project a test bench with a full scale balanced opposed process gas compressor was set up and measurements shall be taken. The measured values will be used for a validation of a thermal simulation model based on the software ANSYS. Also a model test rig is investigated which allows the investigation of temperature along a hollow piston rod in a moving system.

1. INTRODUCTION

The demand for oil free gases has increased during the last decades. On the one hand the gas flow should not include any oil particles because of safety issues. On the other hand hygienic reasons have priority especially in pharmaceutical, chemical and food industry. The delivery of gas with high purity at high pressures is only possible with non-lubricated pistons and packings. Due to higher friction and the omission of the cooling effect of the lubricant, several components of dry running compressors have much higher temperatures compared to those in lubricated compressors.

Especially the piston rod packing heats up enormously because of the friction with the piston rod. These high temperatures lead to a reduction of the lifetime of the packing elements. So the task for non-lubricated piston compressors is to reduce influences on the packing, rider rings and the piston rings, that enlarge the possibility of an early wear out and failure of the sealing components.

To define this radial wear Δr the approach of Kriegel (Kriegel, 1977) can be used according to equation (1) and (2):

$$\Delta r = L \cdot \bar{p}_r \cdot K \quad (1)$$

$$K = K_0 \cdot K_T \cdot K_C \cdot K_R \cdot K_\varphi \quad (2)$$

Here factor K_0 is a characteristic value for the material at reference conditions. The other factors depend on the reference conditions itself: temperature of the sliding path T , average piston speed c_m , roughness of the sliding path R and dew point of the working medium φ . So the goal of research must be to reduce every factor to its minimum.

In fact, we cannot change the type of gas thus K_ϕ . Also, the speed is given in most cases due to the drive. During the last decades there was a lot of optimization on the roughness of the sliding path. The result was, that a rough surface is negative and also smooth is negative, so here an optimum range for K_R has been found. Thus the temperature of the sliding path and K_T are the remaining condition respectively factor to be improved.

A lot of research was made within the last years to develop cooling concepts for the piston rings and packing. These arrangements include mainly cooling channels inside the housing of the packing or in the cylinder that provide a better heat transfer from the sealing elements to the surrounding. They all have the same problem: a separate cooling cycle is necessary for the compressor. This means: more costs, more parts, problems of leakage with the cooling medium, problems with plugging of the pipes and so on. All of these issues led to the development of a new patent for the reduction of the piston rod temperature.

2. INTERNAL COOLING CONCEPT FOR RECIPROCATING COMPRESSORS

2.1 Principle of Operation

The application of a modified type of heat pipe is the basis for the developed cooling concept. The function of a basic heat pipe is shown in figure 1:

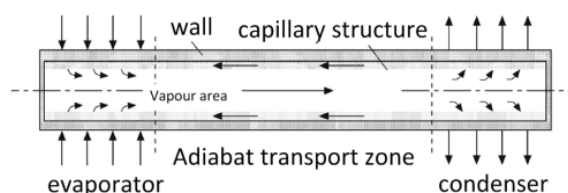


Figure 1: Setup of a heat pipe ⁶

A heat pipe is a heat exchanger with high heat transfer effect. A liquid medium is transported to an area of high temperatures and evaporates. The resulting vapour flows back to the “cold end” of the heat pipe and condenses. So the cycle is closed and can start anew. The vapour flow to a colder area and condenses. The only “critical” factor is the transport of the liquid phase to the “hot end”. In most heat pipes this is done by capillary structures (see figure 1). The liquid is transported by the capillary forces in most cases at the internal wall of the heat pipe. Between the evaporator and condenser is – in most cases – the adiabatic transport zone.

Heat pipes have the advantages of a huge heat transport capability (also at small temperature differences), short reaction time and a compact and simple design. The application ranges from temperatures of 5 K to 2200 K.

This concept is now adapted for the temperature reduction of the piston rod and of the piston of reciprocation compressors.

2.2 Application to compressor technology

A concept was created with the focus on piston and piston rod cooling of reciprocating compressors oscillating and translational movement of the piston and the piston rod (Patent DE 10 2007 00 652 A1). Nowadays there is a large number of non-lubricated piston compressors in the market. In most cases the sealing elements of the piston and packing are made of PTFE-compounds. Due to the absence of oil in the area of the piston rod sealing and of the piston there is no efficient heat transport to the surrounding. Another factor is that the frictional heat between the contact partners (piston rod – packing and piston – cylinder liner) is larger than in lubricated compressors. For this reason a cooling concept had to be developed. Of course there are several options that can be found on the market. The main principle is the cooling of the housing of the packing and the cylinder by utilizing cooling water. Using these methods means the deployment of a separate heat exchanging unit for the compressor. These separate units need space, driving energy and cause costs for the operator. That is the reason to develop an idea for an internal cooling of the piston and piston rod without any separate unit (see figure 2). Piston (1), piston rod (2) and cross head (4) are designed with internal cavities, which are connected to each other. Enclosed inside this cavity is a working fluid for the transport of heat. This leads to an efficient heat transport between the hot parts (piston and upper part of the piston rod) and the colder crosshead.

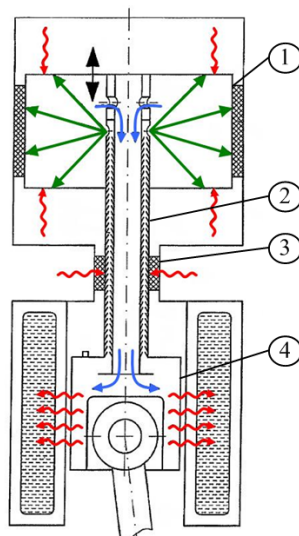


Figure 2: Cooling Device for Piston compressors (Patent DE 10 2007 00 652 A1)

In figure 2 the piston rod is divided into two parts: one channel for the liquid and one channel for the vapour. This shows only one possibility of the internal setup (here with internal structures). Other options are: just one channel as the basic and most simple design and also designs with check valves that should support the direction of the fluid flow. The main principle is that of a heat pipe. The difference is that no capillary forces are needed because of the movement of the piston. The liquid from the crosshead area is tossed to the areas of high temperature – the piston rod and the piston. The piston rod is used for transport from the cross head to the piston. The liquid phase of the cooling fluid is transported from the cross head to the packing area of the piston rod and to the piston itself where it evaporates due to the high temperatures that occur during the compression process. The vapour flows back to the cold end of the system – the cross head – which dissipates the gained heat to the surrounding. This heat dissipation can be supported by an appropriate design of the cross head.

So the heat pipe effect is an efficient tool to be applied to non-lubricated compressors. The difference between the described “basic” heat pipe and the principle that is used here is that there are no capillary structures. In reciprocating piston compressors there is no need for capillary forces due to the fact that the movement of the piston and the acceleration itself feeds the liquid phase of the working medium to the hot areas (Patent DE 10 2007 00 652 A1).

The focus of the concept is the piston and piston rod cooling. For the first research steps the cooling of the piston rod became the focus of attention. Due to the fact that the connections piston rod – piston and piston rod – cross head are heavily strained the first stages of the investigation only consider a cavity inside the piston rod. The remaining compressor components are not changed. The investigation includes an examination of a real compressor system, which is a balanced opposed compressor. An additional model test rig has been set up where basic investigations on the internal heat transfer coefficient are made. The measurements will be used to validate the simulation tool.

3. EXPERIMENTAL ANALYSIS WITH A MODEL TEST RIG

3.1 Set up

A real compressor system including the piston, piston rod, sealing elements and cross head is only hardly accessible. Due to the fact that the piston rod moves back and forth and rubs against the sealing elements a direct temperature measurement is rather difficult. There are several ways of measuring the piston rod temperature indirectly, but the accuracy of these results is controversial (Feistel, 2001). Also a definition of the frictional heat input of every sealing element and the amount of heat that flows into the sealing element or the opposite metal is hard to find.

That is the reason why pre-investigations were made with the help of a model test rig called “Alpha”. These experiments deal with the thermal impact of a piston rod that is designed as a heat pipe with an accurate measurement of the temperatures. A crankshaft driving mechanism was used for the construction of test rig “Alpha”:

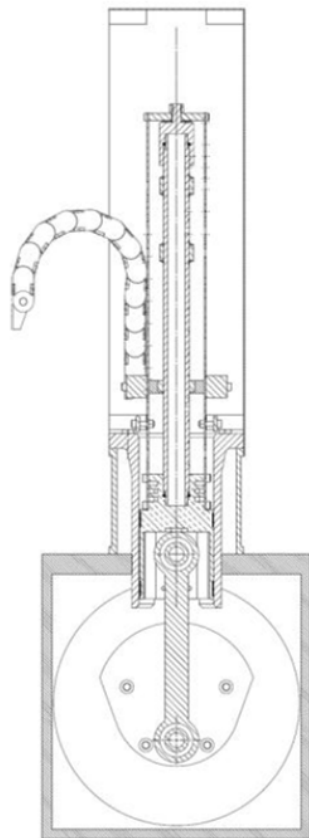


Figure 3: Section View of Test Rig “Alpha”

The pipe is driven by an engine with a variable speed up to 600 min^{-1} . The stroke of the driving mechanism is 100 mm. To reduce the oscillating mass the pipe is made of titanium with an external diameter of 35 mm and an internal diameter of about 24 mm. At an interval of 30 mm thermocouples measure the temperature of the pipe in a depth of 2 mm. Also two mantled thermocouples are placed inside the cavity for the detection of the internal temperatures of the working fluid. The heat input of the packing and the working chamber are simulated by two heating devices in the upper third of the titanium pipe.

The heating devices are mounted to the pipe and create the typical heat input of the packing and piston rings. The final set up of test rig “Alpha” is shown in figure 4. A device called energy chain supports the wires of the thermocouples and heating devices leading from the oscillating pipe to the test rigs frame structure.

The pipe itself is mounted at the bottom to the crosshead that was designed especially for this test rig. A socket with a valve for evacuation and filling of the pipe is mounted to the upper end. To adjust the heat input of the heating devices an adjustable transformer is used. The pipe is completely isolated so that the heat loss can be minimized. As described the heat input is defined, furthermore there is a heat flow to the upper and the lower part of the pipe and also inside the cavity.

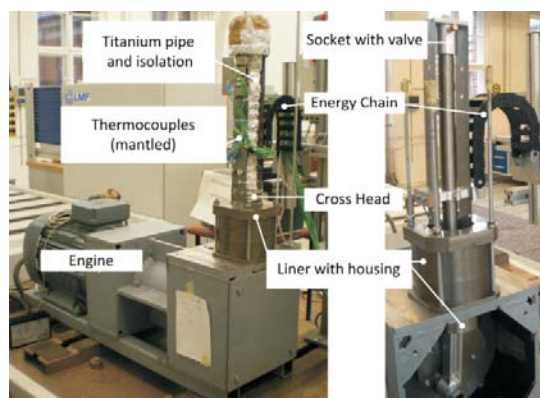


Figure 4: Set up Model Test Rig "Alpha"

The temperatures at the different parts of the pipe are measured and thus the heat flow can be defined. A difficulty is the heat loss to the surrounding. To reduce it as far as possible an effective isolation was installed and the temperatures on the outside surface are measured directly after stopping of the machine.

3.2 Results and Perception

The first stage of the experiments with test rig "Alpha" was the determination of the heat transfer coefficient at the outside of the isolated pipe. During these tests there was only air inside the pipe. Due to the low heat conductivity of air the influence of the heat transfer inside the pipe is neglected.

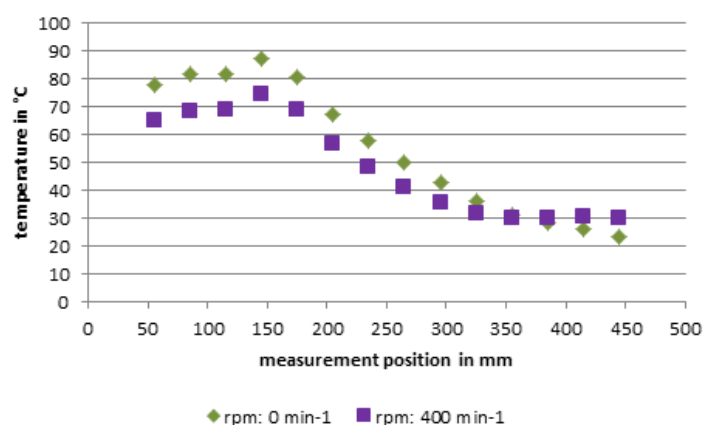


Figure 5: Measurement Test Rig "Alpha", air-filled titanium pipe

Figure 5 shows the results of 14 measurement points of the surface (depth: 2mm) of the titan pipe. The higher temperatures were measured in the area of the fixed heating devices. Then the temperature decreases in the direction of the cross head. The measurements of the temperatures of the pipe, inside the pipe and on the outside of the isolation were taken at different speeds and different heat input rates. The results were evaluated with the help of a steady state thermal ANSYS simulation which will be explained in chapter 4.1. These measurements are necessary to determine the heat transfer coefficient on the outside of the isolated pipe at different speeds. With this information it is possible to perform an energy balance around the system and find the directions of the heat from the heating devices. These values for the outer heat transfer coefficients α_a at different speeds will be another boundary condition for the calculation of the internal heat transfer coefficient when the pipe is filled with a cooling fluid.

The next step was the use of a fluid that has a better heat transport efficiency. The cooling concept should be possible to be realized in all different types of reciprocating compressors, so the use of a non-dangerous working fluid is desired. The first approach was a water-air mixture. A certain amount of water was filled into the titanium pipe and the measurements started once again at different speeds and different heat input rates. The result of such a measurement is shown in figure 6.

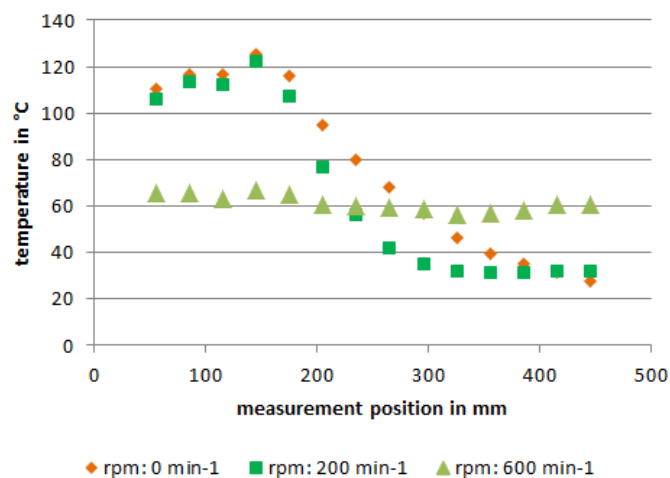


Figure 6: Measurement Test Rig “Alpha”, internal cooling with water-air-mixture

The location of heat input can easily be seen. At downtime the highest temperatures were recorded. The effect of the cooling is only small at a small speed of 200 min^{-1} . This leads to the assumption, that there is a certain range of speed of compressors where the cooling effect doesn't work in an efficient way and that the cooling is dependant from the acceleration of the piston rod. At 600 min^{-1} the internal cooling works obviously. The high temperature levels do not exist anymore but there is almost a constant temperature along the pipe.

4. MODELLING OF THE TEMPERATURE DISTRIBUTION

4.1 Model test Rig “Alpha”

For a detailed description of the temperature conditions inside the pipe and on the surface of the pipe a coupling of measured values and calculation is necessary. The analysis was made with a steady state thermal ANSYS simulation.

This analysis was made step by step – once with the user interface ANSYS Workbench, which is a graphical interface and with ANSYS APDL (Klotsche,2012), a computer language based ANSYS version. Working with ANSYS APDL allows a much easier definition of suitable parameters. Although the handling seems – at a first glance – more complicated than with ANSYS Workbench, it was recommended to use it for solving the calculation because the interaction with the ANSYS program via batch files is possible.

Due to the fact that almost every component of test rig “Alpha” is rotationally symmetric, a 2D-calculation is sufficient. This leads to a reduced calculation time and thus it is possible to set a finer grid for the calculation. So the first step was to transfer the 3D-design into a 2D-model (see figure 7).

With the 2D-model, a steady state calculation was possible to simulate the thermal conditions of the model test rig and to define the heat transfer coefficients inside and outside the titanium pipe.

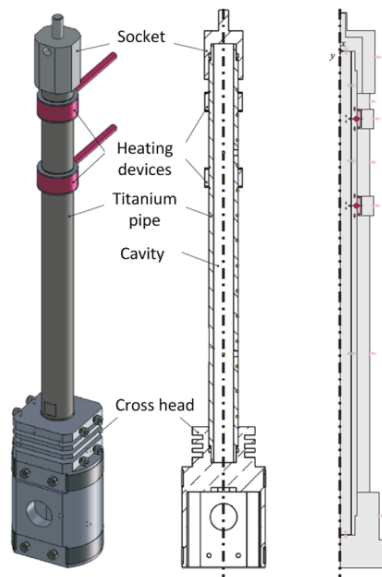


Figure 7: Transfer 3D-model to 2D-model (Klotsche, 2012)

The first step for the simulation was the examination of the heat transfer coefficient on the outside of the isolation of the pipe at different speeds and heat input rates. The assumption was made that – at similar temperatures on the surface – the heat transfer coefficient on the outside is independent from the temperature difference between the surface and the surrounding (Lindner-Silvester, 2007).

So if the heat transfer coefficient on the outside of the pipe is known, future changes of the temperature of the pipe surface can be affiliated to the effect of the cooling medium – which means the internal heat transfer coefficient.

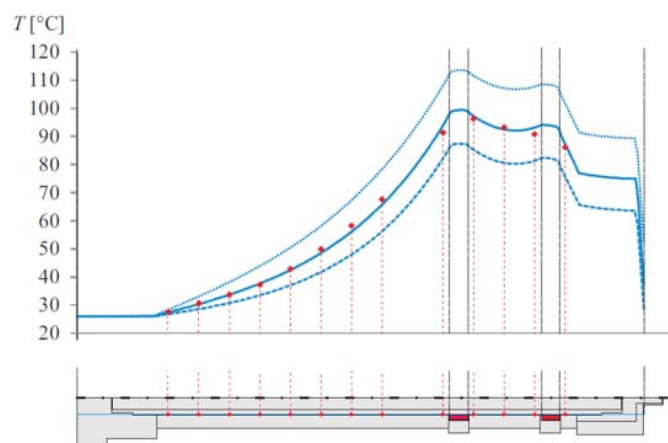


Figure 8: Evaluation of external heat transfer coefficient by parameter study (Klotsche, 2012)

With the help of the measured values and a parameter study it was possible to detect the correct heat transfer coefficients on the outside of the pipe (see figure 8). Also a study for an optimal grid was made considering the accuracy of the results and the calculation time and a good optimum could be found.

With this knowledge it is now possible to evaluate the processes inside the pipe which is one main part of future investigation.

4.2 Process Gas Compressor

A balanced opposed compressor was set up to evaluate the cooling method at a real system (see figure 9 and 10).

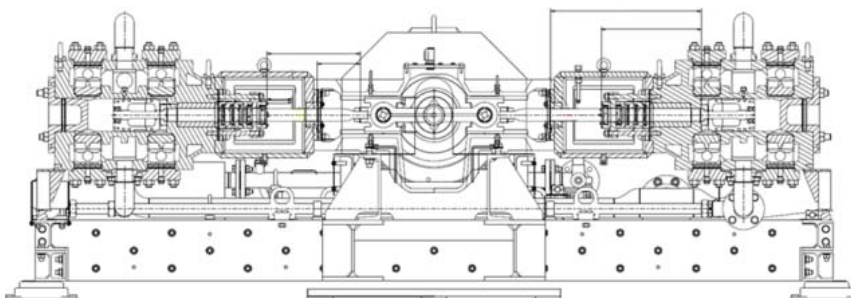


Figure 9: Section View: Balanced opposed compressor (construction LMF)

The compressor is a 1-stage, double acting machine with two cylinders. It is set up with temperature and pressure sensors in the suction, discharge and working chambers.



Figure 10: Balanced opposed compressor

Inside the packing temperatures and pressures after each sealing ring and the temperature of the piston rod are detectable with thermocouples. The thermocouples measure the temperatures just of the piston rod (Patent EP 1 840 544 A1). Also the temperature of the oil surrounding the crosshead and the leakage and discharge gas can be measured. The theoretical part of the simulation will be described in the next passage.

Also for the simulation of the thermal behaviour of the balanced opposed compressor the tool ANSYS APDL was used. In this case a transient thermal simulation was set up. A transient calculation was chosen because of the assumption that the fast periodic changes of frictional heat – owed to the changing pressure inside the sealing chambers – would lead to a transient behaviour of the surface temperature of the piston rod.

Once again the 3D-model was transferred into a 2D-model with a rotationally symmetric calculation. With the help of a calculation of the conditions inside the working chamber and first assumptions about the impact of the sealing elements and their frictional heat an excel-sheet was prepared as an input list for the transient boundary conditions of the piston rod.

In this case no frictional heat was calculated in ANSYS but the assumed amount of heat that is produced by the friction is an input value (area related heat q) for the transient simulation. Also the influence of the grid structure was considered which has large impact on the accuracy of the calculated values and the calculation time.

The programmed batch file makes it possible to define a value for the distance between the calculated crankshaft angle and the number of periods that should be calculated. In figure 11 the results of a transient simulation of the surface temperature of the piston and the piston rod at 1450 min^{-1} and a discharge pressure of 3 bar is shown (with different grid sizes).

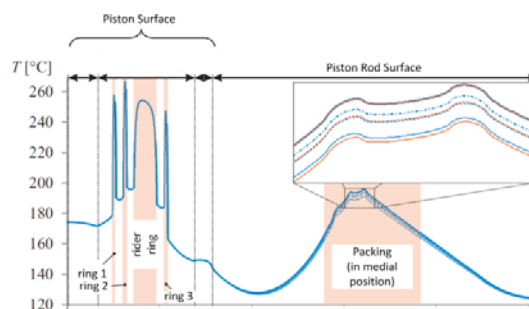


Figure 11: Transient Thermal Simulation balanced opposed compressor (Klotsche, 2012)

To reduce the calculation time it is necessary to define the starting values of each node as close as possible to the stationary end condition. After several simulations with an optimized net it was found that the transient calculation is not necessary if the starting conditions and the averaged boundary conditions are as close as possible to reality. The effect of the transient boundary conditions on the temperature gradient in radial direction within the first millimetres is smaller than assumed because of the heat conduction and inertia of the piston rod material.

5 CONCLUSION

The method of an internal cooling constitutes an efficient and simple design for a temperature reduction of the piston rod. Because of the reduced sliding path temperature an increase of the lifetime of the sealing elements inside the packing is expected. Within a research project several investigations are taken. Two test rigs were set up for measurements and the results are evaluated by the FEM software ANSYS. The influence of several parameters can be evaluated and so an outlook for other machine types or boundary conditions can be made in the future.

ACKNOWLEDGEMENTS

At this point I want to thank the EFRC for their great support during the last years and for believing in this project! Special thanks also go to LMF for their great support during this project and for installing the test compressor in our facility, which represents a major contribution to the experimental analysis of the cooling concept and to STASSKOL and HOERBIGER for their contributions!

REFERENCES

- Feistel, N., 2001, Comparative measurements of the piston rod surface temperature during operation; *2nd symposium EFRC*; 2001
- Klotsche, K., 2012, Simulation stationärer und instationärer Wärmeleitung und Wärmeübergang in der Kolbenstange zur Bestimmung der Kolbenstangentemperatur mit ANSYS; Großer Beleg; TU Dresden
- Kriegel, G., 1977, Beitrag zur Berechnung des Radialverschleißes von Trockenlaufkolbenringen; Dissertation Technische Universität Dresden
- Lindner-Silvester, T., 2007: Advances in fundamental understanding of the dynamic sealing action in packing systems; *5th EFRC Conference*
- Patent DE 10 2007 00 652 A1
- VDI_Gesellschaft Verfahrenstechnik und Chemieingenieurwesen (Hrsg.), 2006, *VDI-Wärmeatlas*, 10. Auflage
- Construction and information provided by Leobersdorfer Maschinenfabrik (LMF), Austria
- Patent EP 1 840 544 A1