Investigation into the effects of surface condensation in steam-driven twin screw expanders

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Abstract. During the operation of twin screw expanders with slightly superheated vapours or even two-phase fluids, surface condensation on machine parts occurs during the filling period and the expansion phase when the working fluid is in contact with cooler inner surfaces. This heat exchange from the working fluid to adjacent machine parts effects the working cycle and the efficiency of these machines. Short time scales and the periodicity of the process indicate the condensation process is best described by models for dropwise condensation. In this paper the effects of surface condensation on the operation of twin screw expanders are initially discussed in a simulation-based investigation. Chamber model simulation coupled with a thermal analysis is used for the thermodynamic simulation, whereby heat transfer coefficients are systematically varied. It is found that during the inlet phase condensate emerges on the inner surfaces of the machine being substantially cooler than the working fluid. This results in a higher mass being trapped within the working chamber and, thus, an increasing mass flow rate of the machine. An increase in power output is, however, not observed. The results obtained from chamber model simulations are finally compared against experimental data of a screw expander prototype.

1. Introduction
Internal leakage and throttling effects during the inlet and discharge phase are unavoidable losses reducing the overall efficiency of screw machines. Typically, in screw machine design those are kept to a minimum by optimising the machine geometry for specific operation parameters. The phase change of vaporous fluids within the working chambers of screw expanders represents a third impact factor on the overall efficiency of screw expanders, which needs to be considered during the design and simulation process. On the one hand, the changing fluid properties occurring when the binodal curve is crossed need to be considered, while on the other, the condensation heat transfer impacts the general energy conversion during the working cycle. It has been shown that non-equilibrium effects in the formation of fog during the expansion phase only have a minor influence on the integral expander parameters [1, 2]. However, a theoretical study indicates that the high heat transfer coefficients in surface condensation result in notable amounts of condensate being trapped within the working chambers during the filling period, which substantially increases the mass flow rate of the screw expander while the power output remains unchanged [3]. This effect will be investigated further by means of chamber model simulation with an enhanced model in this study.
2. Surface condensation
Condensation generally describes the phase change of a fluid from gas to liquid, whereby a significant release of latent heat occurs. This process is further differentiated into surface condensation and spontaneous condensation, where in the latter, a dispersed homogeneous phase emerges within the fluid. Condensation phenomena on surfaces, which are the main subject of discussion in this paper, are characterised by the accumulation of droplets or a condensate film, which wet the cooler surface, and the transfer of latent heat from the fluid to the adjacent solid.

Surface condensation occurs when a vaporous fluid is in contact with a cooler wall whose temperature is below the saturation temperature of the fluid in thermodynamic equilibrium. In contrast to spontaneous condensation, where the latent heat of the fluid is released to the surrounding gas phase, the latent heat is transferred through the wall. If a condensate film is formed, the condensate layer acts as an additional thermal resistance and phase transition of the vapour takes place on the vapour-liquid interface. If the surface conditions prohibit complete wetting, the solid is only partially covered with condensate droplets, which grow in time. Due to external forces or shear stresses, droplets will eventually move along the wall while absorbing other droplets along their way. This mechanism again creates a partially dry surface with a subsequent re-initialisation of the dropwise condensation process. In general, maximum heat transfer is reached in dropwise condensation.

Fundamental work on film condensation was carried out by Nußelt, who presented an analytical model for laminar filmwise condensation on a vertical wall [4]. His model was later improved by consideration of the temperature-dependency of film fluid states [5], formation of waves on the film surface [6], and film turbulence [7]. An approach to modelling the improvement of heat transfer in cocurrent flows of condensate and vapour was presented by Numrich [8]. Condensation heat transfer coefficients are typically determined from dimensionless heat transfer equations based on empirical studies. Since the physical mechanisms and, thus, the choice of the correct empirical formulation, highly depend on the mode of the flow, Tandon et al. proposed a flow regimes map for condensing two-phase flow [9].

It has been found that film condensation is the predominant effect in condenser applications. However, in the case of low surface energy of the solid or when non-wetting layers or hydrophobic materials are added to surfaces, dropwise condensation, characterised by heat transfer coefficients up to ten times higher than for film condensation, can be achieved [10].

Figure 1: Coupled simulation process with chamber model simulation (left) and thermal analysis (right).
One model proposed postulates that vapour condenses on a microscopic film between droplets, which act as condensate collectors. Due to its small length scale, the effect of the thermal boundary layer is negligible; this explains the extremely high heat transfer [11]. Other models are based on nucleation processes in surface cavities. However, the question of whether or not a thin film forms between droplets is as yet unresolved. Furthermore, an integral heat transfer equation explaining all tendencies of published experimental data has yet to be developed [12].

### 3. Modelling and simulation

In this study an unsynchronised dry prototype screw expander is investigated, and chamber model simulation is used for the calculation of thermodynamic characteristics. Heat exchange between the working fluid and machine parts as well as thermal conduction in the rotors and the housing are included in the simulations. Thermal deformation is, however, not included. Since the accurate calculation of chamber states requires for small time steps while changes in rotor and housing temperatures develop over a longer period, the chamber model simulation and the thermal analysis of the solids are executed in separated co-simulations. Figure 1 shows the coupled simulation process. The non-commercial simulation tool used in this study was developed at the Chair of Fluidics at TU Dortmund University.

In chamber model simulation the working chambers of positive displacement machines are considered as non-dimensional control volumes (fluid capacities) that vary throughout the working cycle. Solid heat capacities represent the machine parts. Heat or mass flow connections between those capacities allow the exchange of mass and energy. The geometries of capacities and connections are typically described as a function of the male rotor angle. Properties of the capacities and flows are calculated at discrete rotor angles iteratively [13]. Although this non-dimensional approach lacks information on the three-dimensional flow within the screw machine, high simulation quality can be reached with detailed sub-models for the behaviour of connections and capacities, respectively.

In this study a two-chamber model is applied, which distinguishes between male and female chambers. For each small time step the pressure-volume work \( W \) is calculated initially for each fluid capacity at a constant specific entropy \( s \). It is determined as follows with the static pressure \( p \) and the chamber volume \( V \).

\[
W = \left( -\int p \, dV \right)_s
\]  

(1)

Mass flows between two fluid capacities through a connections with the cross-sectional area \( A_f \) are considered as quasi-stationary orifice flows with the flow coefficient \( \alpha_f \) being the ratio of the actual mass flow rate \( \dot{m}_{FF} \) to the isentropic mass flow rate. The isentropic density \( \rho_s \) and difference in specific enthalpy \( \Delta h_s \) are calculated from the source entropy and the sink static pressure. Heat transfer in clearances is not taken into account in this work. The calculation routine implemented in the simulation tool additionally checks for choked flows and eventually balances pressures. If required, fluid states of preceding chambers are determined from the previous iteration. Connections to succeeding chambers are considered according to the periodicity of working cycles.

\[
\dot{m}_{FF} = \alpha_f \dot{m}_s = \alpha_f A_f \rho_s \sqrt{2 \Delta h_s}
\]  

(2)

The heat exchange \( \Delta Q_{FS} \) between fluid capacities and solid heat capacities through a surface \( A_h \) is calculated with the heat transfer coefficient \( \alpha_h \), the temperature difference \( \Delta T \), and the time increment \( \Delta t \). The individual temperature of each solid heat capacity (housing or rotor segment) is held constant during the chamber model simulation.

\[
\Delta Q_{FS} = \alpha_h A_h \Delta T \Delta t
\]  

(3)
The chamber states are calculated for every time step based on changes in internal energy $U_F$ and mass $m_F$, respectively. The integral energy conversion for the control volume is hence denoted as follows with $\Delta H_{t,F}$ being the change of total enthalpy within the fluid capacity due to mass exchange. Heat increments exchanged between the fluid and solid heat capacities (index $i$) and mass flows through clearances and machine openings (index $j$) are individually calculated for all connections in every time step.

$$\Delta U_F = \Delta W + \sum \Delta Q_{FS,i} + \Delta H_{t,F} = \Delta W + \sum \Delta Q_{FS,i} + \sum \Delta t \dot{m}_{FF,j} h_{t,j}$$

(4)

The convective heat exchange per solid heat capacity (averaged over the working cycle) serves as boundary condition for the coupled thermal analysis, which includes thermal conduction between the solid segments. Figure 2 shows the sliced model of the rotors and housing, respectively. The separation of the housing in helical elements is based on the assumption that the temperature gradients of the housing correlate with those of the working fluid. Due to the high rotational speed, the temperature field of the rotors is assumed to develop in axial direction from the high-pressure plane to the low-pressure plane. The separation of the housing and rotors according to the expected temperature fields is supported by Buckney’s results for time averaged temperature distributions, who mapped cycle temperatures on housing and rotor surfaces of screw compressors, however, without taking into account conduction within the materials [14].

The average heat transfer coefficient and the average temperature of the working fluid being in contact with a solid heat capacity are transferred from the chamber model simulation to the thermal analysis. The conduction of heat $Q_{SS}$ between solid heat capacities through a contact area $A_h$ is calculated with Fourier’s law, where $D$ is the centerline distance between two segments.

$$\Delta Q_{SS} = \frac{\lambda A_h \Delta T \Delta t}{D}$$

(5)

Temperature changes of the solid heat capacities are calculated based on the conservation of energy with $c$ being the material’s specific heat capacity and $m$ being the mass of the capacity.

$$\Delta T_S = \frac{\Delta U}{c m} = \frac{\sum \Delta Q_{SS,i}}{c m}$$

(6)
Table 1: Expander data

| characteristic       | value       |
|----------------------|-------------|
| rotor profile        | SRM A       |
| number of lobes      | 4 (male) / 6 (female) |
| maximum chamber volume | 26254 mm$^3$ |
| built-in volume ratio | 4.0         |
| male rotor tip diameter | 58.9 mm    |
| axis to axis distance | 46.2 mm     |

The thermal analysis routine is executed for a defined period of time after which the chamber model simulation is re-initiated with the latest rotor and housing temperatures. A state of overall convergence is achieved when a steady state is detected in the thermal analysis.

3.1. Simulation parameters

The calculation model presented in this study requires both numerical and machine parameters. Fluid states are calculated in thermodynamic equilibrium, and the fluid model selected is Pollak’s fundamental equation of state [15]. The working cycle is discretised with a defined angle step of $\Delta \varphi = 0.03^\circ$. For the thermal analysis and modelling of heat transfers the rotors are equally separated along the rotational axis into 10 segments each, whereas the male and female housing are separated into helical elements. Considering the greater temperature range, a slightly finer separation is selected. Here, width of the helical elements is equivalent to half of the corresponding rotor separation angle, which results in 11 male housing segments and 14 female housing segments. Relevant information on the machine geometry is shown in table 1. The flow coefficients for internal leakage paths and openings to the inlet and outlet ports are $\alpha_f = 0.8$ for all calculations in this study. Clearance heights are in the range from 0.05 to $0.08 \cdot 10^{-3}$ m.

3.2. Heat transfer in surface condensation and boiling

Heat transfer coefficients are required as input parameters for the chamber model simulation in order to calculate the heat exchange between the working fluid and the solid heat capacities. A discussion whether or not condensation models documented in the literature are applicable to twin screw expanders is provided in [3]. The well-established empirical models do not necessarily meet the complex flow situation and the highly unsteady operation of twin screw expanders. Due to this uncertainty, heat transfer coefficients are systematically varied in the calculations presented in this study.

The mode of surface condensation (dropwise or filmwise) needs, however, to be considered in order to choose adequate values of heat transfer coefficients. It is assumed that the initial mechanisms in filmwise condensation are identical to those in dropwise condensation [16, 17]. Wenzel indicates the period for the formation of primary droplets being $10^{-4}$ to $10^{-3}$ s, which is comparable to the expander cycle time [18]. Since accumulated condensate on the inner machine surfaces is expected to be entrained by the high-velocity leakage flows or to be evaporated at a later point during the working cycle, it can be assumed that the condensation process is re-initiated with every working cycle. Under these assumptions, the process of surface condensation in twin screw expanders seems to be best represented by the models for dropwise condensation. The magnitude of heat transfer coefficients is estimated based on Rose’s correlation, which calculates the heat transfer coefficient in dropwise condensation based on the subcooling of the surface [19].
Moreover, the temperatures of the housing and rotor segments strongly depend on the cycle-averaged temperatures of the fluid. Consequently, situations will occur during the working cycle where the wet fluid is in contact with warm machine surfaces, which will result in boiling. Provided that these situations occur during the late expansion phase and the discharge period, this re-evaporation of condensate has little influence on the operational behaviour of the twin screw expander. It can be expected that heat transfer in boiling is in the magnitude of that in surface condensation, this being the reason why heat transfer coefficients are held constant throughout the working cycle [20].

4. Case study

In the following a case study is presented in order to show the general effects of surface condensation on the operation of an exemplary twin-screw expander. The objective is to determine the time frame within the working cycle in which surface condensation occurs and to identify the corresponding heat transfer surfaces. The findings in this section refer to an inlet pressure \( p_{in} = 4 \times 10^5 \) Pa, outlet pressure \( p_{out} = 1 \times 10^5 \) Pa, expander speed \( n = 8000 \text{ min}^{-1} (u = 24.67 \text{ m s}^{-1}) \). The working fluid is superheated steam with \( \Delta T_{sh} = 10 \) K. The condensation heat transfer coefficient is set to \( \alpha_h = 1 \times 10^5 \text{ W m}^{-2} \text{ K}^{-1} \).

Figure 3 compares the indicator plots of the diabatic simulation, where surface condensation is taken into account, and the adiabatic simulation. In both cases the pressure history shows similar characteristics. The chamber pressures during the filling period almost reach inlet conditions, which is caused by moderate inlet throttling due to the low expander speed. In the late expansion phase leakages from succeeding chambers create a minor recovery effect. The chamber pressure of the adiabatic process in general develops on a slightly higher level. The heat exchange during the inlet and expansion phases causes an additional pressure drop to that driven by pressure-volume work. However, despite the high magnitude of heat exchange these differences are small, which results in the inner power being only 4.2% smaller compared to the adiabatic calculation. In contrast to that, the mass flow rate is increased by 9.7%. During the filling period, condensate emerges on the machine surfaces when the temperatures of the solids are below the fluid’s saturation temperature. As long as the chamber is still connected to the high pressure pipe via the inlet port, this loss in the vapour phase is compensated by an increased inflow. Consequently, a higher mass is trapped within the chamber at the end of the

Figure 3: Comparison of indicator plots for adiabatic process and process with surface condensation taken into account.
Figure 4: Plot of fluid temperature, temperatures of female rotor segments, and vapour mass fraction.

filling period in the diabatic simulations.

Figure 4 shows the corresponding temperature plots of the fluid and the female rotor segments. Even the rotor segment nearest to the high-pressure plane is in contact with the working fluid almost throughout the complete expander cycle. Significant temperature differences between the working fluid and the rotor segments occur during the filling period and the expansion phase, causing an almost instantaneous onset of surface condensation. Moreover, the transfer of latent heat in condensation or evaporation (during the late expansion phase, when surface temperatures are greater than the saturation temperature of the working fluid) cause no significant warming or cooling of the fluid whereby temperature differences remain great. The surface condensation of steam during the filling period and expansion phase results in the vapour mass fraction being smaller than one. During the later working cycle, when the fluid is in contact with warmer machine parts, the vapour mass fraction increases and reaches unity.

The temperature distribution in the rotors and the simplified housing segments is depicted

Figure 5: Temperature distribution in machine parts.
in Figure 5. The temperatures on the male rotor closely match those of the female rotor. Other than the rotating solid heat capacities, the housing segments are assigned to a relatively short period of the working cycle. Consequently, the housing temperature distribution correlates well with the temperature of the working fluid. Small differences between segment temperatures and averaged fluid temperatures indicate the high heat exchange in surface condensation (and evaporation) dominating over conduction.

Since the thermal analysis is based on the conservation of energy and heat exchange to the ambient is neglected, which is feasible considering the magnitude of heat transfer in condensation and boiling, it can be differentiated between condensation surfaces and evaporation surfaces. In particular, surface condensation is expected to occur on the rotor surfaces during the filling period and expansion phase. During the discharge phase rotor temperatures are for the most part greater than the fluid temperature, causing the previously emerged condensate to re-evaporate. Similar effects are observed on the housing surfaces. However, the temperature differences between the working fluid and the housing surfaces are smaller.

5. Operational behaviour and validation against experimental data
In addition, the operational behaviour of the screw expander prototype is investigated by means of the coupled chamber model simulations. Figure 6 shows the integral results of the calculations compared to experimental data. The absolute values of inner power calculated increase with expander speed. Discrepancies between the inner power simulated and the effective power obtained from the experiment result from mechanical losses (friction) not being considered in the calculations. The deviation increases, as expected, with expander speed. The influence of surface condensation (during the inlet period and the expansion phase) on the inner power calculated increases with expander speed.

The mass flow rate is linearly increased with expander speed. The calculations not considering heat exchange between the working fluid and machine parts clearly underestimate the experimental data. The accordance between the simulations and the experimental data is enhanced when taking into account the heat transfer between working fluid and machine parts and, consequently, the condensation-induced increase in mass flow rate. Values calculated with a heat transfer coefficient $\alpha_h = 4 \cdot 10^5 \text{ W m}^{-2} \text{ K}^{-1}$ adequately represent the operational behaviour of the screw expander investigated.

![Figure 6](image_url)  
**Figure 6:** Characteristic diagram with power (black lines) and mass flow rates (grey lines). $p_{in} = 4 \cdot 10^5 \text{ Pa}$, $p_{out} = 1 \cdot 10^5 \text{ Pa}$, $\Delta T_{sh} = 10 \text{ K}$. 
6. Conclusion

Twin screw expanders operating with moderately superheated steam are likely to be affected by surface condensation. During the filling period and the early expansion phase the inner surface temperatures are below the steam’s saturation temperature, which leads to emerging condensate on these surfaces. In particular the significant temperature differences between the fluid and the rotor surfaces as well as the high heat transfer coefficients result in condensate being trapped in the working chamber at the end of the filling period, thus increasing the mass flow rate of the machine. A higher power output is, however, not to be expected.

The thermal analysis indicates the heat transfer in the initial condensation and the subsequent evaporation of liquid, when inner surface temperatures are greater than the fluid temperature, dominating over conduction in the machine parts. The short cycle time of twin screw expanders and the flow pattern within the working chambers suggest the mode of surface condensation being dropwise condensation.

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## List of symbols

| Symbol | Unit       | Property                      | Subscript | Property     |
|--------|------------|-------------------------------|-----------|--------------|
| $A$    | m$^2$      | area                          | $f$       | fluid flow   |
| $c$    | J kg$^{-1}$| specific heat capacity        | $F$       | fluid capacity|
| $D$    | m          | distance                      | $h$       | heat flow    |
| $h$    | J kg$^{-1}$| specific enthalpy             | $i$       | index: heat exchange |
| $H$    | J          | enthalpy                      | $j$       | index: mass exchange |
| $m$    | kg         | mass                          | $in$      | inlet        |
| $\dot{m}$ | kg s$^{-1}$ | mass flow rate                | $out$     | outlet       |
| $n$    | s$^{-1}$   | expander speed                | $s$       | isentropic   |
| $p$    | Pa         | pressure                      | $S$       | solid heat capacity |
| $Q$    | J          | heat                          | $sh$      | superheated  |
| $s$    | J kg$^{-1}$ K$^{-1}$ | specific entropy            | $t$       | total        |
| $t$    | s          | time                          |           |              |
| $T$    | K          | temperature                   |           |              |
| $u$    | m s$^{-1}$ | tip speed                     |           |              |
| $U$    | J          | internal energy               |           |              |
| $V$    | m$^3$      | volume                        |           |              |
| $p$    | Pa         | pressure                      |           |              |
| $W$    | J          | pressure-volume work          |           |              |
| $\alpha_f$ | –   | flow coefficient               |           |              |
| $\alpha_h$ | W m$^{-2}$ K$^{-1}$ | heat transfer coefficient     |           |              |
| $\Delta$ | –   | difference                     |           |              |
| $\lambda$ | W m$^{-2}$ K$^{-1}$ | thermal conductivity         |           |              |
| $\phi$ | $^\circ$   | rotor angle                   |           |              |
| $\rho$ | kg m$^{-3}$ | density                      |           |              |