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Numerical modeling of wave rotor pressure exchanger for supercritical CO_2 Brayton cycles

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Abstract: The supercritical CO_2 Brayton cycle is a novel option for providing efficient and clean power. The pressure exchange between the low-pressure and high-pressure sides consumes much of the energy provided by the cycle. To reduce the back-work ratio, the use of wave rotors to exchange pressure has been proposed as an alternative to turbomachinery in an attempt to reduce rotational speed and shaft work required. However, modeling this method remains a key challenge. In this work, a compressible numerical model for the pressure exchange process in a wave rotor running at 16550 rpm. The mode provides a reliable method to predict the mass flow rates of high-pressure and low-pressure ports at various cycle phases. Results at a pressure ratio of 1.93 indicate that flow velocities at inlet and exit ports are sufficiently high to drive a significant mass flow rate.
Keywords: Pressure exchanger, Wave rotor, Brayton cycle, Fluid dynamics

1. Introduction

Efficient Power generation is critical to the global economy (Guo et al., 2022). Supercritical carbon dioxide Brayton cycles are an emerging power generation cycle (Liu et al., 2019). This closed cycle employs CO_2 as a working fluid for its low cost and as a potential Carbon sink to help reduce greenhouse effects (Crespi et al., 2017). The high density of carbon dioxide near its critical point results in a significant boost to the power density over conventional air cycles. In addition, a lower back-work ratio is expected due to the low compressibility factor of CO_2 under these conditions (Chan et al., 2023). A supercritical CO_2 Brayton cycle typically includes a generation unit consisting of a compressor and a turbine coupled together. The main issue with the supercritical cycle is that the high density of CO_2 requires a low flow rate, resulting in very high rotational speed requirements of the compressor and turbine (Conboy et al., 2012). This requires a significant change in design and presents very high structural and material requirements (Pasch et al., 2012).

As a solution to this issue, (Chan et al., 2023) proposed the use of a wave supercharger as a pressure exchange device in place of the compressor coupled with the back-work part of the turbine. This device results in an increase in the pressure in the upstream part of the cycle without the need for a shaft or complex multistage turbomachinery. However, the high rotational speed and high velocities involved require special consideration during modeling to accurately evaluate the reliability of proposed designs.

In this work, a numerical model is developed to simulate the flow and energy interactions inside a wave rotor. As a pressure wave moving at the speed of sound is expected to develop along with many reflections, the simulation considers time steps close to the Courant–Friedrichs–Lewy criteria with a detailed spatial discretization. The development of the flow rate and fluid variables is analyzed over a full cycle.

2. Materials and Methods

A numerical model of the supercharger is developed to study the exchange of mass flow between the ports on the two sides of the cycle.

A schematic of the device is shown in Figure 1a. The wave rotor pressure exchanger consists of a number of long channels spanning the distance between two end flanges. The end flanges have ports connected to the inlet and outlet flow paths at different pressures. The port configuration is such that the low-pressure flow is introduced to the channel, and then the flow at the highest pressure and temperature is allowed to flow in so that its pressure creates a pressure wave. The ports are then closed to allow the pressure wave to propagate and increase the pressure to the cycle high pressure. Port 2 then opens to the high-pressure side upstream of the energy input part of the cycle, allowing the pressure to fall in the device, and the flow to fall to the conditions of the low-pressure side.

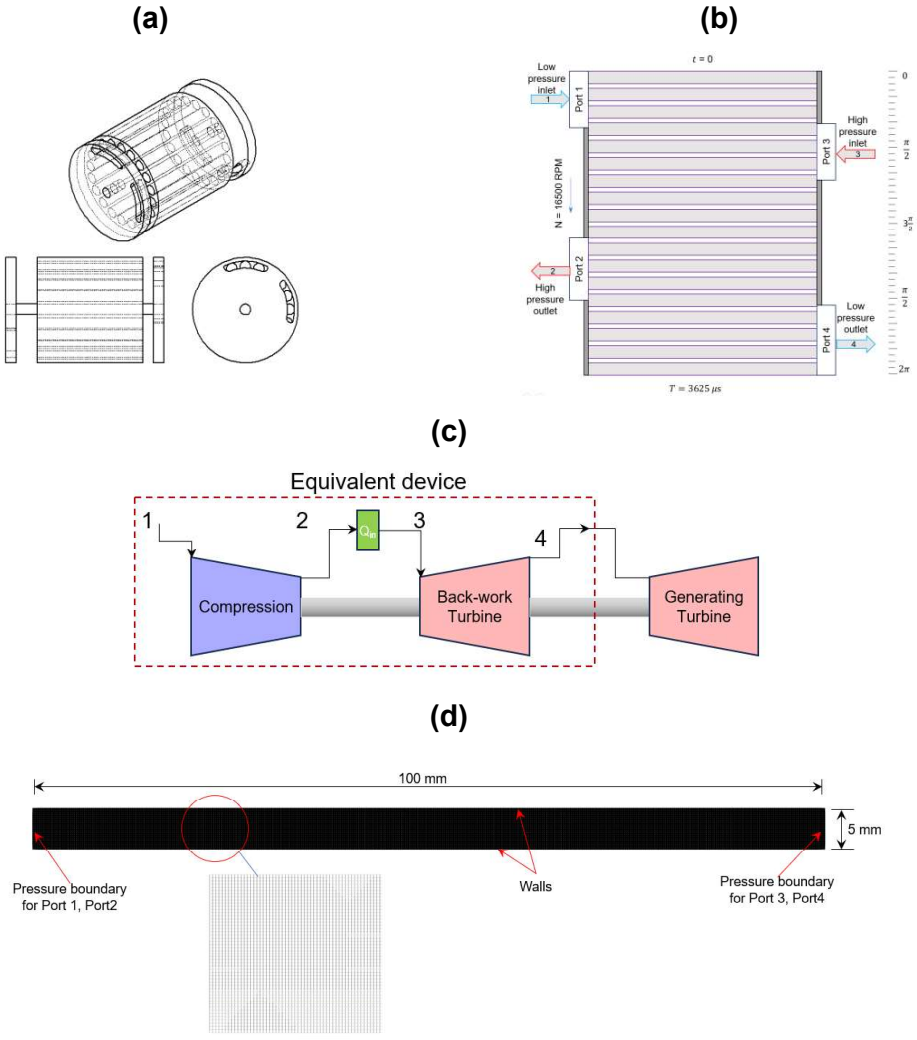


Figure 1. (a) A schematic of the device, (b) the modelled arrangement of the channels at different cycle phases, (c) the equivalent turbomachinery arrangement, and (d) the computational grid of the channel showing boundaries.

A numerical model based on the unwrapping of the path of a single channel across a full cycle. The single channel effectively represents all other channels with a delay according to the phase as in Figure 1b. The model considers the development of the flow parameters as the boundary conditions of a single channel change between wall conditions and port conditions on the end flange sides. The walls of the channels are considered adiabatic walls as the short time span of the cycle should not allow for heat exchange across the walls. The numerical mesh consists of $5x10^4$ orthogonal cells. The effect of the number of cells on the results is minimal at the considered conditions as a mesh convergence study concluded. The channel is 100 mm x 5 mm and is a two-dimensional channel with no consideration of the change in the depth direction. Results show that the behavior is nearly one-dimensional.

A second-order time discretization scheme is used with a time step of $1.81x10^{-6}$ s. This allows for 2,000-time steps per cycle. This value was selected to sufficiently resolve the solution time to allow for accurate calculation of the pressure wave, as the convergence is typically dominated by the Courant–Friedrichs–Lewy number rather than the flow Courant number which is typically lower (Abdelmwgoud et al., 2017). At an element size of 0.1 mm and an acoustic velocity of 443 m/s, the Courant–Friedrichs–Lewy number is less than 10, which is tolerated by the implicit solver.

According to the cycle conditions, the pressure and temperature at the different ports are set to the conditions at the pressure and temperature like those specified by (Chan et al., 2023) for comparison.

Table **1** indicates the conditions of the found boundaries at a pressure ratio of 1.93.

Table 1. The boundary conditions at different ports

Boundary	Type	p [Pa]	T [K]
port1	pressure inlet	7.775	335
port2	pressure outlet	13.900	
port3	pressure inlet	13.500	810
port4	pressure outlet	7.400	

To simulate the operation of the cycle, the port conditions are varied across the cycle according to the position of the channel. At a rotational speed of 16,550 rpm, the full cycle repeats after 3,625 µs. Table 1 shows the detailed development of the boundary conditions across a full cycle.

The fluid properties of supercritical CO_2 are based on the National Institute of Standards and Technology (NIST) Thermodynamic and Transport Properties of Refrigerants and Refrigerant Mixtures Database (Lemmon et al., 2018). Carbon dioxide is considered as a real gas with properties dependent on the pressure and temperature. The Navier-Stokes equations are solved using Ansys fluent density-

based implicit solver. The gradients are discretized using a linear upwind second-order scheme.

Table 2. Boundary condition variations across a complete cycle

step #	*t* [μs]	event	left boundary	right boundary
0	0	port1 opens	port1	wall
1	362	port3 opens	port1	port3
2	580	port1 closes	wall	port3
3	942	port3 closes	wall	wall
4	1486	port2 opens	port2	wall
5	1848	port4 opens	port2	port4
6	2392	port2 closes	wall	port4
7	2755	port 4 closes	wall	wall
8	3625	cycle restarts	port1	wall

3. Results

Figure 2 shows the change of the main flow variables over a full cycle. In Figure 2a, the pressure wave is evident as the driver of the flow. At first, the pressure is stable at the low-pressure value. Once port 2 opens, the pressure wave propagates from the right side over two stages, where the reflection of the wave is seen to propagate and cause an increase in the pressure. Several reflections can be seen over the phases between π and $3\pi/2$.

In Figure 2b, the temperature field can be seen as more stable, where the inlet air at port 2 sweeps the channel. This is followed by the opposite flow direction where the channel is then filled with the higher temperature gas, and then the cycle repeats.

In Figure 2c, the density field can be seen. While the acoustic waves can still be seen, the density changes are milder than pressure due to the lower expansion coefficient of CO_2 close to the critical pressure.

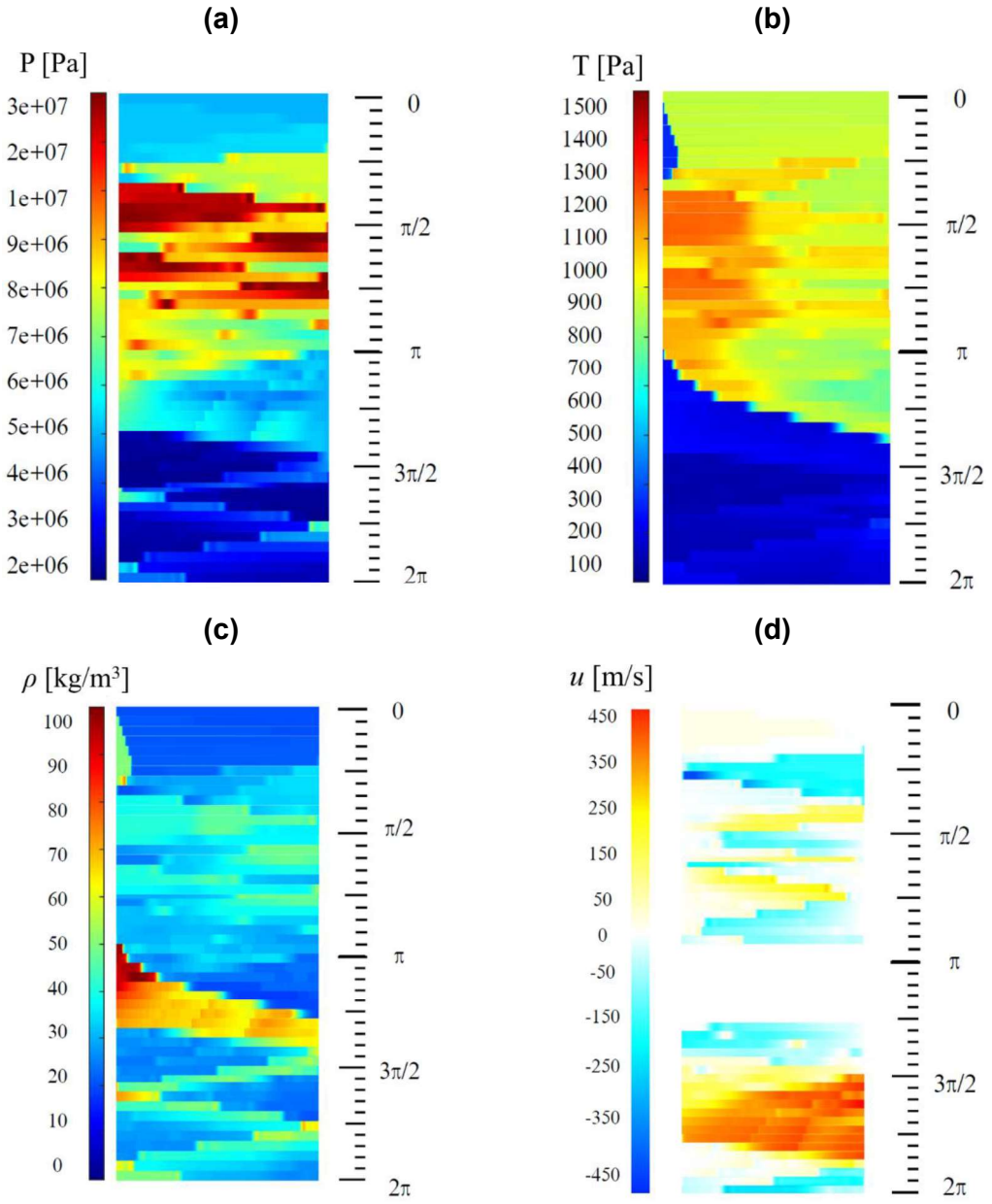
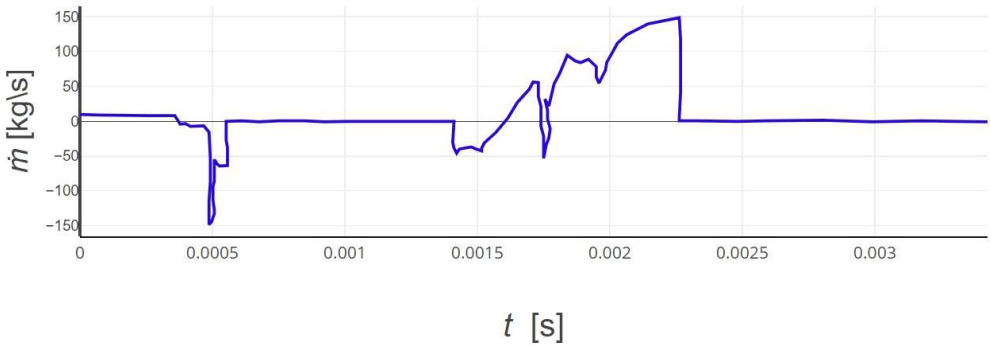


Figure 2. The development of different flow variables across the cycle (a) pressure, (b) temperature, (c) density, and (d) longitudional flow velocity.

In Figure 2d, the flow velocity is seen to alternate between the two ports, effectively driving the mass flow rate across the channel while the pressure is exchanged.

(a) Ports 1 and 2



(b) Ports 3 and 4

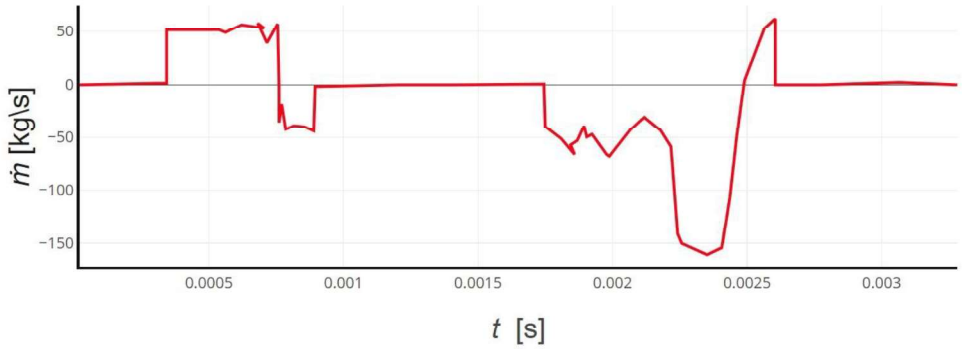


Figure 3. The flow rate across ports of the device over a complete cycle

The mass flow rates across the two end flanges of the channel are shown over a cycle in Figure 3. The results show that the exchange rate of mass between the two ports is not linear or consistent. For instance, the overlap between ports 1 and 2 results in a flow reversal such that some of the low-pressure flow passed from port 1 is expelled through that port before it closes. This effect is increased due to the higher density and lower compressibility of CO_2 at the higher pressure, despite the temperature difference. The bulk of mass transfer across the ports occurs over the opening of ports 3 and 4. Moreover, the closure of port 4 is accompanied by a reversal of the mass flow rate direction, probably a result of a reflected pressure wave that drives the flow outside.

4. Conclusion

In this work, a numerical model is developed to simulate the development of the flow inside a pressure exchanger device for supercritical CO_2. A two-dimensional model of a single channel is simulated over a complete cycle. Results show that the flow is dictated by pressure wave reflections, which have a suppressed effect on density and temperature due to the low compressibility of CO_2. The overlap between port openings, combined with reflected waves, leads to reversed flow at the ports at some point.

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