Influence of water injection on the operating behaviour of screw expanders

Experimental investigation

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Abstract

This paper presents the results of experimental investigations into hot air driven screw expanders without timing gears where water is injected into the inlet pipe. Expander geometry was held constant throughout these experimental investigations. In this context, the characteristic diagram of the test expander is recorded by varying the operating parameters of the screw expander as well as the volume flow rate and temperature of the injected auxiliary liquid. To evaluate the operational behaviour of the test screw expander, characteristic numbers are used considering the working fluid as a humid-air-water mixture.

Water injection into the inlet pipe of screw-type expanders results generally in sinking air mass flow and delivery rate respectively due to the clearance sealing effect. In this context, effective isentropic efficiency with regard to an isentropic change in state of unsaturated humid air generally increases at lower male rotor circumferential speeds. Considering effective isentropic efficiency with regard to humid-air-water mixture, only moderate increase in efficiency at highest injected water volume flow rates at circumferential speeds up to u_{MR} = 30 m·s^{-1} can be observed. At higher male rotor circumferential speeds hydraulic losses increase and effective isentropic efficiency declines compared with a dry-running operation.

1 Introduction

Screw-type expanders offer high potential for energy conversion in lower and medium power ranges, for instance as expansion engines in Rankine cycles[1] for exhaust heat recovery. With the aim of minimising internal leakages within the expander and lubricating moving machine parts, an auxiliary liquid can either be carried with the main flow or be fed to the screw-type expander. Injecting an auxiliary fluid into screw machines is a common method to

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[1] The Rankine cycle is a thermodynamic cycle that is run as a closed loop. In Rankine cycles an operating fluid, water or organic fluid, is continuously evaporated and condensed.

improve machine performance, reduce its thermal stress and fluid temperature, and to afford the operation of the screw rotors without using complex expensive timing gears. In such cases, the auxiliary fluid acts as a cooling medium within the working chamber, lubricates the contact scopes of both rotors, and seals clearances between moving components and casings. Auxiliary fluids can also reduce noise while a screw machine is in operation. The most commonly used auxiliary fluid is oil.

Whereas injection of oil is a common method in screw-type compressors without timing gears, water as an auxiliary fluid offers sufficient potential for investigation. There are commercial water injection concepts for screw-type compressors with timing gears whose aim is to reduce gas temperature and thermal stress in the working camber. Investigations into water injection into screw-type expanders with timing gears were done by Zellermann [1]. Rinder has shown the advantages and disadvantages of water as an injected fluid with regard to a screw-type compressor without timing gears. The main advantages include: oil-free air, saving of energy, no danger of condensate in the oil-separator vessel and in the bearing lubricant, low temperatures, no fire hazard, no environmental pollution, and high pressure ratios in one stage. Negative effects in compressor applications are separate bearing-lubrication-systems, problems sealing of the bearing casing, inadequate lubrication of the rotor flanks, expensive materials for rotors, stainless materials for housings, highly accurate manufacturing, poor rotor clearance sealing, higher noise levels compared to oil, high water consumption, and steam saturated air at the outlet, as well as problems with biological contamination. [2]

Some of these effects are also relevant for screw-type expanders. The most important expected advantage of water injection into screw-type expanders is sealing of clearances as well as lubrication of the hardened and hard-coated screw rotors during operation. Injected water reduces temperature drops during expansion which result in a slight difference between the inlet and outlet temperature of the working fluid, as well as lower thermal stress alongside the expander.

2 Experimental equipment

The experimental rig and the test screw-type expander are introduced below. In addition, an overview of the test parameters is given.

2.1 Test rig and measurement equipment

The schematic figure of the test setup and measurement supply is shown in **Figure 1**. The experimental rig is designed particularly for the investigation of screw-type expanders and consists of two open loops. On the one hand, hot pressurised air is provided by a hot air test rig based on the open Joule cycle[2]. The hot air test rig consists of a compressor unit, pressure vessel, filter unit, electrical preheater, as well as pressure and flow control valves. To control the inlet pressure of the screw-type expander, a bypass is used before pressurised air flows into the preheater where the hot air inlet temperature is set. On the other hand, a second loop delivers the auxiliary fluid. Water is injected into the high pressure pipe in front of the screw expander inlet domain. The water flow is set by a pressure vessel with a maximum air pressure cushion of $p_{max} = 10 \cdot 10^5$ Pa and a flow control valve. The water is preheated by means of a boiler at a maximum temperature of $\vartheta_w = 85$ °C. Thus, no vaporisation in the water cycle is expected.

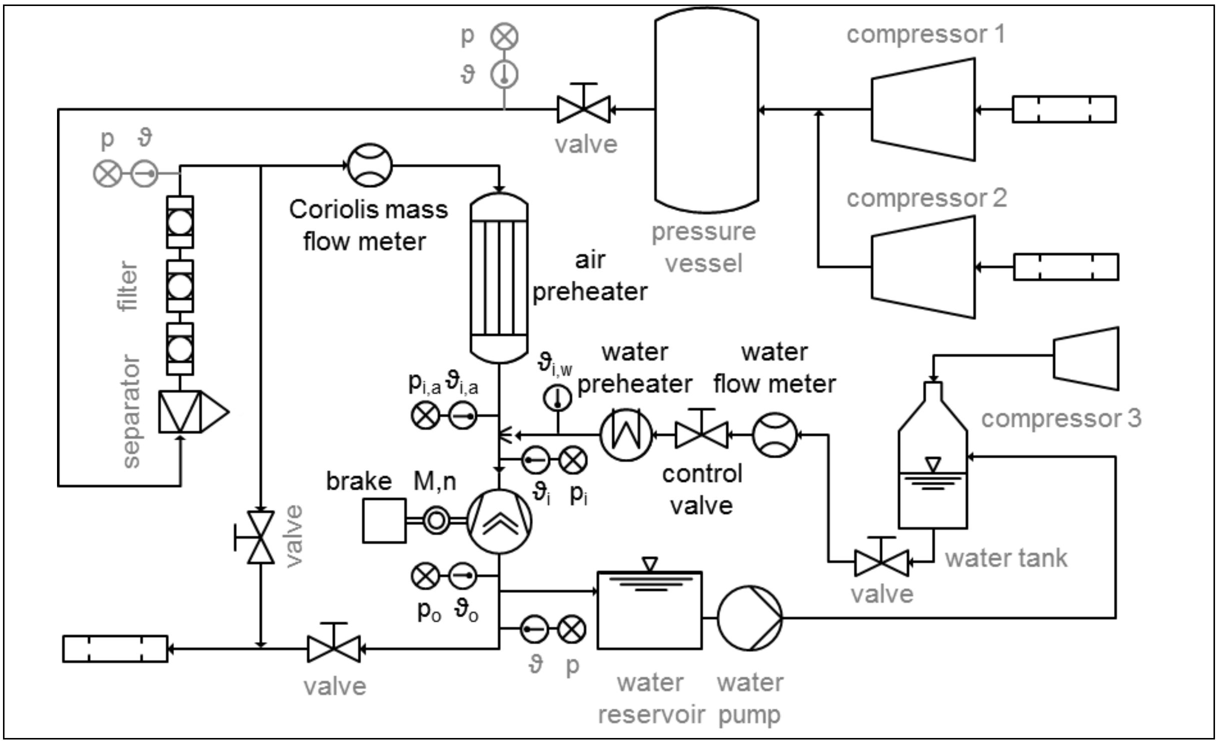

Figure 1: Experimental rig and measurement instruments

Diverse measurement devices embedded in the test rig record information about the fluid state and the test screw-type expander. Insulated thermocouples and static relative pressure transducers are installed at different points of the test rig. Besides monitoring pressure and temperature metering points, the mean inlet static pressure of the gaseous working fluid is metered. The mean static pressure of the air-water mixture is recorded after water is injected both in the high and low pressure ports of the test expander. Insulated thermocouples are

[2] The Joule cycle is a thermodynamic cycle that is usually run as an open system. It is also sometimes known as the Brayton cycle.

installed in the centre of the pipe on each metering point to measure the overall fluid temperature at different states. To record the characteristic map of the test screw-type expander, the rotational speed and driving torque are also metered by means of a torque transducer.

2.2 Test screw-type expander and test parameters

Within the framework of this experimental investigation, a screw-type expander without timing gears (**Table 1**) was installed. It is a further development of a twin screw charger GL 51 designed at Chair of Fluidics, [3]. In general, optimisations of the reference screw machine had been done with respect to the specific operating conditions required for the screw expander's application with high inlet pressures and internal pressure ratios respectively. In this context, the inner volume ratio can be varied from $v_i = 2.5$ up to $v_i = 6$ by means of flexible inlet area modules in order to set different "designed" operating conditions[3] with respect to inlet pressure levels at maximum expander efficiency. For the initial experimental investigations presented in this paper, the inner volume ratio is set to $v_i = 2.5$. Further geometric parameters of the test screw-type expander are given in Table 1.

Table 1: Parameters of the test screw expander

designation	**unit**	**male**	**female**
rotor profile	[-]	modified asym. SRM	
number of lobes z	[-]	3	5
rotor length l	[mm]	101	
axis-center distance a	[mm]	51.2	
diameter d	[mm]	71.8	67.5
wrap angle φ	[°]	200	-120
rotor lead s	[mm]	181.8	-303
internal volume ratio v_i	[-]	2.5	
displaced volume per male rotor rotation V	[cm^3]	286	
front gap height h_{fg}	[mm]	0.1	
housing gap height h_{hg}	[mm]	0.08	

The region of interest for the screw-type expander with respect to an air-water mixture flow is illustrated in **Figure 2**. With reference to the energy balance over the screw-type expander, it is assumed there is no heat flux into the atmosphere and the expander operates adiabatically. Within the framework of this experimental investigation, both the system and the screw expander's operating parameters were varied. The inlet pressure is set from $p_i = 2{\cdot}10^5$ Pa to $p_i = 6{\cdot}10^5$ Pa for a constant back pressure of $p_o = 1{\cdot}10^5$ Pa. The air inlet temperature varies in a range between $\vartheta_{i,a} = 60$ °C and $\vartheta_{i,a} = 120$ °C; water gets preheated

[3] In "designed" operation conditions, the lowest pressure in the working chamber before discharge equals the back pressure, so no over- or underexpansion occurs.

from $\vartheta_{i,w} = 45$ °C to the maximum temperature of $\vartheta_{i,w} = 85$ °C for constant air inlet pressure $p_i = 4 \cdot 10^5$ Pa and temperature $\vartheta_{i,a} = 90$ °C. Auxiliary fluid flow is restricted to $\dot{V}_w = 7.5\ \mathrm{l \cdot min^{-1}}$, where four nozzle sizes are used with respect to the maximum pressure in the water cycle of $p_{max} = 10 \cdot 10^5$ Pa and thus the restricted water flow range per nozzle. Maximum expander rotational speed is $n_{MR} = 20000\ \mathrm{min^{-1}}$ which equals a male rotor circumferential speed of $u_{MR} \approx 75\ \mathrm{m \cdot s^{-1}}$.

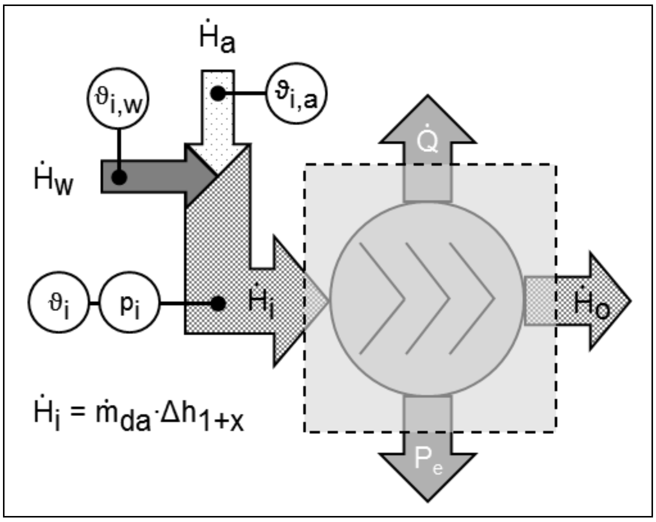

Figure 2: Expander control volume

3 Characteristic numbers for screw-type expanders

The operational behaviour of screw-type expanders can be evaluated by means of characteristic numbers. Effects such as the sealing of expander clearances, inlet pressure drops, hydraulic losses, etc., can be evaluated integrally by taking the system and expander operating parameters into account. The characteristic numbers used here are delivery rate and effective isentropic efficiency.

3.1 Delivery rate

Delivery rate is a characteristic number which describes different loss mechanisms such as internal or external leakages, inlet pressure drops during the chamber filling, or thermal effects within a displacement machine. For a screw-type expander, the delivery rate is defined as:

$$\lambda_L = \frac{\dot{m}_a}{\dot{m}_{th}}, \tag{1}$$

at which $\dot{m}_a$ is the measured humid air mass flow (Coriolis mass flow meter) and $\dot{m}_{th}$ the theoretical air mass flow with:

$$\dot{m}_{th} = V_{th,ex} \cdot \rho_{da,i} \cdot n_{MR} \cdot z_{MR}. \tag{2}$$

Indeed, a certain chamber volume is displaced resulting from water injection. Nevertheless, this can be ignored due to the relatively high volume fraction of the air which is over 99 % even at the lowest air mass flow and highest injected water volume flow rate. Dry air is regarded as ideal gas and its density $\rho_{da,i}$ is considered with reference to calculated subcooled temperature T_i after water was injected.

3.2 Effective isentropic efficiency

Effective isentropic efficiency is a characteristic number which relates an expander's effective power to the isentropic change in the state of the working fluid. With respect to operating water injected screw-type expanders, an isentropic change in state represents the maximum energy available in the working fluid. Hence when the working fluid is a humid-air-water mixture, effective isentropic efficiency is defined as:

$$\eta_{e,s} = \frac{P_e}{\Delta\dot{H}_s}. \tag{3}$$

Isentropic enthalpy flow $\Delta\dot{H}_s$ is defined as the product of dry air mass flow $\dot{m}_{da}$, calculated from the measured air mass flow $\dot{m}_a$, the water load X_{amb}[4)] of the humid air drawn in, and the specific enthalpy difference for humid air $\Delta h_{1+x,s}$ as follows:

$$\Delta\dot{H}_s = \dot{m}_{da}\cdot\Delta h_{1+x,s} = \frac{\dot{m}_a}{X_{amb} + 1}\cdot\left(h_{1+x,s,o} - h_{1+x,i}\right). \tag{4}$$

In this context, the specific isentropic enthalpy h_{1+x} for a humid-air-water mixture can be calculated with reference to water load X, [4]. Hence, the specific enthalpy of unsaturated humid air ($X < X_{sat}$) is defined as:

$$h_{1+x} = c_{p,da}\cdot\vartheta + X\cdot\left(r_w + c_{p,ws}\cdot\vartheta\right). \tag{5}$$

The specific enthalpy of saturated humid air ($X \geq X_{sat}$) with reference to water in the mixture can be calculated as:

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[4] Water load $X = \frac{\dot{m}_w}{\dot{m}_{da}}$ represents the ratio of water mass (flow) to dry air mass (flow). [4]

$$h_{1+x} = c_{p,da} \cdot \vartheta + X_{sat} \cdot \left(r_w + c_{p,ws} \cdot \vartheta\right) + \left(X - X_{sat}\right) \cdot c_w \cdot \vartheta. \qquad (6)$$

For temperatures $\vartheta < 0$ °C the calculation of the specific humid air enthalpy is referred to ice condensation as follows:

$$h_{1+x} = c_{p,da} \cdot \vartheta + X_{sat} \cdot \left(r_w + c_{p,ws} \cdot \vartheta\right) - \left(X - X_{sat}\right) \cdot \left(r_{ice} - c_{ice} \cdot \vartheta\right). \qquad (7)$$

Dry air, water steam, water, and ice heat capacity are assumed to be constant with $c_{p,da} = 1004.6$ J·kg^{-1}·K^{-1}, $c_{ws} = 1863$ J·kg^{-1}·K^{-1}, $c_w = 4185$ J·kg^{-1}·K^{-1}, and $c_{ice} = 2070$ J·kg^{-1}·K^{-1}. Specific solidification enthalpy r_{ice} at triple temperature ($\vartheta_{tr} = 0.01$ °C) and specific vaporisation enthalpy r_w at triple temperature ($\vartheta_{tr} = 0.01$ °C) are set to $r_{ice} = 333.4$ kJ·kg^{-1}and $r_{ice} = 2500.9$ kJ·kg^{-1}. The homogeneous expander inlet temperature ϑ_i of the mixture can be determined iteratively from the input parameters ambient water load X_{amb}, air inlet temperature $\vartheta_{i,a}$, water injection temperature $\vartheta_{i,w}$, air mass flow $\dot{m}_a$, as well as water mass flow $\dot{m}_w$. For this reason, the specific fluid mixture enthalpy according to Eq. 6 is equated with the sum of specific enthalpy of the humid air drawn in and the injected water. Thus, the state of the working fluid after water injection is known. The isentropic fluid mixture temperature and enthalpy at the expander outlet can be iteratively calculated with regard to constant specific entropy at the expander's inlet and outlet. [4]

4 Results

The results of this experimental investigation into the operational behaviour of a screw-type expander are presented below. At first, the influence of the system parameters on the operation of a non-injected machine is shown. Afterwards, the impact of inlet pipe water injection is shown by means of the change in the characteristic numbers defined in Section 3.

4.1 Dry-running screw-type expander

The characteristic values of a hot air driven screw-type expander without timing gears depend directly on the operating parameters. Initially, expander mass flow and effective power are presented in **Figure 3** as a function of inlet pressure and temperature as well as circumferential speed. Basically, expander mass flow rises on the one hand with inlet pressure due to a higher gas inlet density and, on the other hand, with circumferential speed due to an increasing number of working cycles.

As shown in Figure 3, inlet pressure and expander circumferential speed also considerably influence expander effective power. A general increase in effective power for constant circumferential speeds and increasing inlet pressure results from both the ensuing rise of the mass flow and a growing specific enthalpy difference between the expander's inlet and outlet. For fixed inlet pressure, expander power firstly rises as a function of the circumferential speed due to increasing number of working cycles and expander mass flow. Afterwards, expander effective power regressively increases at higher circumferential speeds due to increasing inlet throttling and mechanical losses as well as overexpansion in the working chamber. The overexpansion results from an increasing inlet pressure drop due to the reduced chamber filling time and a relatively low pressure level at the beginning of the expansion in the working chamber for a fixed inner volume ratio. Hence, at the lowest inlet pressure investigated expander power is maximum at about $u_{MR} = 45\ m{\cdot}s^{-1}$ and declines at higher circumferential speeds. Varying hot air inlet temperature reveals a minor impact on expander effective power. With respect to mass flow, a slight decrease occurs at higher inlet temperatures due to lower fluid density, which compensates for the rise of the useable enthalpy difference between the expander's inlet and outlet. Thus, effective power remains nearly constant.

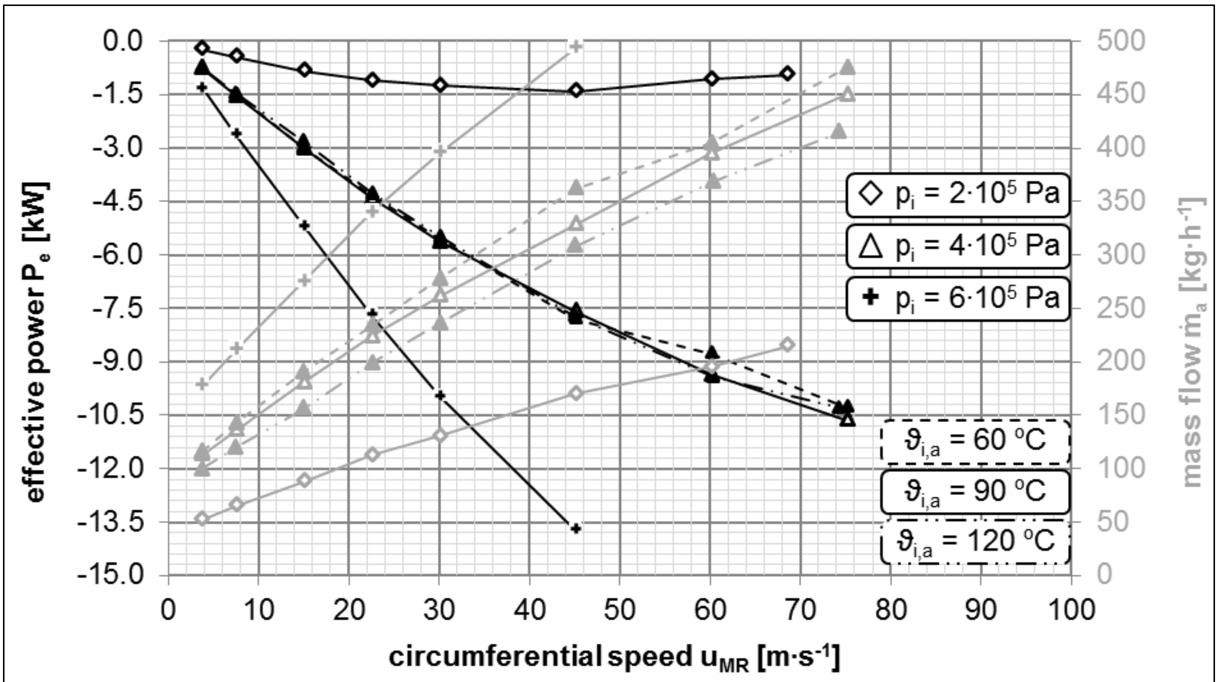

Figure 3: Effective power Pe and mass flow $\dot{m}_a$ of a dry-running screw-type expander depending on circumferential speed, inlet pressure, and temperature

In this context, the most relevant loss mechanisms concerning the operation of screw-type expanders – inlet pressure drops, mechanical losses, and internal leakages – can be explained by means of effective isentropic efficiency (Eq. 3) as well as delivery rate (Eq. 1), **Figure 4**. Low circumferential speeds and high inlet pressure levels are characterised by relatively low efficiency and high delivery rates. An increase in circumferential speed reduces internal leakages, where delivery rate decreases and efficiency initially rises. Then, increasing rotor circumferential speed induces increasing inlet pressure drops and mechanical losses as well as a potential overexpansion, especially in case of low inlet pressure levels. The result is a regressively increasing efficiency or even declining efficiency values. Hence with regard to optimum effective isentropic efficiency, male rotor circumferential speeds $u_{MR} > 50\ m \cdot s^{-1}$ at inlet pressure levels $p_i \geq 4 \cdot 10^5$ Pa correspond to a range with respect to the operation of dry-running screw-type expanders. With regard to variation of the inlet temperature, no significant impact can be determined either on the effective isentropic efficiency or on the delivery rate. Further detailed analysis of dry-running screw-type expanders were carried out by Hütker [5] and Kovacevic [6].

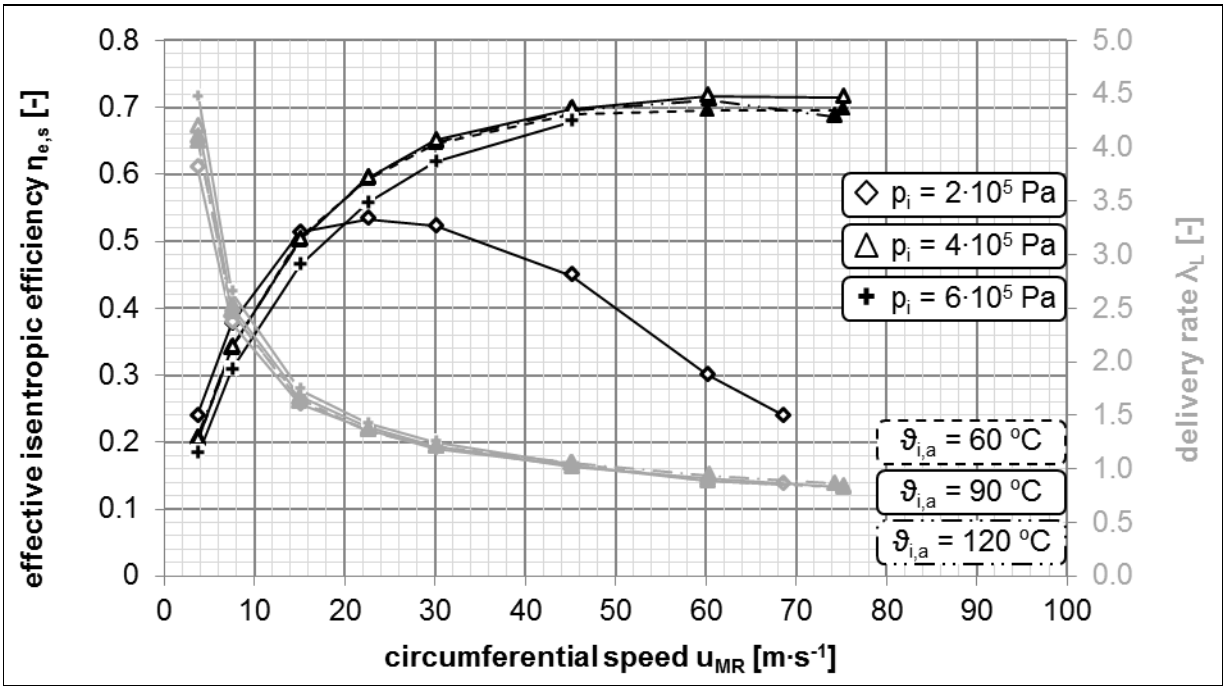

Figure 4: Effective isentropic efficiency $\eta_{e,s}$ and delivery rate λ_L of a dry-running screw expander depending on circumferential speed, inlet pressure, and temperature

4.2 Water injected screw-type expander

Before evaluating the influence of water injection on the operation of the screw-type expander investigated, the thermodynamic effects resulting from the interaction between the hot air drawn in and the injected water are discussed. From a thermodynamic point of view, it is important to know how humid air behaves when it comes into contact with water. Even the ambient air drawn in has a certain liquid content which affects the properties of the working fluid significantly. It is not appropriate to consider hot compressed air exclusively as dry air due to the interaction between air and water. On the one hand, the interaction consists of material exchange, in which water is evaporated by unsaturated humid air, whereby the liquid water content decreases. In this case, the enthalpy of humid air declines, which is associated with a decrease in temperature. On the other hand, enthalpy exchange in terms of heat flux between both saturated humid air and water involves a change in state during the entire working cycle. Hence, the thermodynamics of a humid-air-water mixture have a significant influence on the screw expander operation. While in a (quasi) stationary case such as water injection into the high pressure pipe the interaction between humid air and water is well predictable, the primarily transient expander working cycle does not allow a simple approach to calculate state change of the fluid mixture within a working cycle. Nevertheless, an integral consideration of the fluid mix state at the expander inlet and outlet is possible.

The thermodynamic interaction between the two fluids is exemplarily illustrated in **Figure 5** with reference to the mixture temperature ϑ_i after water is injected into the pressurised hot air pipe both for measurement[5] as well as for theoretical calculation. The theoretically calculated temperature considers the measured air and water state before water injection at $\vartheta_{i,a}$ = 90 °C and $\vartheta_{i,w}$ = 85 °C as well as $\vartheta_{i,w}$ = 45 °C.

Each calculation of mixture temperature concerns the corresponding relative humidity of the air drawn in at the time of measurement ($X_{amb} \approx 0.005$). At first, a general temperature drop results from the injection of hot water into hot humid air flow ($\vartheta_{i,a}$ = 90 °C) due to the required vaporisation enthalpy. With regard to injected water volume flow, the temperature increases for the rising amount of water at $\vartheta_{i,w}$ = 85 °C due to increasing heat exchange from the hot water to the saturated humid air. Similarly, a lower circumferential speed accords higher mixture temperature due to lower mass flow rate of the working fluid air and, therefore, longer time period for heat exchange. Furthermore, lower water injection temperatures result in lower mixture temperatures due to lower heat exchange. A further reduction of water temperature to $\vartheta_{i,w}$ = 45 °C for constant hot air conditions results in an inversion of heat flow from the saturated humid air to the colder water so that the mixture temperature decreases with rising injected water volume flow.

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[5] Mixture temperature is measured in a DN80 high pressure pipe in a distance of 92 mm from the water injection nozzle.

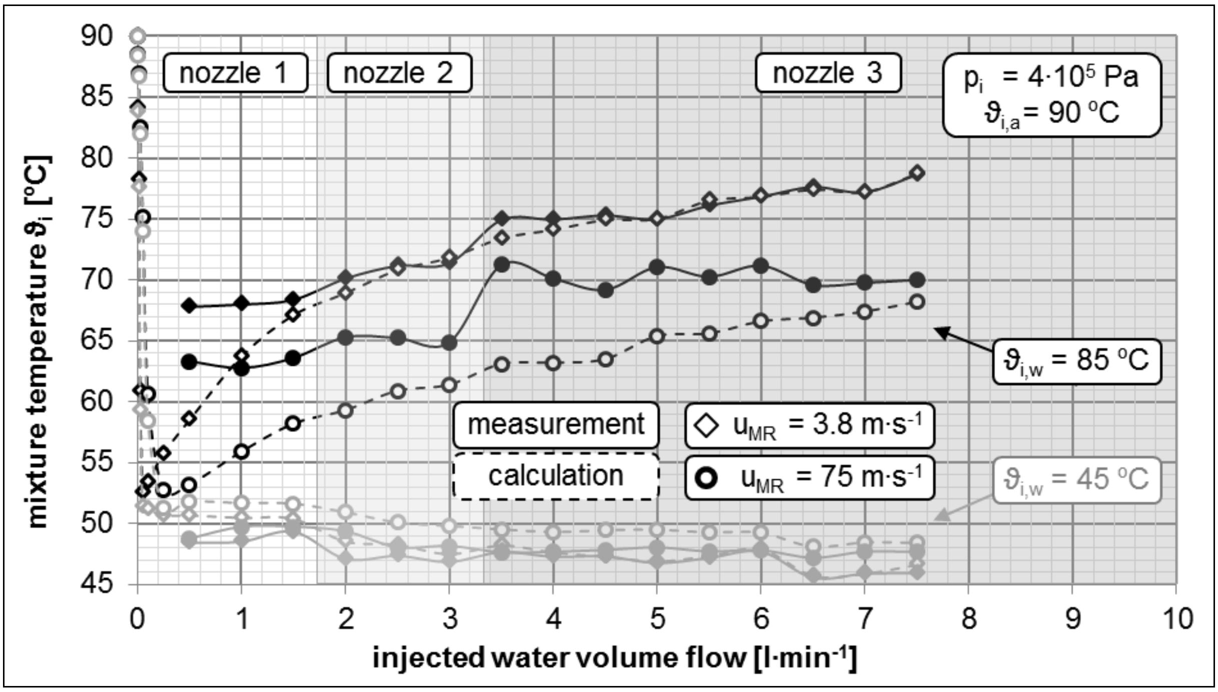

Figure 5: Measured and calculated fluid temperatures ϑ_i after water injection into the hot air pipe

Apparently, a further parameter affects the fluid mix state considering the metered mixture temperature. The spraying quality (e.g. droplet size, type of resulting fluid flow) which results from the pressure difference at the nozzle shows a significant effect on the resulting measured temperature. The spraying quality for a constant nozzle size at low volume flow rates is relatively low. In Figure 5, the scope of application for each nozzle is shown in order to explain the unsteady trend of the metered temperatures as a function of the water volume flow. Here, nozzle 1 has the smallest outlet diameter and nozzle 3 has the largest. The deviation between the calculated and measured temperatures is due to the fact that the mixture is not in equilibrium yet at the measurement point. This effect was tested by varying the measurement point (not shown here) where a bigger distance between the nozzle outlet and temperature measurement point results in declining deviance, and vice versa.

In this context, a state equilibrium can be expected in the inlet port, and it can be assumed that the calculated mixture temperature in Figure 5 represents the working fluid temperature. The following two-phase-mixture expansion in the screw expander is characterised by material and heat exchange between both fluids resulting in higher temperature levels compared with dry-running operation.

To exemplify how water injection integrally affects the operational behaviour of the test screw-type expander, the experimental results are evaluated in detail by means of characteristic numbers delivery rate and effective isentropic efficiency. For this reason, the parameters water volume flow and temperature were varied at constant hot air inlet pressure $p_i = 4 \cdot 10^5$ Pa and temperature $\vartheta_{i,a} = 90$ °C.

Delivery rate

The change in delivery rate dependent on the injected water parameters volume flow rate and temperature as well as on the expander circumferential speed is shown in **Figure 6**. It is noticeable that the theoretical chamber gaseous volume $V_{th,ex}$ remains almost constant due to the still very high air volume fraction (over 99 %) even at the highest water volume flow and lowest air mass flow despite the displaced chamber volume by the liquid fluid. Compared to dry-running operation, delivery rate decreases permanently for increasing water volume flow and constant circumferential speed. The biggest change in delivery rate occurs at $u_{MR} = 3.8\ m \cdot s^{-1}$ and declines with rising circumferential speed. Considering varying water temperature, a negligible deviance of delivery rate can be determined.

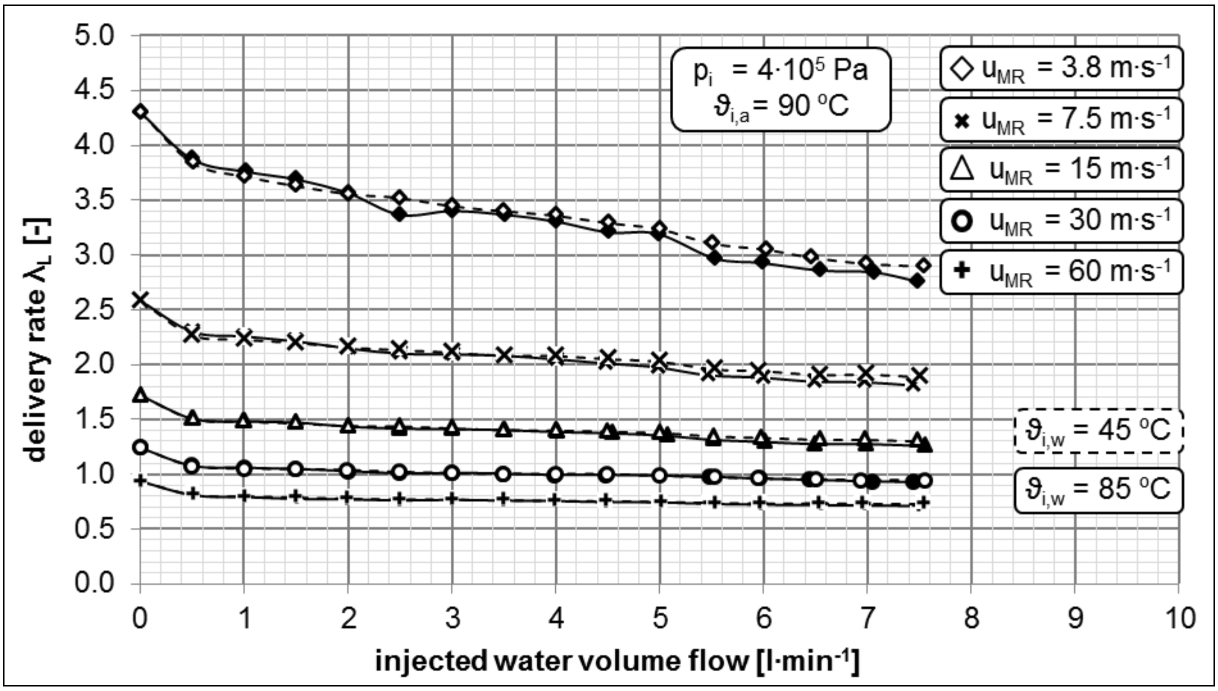

Figure 6: Delivery rate λ_L for a water injected screw-type expander with regard to the calculated subcooled inlet temperature ϑ_i (see Figure 5) and the working fluid as dry air

The mechanisms responsible for the decrease in delivery rate include blocking of the clearances by the liquid, inlet throttling losses, change in fluid density in working chamber and at clearance inlet, and change in speed of sound with regard to different fluid temperatures and a two-phase flow. In the first instance, the assumed sealing effect of the auxiliary fluid on the clearances reduces the air flow passing through the expander. Similar to

Figure 4, the accompanying change in working fluid density resulting from the drop in inlet port temperature was eliminated as a possible integral effect on the change of the delivery rate by definition with reference to the displaced mass per male rotor revolution. In this context the influence of water injection temperature on delivery rate can be neglected. Slightly increasing delivery rate at $\dot{V}_w > 4\ l \cdot min^{-1}$ and low male rotor circumferential speeds can be explained presumably by a not considered external leakage, since delivery rate is very sensitive to minor changes in measured air mass flow.

Considering a homogenous two-phase flow, there might be a significant drop in speed of sound [7], which results in a presumable increasing clearance sealing effect and rising inlet throttling losses. In contrast to theoretical concerns, the water-air-mixture flow considered within the framework of this experimental investigation is characterised presumably by relatively large water droplet diameters and a relative velocity of both fluids. This fact implies non-homogenous flow. Nevertheless, an increase in inlet throttling could be assumed pointing toward the need for further experimental and theoretical investigations.

Effective isentropic efficiency

The change in effective isentropic efficiency that results from water injection as opposed to dry-running screw-type expanders is presented for constant hot air conditions at $p_i = 4 \cdot 10^5$ Pa and $\vartheta_{i,a} = 90$ °C. In this context, two different definitions of an isentropic change in state are considered. At first, measured effective power is referred to an isentropic enthalpy flow change considering unsaturated humid air, **Figure 7**. In this case, specific enthalpy at the expander's inlet and outlet is calculated exclusively according to Eq. 5 for an ambient water load $X = X_{amb}$ with regard to subcooled mixture temperature T_i after water was injected. Afterwards, effective isentropic efficiency $\hat{\eta}_{e,s}$ is calculated with reference to an isentropic change in the state of the humid-air-water mixture, **Figure 8**. Here, the isentropic change in enthalpy flow of each mixture component is considered. In Figure 7 and Figure 8, the change in effective isentropic efficiency is shown for a lower and a maximum injected water volume flow rate of $\dot{V}_w = 1.5\ l \cdot min^{-1}$ and $\dot{V}_w = 7.5\ l \cdot min^{-1}$ at different water injection temperatures. Positive values of $\Delta\eta_{e,s}$ and $\Delta\hat{\eta}_{e,s}$ correspond to an increase of effective isentropic efficiency after water injection compared with dry-running operation. Negative values accord to decreasing expander efficiency.

In general, two significant effects on the effective isentropic efficiency can be specified as result of water injection concerning the isentropic change in enthalpy of unsaturated humid air, Figure 7. On the one hand, water injection primarily results in increasing efficiency values compared to dry-running operation at low male rotor circumferential speeds. This positive influence is attributed to the sealing of clearances by the liquid so that disadvantageous air clearance flow can be reduced. Therefore at lower male rotor circumferential speeds, increasing water injection volume flow at constant injected water temperatures corresponds to higher efficiency values. On the other hand, hydraulic losses due to shear stress within the clearances and due to momentum exchange between the high viscous auxiliary fluid and the rotating lobes influence the operation of water injected screw-type expanders [8]. Hence, increasing water volume flow rates correspond to a decreasing benefit and even negative change in efficiency compared to dry-running operation at high male rotor circumferential speeds. Considering injected water temperatures at constant water volume flow rates, a general increase in effective isentropic efficiency can be observed at higher temperature levels. This effect can be traced back to declining water viscosity and thus a decreasing liquid friction in working chambers and clearances.

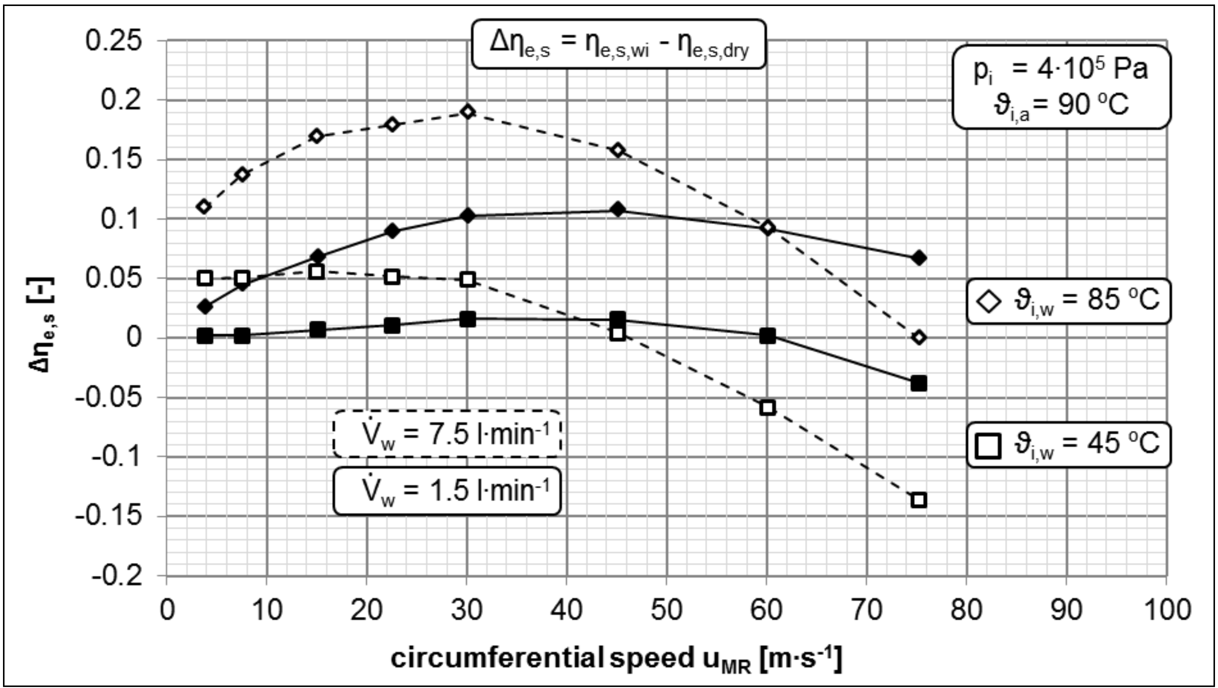

Figure 7: Change in effective isentropic efficiency $\Delta\eta_{e,s}$ after water injection into the high pressure inlet pipe of screw-type expanders with regard to an isentropic change in enthalpy flow of unsaturated humid air ($X = X_{amb}$)

In Figure 8, the change in effective isentropic efficiency $\Delta\hat{\eta}_{e,s}$ with regard to an isentropic change in enthalpy of the humid-air-water mixture is presented. In this context, the isentropic change in enthalpy flow of each mixture component is considered. Compared to the efficiency values presented in Figure 7, minor efficiency benefits of water injection are observed. On the one hand, low water volume flow rates basically result in decreasing effective isentropic efficiency compared to dry-running operation. On the other hand, an increase in effective isentropic efficiency $\hat{\eta}_{e,s}$ can be observed at $\dot{V}_w = 7.5\ l\cdot min^{-1}$ up to a male rotor circumferential speed of nearly $u_{MR} = 30\ m\cdot s^{-1}$. As mentioned in Figure 7, this positive influence is attributed to the blocking of the clearances by the liquid so that disadvantageous air clearance flow can be reduced. Decreasing water amount reduces the clearance sealing so that the positive effect declines. At high circumferential speeds, efficiency drops compared to dry-running operation result from rising hydraulic losses. An increasing amount of injected water also corresponds to higher hydraulic losses and hence lower efficiency values.

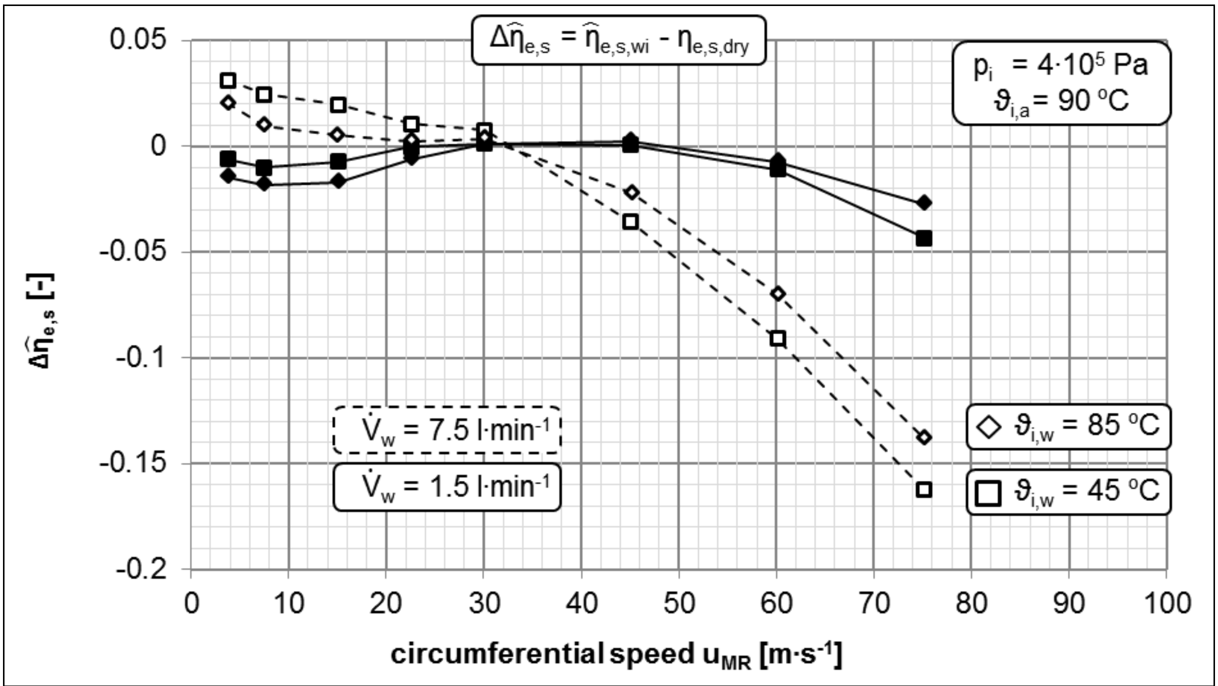

Figure 8: Change in effective isentropic efficiency $\Delta\hat{\eta}_{e,s}$ after water injection into the high pressure inlet pipe of screw-type expanders with regard to an isentropic change in enthalpy flow of a humid-air-water mixture

Considering injected water temperature in Figure 8, two opposed effects are observed. At first, positive change in effective isentropic efficiency $\Delta\hat{\eta}_{e,s}$ increases at declining water temperatures and low rotor circumferential speeds. In contrast, decreasing water injection temperatures result in lower efficiency values at high male rotor circumferential speeds. The positive effect of increasing water temperatures at high male rotor circumferential speeds can be explained by declining liquid viscosity and thus a decrease in liquid friction in working chambers and clearances. Since in Figure 6 no significant influence of injected water temperature on delivery rate and thus clearance mass flow is detected, it can be assumed that further effects on the operational behaviour of water injected screw-type expanders exist. Besides the clearance sealing effect and the hydraulic losses, a persistent interaction between the components of the working fluid mixture during the two-phase expansion additionally influences the screw expander's operation. The effects include for instance heat flow between water and humid air or phase change as a result of vaporisation or condensation depending on the operating parameters pressure and temperature. Hence, further investigations are required in order to extensively evaluate the effects on the operational behaviour of water injected screw-type expanders.

Since relatively small clearance heights in the test screw-type expander are set, further potential with regard to increasing effective isentropic efficiency could be expected. Clearance heights of 0.1 mm for the front gap and 0.08 mm for the housing gap seem to deliver relatively high efficiency rates even in dry-running operation. Therefore in order to further understand the influence mechanisms on water injected expanders, experimental investigations with respect to variation of clearance heights will be carried out. The aim is to distinguish different effects in detail. For instance, the theoretical investigation of incompressible clearance flow presented by Gräßer [9] shows that hydraulic losses within clearances in some cases may be nearly neglected. In this context, the primary hydraulic losses occur in the working chamber and not in the clearances, which can be proved within the framework of further experimental investigations. For example, it could be assumed that injection of highly viscous fluids such as oil could increase the clearance blocking at low male rotor circumferential speeds, where hydraulic losses do not predominate the sealing effect of the auxiliary fluid. Furthermore injection of auxiliary fluids with more advantageous properties with regard to the thermodynamic interaction with air could also result in increasing efficiency values.

5 Summary and Outlook

The results of an experimental investigation concerning the influence of water injection on the operational behaviour of screw-type expanders are presented. Unless hot humid air drawn in is not saturated, injected water partially evaporates and vaporisation enthalpy is released, so that mixture temperature declines. The following two-phase-mixture expansion is characterised by material and heat exchange between both fluids resulting in higher temperature levels compared with dry-running operation.

Water injection into the inlet pipe of screw-type expanders results generally in sinking air mass flow and delivery rate respectively due to the clearance sealing effect by injected water. In this context, effective isentropic efficiency with regard to an isentropic change in state of unsaturated humid air generally increases at lower male rotor circumferential speeds. Considering effective isentropic efficiency with reference to an isentropic enthalpy change of humid-air-water mixture only a moderate increase at highest injected water volume flow rates at circumferential speeds up to u_{MR} = 30 m·s^{-1} and low liquid temperatures can be observed. At higher male rotor circumferential speeds, there is an increase in hydraulic losses especially at low water temperatures and high viscosity respectively, which reduce the efficiency compared with a dry-running operation. In general, injection of auxiliary fluids with thermodynamically more advantageous properties with respect to the interaction with humid air is expected to result in a further increase in effective isentropic efficiency.

For a better understanding of the physical technical mechanisms in water injected screw-type expanders, further experimental investigations for variable clearance heights, pressure measurement in working chamber and clearances, and water injection into the screw-type expander will be carried out. Moreover high speed pictures are to be taken in order to better understanding of the liquid distribution within the expander’s working chamber. Complementary, simulations of screw-type expanders under consideration of the thermodynamic interaction between working and auxiliary fluid should be carried out for a detailed investigation of expander operation. Furthermore, investigations on screw-type expanders in energetically more appropriate thermodynamic cycles, such as Rankine cycles, should be carried out to extensively evaluate the potential of auxiliary liquid injection into screw-type expanders considering the system thermal efficiency.

6 Nomenclature

Symbols

a	axis-center distance	[m]
c	specific heat capacity	[J·kg^{-1}·K^{-1}]
d	diameter	[m]
h	height	[m]
h	specific enthalpy	[J·kg^{-1}]
$\dot{H}$	enthalpy flow	[W]
l	length	[m]
M	torque	[Nm]
$\dot{m}$	mass flow	[kg·s^{-1}]
n	rotational speed	[s^{-1}]
p	pressure	[Pa]
P	power	[W]
r_{ice}	specific solidification enthalpy at triple temperature (ϑ_{tr} = 0.01 °C)	[J·kg^{-1}]
r_w	specific vaporisation enthalpy at triple temperature (ϑ_{tr} = 0.01 °C)	[J·kg^{-1}]

Subscripts

a	humid air drawn in
amb	ambient
c	clearance
da	dry air
dry	dry-running expander
e	effective
ex	begin of expansion
fg	front gap
hg	housing gap
i	inlet/inner/internal
MR	male rotor
o	outlet
p	isobaric, pressurise
s	isentropic
sat	saturation
th	theoretical

s	rotor lead	[m]	tr	triple
T	temperature	[K]	w	water
u	circumferential speed	$[m \cdot s^{-1}]$	wi	water injection
v	volume ratio	[-]	ws	water steam
V	volume	$[m^3]$	1+x	humid-air-water mixture
$\dot{V}$	volume flow	$[m^3 \cdot s^{-1}]$		
X	water load	[-]		
z	number of lobes	[-]		
ρ	density	$[kg \cdot m^{-3}]$		
φ	wrap angle	[°]		
ϑ	temperature	[°C]		
η	efficiency	[-]		
$\hat{\eta}$	efficiency with regard to an isentropic change in state of unsaturated humid air	[-]		
λ_L	delivery rate	[-]		

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