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Influence of Thermal Deformation on the Characteristic Diagram of a Screw Expander in Automotive Application of Exhaust Heat Recovery

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ABSTRACT

The present paper responds to the operating behavior of an exemplarily selected screw expander for exhaust heat recovery from combustion engines in the lower power range. The application in vehicles for the purpose of heat recovery, which is characterized by the small machine dimensions due to relatively low system mass flows, is a still unexplored area for the screw-type machine. The screw expander is to be used in an organic Rankine cycle (ORC). Compared to other expander designs in general screw expanders are characterized by high efficiency together with relatively low geometrical dimensions. Within the presented investigation mainly the performance characteristic of a screw expander, geometrically designed for a particular operating point, is determined as a function of the system parameters – inlet pressure and temperature – as well as the rotational speed of the expander. Here, based on an iterative coupling of thermodynamic and thermal simulations the influence of the thermal deformation on the machine performance in particular is analyzed. The results of the thermodynamic simulation, mainly based on so-called chamber models, represent the thermodynamic and fluid dynamic performance of the screw machine by means of mass and energy conservation. The information obtained in this way about the temperature distributions and the heat fluxes provides a basis for the subsequent thermal simulation using a finite element method (FEM) calculation. The resulting thermally deformed machine, whose performance-related clearance heights are now changed, is used for the next iteration step within the thermodynamic simulation.

1. INTRODUCTION

An approach to a more efficient use of energy from fossil fuels in vehicles is the utilization of waste heat of combustion engines. This may occur, for example, in an organic Rankine cycle (ORC), where a working fluid, which is compressed, absorbs the heat of the exhaust gas via a heat exchanger and then provides usable energy in an expansion machine in the form of mechanical shaft work. An important component in this process is the expansion machine which can be realized in most different designs.

In this paper, the potential of the screw-type expansion machine in the organic Rankine cycle shall be discussed. In the framework of the machine analysis, the focus is on the determination of the characteristic diagram of a machine which is geometrically designed for constant system mass flow as a function of the system parameters inlet pressure and temperature and in dependence of the machine rotational speed. Based on an iterative coupling of
thermodynamic and thermal simulation calculations, in particular the influence of the machine’s thermal deformation and the resulting gap height change on the machine characteristics is analyzed. However, the knowledge of the changed gap heights in dependence of the operating point is not only interesting for the precise forecast of the engine’s operating behavior, but also for a statement regarding the operating safety. Thus the design of the gap heights of the “cold” machine plays an important role in the construction of a screw-type engine. Due to the thermal strain, the thermal interaction between operating fluid and component during operation has an indirect impact on the operating behavior of the engine. At the same time, due to the heat exchange between fluid and component during a working cycle it has a direct influence, too. Concerning this issue, the aspects regarding the design of dry running rotary displacement vacuum pumps has been described by Wenderott (2001), which is shown in Fig. 1.

2. THEORETICAL BASICS

For the utilization of waste heat in automotive applications, a Clausius Rankine cycle as a closed thermodynamic cycle shall be applied. When using the waste heat of a commercial vehicle, heat exchangers are integrated in the exhaust gas system to enable the utilization of the thermal energy of waste gas. The Rankine system consists of the key components evaporator, superheater where necessary, expander, condenser and pump. A heat exchanger extracts heat from the waste gas by evaporating the working fluid of the Rankine cycle. The evaporated and overheated working fluid reaches the expander, where the pressure and thus a part of the thermal energy of the working fluid is converted into mechanical energy and transferred to the expander shaft. The latter is linked to a generator which, for example, can be connected to the hybrid system of the vehicle. In the here provided ORC ethanol is used instead of water as working fluid which, in contrast to water, is characterized by more favorable thermodynamic properties, e.g. the lower evaporation enthalpy.

In previous research works, the potentials of expansion machines in displacement design have already been described and verified for different performance ranges. At the Chair of Fluidics, the application of a screw-type engine for utilization in a Clausius Rankine cycle was surveyed for the first time by Dreißig (1990) who gave a detailed description of the operating behavior. Later at the University of Dortmund, a prototype plant with a single-stage dry running screw-type steam engine was operated in the Clausius Rankine cycle. Screw-type expanders with fluid injection were examined by Zellermann (1996). The wide application range was surveyed and disclosed, for example, through the application of a screw-type expander in a GASSCREW (von Unwerth, 2002) or through the injection of pressure water with subsequent flash evaporation (Kliem, 2005). Apart from the prototype plant in Dortmund, a two-stage screw-type expander has been successfully applied in a biomass-fired combined heat and power unit in Hartberg/A since 2003. Currently, all commercially available screw-type expanders are based on screw-type compressors that are adapted in terms of design and energy consumption. At the Chair of Fluidics, the development of a single-stage screw-type expander for overheated water vapor and specified plant conditions is presently being pushed.

3. SIMULATION TOOLS AND PROCEDURE

The simulation calculations in this work are realized by iterative coupling of two simulation tools. In the framework of the theoretical investigations of the thermodynamic and fluid-mechanical operating behavior of screw-type expanders, a simulation program based on multiple chamber models (KaSim), developed at the Chair of Fluidics, is applied. The thermal simulation, thus the calculation of the temperature distribution in the components as well as the resulting thermal expansion, is carried out by the FEM program ANSYS. The automation capability is given due to the ANSYS Parametric Design Language (in short: APDL) based on which scripts can be generated that are able to be processed in an automated way (Müller and Groth, 2007).
3.1 KaSim
The simulation of rotary displacement machines on the basis of a chamber model represents a recognized method for their analysis and further development. The chamber model method is primarily based on the common characteristic of all displacement machines, on one or more cyclically changing working chambers. For the purpose of simplicity, it is assumed that spatial gradients in the intensive state variables are negligibly small and the fluid condition within a working chamber can thus be regarded as homogeneous. Based on this principle, the fluid condition is described by the extensive state variables of energy and mass and the related working chamber volume, Fig. 2. A thermodynamic change of the fluid state can occur e.g. by means of volume change, heat convections or mass and energy flows. The calculation by the time-stepping method is made under consideration of the law of conservation of mass and energy (Janicki, 2007).

The simulation program KaSim represents an implementation of the chamber model method and combines this with the advantages of object oriented programming. A fundamental distinction is made between capacities and connections. Capacities represent a storage for different physical manifestations, whereas connections enable an exchange between the capacities and, in their state, they are at any time defined by the related capacities, Fig. 2. The working chambers of the rotary displacement machine are an example for a finite fluid capacity with time-variant volume; the gaps are an example of a connection for the exchange of mass and enthalpy (Kauder et al., 2002).

3.2 ANSYS-KaSim Coupling
The structure-mechanical simulation of the screw-type engine is carried out by means of ANSYS. This simulation program is responsible for the meshing of the model and the calculation of temperature and displacement fields. The coupling of the two simulation programs KaSim and ANSYS requires the availability of suitable interfaces. This ensures the transfer of data and information between the two simulation programs. The structure of the interfaces will be explained in the following.

Structure and interconnection of the individual programs in the simulation cycle are shown in Fig. 3. MapKaSimHeatFlowToMesh is the interface from KaSim to ANSYS, while the three programs MapValuesToMesh, MapTemperatureToKaSim and MapDisplacementToKaSim are the interface from ANSYS to KaSim. A further program, ExitCondition, controls the simulation process and provides variable stop criteria.

The process demonstrates that basically three types of boundary conditions need to be exchanged between ANSYS and KaSim. KaSim calculates the thermodynamic processes of the working fluid during one working cycle. Based on a predetermined temperature distribution in the components and the model selected for a convective heat transfer in the working chamber or the gap, KaSim calculates the respective heat flux between the working fluid and the component. The heat flows need to be prepared in an appropriate manner and ANSYS must
be provided as the boundary condition for the ensuing calculation of the temperature distribution and the displacement field.

The output of ANSYS comprises a temperature distribution of the components and a displacement field caused by thermal stress. A temperature change in the areas that limit the working spaces has a direct impact on the heat transfer between component and working fluid, which causes changes in the heat flows. For this reason, component surface temperatures changed by KaSim need to be provided. The displacement fields influence the gap of the machine under consideration. The changes in gap height influence the flow cross-section and thus also the flow speed and mass flow inside the gap. Therefore, KaSim needs to be provided with changed gap dimensions. A detailed description of the interfaces is given in the following section. In this context, particular attention is paid to the physical-geometrical modeling of heat capacities and the changed gap dimensions as well as to the software interfaces that serve as a tool for information exchange.

The modeling of the heat capacities is outlined in Fig. 4 with the help of a female rotor. In general, the rotors are expected to have a heat flux gradient towards the z-axis from the high pressure to the low pressure side. Therefore, a so-called “disk-based” model is being developed, with the component along the z-axis being subdivided into several disks with regard to the XY-level. In this way, a heat flow curve depending on the z-coordinate can be determined; here the fineness of modeling can be varied through the number of disks. This classification is particularly suitable for rotating components, thus for rotors for which constant heat flow in circumferential direction is to be assumed. Since when modeling the heat capacities of the housing parts a distinction is also made between male and female rotor housing, it is also possible to map a heat flux towards the x-axis. In case of a disk-based approach however, the consideration of a temperature gradient in y-direction is not possible.

3.2.1 Interfaces of the heat flows KaSim-ANSYS: As a result of the KaSim calculation, the heat flow is output for each modeled heat capacity (“result_HT.ssd”). Independent of the modeling method, the heat fluxes must be processed in an appropriate way in order to be read in into ANSYS. This task is taken over by an interface which is listed in the program flow under the name MapKaSimHeatFlowtoMesh. For this purpose, the meshes of the components (“Bauteil1.ansys”, etc.) and the heat flows per capacity calculated by KaSim (“result_HT.ssd”) are required. Initially, the program extracts those surface meshes from the volume meshes which are in contact with the working chamber. Furthermore, all nodes from these surface meshes are allocated to the respective heat capacities. Fig. 4 shows the procedure of converting the heat flux of a heat capacity into an arbitrary finite element mesh. For this purpose, a disk-based modeling with a female rotor is used as an example.

The heat flow density \( \dot{q}_{HC} \) can be determined by means of the heat flux \( \dot{Q}_{HC} \) of the heat capacity calculated in KaSim and the surface content \( A_{HC} \):

\[
\dot{q}_{HC} = \frac{\dot{Q}_{HC}}{A_{HC}}. \tag{1}
\]

Thus the heat flow \( \dot{Q}_{\text{node}} \) at an arbitrary node of the finite element mesh can be determined by means of the heat flow density \( \dot{q}_{HC} \) and the area \( A_{\text{node}} \).
\[ \dot{Q}_{\text{node}} = \dot{q}_{HC} \cdot A_{\text{node}} = \dot{q}_{HC} \cdot \sum_{i=1}^{n} \left( \frac{A_i}{k_i} \right). \] (2)

The area \( A_{\text{node}} \) is calculated by adding the proportionate area of a node to the adjacent element area. In this case, \( n \) is the number of the adjacent element areas and \( k \) the number of the adjacent nodes of the respective element area. Therefore, an even distribution of the heat flows in ANSYS is possible, which is independent of the structure and the local fineness of the mesh. At this point, the program outputs a list of the node numbers with the related heat fluxes corresponding to a format which is readable for ANSYS (“heatflow.txt”).

3.2.2 Linking of the mesh data with the structural mechanical solution: The result of the ANSYS calculation (“temperature.ansys” and “displacement.ansys”) and of the subsequent data processing by the interface tool MapValuesToMesh represent the actual component mesh along with the temperature and the displacement vector of each node.

3.2.3 Interface of the temperature information ANSYS-KaSim: In order to enable a consideration of the temperature change of the heat capacity in KaSim, the new temperature of each capacity which results from the linked model with the node information needs to be calculated. Also in this case a conversion is necessary, since ANSYS outputs information per node while KaSim requires information per capacity. This task is taken over by an interface which is listed in the program flow under the name MapTemperatureToKaSim. The program in turn extracts those surface meshes from the volume meshes which are in contact with the working chamber and allocates the nodes to the respective heat capacities. The procedure of calculating the heat capacity temperature is similar to the procedure of MapKaSimHeatFlowToMesh. This consists of information on all node temperatures \( T_j \) weighted according to the element surfaces (see Fig. 4). Here, \( m \) is the number of finite element nodes of the heat capacity under consideration:

\[ T_{HC} = \sum_{j=1}^{m} T_j \cdot A_{\text{node}} \cdot \sum_{i=1}^{n} \left( \frac{A_i}{k_i} \right) \cdot \frac{A_{HC}}{A_{HC}}. \] (3)

In this way, it is possible to determine a heat capacity temperature which is independent of the structure and the local fineness of the finite element mesh. Similar to the MapHeatFlowToMesh program already described, here it is also a configuration file (“config_MTK”) that provides the necessary information, such as file name of the meshes, file name of the output and in case of a disk-based analysis, the subdivision of the machine by means of \( z \)-coordinates. This configuration file is generated and the user can make individual changes to the settings. The output of the program consists of an initialization file for KaSim, in which the temperature of each heat capacity is stored (“initHeatCaps.ssd”).

3.2.4 Interface of displacement information ANSYS-KaSim: At this point, the calculation of new gap heights by MapDisplacementToKaSim is addressed. Since, in contrast to the finite element mesh, the discretization for the description of the geometry is spatially finely resolved, the displacements of individual points need to be interpolated. For the interpolation of the displacement at a point \( P \), the finite element mesh, including the displacement vectors of all nodes, is available. The interpolation of the displacement occurs with the help of barycentric coordinates. An introduction in this topic is given, for example, by Farin and Hansford (2003). The method is explained in Fig. 5 (a) using a triangular surface element as an example. In the work flow, the surface element which encloses the point under consideration on the surface of the component is determined at first. The sketch in Fig. 5 (a) shows that point \( P \) can be mapped by using the triangulation points A, B and C. Therefore, the coefficients \( \alpha, \beta \) or \( \gamma \) are required which describe the position of point \( P \):

\[ P = A + \beta \cdot (B-A) + \gamma \cdot (C-A) \quad \text{with} \quad \alpha = 1 - \beta - \gamma. \] (4)

Accordingly, Eq. (4) can be simplified to the core equation of the barycentric coordinates:
\[ P(\alpha, \beta, \gamma) = \alpha \cdot A + \beta \cdot B + \gamma \cdot C \]  

(5)

With the help of Eq. (4), a linear equation system can be established without problems so that the coefficients \( \beta \) and \( \gamma \) can be determined. In this way, the Cartesian coordinates of point P can be converted into barycentric coordinates. The verification of the question described at the beginning, i.e. in which surface triangle point P is situated, can also be realized by simply using barycentric coordinates; it merely requires the examination of the algebraic signs. Thus, point P will be outside the triangle in case one of the coefficients \( \alpha \), \( \beta \) or \( \gamma \) is negative. Barycentric coordinates can be used not only to describe the position of point P, they also serve as a means for the linear interpolation of a function value of point P which is predetermined by the points A, B and C. In this case, the displacements at the nodes A, B and C can be used to calculate the displacement at point C:

\[ \vec{v}_P = \vec{v}_A + \beta \cdot (\vec{v}_B - \vec{v}_A) + \gamma \cdot (\vec{v}_C - \vec{v}_A) \]  

(6)

In case asymmetric deformation of two components occurs, the simple vector addition with the ensuing calculation of the new distance between the point pair does not seem to be appropriate for the determination of the new gap height. Therefore, a simplified approach is chosen, which is shown in Fig. 5 (b). At first, the vectors \( n_{P_1} \) and \( n_{P_2} \) are determined, which result from the respective point on the component surface and the point of the shortest distance on the opposite component. These vectors are unit vectors so that the new gap height \( h_{gap} \) along with the “cold” gap height \( h'_{gap} \) result in the following equation:

\[ h_{gap} = h'_{gap} - \left( n_{P_1} \cdot (\vec{v}_{P_1} - \vec{v}_{P_1}) + n_{P_2} \cdot (\vec{v}_{P_2} - \vec{v}_{P_2}) \right). \]  

(7)

In this way, it is possible to determine the height of each gap along its path. Based on the thus determined curve, the average height is calculated, which will be stored for a number of rotor positions predetermined by the user and saved in the chamber model. For the determination of the gap path in different rotor positions, the finite element mesh of the rotors with the displacement vectors must be turned around the rotation axis. Based on the described approach, negative gap heights and, respectively, contact between two components may be the result. The collision of the components may also be used as a stop criterion in the framework of an operational safety inspection.

4. EXEMPLARY EXPANDER AND BOUNDARY CONDITIONS FOR THE SIMULATION

The parameters of the exemplary screw-type expander are summarized in Table 1. In the run-up, the screw-type expander was designed based on an adiabatic calculation for a relatively low constant ethanol mass flow in the ORC. A decisive feature of the expander is its relatively high internal volume ratio \( v_i = 8.0 \). On the one hand, the utilization of high pressure gradients is aspired with the help of the large volume ratio. On the other hand, it is possible to increase the stroke volume and thus the machine dimensions by means of strong throttling and the reduction of density before the expansion starts. Otherwise the machine dimensions will remain limited because of the relatively low ORC mass flow and would considerably complicate the already demanding implementation of the design. Due to the selected internal volume ratio, the chamber filling extends over a relatively short rotation angle range of the male rotor from \( \alpha_{MR} = 76^\circ \) to \( \alpha_{MR} = 148^\circ \). The inlet pressure in the design point of the expander is about \( 40 \cdot 10^5 \) Pa at a temperature of \( 252^\circ \) C, the male rotor rotational speed is selected at \( n_{MR} = 20000 \) min\(^{-1}\). In this case, it is a synchronized screw-type machine, for which the adjustment of the two rotors towards each other is
realized by means of a gear pair. The operation of the screw rotors is contactless and the torque is transmitted by the synchronization gear.

Ethanol is used as working fluid because of its thermodynamic properties and advantages over water. In contrast to water, ethanol has a significantly lower evaporation enthalpy which, in turn, is accompanied by lower heat supply for evaporation. Furthermore, ethanol shows a low entropy range on the saturated vapor line so that falling below the saturated vapor line and the entry in the wet-steam region during expansion can already be avoided by moderate overheating. The maximum pressure in the system (pump outlet pressure) is varied in the range between 5·10⁵ Pa and 40·10⁵ Pa. The ethanol is overheated at a constant temperature of 252° C K independent of the inlet pressure. As a result, even at a maximum inlet pressure of 40·10⁵ Pa and in case of isentropic expansion the saturated vapor line would not be underrun and pure ethanol vapor will be expanded. Due to the presetting of constant condensation parameters (condensation temperature or pressure), the machine outlet temperature is kept at a constant level of about 70°C. The respective counter pressure amounts to 0.806·10⁵ Pa.

For the ensuing calculation of the component deformation due to thermal stress, adequate boundary conditions of the components must be predetermined in ANSYS. This requires the bearing positions of the real machine to be transferred to the model in a suitable manner. The predetermined boundary conditions for the simplified model are illustrated in Fig. 6. Since there is a fixed bearing on the shaft shoulders at the high-pressure side of the rotors, a boundary condition is set which does not permit displacements towards all three spatial directions. Position and width of the fixed bearings are determined by the position of the ball bearings. The fixed bearings have a relatively large distance to the rotor front side, since in-between, it is provided for a suitable sealing for the separation of the working chamber, which is flown through by ethanol, from the oil-lubricated gear/bearing chamber. For the floating bearing on the low pressure side of the rotors, a boundary condition is set which does not permit displacements as to the x- and y-coordinate direction. The housing is determined to be at the shaft outlet level with reference to the z-axis. Since the automated modeling for KaSim does still not allow displacements of the axis, a coupling boundary condition between the rotors and the housing is not predetermined in the framework of the ANSYS simulation. Another boundary condition is the imposition of a constant fluid temperature of 252° C in the nozzle at the high-pressure side of the machine. This corresponds to the inlet temperature of the ethanol in the thermodynamic simulation which always exists in the pressure nozzle of the expander. In the present case, different materials are used for housing and rotors. The material properties of 19Mn5 are chosen as rotor material, while the material properties of C45E are used for the housing.

### 5. INVESTIGATION RESULTS

In the following chapter, the results of the theoretical investigation of the operating behavior of the selected screw-type expander are explained. For this purpose, the inner power of the expander is discussed in the framework of an adiabatic thermodynamic calculation as a function of the conveyed mass flow. From a fluid-mechanical point of
The effect the gap height change, caused by the thermal deformation, has on the expander operation is demonstrated in Fig. 8 by means of the indicator diagram for a female rotor chamber (Fig. 8 (a)) and by the gap mass flows for a representative chamber during chamber filling and expansion (Fig. 8 (b)). Here, the results of the adiabatic calculation are exemplarily compared to those of the diabatic one after the completion of the deformation calculation for an inlet pressure of 40·10^5 Pa and a male rotor rotational speed of n_{MR} = 20000 min^{-1}. It is noticeable that the diabatic expander is always characterized by a higher pressure as a function of the male rotor angle. In both cases, a supercritical, blocked chamber filling and thus a relatively low differences in the expander mass flow can be assumed. For the diabatic case, the higher pressure level in the chamber and respectively the higher internal power
can be explained by the decrease of the integral gap mass flow (Fig. 8 (b)) for a representative working chamber during the chamber filling and the ensuing expansion. The various gaps show a different flow-through characteristic which tends to correspond to the gap height change. The gap mass flow is directly proportional to the size of the flown-through gap area, of the fluid density in the chamber at the high-pressure side and of the pressure ratio at the gap. Depending on the pressure ratio, the flow can be either subcritical or supercritical and, in the second case, there is no dependence of the gap flow on the pressure ratio.

The different gap mass flows for a constant rotation angle of the male rotor occur due to the different geometrical and thermodynamic conditions at the gap boundaries. The profile meshing gap and some of the front gaps create a direct connection between a chamber which is still filled or at a high pressure level and a displacing chamber on the expander outlet side, due to which the whole pressure ratio \( \Pi = p_i/p_o \) of the machine exists at these gaps. In view of the usually high pressure gradient, a blocked flow, i.e. a maximum gap mass flow, exists. In particular during the expansion in the working chamber, the profile and the rest of the front gaps connect two adjacent volumes that are still closed towards the low-pressure side. Consequently, a gradation of the pressure ratio at the gap and partly a smaller subcritical gap flow are the result. The second significant effect leading to an increase in pressure and power for the adiabatic expander is the heat flux which is integrally transferred from the housing to the working fluid. This heat flux results from the presetting of the temperature boundary condition in ANSYS, which corresponds to the inlet temperature of 252° C. The heat flows conveyed from the ethanol into the rotors can be assessed as negligibly low due to the minimal temperature difference between fluid and component temperature.

A quantitative analysis of the machine deformation is exemplary carried out in Fig. 9 with the help of the height change of a male and female rotor front gap on the expander high-pressure side for a rotor position \( \alpha_{MR} = 180^\circ \) relevant for the working chambers during the transient expander operation. It can be seen that the front gap height shows a significant decrease. In part, the front gap at both the male and the female rotor side becomes zero (iteration 5) during the start-up procedure, which can be ascribed to the expansion of the housing which, at the beginning, is faster than that of the rotors due to the immediate heat input into the housing caused by the preset temperature boundary conditions in the expander pressure port. The heat flux caused by the temperature difference between working fluid and the rotating components is solely responsible for the increase of the rotor.

**Figure 8:** Indicator diagram for a working cycle (a) of an exemplary female rotor chamber and gap mass flows of different gaps during the filling and the expansion of a representing expander working chamber (b)

**Figure 9:** Gap height change at an exemplary male and female front gap (hp) as a function of the number of iteration steps for \( \alpha_{MR} = 180^\circ \)
temperature and its ensuing expansion. As a result of the expansion in the closed chamber, the ethanol pressure as well as temperature is at a significantly lower level, which is the reason for the moderate rotor expansion. Under real conditions, this is synonymous with a contact of the rotors with the casing and thus with a machine failure. However, this effect could be supported by better temperature control of the housing or a moderate increase in the inlet temperature. Furthermore, in the context of the machine design, particular attention should be paid to the design of the machine gap to prevent a potential contact between the rotors and the casing. Therefore, the front gap height of 0.05 mm, chosen in the framework of this theoretical investigation, proves to be assessed as too small and shall be reviewed within the scope of the following design activities. In addition, constructive measures can be taken which include a reduction of the shaft shoulder provided for sealing purposes between the bearing positions and the rotor front. This shall help to minimize the influence of the different component expansions on the reduction of the front gap heights.

6. SUMMARY AND OUTLOOK

The investigations within the framework of this paper showed that a stationary, diabatic expander is characterized by significantly higher power than the adiabatic machine, which is mainly attributable to the decrease in the relevant gap heights. Another effect resulting in an increase in power is the heat input from the warmer machine housing into the working fluid. The relevance of the gap height change with regard to the operational reliability is illustrated by taking the front gap of the high-pressure side as an example. Due to the current state of the available software for the generation of chamber models, the coupled KaSim-ANSYS simulation calculations are carried out without taking into account the interaction of the displacements of the rotors and the housing, which may have a significant effect on the representation accuracy of the gap height changes. For this reason a suitable strategy in the framework of the chamber model generation must be developed. Another aspect to be considered in future works is the influence of heat transfer in the gap due to higher flow speeds and thus its higher heat transfer coefficients on the temperature distribution and expansion inside the screw-type expander.

NOMENCLATURE

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</table>

Subscripts

HC heat capacity
i inlet/inner/internal
MR male rotor
o outlet
REFERENCES


