ANALYSIS OF AN EJECTOR EXPANSION DEVICE IN A TRANSCRITICAL CO₂ AIR CONDITIONING SYSTEM

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ABSTRACT

A two-phase flow ejector model was developed and validated using experimental results. An ejector expansion transcritical CO₂ air conditioning system simulation model was developed by incorporating the two-phase flow ejector model into an existing transcritical CO₂ air conditioning system simulation model. Parametric studies were performed to investigate the effects of ejector design parameters on the system performance of an U.S. Military Environmental Control Unit (ECU) for varying operating conditions. It was found that when both systems have the same gas cooler, evaporator and compressor, the ejector expansion system could have 11% higher COP and a 9.5% higher cooling capacity over the basic transcritical system.

1. INTRODUCTION

Research studies on transcritical carbon dioxide refrigeration systems have drastically increased in recent years because carbon dioxide is being advocated as one of the natural refrigerants to replace CFCs and HCFCs in vapor compression systems. This is mainly due to the excellent thermodynamic and transport properties, and other environmentally friendly characteristics of carbon dioxide. However, lower efficiencies of the basic transcritical carbon dioxide refrigeration cycle compared to its HCFCs and HFCs counterparts is a major hindrance for the technology to make progress towards widespread practical applications. To improve the efficiency of transcritical carbon dioxide refrigeration systems, various innovative ideas and techniques have been put forward, e.g., the use of micro-channel heat exchanger, optimal control of high side pressure, and the use of expansion work recovery machines. The thermodynamic analysis of the transcritical carbon dioxide refrigeration cycle indicates that recovery of the expansion losses that occur during the isenthalpic expansion process can be one of the key issues to improve the system efficiency (Robinson, 1998) (Brown, 2002). Among various expansion work recovery schemes, the ejector has the advantages of simplicity, reliability and availability compared to other devices.

Although ejectors have been widely used in the refrigeration and other industries for many years, the primary ejector applications use single-phase working fluids. In comparison, only few studies can be found in the literature on two-phase flow ejectors, as used in ejector expansion refrigeration cycles. However, the design parameters and the operation conditions of a transcritical two-phase flow ejector are significantly different than the ones for a single-phase application. In addition, the interaction of the ejector expansion device with other system components such as compressor, gas cooler and evaporator is not well understood. Thus, a detailed theoretical and experimental investigation of a two-phase flow ejector has been performed. For this purpose, an ejector expansion transcritical CO₂ air conditioning system simulation model has been developed and parametric studies were performed to investigate the effects of ejector design parameters on system performance for varying operating conditions.

2. TWO-PHASE FLOW EJECTOR MODEL

A two-phase flow ejector model has been established by combination of the one-dimensional mass, momentum and energy conservation equations with critical flow conditions of two-phase flow. The
working processes of an ejector are shown in detail in Figures 1. The motive stream expands in the motive nozzle from the high pressure $P_1$ to the receiving chamber pressure $P_b$. The enthalpy reduces from $h_1$ to $h_{mb}$ and the velocity increases to $u_{mb}$. The suction stream expands in the suction nozzle from pressure $P_2$ to $P_b$. The enthalpy reduces from $h_2$ to $h_{sb}$ and the velocity increases to $u_{sb}$. The two streams mix in the mixing section and become one stream with pressure $P_m$ and velocity $u_{mix}$. This stream further increases its pressure to $P_3$ in the diffuser by converting its kinetic energy into internal energy.

![Figure 1: Schematic of the Ejector Working Processes](image)

2.1. Model of Motive Nozzle
The major assumptions for the motive nozzle flow model are listed as follows:
- The flow inside the motive nozzle is a steady, one dimensional flow.
- The nozzle is a converging nozzle and its throat is at its exit.
- At the nozzle throat, the flow reaches the critical flow condition.

The equations for motive nozzle model are:

$$\eta_m = \frac{h_1 - h_{mb}}{h_1 - h_{sat}}$$  \hspace{1cm} (1)
$$h_i = h_i + \frac{V_i^2}{2}$$  \hspace{1cm} (2)
$$m_m = \rho_i A V_i$$  \hspace{1cm} (3)
$$\rho_i = \frac{1}{\frac{x_i}{\rho_{g,s}} + \frac{1-x_i}{\rho_{f,s}}}$$  \hspace{1cm} (4)
$$V_c = \left( \frac{v_m^2 (h_g - h_f)}{(v_g - v_f)(h_m - v_m) - v_m'(h_g - h_f)} \right)^{1/2}$$  \hspace{1cm} (5)
$$v_m' = v_{xg}' + (1-x)v_f', \hspace{1cm} h_m' = x_{hg}' + (1-x)h_f'$$  \hspace{1cm} (6)
$$v_g' = \left( \frac{\partial v_g}{\partial p} \right)_s, \hspace{1cm} v_f' = \left( \frac{\partial v_f}{\partial p} \right)_s, \hspace{1cm} h_g' = \left( \frac{\partial h_g}{\partial p} \right)_s, \hspace{1cm} h_f' = \left( \frac{\partial h_f}{\partial p} \right)_s$$  \hspace{1cm} (7)

Equation (1) is the definition of the motive nozzle isentropic efficiency. Equation (2) defines the energy conservation. Equation (3) is the mass conservation. Equation (5), (6) and (7) defines the speed of the two-phase flow when the critical flow conditions as determined with Katto’s principle (Katto, 1968, 1969) for homogeneous equilibrium two-phase flow are reached.
In order to determine the critical mass flow rate through the motive nozzle for a given isentropic efficiency, the pressure $P_t$ of the two-phase flow at the nozzle throat has to be assumed first. Using Equations (1) and (2), the enthalpy $h_t$ and velocity $V_t$ at the throat can be determined. The quality $x_t$ can be determined with $P_t$ and $h_t$. The critical speed $V_c$ can be calculated and compared with $V_t$. The assumed pressure $P_t$ will be iterated until $V_c$ and $V_t$ agree with each other within the convergence criteria. The critical mass flow rate can be calculated using Equation (3).

2.2. Model of Suction Nozzle

In a real ejector, the suction nozzle is typically just a suction chamber. However, to simplify the analysis, the expansion process from the suction inlet to the mixing section inlet is treated in the same way as the expansion process of a converging nozzle. The equations for the suction nozzle flow model are:

\[ \dot{m}_s = \varphi \dot{m}_m \] (8)

\[ \eta_s = \frac{h_s - h_b}{h_t - h_{p,ix}} \] (9)

\[ h_s = h_b + \frac{V^2}{2} \] (10)

\[ \dot{m}_s = \rho_b A_b V_b \] (11)

Equation (8) provides the definition of ejection ratio $\varphi$. When the motive nozzle mass flow rate $\dot{m}_m$ is known, the mass flow rate through the suction nozzle can be calculated using Equation (8). The pressure and velocity at the suction nozzle exit can then be determined using Equation (9), (10) and (11).

2.3. Model of Mixing Section

A schematic of the mixing section of the ejector is shown in Figure 3.

To simplify the model of the mixing section, the following assumptions are made:

- At the inlet plane A-A, the motive stream has a velocity of $V_t$, a pressure of $p_t$, and occupies the same area as motive nozzle exit area $A_t$.
- At the inlet plane A-A, the suction stream has a velocity of $V_b$, a pressure of $p_b$, and occupies the same area as suction nozzle exit area $A_b$.
- At the outlet plane B-B, the flow becomes uniform and has a velocity of $V_m$ and a pressure of $p_m$.
- The mixing section has a constant area $A_m$, which is equal to summation of the motive nozzle exit area $A_t$ and the suction nozzle exit area $A_b$.

The equations for the mixing section flow model are:

\[ \rho_t A_t V_t + \rho_b A_b V_b = \rho_m A_m V_m \] (12)

\[ p_t A_t V_t^2 + p_b A_b V_b^2 + \rho_b A_b V_b^2 = p_m A_m V_m^2 + \rho_m A_m V_m^2 \] (13)
\[ \dot{m}_m (h_t + \frac{V_t^2}{2}) + \dot{m}_s (h_h + \frac{V_h^2}{2}) = (\dot{m}_m + \dot{m}_s) (h_m + \frac{V_m^2}{2}) \] (14)

### 2.4. Model of Diffuser

The equations for the diffuser flow model are:

\[ Ct = \frac{P_d - P_w}{\frac{1}{2} P_m V_m^2} \] (15)

\[ Ct = 0.85 \rho_m \left[ 1 - \left( \frac{A_m}{A_d} \right)^2 \right] \left[ \frac{x_m^2}{\rho_{g,m}} + \frac{(1 - x_m^2)}{\rho_{f,m}} \right] \] (16)

\[ \dot{m}_m h_t + \dot{m}_s h_h = (\dot{m}_m + \dot{m}_s) h_d \] (17)

Equation (15) provides the definition of the pressure recovery coefficient. Equation (16) is based on the correlation proposed by Owen et al. (1992). Equation (17) is the energy conservation equation of the ejector.

Using the four flow models listed above, the ejector model predicts the mass flow rate through the motive nozzle and the pressure at the ejector outlet for given motive and suction stream inlet conditions, a specified ejection ratio, and given isentropic efficiencies of the motive nozzle and suction nozzle.

### 3. VALIDATION OF TWO-PHASE FLOW EJECTOR MODEL

To validate the two-phase flow ejector model, a prototype ejector was designed and built, and installed in an experimental system of an U.S. military Environmental Control Unit (Li 2006). A schematic of the test setup is shown in Figure 4. The test data recorded with the two-phase flow ejector experimental setup are the pressures, temperatures, and CO₂ mass flow rates at the motive nozzle and suction nozzle inlets as well as the CO₂ pressure at the ejector outlet. To validate the two-phase flow ejector model, the motive nozzle and suction nozzle efficiencies had to be determined from the test data. The isentropic efficiencies of both nozzles were determined by matching the model predictions of the motive nozzle mass flow rate and the ejector outlet pressure to the experimental data of eight test runs. Figure 5 presents the measured isentropic efficiencies of the motive and suction nozzles as a function of the motive nozzle inlet pressure.

It can be seen from Figure 5 that the average value of the motive nozzle isentropic efficiency is 0.946 with a standard deviation of 0.006 and the average value of the suction nozzle isentropic efficiency is 0.2655 with a standard deviation of 0.004. These results indicate that the expansion process in the motive nozzle is almost isentropic, which agrees with other researchers’ observation (Ozaki et al. 2004). However, the isentropic efficiency of the suction nozzle is rather low due to that fact that an off-the-shelf T-connector was used to manufacture the suction nozzle. To improve the performance of the overall ejector, an improvement of the suction nozzle isentropic efficiency is the key factor.

Assuming that the motive nozzle isentropic efficiency is 0.946 and the suction nozzle isentropic efficiency as 0.2655, the two-phase flow ejector model predictions of the motive nozzle mass flow rate and the ejector outlet pressure were compared to the test results of forty-eight individual test runs. The deviation between the predictions and experimental results are shown in Table 1. The two-phase flow ejector model predicts the motive nozzle mass flow rate within 2.9% and the ejector outlet pressure within 11.4%.
4. EJECTOR EXPANSION TRANSCRITICAL AIR CONDITIONING SYSTEM SIMULATION MODEL

For the ejector expansion transcritical air conditioning system as shown in Figure 4, an ejector expansion transcritical CO₂ air conditioning system simulation model was developed by incorporating the two-phase flow ejector model into the existing transcritical CO₂ air conditioning system simulation model, ACCO₂, developed by Ortiz et al. (2003). The ejector expansion transcritical air conditioning system model consists of the compressor model, gas cooler model, evaporator model, and separator model in addition to the motive nozzle model, suction nozzle model, mixing section model, and diffuser model that were developed for the two-phase flow ejector. The following assumptions are made:

- The system operates at steady state.
- The pressure drops and heat transfers in the connecting tubes between different components are neglected.
- All throttling processes are isenthalpic.
- The discharge pressure of compressor and the evaporation pressure are specified.
- Both indoor and outdoor air temperatures and air flow rates are specified.

The computation sequence for the ejector expansion transcritical air conditioning system model is as follows:

1. The operating conditions and design parameters of the system are read in.
2. A pressure $P_{\text{diff}}$ is assumed as the diffuser outlet pressure, which is the same as the separator pressure and compressor inlet pressure.
3. Based on the specified compressor discharge pressure $P_{gc}$, the compressor inlet pressure $P_{\text{comp,in}} = P_{\text{diff}}$, and the compressor inlet quality $x_{\text{comp}} = 1$, the mass flow rate through the compressor $\dot{m}_{\text{comp}}$, the power input to the compressor $\dot{W}_{\text{comp}}$ and discharge temperature $T_{gc}$ are determined using the compressor model.

Table I: Statistic Analysis of Ejector Model Predictions Deviation

<table>
<thead>
<tr>
<th>Item</th>
<th>Mean Deviation</th>
<th>Standard Deviation</th>
<th>Maximum Deviation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Motive nozzle mass flow rate (kg/s)</td>
<td>-1.4%</td>
<td>2.1%</td>
<td>2.9%</td>
</tr>
<tr>
<td>Ejector outlet pressure (MPa)</td>
<td>7.4%</td>
<td>3.9%</td>
<td>11.4%</td>
</tr>
</tbody>
</table>
4. The gas cooler capacity $\dot{Q}_{gc}$, the CO$_2$ temperature and pressure at the gas cooler outlet, and the air temperature and pressure at the gas cooler outlet are determined using the gas cooler model.

5. The mass flow rate of the motive stream $\dot{m}_{nozzle}$ is determined using the motive nozzle model. If the calculation failed, the CO$_2$ motive nozzle inlet pressure $P_n$ is throttled to a lower value until the motive nozzle calculation is successful.

6. If the critical mass flow rate of the motive stream is larger than the mass flow rate predicted by the compressor map ($\dot{m}_{nozzle} > \dot{m}_{comp}$), the motive nozzle is operated at non-critical mode and $\dot{m}_{nozzle} = \dot{m}_{comp}$. The non-critical motive nozzle model is used to calculate the nozzle outlet pressure and velocity. If the mass flow rate of the motive stream is smaller than the mass flow rate predicted by the compressor model, the simulation is discontinued as the nozzle area is too small to operate under given operation conditions.

7. The CO$_2$ mass flow rate through the evaporator is determined from the assumed ejection ratio $\phi$ by $\dot{m}_{evap} = \phi \dot{m}_{nozzle}$.

8. Assuming the refrigerant side pressure drop of the evaporator, $\Delta P_{evap}$, the evaporator inlet pressure can be determined from the specified evaporator outlet pressure as: $P_{evap, in} = P_{evap, out} + \Delta P_{evap}$

9. The CO$_2$ inlet conditions (quality) to the evaporator are determined based on an isenthalpic throttling process from the separator pressure $P_{diff}$ to the evaporator inlet pressure $P_{evap, in}$

10. The evaporator capacity $\dot{Q}_{evap}$, the CO$_2$ temperature and pressure at the evaporator outlet, and the air temperature and pressure at the evaporator outlet are determined using the evaporator model. If the calculated refrigerant side pressure drop of the evaporator is different than the assumed value of $\Delta P_{evap}$, the pressure drop is updated and steps 8, 9 and 10 are repeated until the refrigerant side pressure drop of evaporator converges within a specified tolerance.

11. The suction nozzle model calculations are performed to determine the suction nozzle outlet conditions.

12. The mixing section model calculations are performed to determine the mixing section outlet conditions.

13. The diffuser model calculations are performed to determine the CO$_2$ pressure and quality at the diffuser outlet.

14. The ejection ratio is calculated. If the value of $\phi$ is different than the assumed value, the ejection ratio is updated and steps 7 through 14 are repeated until the ejection ratio converges within a given tolerance.

15. The calculated diffuser outlet pressure $P_{diff, out}$ is compared to the assumed diffuser outlet pressure $P_{diff}$. If the pressures are not equal, the diffuser outlet pressure $P_{diff}$ is updated and steps 3 through 15 are repeated until the pressure converges within a given tolerance.

5. PARAMETRIC STUDY

A parametric study was performed to investigate the effects of the design parameters of the ejector expansion device on the performance of the ejector expansion transcritical CO$_2$ air conditioning system for a U.S. military ECU. The operating conditions are 26.7 °C indoor air temperature with 50% indoor air relative humidity and 35 °C outdoor air temperature. The rated cooling capacity is 10.55 kW (36000 Btu/h). Both the basic transcritical CO$_2$ system and the ejector expansion transcritical CO$_2$ system are assumed to have the same gas cooler, evaporator, compressor and fans. The fan power inputs for both systems are assumed to be 1.6 kW based on the U.S. military ECU design. The gas cooler and evaporator coils are made of micro-channel heat exchanger slabs with the geometric parameters listed in Table 2. The compressor efficiencies and displacement volume
are listed in Table 3. During the simulation, both systems are assumed to have the same compressor discharge pressure and evaporator outlet pressure.

Table 2: Configuration of Evaporator and Gas Cooler for ECU

<table>
<thead>
<tr>
<th></th>
<th>Evaporator</th>
<th></th>
<th>Gas Cooler</th>
<th></th>
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</thead>
<tbody>
<tr>
<td>Width (mm)</td>
<td>440</td>
<td>Height (mm)</td>
<td>700</td>
<td>Height (mm)</td>
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<tr>
<td>Depth (mm)</td>
<td>50</td>
<td>Tube Rows</td>
<td>70</td>
<td>Tube Rows</td>
</tr>
<tr>
<td>Flow Circuit</td>
<td>140 X 1</td>
<td>Tube Columns</td>
<td>2</td>
<td>Tube Columns</td>
</tr>
<tr>
<td></td>
<td>1240</td>
<td></td>
<td>480</td>
<td></td>
</tr>
<tr>
<td></td>
<td>50</td>
<td></td>
<td>48</td>
<td></td>
</tr>
<tr>
<td></td>
<td>2</td>
<td></td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>Flow Circuit</td>
<td>24 X 4</td>
<td></td>
<td></td>
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Table 3: Compressor Efficiency and Displacement Volume

<table>
<thead>
<tr>
<th>Displacement volume (m³/h)</th>
<th>Volumetric Efficiency</th>
<th>Overall Isentropic Efficiency</th>
<th>Mechanical Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.3858</td>
<td>0.8</td>
<td>0.6</td>
<td>0.75</td>
</tr>
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</table>

The effect of the ejector dimensions on the system performance was studied by changing the motive nozzle exit diameter. For this parametric study, it was assumed that the diameter of the mixing section is 4 mm and the diffuser exit diameter is 8 mm. The motive nozzle exit diameter was changed from 1.4 mm to 1.65 mm and then to 1.8 mm. The discharge pressure is specified at 10.5 MPa. The COPs and cooling capacities of the basic CO₂ based system and the ejector expansion CO₂ based system with three different motive nozzle throat sizes are shown in Table 4. It can be seen from Table 4 that an optimum motive nozzle throat diameter exists for the specified system configuration, where the system reaches maximum COP and cooling capacity. Thus, during the additional simulations, the motive nozzle throat diameter was kept constant at 1.65 mm.

The effect of the mixing section diameter of the ejector expansion device on the system performance was also investigated. The mixing section diameter was changed from 4 mm to 5 mm and then to 6 mm. The motive nozzle throat diameter was kept constant at 1.65 mm and the diffuser exit diameter was kept constant at 8 mm. The COPs and cooling capacities of the basic CO₂ based system and the ejector expansion CO₂ based system with three different mixing section diameters are shown in Table 5. It can be seen from Table 5 that the ejector expansion CO₂ based system reaches maximum COP and capacity when the mixing section diameter is 5 mm. It can also be seen that the ejector expansion CO₂ based system has a 7.9% higher COP and a 7% higher cooling capacity in comparison to the basic CO₂ based system.

The suction nozzle efficiency was fixed at 0.266 during the above simulations, which is based on the test results of the prototype ejector expansion device. Assuming that higher suction nozzle efficiencies can be achieved with a better design of the ejector expansion device, the effect of the suction nozzle efficiency on the system performance was also investigated. Table 6 lists the COPs and cooling capacities of the basic CO₂ based system and the ejector expansion CO₂ based system with three different suction nozzle efficiencies. It can be seen that the ejector expansion CO₂ based system could have an 11% higher COP and a 9.5% higher cooling capacity in comparison to the basic CO₂ based system if the suction nozzle efficiency increases to 0.5. Therefore, during the additional simulations, the suction nozzle efficiency was increased to 0.5.

Table 4: Effect of Motive Nozzle Throat Diameter

<table>
<thead>
<tr>
<th></th>
<th>Basic</th>
<th>Dt = 1.4 mm</th>
<th>Dt = 1.65 mm</th>
<th>Dt = 1.8 mm</th>
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<tbody>
<tr>
<td>COP</td>
<td>1.65</td>
<td>1.67</td>
<td>1.72</td>
<td>1.68</td>
</tr>
<tr>
<td>Qe (kW)</td>
<td>11.50</td>
<td>11.64</td>
<td>11.96</td>
<td>11.79</td>
</tr>
</tbody>
</table>
The cooling COPs and cooling capacities of the CO₂ based systems with and without an ejector expansion device as a function of the compressor discharge pressure are shown in Figure 6 and Figure 7, respectively. It can be seen from these figures that the COPs of both systems slightly decrease with an increase of the discharge pressure, while the cooling capacities of both systems increase with an increase of the discharge pressure. In addition, it can be seen from Figures 6 and 7 that the COP and cooling capacity of the ejector expansion CO₂ based system are approximately 11% and 9% higher, respectively, than the ones of the basic CO₂ based system. Furthermore, it has to be noted that the ejector expansion system can not operate at discharge pressures of 10 MPa or lower because the increase of the compressor mass flow rate exceeds the motive nozzle critical mass flow rate for the given design parameters and operating conditions. Based on these observations, a high-side pressure of 10.5 MPa is recommended for the given design parameters and operating conditions.

![Figure 6: COP of CO₂ based systems with and without ejector expansion versus discharge pressure](image1)

![Figure 7: Cooling capacity of CO₂ based systems with and without ejector expansion versus discharge pressure](image2)

### 6. CONCLUSIONS

A two-phase flow ejector expansion device model was developed and validated with experimental results. It was found that the motive nozzle expansion process had an isentropic efficiency of 95% but the suction nozzle had a very low isentropic efficiency of 26%. An ejector expansion transcritical air conditioning system simulation model was developed. Parametric studies were performed to investigate the effects of ejector design parameters on the system performance. It was found that when both systems have the same gas cooler, evaporator and compressor, the ejector expansion system could have 11% higher COP and a 9.5% higher cooling capacity over the basic system for the given design of a U.S. Military ECU and operating conditions.
## NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Units</th>
<th>Subscripts</th>
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<tbody>
<tr>
<td>$A$</td>
<td>area</td>
<td>(m$^2$)</td>
<td></td>
</tr>
<tr>
<td>$C_t$</td>
<td>pressure recovery coefficient</td>
<td>(-)</td>
<td>b</td>
</tr>
<tr>
<td>$D$</td>
<td>diameter</td>
<td>(m)</td>
<td>d</td>
</tr>
<tr>
<td>$h$</td>
<td>enthalpy</td>
<td>(kJ/kg)</td>
<td></td>
</tr>
<tr>
<td>$i$</td>
<td>inlet</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$m$</td>
<td>mass flow rate</td>
<td>(kg/s)</td>
<td>m</td>
</tr>
<tr>
<td>$p$</td>
<td>pressure</td>
<td>(Pa)</td>
<td>s</td>
</tr>
<tr>
<td>$s$</td>
<td>suction nozzle</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$t$</td>
<td>motive nozzle exit</td>
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<td></td>
</tr>
<tr>
<td>$x$</td>
<td>quality</td>
<td>(-)</td>
<td></td>
</tr>
<tr>
<td>$\phi$</td>
<td>ejection ratio</td>
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<tr>
<td>$\eta$</td>
<td>isentropic efficiency</td>
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<tr>
<td>$\rho$</td>
<td>density</td>
<td>(kg/m$^3$)</td>
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## REFERENCES


