Effects of surface condensation in an idealised steam-driven screw expander

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Abstract. Condensation and its effects on turbo machinery operation are well understood and have been widely investigated. However, little scientific work on condensation in rotary positive displacement machines has been published. Since the robustness of screw expanders allows for expansion of slightly superheated vapours or even two-phase fluids, condensation on machine parts needs to be considered during design and simulation of these machines. In this paper the general effects of surface condensation of water on the machine parts of an idealised screw expander are discussed. Diabatic chamber model simulation is used for the thermodynamic simulation of operational behaviour. The effect of surface condensation on energy conversion and the delivered mass flow rate is analysed. Furthermore, a comparison of adiabatic and diabatic simulation of steam expansion in screw expanders is given in order to quantify condensation losses. Typical operating parameters are widely varied in simulation so as to identify influential factors on the condensation process. It is found that surface condensation, which is driven by heat exchange from the working fluid to adjacent machine parts, slightly raises the mass flow rate of the machine. For low expander speeds a reduction in isentropic efficiency can be expected due to a condensation induced pressure drop during the expansion phase.

1. Introduction
Rotary twin screw machines are robust and are therefore commonly operated with two-phase working fluids such as gas-oil mixtures in compressor applications. Due to the design similarities of screw compressors and expanders, the presence of condensate in the working chambers of expanders is not an issue for reliable machine operation; this allows expansion of vaporous fluids with minor superheating or even wet vapours. Thus, utilising screw expanders for power generation in Rankine cycles with low temperature heat sources is a promising concept. However, for performance prediction and correct design, it is necessary to take condensation into account in simulations. It has been found that screw expanders can be affected by the effects of spontaneous condensation. However, theoretical considerations indicate the impact on energy conversion is relatively small [1]. In this study, a simulation based investigation into the general influence of surface condensation in a steam-driven screw expander is reviewed.

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, e.g. formation of clouds. 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 [2]. His model was later improved by consideration of the temperature-dependency of film fluid states [3], formation of waves on the film surface [4], and film turbulence [5]. An approach to modelling the improvement of heat transfer in cocurrent flows of condensate and vapour was presented by Numrich et al. [6]. Condensation heat transfer coefficients are typically determined from dimensionless heat transfer equations based on empirical studies. Since the physical mechanisms, and thus 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 [7].

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 [8]. One model proposed postulates that vapour condenses on a microscopic film between droplets, which acts as condensate collector. Due to its small length scale, the effect of the thermal boundary layer is negligible; this explains the extremely high heat transfer [9]. 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 [10].

3. Diabatic simulation of rotary positive displacement machinery
Chamber model simulation has been found to be an appropriate method for the thermodynamic simulation of rotary positive displacement machines. In such simulations, geometry (e.g. chamber volume history, machine port cross-sectional area, leakage paths) as a function of the angular rotor position is provided as input to a non-dimensional model. Chamber states for each time step are determined on the basis of the conservation of mass and energy, where transport processes of energy and mass are calculated with a quasi-stationary approach.

Keller investigated thermal deformation of the rotors and casing of screw expanders, and proposed a model to calculate heat exchange between the working chambers and the surrounding solids [11]. A representative velocity was derived from the machine speed and rotor geometry; this approach allowed use of dimensionless heat transfer correlations for turbulent gas flow. Von Unwerth extended the model by considering convective heat transfer for the casing of screw expanders [12]. In addition, Rohe used plane wall correlations to calculate the heat exchange between the working fluid and machine parts of screw vacuum pumps [13]. On the basis of these investigations, Hütker determined related losses and their effect on efficiency ratings of
discretisation of machine geometry (constant $\Delta t$)
pressure-volume work (constant $s$) $\rightarrow p, T, v, u$
external mass exchange (leakage, ports) $\rightarrow \Delta m, H_{in}, H_{out}$
external heat exchange $\rightarrow \Delta U$
calculation of chamber states $\rightarrow p, T, v, u, s$
calculation of integral results $\rightarrow P, m$

**Figure 1.** Process for diabatic chamber model simulation.

screw expanders. Losses investigated included external heat loss and increasing leakage caused by thermally induced changes of clearance heights [14]. However, these investigations were performed for dry-running screw machines, and, consequently, the proposed heat exchange models are only valid for single-phase gaseous fluids.

**4. Simulation model**

In this investigation, chamber model simulation is used to analyse a screw expander process with external heat exchange. An idealised expander cycle is investigated in order to show the full impact on operation and to isolate surface condensation from typical loss effects such as leakage and throttling. The following changes of state are postulated for the idealised cycle: an isobaric inlet phase at inlet pressure, an expansion phase with constant chamber mass, and an isobaric discharge phase at a given back pressure. In the case where heat transfer is suppressed, the idealised process becomes isentropic. Thermodynamic properties are provided from a fundamental equation of state where the fluid states are calculated from an empirical formulation of Helmholtz free energy.

In this theoretical study, the exchanged heat $Q$ between the working fluid and surrounding machine surfaces $A$ within one simulation time step $\Delta t$ is considered during the inlet and expansion phases by applying heat fluxes $\dot{q}$ corresponding to typical surface condensation modes. $\alpha$ is the heat transfer coefficient, and $\Delta T$ is the temperature difference between fluid and solid.

$$\dot{q} = \frac{Q}{A \Delta t} = \alpha \Delta T$$

In dimensionless notation heat transfer is expressed with the Nusselt number $Nu$, where $L$ is a characteristic length and $\lambda$ is the thermal conductivity of the fluid.

$$Nu = \frac{\alpha L}{\lambda}$$

The calculation routine for the diabatic chamber model simulation is depicted in **figure 1.** Changes of state in the working chamber are calculated in two successive steps. Initially, an isentropic expansion with constant trapped chamber mass $m$ is performed. Based on the isentropic change of states, mass inflow and outflow as well as heat transfer are calculated; these lead to changes in specific internal energy $u$ and specific volume $v$. The final chamber states
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                    |

are ultimately calculated from the equation of state with those inputs. For the idealised filling period at constant inlet pressure, the inlet mass increment is iteratively calculated. Indicated work $W_i = -\oint p \, dV$ and indicated power $P_i = n \, \varepsilon_{male} \, W_i$ are later determined by the integration of pressure history with respect to the chamber volume. The basic principle of chamber model simulation and formulations for the conservation of energy and mass are well documented, for instance, in [15].

4.1. Simulation parameters and machine geometry
The calculation model presented in this study requires both numerical and machine parameters. The expansion phase is discretised with a defined time step. A sensitivity analysis with regard to indicated power $P_i$ implies $\Delta t = 1.0 \cdot 10^{-6} \, s$ is sufficiently small for the parameter range under consideration. The fluid model selected is Pollak’s fundamental equation of state; fluid states are calculated in thermodynamic equilibrium [16].

Relevant information on the machine geometry, which determines the volume curve and links tip speed with rotational speed, is shown in Table 1. The corresponding plots of chamber volume and heat transfer areas to the rotors and casing are depicted in figure 2. It can be noted that heat transfer areas on the male rotor side are, in general, greater than on the female rotor side, but areas connecting the female chamber to the female rotor emerge at an earlier time. In addition, with the male rotor operating at higher speeds, greater heat transfer coefficients can be expected. This imbalance of thermal load is further amplified since the surface connection from the female rotor chamber to the male rotor is greater than vice versa.

5. Expander cycle with surface condensation
Initially, the general influence of surface condensation on an idealised expander cycle is discussed. There are two fundamental differences in comparison of heat exchange between working fluid and machine parts in dry-running screw machines with single-phase fluids. On the one hand, the release of latent heat in close proximity to the heat transfer surface induces significantly higher heat transfer coefficients. On the other hand, condensation includes mass transfer from the gaseous to the liquid phase, where, due to the difference in density, a substantial part of the trapped mass might be accumulated in a small liquid volume within the chamber. This becomes obvious when plotting the vapour volume fraction $\epsilon$ of wet steam over the steam quality, figure 3. Even with a substantial reduction of steam quality, by far the largest part of the chamber volume is still filled with vaporous fluid.

The conservation of energy for a working chamber of the screw expander, neglecting gravitation effects, can be expressed as follows:

$$\Delta U = Q + W + H_{t,in} - H_{t,out}$$  \hfill (3)
Surface condensation is always combined with heat transfer to the surrounding surface \((Q < 0)\), thus reducing internal energy of the fluid within the working chamber. For an idealised process, the trapped chamber mass remains constant during the expansion phase. Therefore, condensation induced heat transfer results in an additional pressure drop. In order to quantify this effect, the partial derivative of pressure with respect to specific internal energy at constant specific volume (constant trapped mass and constant chamber volume) is evaluated:

\[
\left( \frac{\partial p}{\partial u} \right)_v
\]  

The result is depicted in figure 4. The impact of surface condensation on pressure history and, in turn indicated work, is potentially higher at greater pressure and for lower steam quality. When comparing the effect of heat output from the working chamber on the pressure drop in the two phase region and for superheated fluid states, greater sensitivity can be expected for moderately superheated steam. The general influence of heat outflow on chamber pressure decreases again for well superheated states.

During the inlet phase, when the working chamber is, by definition, not a closed thermodynamic system, surface condensation decreases the specific volume of the working fluid and hence raises the expander mass flow. However, due to the high condensation enthalpy, this effect requires extremely high heat output for a considerable impact. The heat loss induced pressure drop during the inlet phase is eventually compensated for by a higher inflow.

A comparison between the idealised indicator plot of an adiabatic working cycle and that of a diabatic working cycle for the selected twin screw expander is given in figure 5. Here, the heat flux is set to an exemplary value. Obviously, only minor differences in pressure history are observed. Despite a substantial amount of heat being transferred from the working fluid to the casing and rotors, a notable drop in chamber pressure is only seen late in the expansion phase. Consequently, the (ideal) indicated work of the machine is only slightly reduced. Since the expander is fed with saturated steam, heat losses during the inlet phase reduce steam quality in the working chamber. The resulting reduction of the specific volume increases the amount of fluid being trapped at the start of expansion, which in turn increases the mass flow rate. However, due to the high specific latent heat of water, high heat outflow is required for a
Figure 5. Comparison of indicator plots for adiabatic and diabatic idealised expander cycle, \(p_{in} = 6 \cdot 10^5\ \text{Pa}, \ p_{out} = 1 \cdot 10^5\ \text{Pa}, \ T_{in} = 431.98\ \text{K}\) (sat. steam), \(u = 60\ \text{m s}^{-1}, \ \dot{q} = 6 \cdot 10^5\ \text{W m}^{-2}\).

Figure 6. Virtual thickness of condensate film during inlet and expansion phase, \(p_{in} = 6 \cdot 10^5\ \text{Pa}, \ p_{out} = 1 \cdot 10^5\ \text{Pa}, \ T_{in} = 431.98\ \text{K}\) (sat. steam), \(u = 60\ \text{m s}^{-1}, \ \dot{q} = 6 \cdot 10^5\ \text{W m}^{-2}\).

substantial increase in mass flow rate. At this exemplary operating point, the simulated mass flow rate is only increased by 1.2% compared to the adiabatic idealised expander cycle.

In order to quantify emerging liquid during expansion, a virtual film thickness \(\delta_v\) is introduced. This is the thickness of the film that would be formed if all of the trapped condensate volume \(V_c\) were uniformly distributed over the heat transfer surfaces.

\[
\delta_v = \frac{V_c}{A} = \frac{(1 - \epsilon)V}{A}
\] (5)

Here, \(\epsilon\) is the vapour volume fraction. The characteristic of this virtual film thickness vs. chamber volume for the inlet and expansion phases is plotted in figure 6. The absolute value of heat flow in this exemplary simulation is \(-11.4 \cdot 10^3\ \text{W}\), which is equivalent to 68.6% of the machine’s isentropic power. Nevertheless, the vapour volume fraction being close to unity results in small values of virtual film thickness, and only little interaction between the emerging condensate and the machine parts is expected (e.g. seal of clearances).

From this theoretical consideration, two effects of surface condensation on the working cycle of a screw expander are identified: the reduction of the specific volume of the fluid during the inlet phase, and the heat transfer induced pressure drop during expansion. In addition to the influence of fluid properties, both mechanisms are driven by the amount of heat being transferred from the working fluid to the surrounding surfaces. Here, measurements of surface areas, which depend on the rotor dimension and geometry, temperature differences between working fluid and machine parts, which, to some extent, can be affected by insulation or cooling, the duration of the working cycle, which results from machine speed, and the heat transfer coefficient, which is mainly affected by the flow mode within the expander, are the main factors impacting heat outflow. In the subsequent theoretical study, the magnitude of the heat flow is systematically varied by altering heat fluxes \(\dot{q} = \alpha \Delta T\). Time dependence is considered by altering machine speed.

6. Variation of expander parameters
The influence of decreasing specific volume during the inlet phase due to heat outflow can be quantified by the delivery rate \(\lambda_L\), which is the ratio of actual mass flow rate \(\dot{m}\) to the theoretical
mass flow rate $\dot{m}_{th}$. The theoretical flow rate is calculated from expander speed $n$, number of lobes on the male rotor $z_{male}$, the chamber volume at the beginning of expansion phase $V_{ex}$ and the specific volume at inlet conditions $v_{in}$.

$$\lambda_L = \frac{\dot{m}}{\dot{m}_{th}} = \frac{\dot{m} v_{in}}{n z_{male} V_{ex}}$$  \hspace{1cm} (6)

The impact of surface condensation on the delivery rate of an idealised diabatic expander cycle is plotted in figure 7 for various inlet pressures and expander speeds. Despite choosing high absolute values of heat flux for the simulation, the specific volume at the start of expansion is only slightly affected. Consequently, the impact on delivery rate is moderate for all operating points displayed. In addition to the influence of expander speed, which correlates with the absolute value of heat being transferred from the working fluid to the adjacent solids, the delivery rate increases with decreasing inlet pressures. Although the specific latent heat of steam is lower for higher pressures, the mass trapped in the chamber is dependent on the steam density, which decreases with pressure. Consequently, with the boundary condition of a constant heat flux (heat outflow is proportional to expander speed), the relative change in specific internal energy is greater for lower pressures.

The increasing delivery rate in combination with the surface condensation induced pressure drop (reduced indicated work) during expansion result in a reduced isentropic efficiency $\eta_s$ of the machine under consideration. The isentropic power $P_s$ is calculated from the actual mass flow rate and the specific isentropic enthalpy difference $\Delta h_s$. The isentropic efficiency can furthermore be expressed with $W_i$ being the indicated work and $W_{s,th}$ being the theoretical isentropic work of the idealised expander cycle.

$$\eta_s = \frac{P_i}{P_s} = \frac{P_i}{\dot{m} \Delta h_s} = \frac{n z_{male} W_i}{\lambda_L \dot{m}_{th} \Delta h_s} = \frac{W_i}{\lambda_L W_{s,th}}$$  \hspace{1cm} (7)

Figure 8 shows the isentropic efficiency characteristics for a process with surface condensation in thermodynamic equilibrium. The inlet pressure and back pressure are set to fixed values. Expander tip speed is varied from $u = 30$ to $120 \text{ m s}^{-1}$. Heat flux is varied from $q = 2 \cdot 10^4$ to $8 \cdot 10^5 \text{ W m}^{-2}$. Isentropic efficiency of the adiabatic process is $\eta_s = 0.99$, 

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**Figure 7.** Impact of surface condensation on delivery rate for various expander speeds and inlet pressures, $\Pi = \frac{p_{in}}{p_{out}} = 6$, $\dot{q} = 6 \cdot 10^5 \text{ W m}^{-2}$, $T_{in} = 431.98 \text{ K (sat. steam)}$.

**Figure 8.** Influence of heat transfer on isentropic efficiency of an idealised expander cycle, $p_{in} = 6 \cdot 10^5 \text{ Pa}$, $p_{out} = 1 \cdot 10^5 \text{ Pa}$, $T_{in} = 431.98 \text{ K (sat. steam)}$. 
Figure 9. Calculated values of heat transfer coefficient for annular film condensation [8] and dropwise condensation [17] for steam, \( p = 6 \cdot 10^5 \) Pa.

which is the small loss being caused by under-expansion (see figure 5). The influence of surface condensation on indicated power is moderate for expander operation with saturated steam and efficiency decreases both with increasing heat flux and with reduced expander speed. However, for the operating points displayed, the influence on pressure history is predominant compared to the changing mass flow rates due to heat outflow during the inlet phase. Despite choosing high absolute values of heat flux, the mass flow rate is maximally increased by 3.2\% for the operating points depicted in figure 8 (\( u = 30 \, \text{m} s^{-1},  \dot{q} = 8 \cdot 10^5 \, \text{W} m^{-2} \)).

7. Validity of condensation models for screw expanders

For a precise simulation of surface condensation in screw expanders, the application of a realistic heat transfer model is required. Initially, the mode of condensation (separated or dispersed flow, filmwise or dropwise condensation) needs to be identified. Flow regime maps, as provided by Tandon et. al [7], allow for a first estimate of the condensation process. However, information on the steam flow within the working chamber needs to be provided. The principle of chamber model simulation cannot provide such information, since in this model, thermophysical properties are calculated for homogeneous chamber states at rest. A reference velocity along the helical working chambers, which is in the range of the rotor tip speed, can be calculated from the expander speed and its geometry [11]. These relatively high velocities dominate over gravitational force and indicate the mode being an annular condensation flow or, for high expander speeds, even a mist flow.

Heat transfer models for annular condensation proposed in the literature focus on steady state flow and validation is most often provided for complete phase transition in condenser pipes. However, only partial condensation is expected during the expansion of steam in screw expanders and empirical correlations might provide extremely high values of heat transfer coefficient for the onset of annular condensation due to small film thickness. Figure 9 shows calculated values for annular film condensation in a male rotor chamber and dropwise condensation for saturated steam at \( p = 6 \cdot 10^5 \) Pa. The heat transfer correlation in [17] for dropwise condensation is obtained from experimental studies. For the annular film condensation, the working chamber is treated as a helically wound condenser pipe. Heat transfer coefficients are calculated from a Nußelt correlation documented in [8]. Due to the small film heights, transport properties of the condensate film are calculated for saturation points. The reference vapour velocity along the helix is calculated with Keller’s proposal according to the expander geometry [11] for a male.
rotor tip speed of $u = 30 \text{ m s}^{-1}$. The hydraulic diameter $d_h$ of the pipe is established from a plane orthogonal to the velocity vector within the rotor cavity. The boundary conditions for the film condensation model lead to heat transfer coefficients being greater than for dropwise condensation at high steam quality, which is a questionable result.

Furthermore, it is a debatable point whether or not the assumption of a steady state condensation process is applicable, since at least the surfaces of the housing bore are only in contact with the expander chamber for a finite time frame. Moreover, condensate entrainment is expected in the high velocity flow of the housing clearances, which removes liquid from the housing bore as the rotor tips pass by. This re-exposure of dry surfaces is comparable to the effects occurring in dropwise condensation and potentially increases heat transfer. In addition, centrifugal forces might affect the condensation process, especially on rotor surfaces. In summary, it can be stated that adopting Nußelt correlations for surface condensation documented in literature to the thermodynamic simulation of rotary displacement machines can only lead to a rough approximation of the physical effects expected during the working cycle.

8. Conclusion

The general influence of steam condensation on the surfaces of an idealised screw expander is discussed. Surface condensation in rotary positive displacement machines can be expected when external heat transfer occurs in the two phase region. External heat transfer has two major effects on the energy conversion: The specific volume of the fluid is decreased during the inlet phase, which results in a higher mass being trapped within the working chamber and, consequently, increases the mass flow rate. Furthermore, heat transfer induces an additional pressure drop during the expansion phase. However, the high specific condensation enthalpy of water minimises these two effects. Validation of these findings against experimental data is still pending. Numerous Nußelt correlations for surface condensation are documented in literature. Since those models are developed for steady state flow and present most often a solution for complete phase transition in condenser applications, their validity is limited for surface condensation in rotary displacement machines. Estimation of heat transfer based on these models is, however, possible and indicates high values of heat transfer coefficients for steam-driven screw expanders. Experimental work on condensation processes in rotating systems and consideration of forced condensate entrainment might help gain a deeper insight into the actual surface condensation mechanisms in rotary positive displacement machines.

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List of symbols

| Symbol | Unit       | Property          |
|--------|------------|-------------------|
| A      | m^2        | area              |
| c      | m s^{-1}   | velocity          |
| d      | m          | diameter          |
| h      | J kg^{-1}  | specific enthalpy |
| H      | J          | enthalpy          |
| \dot{m}| kg s^{-1}  | mass flow rate    |
| m      | kg         | mass              |
| n      | s^{-1}     | expander speed    |
| Nu     | –          | Nusselt number    |
| p      | Pa         | pressure          |
| P      | W          | power             |
| \dot{q}| W m^{-2}   | heat flux         |
| Q      | J          | heat              |
| s      | J kg^{-1} K^{-1} | specific entropy   |
| t      | s          | time              |
| T      | K          | temperature       |
| u      | m s^{-1}   | male rotor tip speed |
| u      | J kg^{-1}  | specific internal energy |
| U      | J          | internal energy   |
| v      | m^3 kg^{-1} | specific volume   |
| V      | m^3        | volume            |
| W      | J          | work              |
| z      | –          | number of lobes   |
| \alpha| W m^{-2} K^{-1} | heat transfer coefficient |
| \delta| m          | film thickness    |
| \epsilon| –         | volume fraction   |
| \lambda| W m^{-1} K^{-1} | thermal conductivity |
| \lambda_L| –         | delivery rate     |
| \eta   | –          | efficiency        |
| \mathcal{L}| m        | characteristic length |

| Subscript | Property      |
|------------|---------------|
| c          | condensate    |
| ex         | expansion     |
| h          | hydraulic     |
| i          | indicated     |
| s          | isentropic    |
| sat        | saturated     |
| t          | total         |
| th         | theoretical   |
| v          | virtual       |
| \Delta     | difference, increment |