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Test rig design for experimental characterisation of condensation in sCO_2 compressors

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Abstract

Industrial waste heat recovery provides an opportunity to reduce carbon emissions and meet climate goals. About 1000 TWh of industrial waste heat is generated in Europe alone, of which 30% is above 300 °C. Supercritical CO_2 (sCO_2) cycles offer a competitive advantage to recover high-grade waste heat compared to steam or organic Rankine cycles (ORC). However, several technical challenges in realising sCO_2 cycle components need addressing. One such challenge is the non-ideal gas effects and possible condensation observed in sCO_2 compressors operating near the critical point where numerical investigations alone do not suffice. This paper describes the design of an experimental facility at City, University of London to characterise the flow near the leading edge of an sCO_2 compressor operating near the critical point.

sCO_2 cycle analyses have been carried out to ascertain the significance of operating the compressor near the critical point with the obvious advantage emanating from the fact that the high density of CO_2 near the critical point reduces compressor work. The experimental test rig is a blowdown facility consisting of a high-pressure (HP) tank and a low-pressure (LP) tank with a test section in between them. The test section is comprised of a convergent-divergent nozzle with access for optical interferometric flow measurements. The blowdown test facility is operated as a batch process consisting of two phases. In the blowdown phase, CO_2 expands from the HP tank to the LP tank through the nozzle. The CO_2 from the LP tank is then fed back to the HP tank for the next test run through a compressor in the charging phase, thus allowing closed-loop operation. A target temperature in the HP tank in the range of 35 °C and 55 °C is achieved by electrical heating and electrical/water cooling to simulate different compressor inlet conditions. Similarly, the target pressure ranges from 130 bar to 75 bar and is achieved by adjusting the distribution of charge between the two tanks. A lumped mass model is developed to size the tanks and to simulate the blowdown and charging processes. The tank volumes are 100 litre and 185 litre for the HP and LP tanks respectively for a total CO_2 charge of 100 kg. This results in a blowdown time of approximately 10 seconds at 130 bar high side pressure. Quasi-1D nozzle theory is used to estimate the evolution of nozzle conditions under varying pressure ratios, under the assumption of homogeneous thermal equilibrium flow. The complete expansion of the CO_2 occurs in the nozzle for outlet pressures below 93 bar. Two-phase conditions are observed in the nozzle after a length of 60% with quality exceeding 20% at the nozzle exit.

Nomenclature

A	area (m^2)	T	temperature (m/s)
h	enthalpy (J/kg)	u	flow velocity (m/s)
$\dot{m}$	mass flow rate (kg/s)	V	volume (m^3)
$\dot{Q}$	heat transfer rate (kW)	S,s	entropy (J/kg-K)

Subscripts, Superscripts and Greek Symbols

1-6	thermodynamic state points	th	throat
comp	compressor	sat	saturated
HP	high pressure	ρ	density (kg/m^3)
LP	low pressure		

Introduction

Waste heat recovery (WHR) plays a central role in addressing climate goals. It is estimated that 63% of the overall global energy consumption is lost after combustion and heat transfer processes [1]. The properties of CO_2 in its supercritical state, such as higher density, absence of phase change, reduced compressor work, chemical stability, low global warming potential (GWP), and zero ozone depleting potential means supercritical CO_2 (sCO_2) power cycles are ideally suited for next-generation renewable and WHR power generation applications. Industrial WHR is different from utility-scale power plants, emphasizing affordability and return on investment (ROI) rather than efficiency and work output. The worldwide research on sCO_2 as a working fluid began with greater focus on cold climates which reduced the work of compression. Though this advantage is negated in tropical regions, the benefits such as ROI suffice to pursue sCO_2 for industrial WHR. In a review paper on sCO_2 for high-grade waste heat to power conversion, Marchionni et al. [2], compare sCO_2 power cycles with other conventional heat to power systems. Organic Rankine cycles (ORC) are limited to a heat source temperature of 400 °C due to their flammability and low chemical stability. Steam Rankine cycles are a good alternative to waste heat temperatures in the range 250 °C to 700 °C, but they are suitable for a power range below 10 MW as micro steam turbines are less efficient. sCO_2 WHR cycles on the other hand can utilize waste heat in a broad temperature range from 250 °C to 850 °C with power ranging from few hundred kilowatts to 100 MW with efficiencies comparable to or exceeding that of steam Rankine cycle. The techno-economic assessment of sCO_2 power cycles for WHR has found split heating of the flow after compression as being able to generate higher net power compared to other sCO_2 cycle layouts [3]. Simple recuperated sCO_2, which is the least complex among other sCO_2 cycles, turns out to be the most cost-effective for WHR applications with a specific cost of 770 $/kW$_e$ and a payback period of 1.86 years.

Phase change in sCO_2 compressors operating near the critical point is a major concern due to possible condensation. Some studies indicate that the local expansion at the impeller leading edge can exceed 30% of the overall compressor enthalpy rise [4]. Several numerical analyses have been carried out to assess two-phase flow in sCO_2 compressors [4,5]. Lettieri al [6] carried out experimental investigations to assess and quantify two-phase flow by simulating conditions similar to that observed in a sCO_2 compressor operating near the critical point.

In this work, sCO_2 cycle analysis has been carried out with a specific focus on industrial WHR. The cycle analysis aims to quantify the efficiency of the sCO_2 cycle operating near the critical point and hence define the boundary conditions for the experimental test rig. Subsequently the test rig configuration and component sizing are addressed in detail including the modelling of nozzle where the CO_2 flow expands from supercritical state to two-phase conditions.

sCO_2 cycle analysis

To establish the required operating range of the test facility, namely the operating temperatures and pressures, numerical simulations of a simple recuperated sCO_2 power cycle that emulate realistic heat recovery conditions were carried out. A schematic of the cycle is shown in Fig. 1.

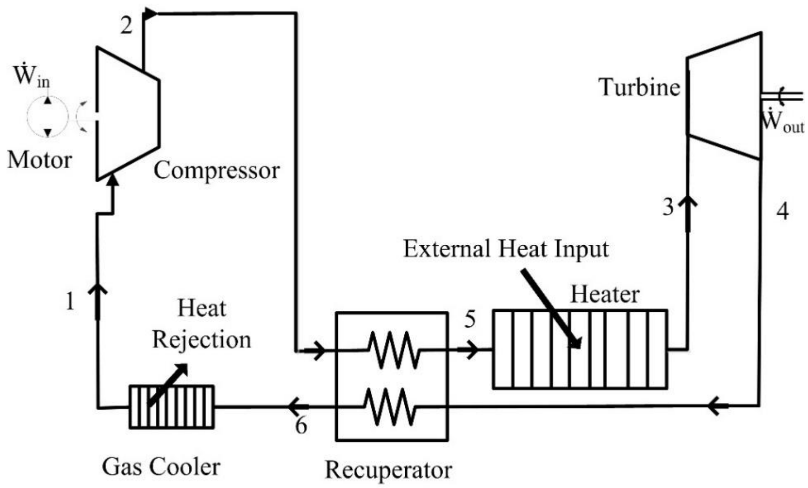

Fig. 1. Schematic of the simple recuperated sCO_2 cycle

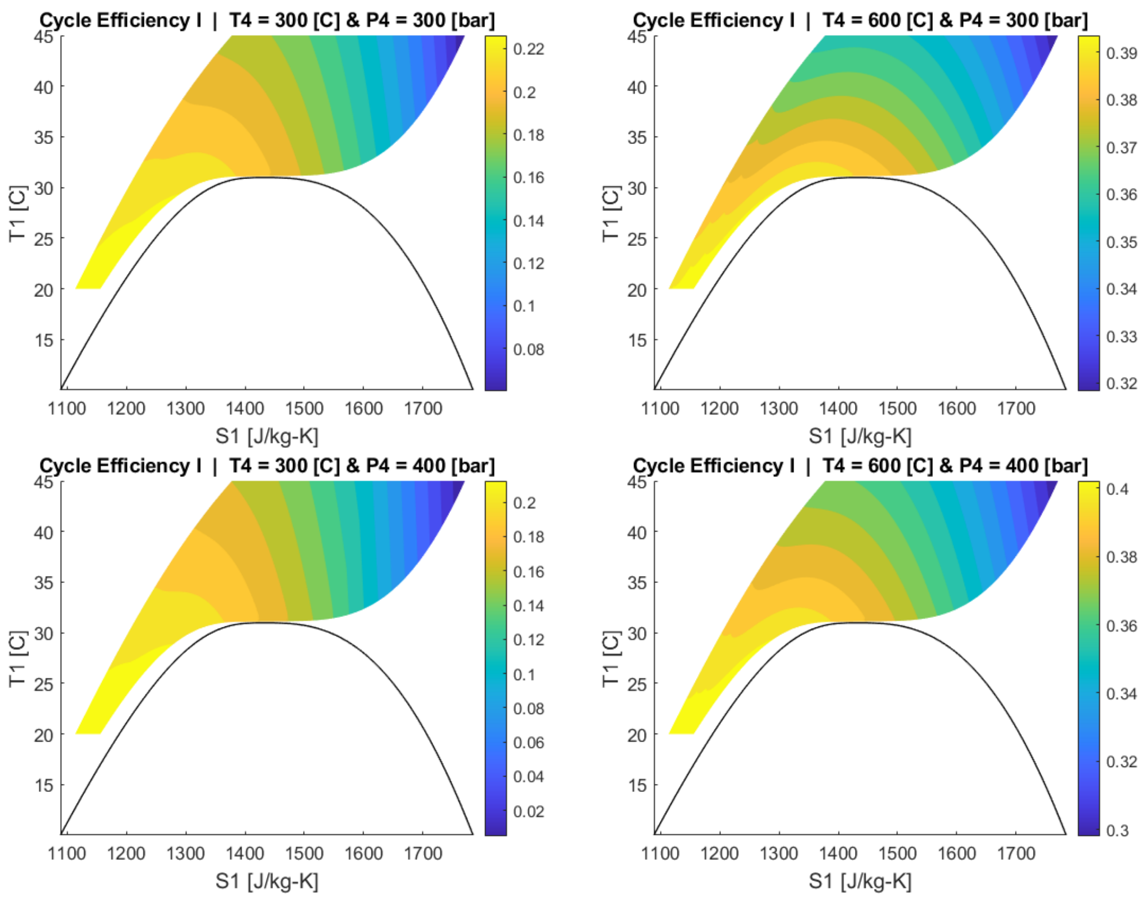

Fig. 2. Cycle efficiency plots of simple recuperated heat recovery power cycle against compressor inlet temperature and entropy with varied turbine inlet conditions

Inlet conditions to the compressor were varied to include the entire subcooled and supercritical regions surrounding the saturation lines. A temperature range of 20 – 45 °C coupled with a pressure range of 74 – 110 bar were considered for compressor suction conditions. Turbine inlet temperatures of 300, 450 and 600 °C were deemed representative of typical heat recovery cycles. Two turbine inlet pressures (300 and 400 bar) were considered to cover an effective cycle pressure ratio range of 2.5 – 5.5. The Span and Wagner equation of state [7] was used to carry out the cycle analysis. The isentropic efficiency of the turbine and compressor stages were set to 85% and 75%, respectively. A 1% drop in total pressure was assumed in each heat exchanger, and the recuperator pinch temperature was maintained at 10 °C.

To identify optimum cycle conditions, efficiency plots were generated for the investigated regions as shown in Fig. 2. The first law cycle efficiency is plotted for various compressor inlet temperature and entropy values. As evident from the results, the cycle performance peaks in the immediate vicinity of the saturation dome at lower pressures, specifically for subcritical temperature values. As compressor inlet temperatures cross into the supercritical region, cycle performance improves with elevated pressures. An increased turbine inlet temperature yields a significant improvement in the cycle efficiency, whereas a higher pressure ratio contributes to a modest change. This is partly due fixed turbomachinery isentropic efficiency values, effectively eliminating the influence of cycle pressure ratio on compressor and turbine stage performance. Nonetheless, accounting for the limitations of the required heat sink, the results suggest that a range of compressor inlet temperatures of 25 – 40 °C coupled with inlet pressures of 75 – 100 bar constitutes the optimal cycle conditions to be covered by the test facility.

sCO_2 blowdown test rig design

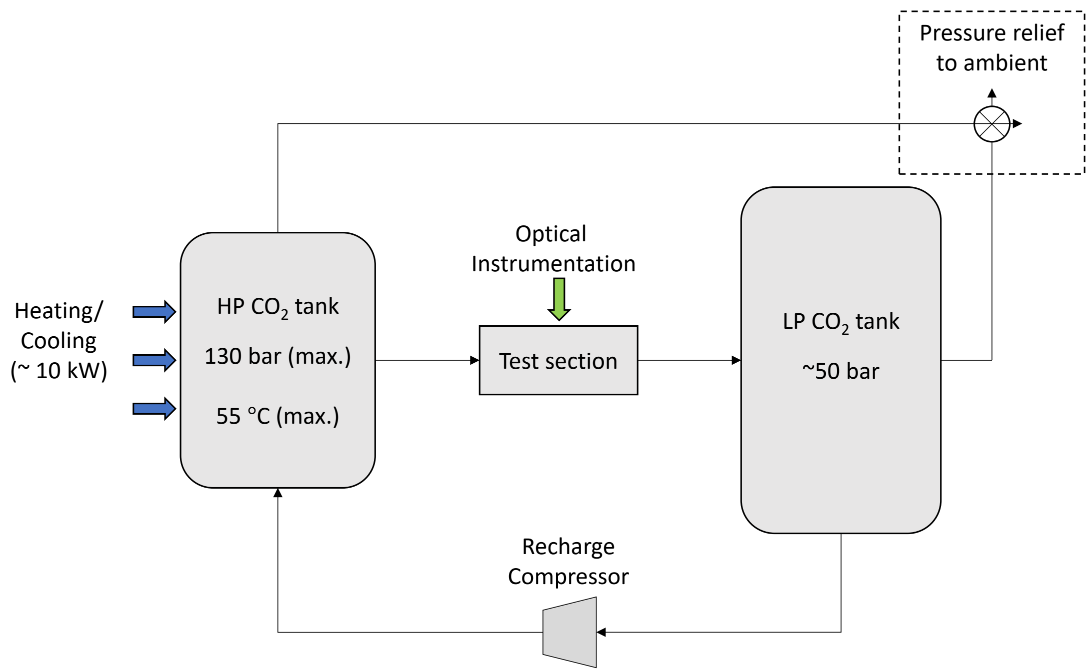

Fig. 3. sCO_2 blowdown test rig schematic

The major components of the test rig are HP tank, LP tank, test section, and recharge compressor. A simplified schematic of the rig is shown in Fig. 3. The HP tank is a jacketed pressure vessel with a provision for heating or cooling to adjust the temperature between the range 32 °C and 55 °C and a maximum pressure of 130 bar. CO_2 expands through the nozzle in the test section to the LP tank. Once the blowdown from the HP tank to the LP tank is completed, the recharge compressor is used to fill the HP tank to the desired pressure and temperature conditions. Hence the test rig operates in two modes, blowdown mode which is the primary mode for testing and the recharge mode.

To quantify the tank sizes, CO_2 charge and blowdown time, lumped mass modelling has been carried out. The modelling assumes a fixed CO_2 charge, a set target HP tank pressure and equilibrium conditions ($P_{HP} = P_{LP}$) at time t = 0. It also assumes bulk properties are constant within each tank and there is no heat loss to the surrounding. The modelling of the charging process is depicted in Fig. 4 (left) and Eqs. 1-4 which describe the energy and mass balance equations used to calculate the mass flow rate and enthalpy entering and leaving the tanks.

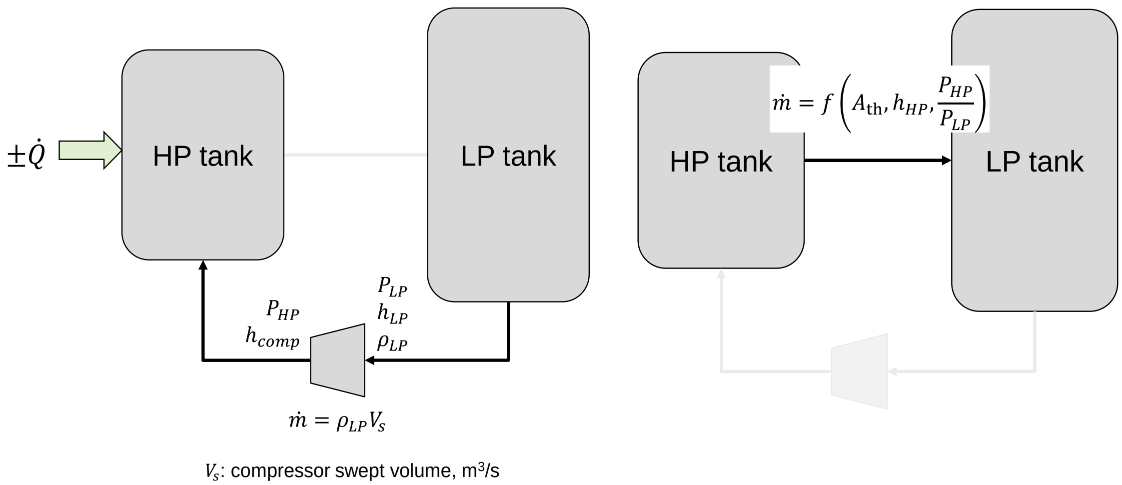

Fig. 4. Modelling of test rig charging (left) and blowdown (right) processes

$$\frac{d(\rho_{HP}V_{HP})}{dt} = \dot{m} \qquad \text{Eq. 1}$$

$$\frac{d(\rho_{LP}V_{LP})}{dt} = -\dot{m} \qquad \text{Eq. 2}$$

$$\frac{d(\rho_{HP}V_{HP}u_{HP})}{dt} = \dot{m}h_{comp} + \dot{Q} \qquad \text{Eq. 3}$$

$$\frac{d(\rho_{LP}V_{LP}u_{LP})}{dt} = -\dot{m}h_{LP} \qquad \text{Eq. 4}$$

In the above, $\dot{m}$ is the mass-flow rate displaced by the compressor, which is in turn is related to the compressor swept volume and the rotational speed, assuming a reciprocating piston compressor. The enthalpy leaving the compressor, h_{comp}, is calculated based on the assumption of an isentropic compression efficiency of 65% acting across the pressure difference between the two tanks. Finally, $\dot{Q}$ corresponds to the heating or cooling load provided by the jacket water.

Similarly, the blowdown process is shown in Fig. 4 (right) and Eqs. 5-8. The blowdown modelling assumes initial HP and LP tank conditions after charging and the simulation is run until equilibrium condition is achieved ($P_{HP} = P_{LP}$). The blowdown time is defined as the time taken to reach this equilibrium condition.

$$\frac{d(\rho_{HP}V_{HP})}{dt} = -\dot{m} \qquad \text{Eq. 5}$$

$$\frac{d(\rho_{LP}V_{LP})}{dt} = \dot{m} \qquad \text{Eq. 6}$$

$$\frac{d(\rho_{HP}V_{HP}u_{HP})}{dt} = -\dot{m}h_{HP} \qquad \text{Eq. 7}$$

$$\frac{d(\rho_{LP}V_{LP}u_{LP})}{dt} = \dot{m}h_{HP} \qquad \text{Eq. 8}$$

In the above, the mass-flow rate, $\dot{m}$, is calculated by first determining if the pressure difference between the two tanks exceeds the critical pressure ratio. If this is the case, the nozzle is assumed to be choked and the mass-flow rate is set based on the choked flow conditions and the throat area assuming isentropic expansion from the HP tanks conditions (i.e., $\dot{m} = \rho^* A_{\mathrm{th}} c^*$). Otherwise, the mass-flow rate is determined based on an isentropic expansion from the HP conditions to the LP pressure and a defined nozzle outlet area.

The evolution of CO_2 properties in the HP and LP tanks for a desired HP target pressure of 130 bar and temperature of 45 °C for charging and blowdown processes is shown in Fig. 5. During the charging process, the CO_2 charge is moved from the LP to the HP tank leading to an increase in the HP tank pressure and temperature while the LP tank pressure and temperature drops. A control strategy with alternating compression and cooling stages may be employed to achieve the desired HP tank pressure and temperature (inset Fig. 5-left). In the blowdown process, compressor is isolated and CO_2 expands from the HP tank to the LP tank through the nozzle.

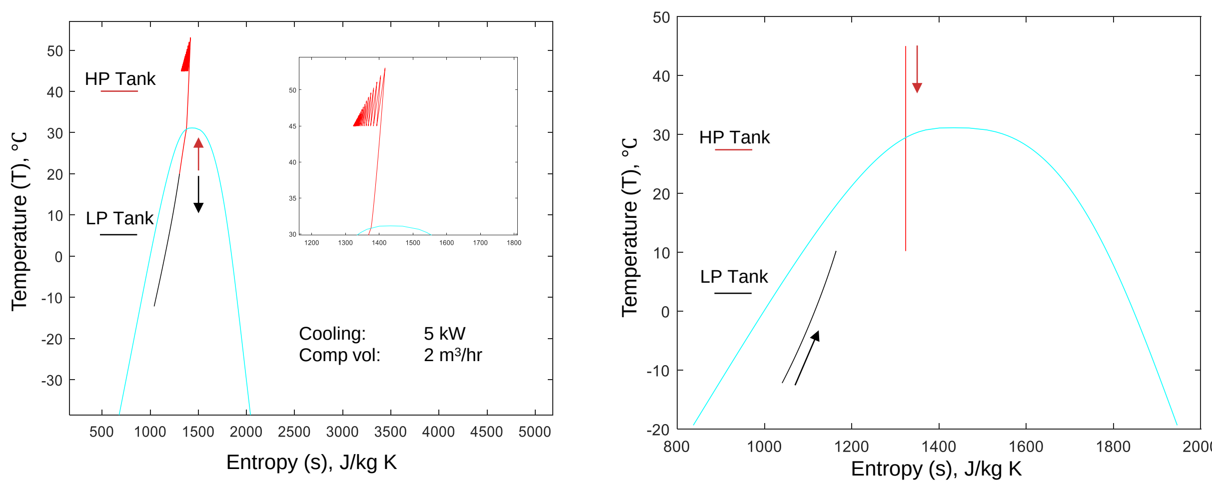

Fig. 5. CO_2 property evolution during charging (left) and blowdown (right) processes

Further to this, the lumped mass model was used to carry out parametric studies to evaluate the optimum CO_2 charge, tank volumes, blowdown time and HP tank cooling power (Fig. 6). The tank vessel volumes have an additional constraint of the overall pressure vessel cost. The optimum values for total CO_2 charge, HP tank volume, LP tank volume have been selected to 100 kg, 100 litre and 185 litre respectively.

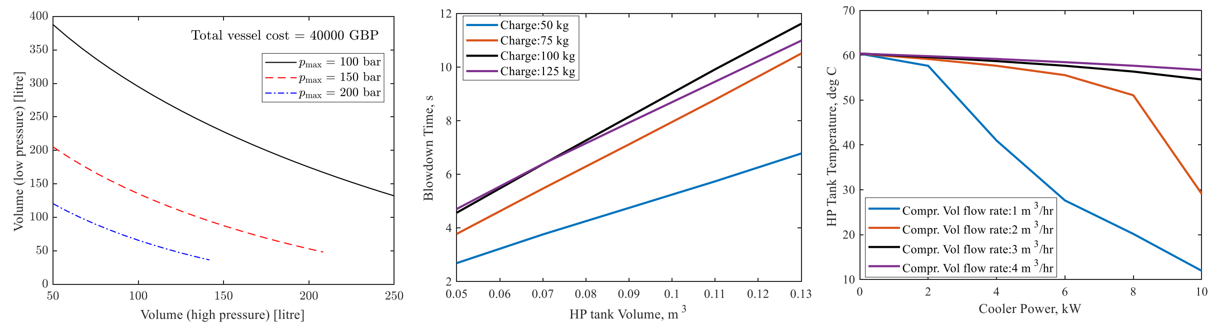

Fig. 6. Parametric studies to evaluate tank volumes, blowdown time and cooler power

sCO_2 blowdown nozzle modelling

The preliminary nozzle design is carried out using quasi-1D nozzle theory to estimate the evolution of nozzle conditions (pressure, vapour quality etc.) under varying pressure ratios across the nozzle. The model assumes thermal equilibrium between the phases (T_{liquid} = T_{vapour} = T_{sat}, c_{liquid} = c_{vapour}) which can be rationalised near the critical point where there is a small difference in density of the two phases and surface tension is low. Under this assumption, the behavior of CO_2 in both the supercritical and two-phase regions is analogous to a single-phase fluid, and thus conventional quasi-1D nozzle theory can be used to establish the conditions within the nozzle for various nozzle pressure ratios. Further to this, it is assumed that the inlet conditions remain constant. The constant inlet condition can be achieved by using an additional regulation valve between the HP tank and the nozzle inlet and this condition is achieved for a fraction of the overall blowdown time. The plots shown in Fig. 7 (left) report the pressure profile for a fixed reservoir condition and variable outlet pressure. For p_{out} < 93 bar it can be observed that normal/oblique shock occurs at the nozzle exit leading to a full expansion within the nozzle, and two-phase conditions expected within the nozzle. The plot Fig. 7 (centre) reports the expected vapour quality in the nozzle. The plot shown in Fig. 7 (right) reports the possibility of shocks transitioning from two-phase conditions upstream of the shock to supercritical conditions downstream of the shock which needs further investigation during the actual testing. Finally, the nozzle model can be used to estimate the evolution in the nozzle conditions during the blowdown process (i.e., varying reservoir conditions).

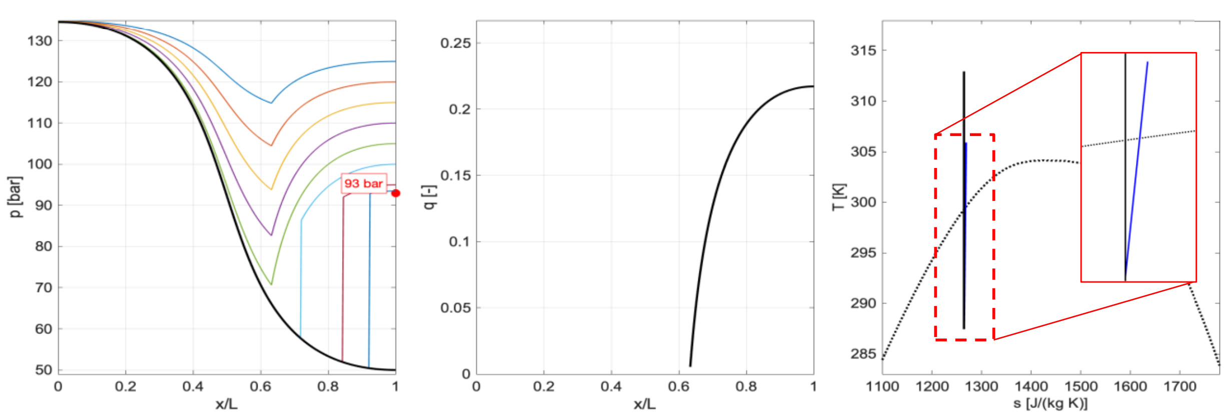

Fig. 7. Evolution of CO_2 conditions across the nozzle during the blowdown process

Conclusion

This work highlights the sCO_2 cycle analysis and test rig design to study the condensation effects in the sCO_2 compressors operating near the critical region. A simple recuperated cycle is considered for the analysis due to its relevance in waste heat recovery. The analyses confirm the maximisation of cycle efficiency near the critical point and hence the sCO_2 compressor operating at this point may be subjected to excursions into the two-phase zone and condensation at the inlet. The test rig design to study this effect is based on batch operation with recharging to operate in a closed loop. A detailed lumped mass model has been developed to model the charging and blowdown processes. The lumped mass model is then utilised to carry out parametric studies to find the optimal values for component sizing. Quasi-1D nozzle theory is used to predict the evolution of CO_2 properties in the nozzle as it expands from the HP to the LP tank in the blowdown process. The model predicts the observation of intended two-phase conditions of CO_2. Currently, the work is being extended to specify and procure the major components and finalise the nozzle/test section design. Optical instrumentation and pressure transducer selection are being evaluated to measure density and static pressure to quantify and analyse two-phase expansion within the nozzle.

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