

# CYCLE DESIGN OF REVERSE BRAYTON CRYOCOOLER FOR HTS CABLE COOLING USING EXERGY ANALYSIS

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**Abstract** The reliability and price of cryogenic refrigeration play an important role in the successful commercialization of High Temperature Superconducting (HTS) cables. For cooling HTS cable, sub-cooled liquid nitrogen (LN<sub>2</sub>) circulation system is used. One of the options to maintain LN<sub>2</sub> in its sub-cooled state is by providing refrigeration with the help of Reverse Brayton Cryo-cooler (RBC). The refrigeration requirement is 10 kW for continuously sub-cooling LN<sub>2</sub> from 72 K to 65 K for cooling 1 km length of HTS cable [1]. In this paper, a parametric evaluation of RBC for sub-cooling LN<sub>2</sub> has been performed using helium as a process fluid. Exergy approach has been adopted for this analysis. A commercial process simulator, Aspen HYSYS® V8.6 has been used for this purpose. The critical components have been identified and their exergy destruction and exergy efficiency have been obtained for a given heat load condition.

## 1. Introduction

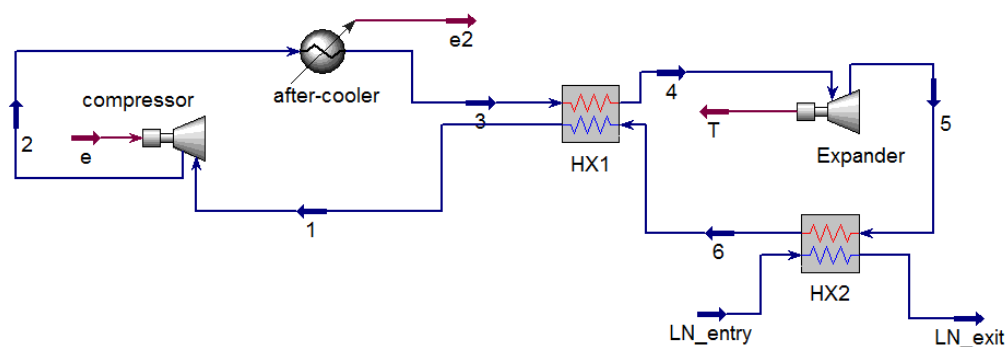
High temperature superconducting power cables are the future of power transmission across world. There are numerous of projects which demonstrate the feasibility of the HTS cable in power transmission [2]. The critical temperature of HTS power cables is in the range of 30 K – 120 K. Resistance offered by HTS cables is negligible when they are operated in below their critical temperature. For cooling HTS cables, sub-cooled liquid nitrogen (LN<sub>2</sub>) circulation system is used. LN<sub>2</sub> is widely used to cool HTS cable because liquid nitrogen is a cheap, nonflammable and a nontoxic cryogen its dielectric strength is high and it supports a large critical current. Sub-cooled LN<sub>2</sub> is thought to be the best cryogen to absorb generated heat from HTS power equipment [3]. One of the options to maintain LN<sub>2</sub> in its sub-cooled state is by providing refrigeration with the help of RBC.

A reversed brayton cycle based refrigerator is a continuous flow recuperative cryo-cooler consisting of an isothermal compressor, adiabatic expander and at least two heat exchangers, one of them being the load heat exchanger (which cools LN<sub>2</sub>). RBC has several advantages over alternative cycle configuration for cryogenic applications. Its advantages are its simplicity, low mass flow rate requirement of refrigerant per unit cooling capacity, small size, light weight, high operating reliability and low level of vibration .



## 2. Modeling

Figure 1 shows the schematic diagram of a 10 kW RBC for sub-cooling of liquid nitrogen from 72 K to 65 K. Compressor and after cooler are operated at ambient condition and other components like He-He heat exchanger (HX1), expansion turbine and He-LN<sub>2</sub> heat exchanger (HX2) are placed in a cryostat. For designing a 10 kW RBC the following assumptions are made i.e. isentropic efficiency of the compressor and turbine is kept constant at 75%, the minimum temperature difference between the warm and cold streams is 5 K for HX1. The pressure drop in each stream is 50 kPa for HX1, 20 kPa for HX2 and 10 kPa after-cooler, sub-cooled liquid nitrogen enters into the cryo-cooler at 72 K and 100 kPa and exits at 65 K and 90 kPa.



**Figure 1.** Schematic diagram of Reverse Brayton Cryocooler

## 3. Theoretical analysis

Sensible heat of sub-cooled LN<sub>2</sub> is used to provide 10 kW of refrigeration to HTS cables. The required mass flow rate of sub-cooled liquid nitrogen in order to provide refrigeration at a rate of 10 kW is given by:

$$\dot{m}_{\text{LN}_2} = \frac{\text{refrigeration effect}}{h_{\text{in}} - h_{\text{out}}} = \frac{10 \text{ kJ/s}}{(146.97 - 132.89) \text{ kJ/kg}} = 0.71 \text{ kg/s} \quad (1)$$

Here,  $h_{\text{in}}$  and  $h_{\text{out}}$  is enthalpy of LN<sub>2</sub> at entry and at exit of He-LN<sub>2</sub> heat exchanger respectively.

### 3.1. Exergy Analysis

The availability or exergy is the maximum useful work obtainable from a system as it reaches the dead state [4]. Quantative evaluation of energy in a cycle or in a process can be done using the first law of thermodynamics. The direction of flow of heat or work is known from the second law of thermodynamics. Second law of thermodynamics dictates that conversion of 100% heat into work is never possible. Exergy analysis helps in finding the following:

- It can be used to determine the type, location and magnitude of energy losses in a system.
- It can be used to find means to reduce losses to make the energy system more efficient.

### 3.2. The total exergy associated with a stream of matter

Equation (2) gives the total exergy associated with a stream [5]

$$ex = (h - h_0) - T_0 (s - s_0) + (1/2)V^2 + gz + ex^{ch} \quad (2)$$

Where  $h$ ,  $s$ ,  $V$  and  $z$ , which are specific enthalpy, specific entropy, velocity and height with respect to a datum respectively, are the properties of the stream at the inlet of the control volume.  $T_o$ ,  $h_o$ ,  $s_o$  are the properties corresponding to the outlet i.e. the restricted dead state,  $ex^{ch}$  is the chemical exergy (it is assumed that chemical exergy is constant throughout the whole process).

### 3.3. Exergy destruction

The part of supplied exergy which is destroyed inside the system, is denoted as  $\dot{Ex}_{dest}$ . This is equal to the product of dead-state temperature and the entropy generation ( $S_{gen}$ );

$$\text{So, } \dot{Ex}_{dest} = T_o S_{gen} \quad (3)$$

Equation for exergy destruction will be given as:

$$\dot{Ex}_{dest} = \underbrace{\left[ \sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out} \right]}_{\text{Exergy\_Change\_massflow}} + \underbrace{\sum_j \left( 1 - \frac{T_o}{T_j} \right) \dot{Q}_j}_{\text{Exergy\_Transfer\_Heat}} - \underbrace{\dot{W}_{cv}}_{\text{Exergy\_Transfer\_Work}} \quad (4)$$

### 3.4. Exergy efficiency

Overall exergy efficiency of the entire process can be expressed using eqn. (5) and exergy destruction and exergy efficiency of different components is given in Table 1 [5].

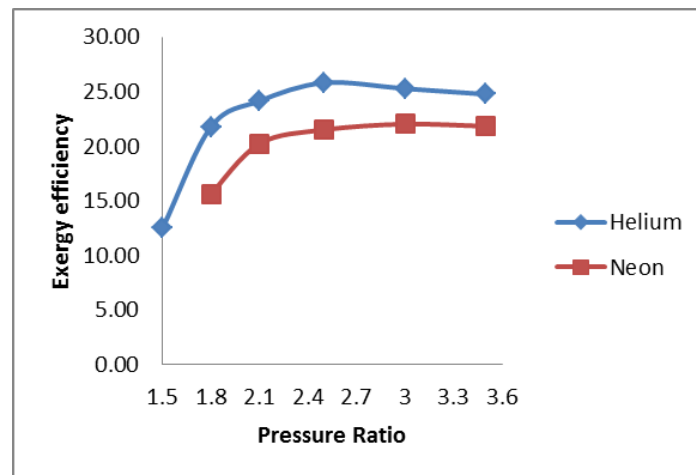
$$\eta_{ex} = 1 - \frac{\Sigma(\text{exergy losses in each components})}{(\text{exergy expenditure})} = 1 - \frac{\Sigma(\text{exergy losses in each components})}{\Sigma(\text{Total work supplied})} \quad (5)$$

**Table 1.** Exergy destruction and exergy efficiency of different components

Equipment	Exergy destruction	Exergy Efficiency
Compressor	$\dot{Ex}_{dest} = \left[ \sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out} \right] - \dot{W}_{comp}$	$\eta_{ExComp} = \frac{\dot{m}(Ex_{out} - Ex_{in})}{\dot{W}_{COMP}}$
After-cooler	$\dot{Ex}_{dest} = \left[ \sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out} \right]$	$\eta_{Excooler} = Ex_{out} / Ex_{in}$
Heat Exchanger	$\dot{Ex}_{dest} = \left[ \sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out} \right]$	$\eta_{Excooler} = Ex_{out} / Ex_{in}$
Turbine	$\dot{Ex}_{dest} = \left[ \sum \dot{m}_{in} ex_{in} - \sum \dot{m}_{out} ex_{out} \right] - \dot{W}_{Exp}$	$\eta_{ExEXP} = \dot{W}_{EXP} / \dot{m}(Ex_{in} - Ex_{out})$

### 3.5. Selection of Refrigerant

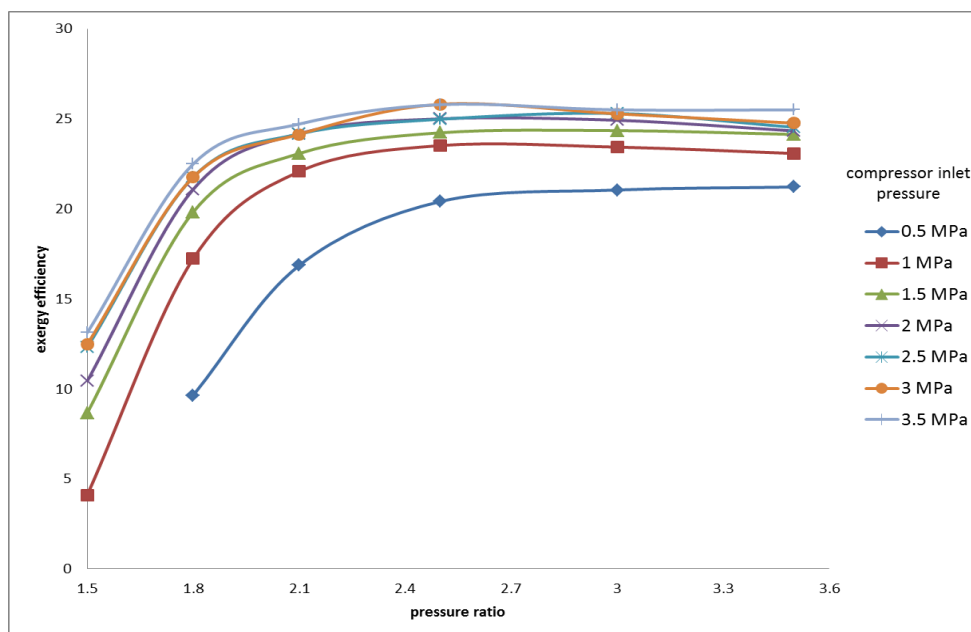
Simulation shows that the helium is more thermodynamically efficient refrigerant than neon. Result from the simulation is shown in Figure 2. This figure shows that the exergy efficiency of helium is more than that of neon for various pressure ratios (compressor outlet pressure/ compressor inlet pressure) for 3 MPa compressor inlet pressure. The variation in their exergy efficiency is due to the real-gas behavior of helium (especially, the pressure-dependence of enthalpy).



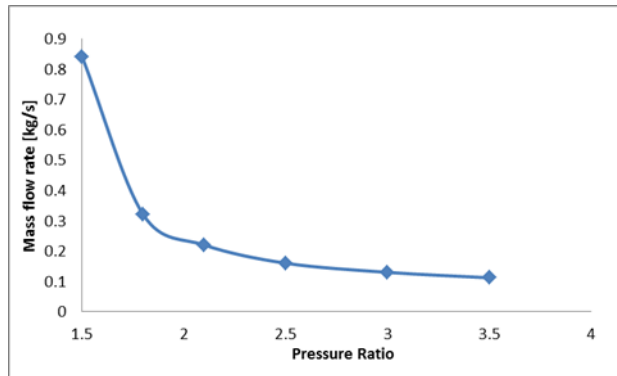
**Figure 2.** Exergy efficiency of RBC as a function of  $r_p$  for 3MPa compressor inlet pressure with helium and neon as refrigerant.

## 4. Results and discussion

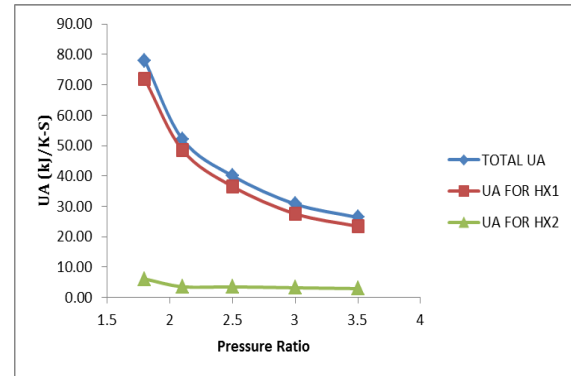
Exergy analysis of 10 kW RBC has been done through parametric analysis, the effects of different process parameters on the performance of the cycle have been simulated in Aspen HYSYS® V8.6. Different sets of pressure ratio (1.5, 1.8, 2.1, 2.5, 3, 3.5) for different compressor charging pressure (0.5, 1, 1.5, 2, 2.5, 3 and 3.5 MPa) is taken into account for simulation of 10 kW RBC and their exergetic efficiency is calculated which is shown in Figure 3.



**Figure 3.** Exergy efficiency as a function of  $r_p$  for various compressor inlet pressure



**Figure 4.** Required mass flow rate of helium as function of  $r_p$  for 3 MPa compressor inlet pressure



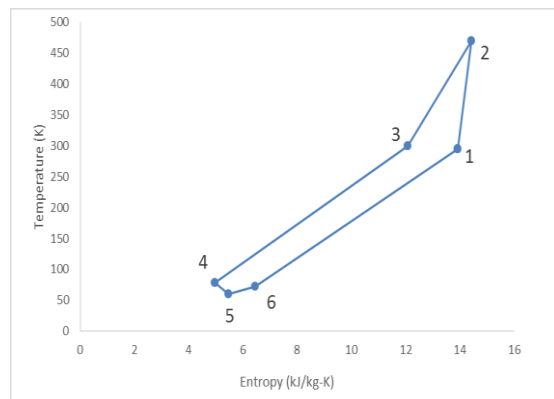
**Figure 5.** UA as a function of  $r_p$  for 3 MPa compressor inlet pressure

Figure 4 shows the variation of mass flow rate of helium as a function of  $r_p$ . As the  $r_p$  increases the required mass flow rate of helium decreases because refrigeration effect of cycle is kept constant at 10 kW. Figure 5 shows the variation of UA (overall heat transfer coefficient, kJ/K-s) value of HX1, HX2 as a function of  $r_p$ . As  $r_p$  increases the size of heat exchanger decreases.

Figure 3 explains the variation of exergy efficiency which is a function of compressor  $r_p$  with respect to compressor inlet pressure. From Figure 3, it is clear that up to  $r_p = 2.5$  exergy efficiency increases and after  $r_p = 2.5$  it decreases slightly. The exergy efficiency at  $r_p = 2.5$  and 3 MPa compressor inlet pressure is 25.80%. As the exergy efficiency is optimum at this condition, therefore operating condition of the designed cycle is taken as  $r_p = 2.5$  and compressor inlet pressure is taken as 3 MPa. For higher value of  $r_p$  ( $r_p > 2.5$ ), the ratio of rate of irreversibility to the work input to the compressor increases and because of this, there is decreasing trend of exergy efficiency for  $r_p > 2.5$ . The complete thermodynamic properties of helium at the proposed designed condition are listed in TABLE 2, and the corresponding temperature-entropy diagram is shown in Figure 6.

**Table 2:** Thermodynamic properties of helium for designed cycle

S.No.	T (K)	P (MPa)	h(kJ/kg)	s (kJ/kg-K)
1	295	3	-18.26	13.91
2	469.2	7.5	888.2	14.42
3	300	7.49	6.129	12.09
4	77.79	7.44	-1163	4.974
5	59.97	3.07	-1250	5.481
6	71.65	3.05	-1188	6.448



**Figure 6:** Designed Brayton-cycle cryocooler on temperature-entropy diagram of helium

Gas bearing turbo expander system is selected for this cycle as it gives stable performance and runs a system for the longer time [7]. Ghosh [8] has described a procedure to obtain the parameter such as diameter of wheel and specific speed of turbine. By using his methodology and the data obtained from the steady state analysis, specific speed of turbine and diameter of turbine wheel is calculated.

**Specific speed ( $n_s$ )**

$$n_s = \omega * \frac{Q^{1/2}}{(\Delta h)^{3/4}} \quad (6)$$

**Specific diameter ( $d_s$ )**

$$d_s = \frac{D * (\Delta h)^{1/4}}{Q^{1/2}} \quad (7)$$

Where,  $\omega$  is rotational speed (rad/s)  $\Delta h$  is adiabatic enthalpy drop across turbine wheel (J/kg)  $Q$  is volume flow rate ( $m^3/s$ ) and  $D$  is diameter of the wheel at inlet (m). Ghosh [8] has suggested the range of  $d_s$  (0.4-0.7) and  $n_s$  (3.5-3.8) for efficient design of radial in-flows turbine.  $n_s$  and  $d_s$  value is assumed to be 3.5 and 0.5 respectively for this paper.

Putting these values in equation (6) and (7)

We get,

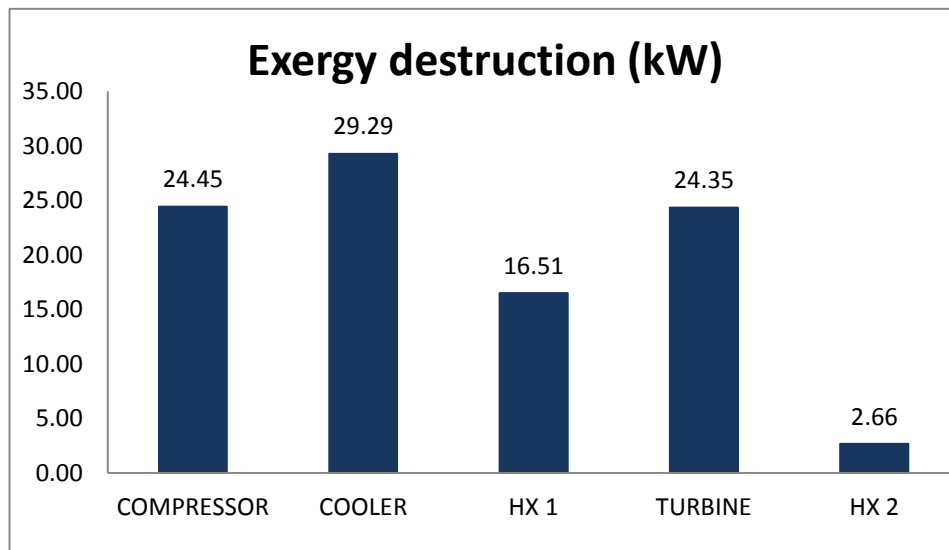
$$D = 0.0535 \text{ m or } 53.5 \text{ mm}$$

$$\omega = 9642 \text{ rad/s or } 92124 \text{ rpm}$$

Aspen EDR is used for designing heat exchangers. Plate fin heat exchangers are chosen for both heat exchangers because they are highly efficient heat exchangers and also they are compact, with approximately ten times the surface area per unit volume as that for conventional shell and tube heat exchangers [6]. Table 3, gives the geometry specification, value, pinch point temperature, metal used in fabrication heat exchangers. UA values calculated from Aspen EDR for HX1 and HX2 are 32.9 kJ/K-s and 4.5 kJ/K-s. Centrifugal compressor can be used for compressing helium gas. Mass flow rate of helium through the compressor is 0.16 kg/s.

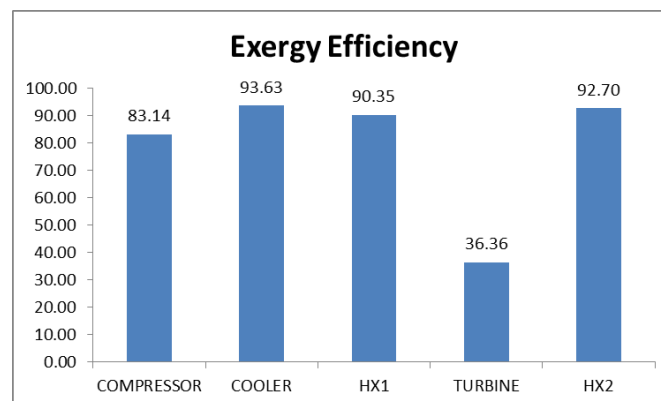
**Table 3:** Result of heat exchangers (HX1 and HX2) design obtained from Aspen EDR

Calculation mode	Design (HX1)	Design (HX2)
Overall heat transfer calculated (kW)	186	10
Pinch point temperature (K)	5 K	0.75 K
UA value of calculated duty (kJ/K-s)	32.9	4.5
Number of exchangers in parallel	1	1
Number of layers per exchanger	104	25
Number of fins	4	4
Core length (mm)	1597.83	601.04
Core width (mm)	596.89	206.15
Core depth(stack height) (mm)	929.9	216.5
Internal (effective) width (mm)	573.89	183.15
Parting sheet thickness (mm)	2	1
Exchanger metal	Aluminium	Aluminium
Exchanger weight (kg)	1315.4	35.2



**Figure 7.** Exergy Destruction of different components and total exergy destruction in cycle at designed condition.

Figure 7 shows exergy destruction of different component at design condition. The bar chart clearly indicates that maximum exergy is destroyed in after-cooler and the minimum exergy is destroyed in He-LN<sub>2</sub> heat exchanger. Net Exergy provided in the cycle is 131.09 kW out of which 97.26 kW is destroyed.



**Figure 8.** Exergy efficiency of different components at designed condition

Exergy efficiency of different components is shown in Figure 8. Turbine is the least exergetic efficient component of the cycle. Exergy efficiency of the cycle can be increased by increasing the exergy efficiency of turbine. Further study will be done in future in-order to increase the exergy efficiency of turbine.

## 5. Conclusion

A rigorous thermodynamic study on RBC is performed to develop 10 kW cryo-cooler. By taking into account the performance of the compressor, expander, and heat exchangers, RBC for sub-cooling liquid nitrogen is simulated in Aspen HYSYS® V8.6. It is found out that mass flow rate requirement of helium decreases as the  $r_p$  increases. Exergy losses, exergy efficiencies of each component are calculated. After-cooler is found out to be the most exergetic efficient component whereas turbine is found out to be the least exergetic efficient component in RBC. Furthermore, RBC is a maintenance-free refrigerator. So, it can be considered as one of the promising solution for HTS cable cooling.

## 6. Acknowledgments

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**NOMENCLATURE**

$ex$	-	exergy per unit mass (kJ/kg)	$z$	-	Height (m)
$ex_{in}$	-	exergy in to control volume (kJ/kg)	$\dot{W}$	-	work (kW)
$ex_{out}$	-	exergy out (kJ/kg)	$\omega$	-	rotational speed (rad/s)
$ex^{ch}$	-	chemical exergy (kJ/kg)	$\Delta h$	-	enthalpy drop across turbine wheel (J/kg)
$\dot{E}x_{dest}$	-	Rate of exergy destruction (kW)	$Q$	-	volume flow rate ( $m^3/s$ )
$h$	-	Specific enthalpy (kJ/kg)	$D$	-	diameter of the wheel at inlet (m)
$U$	-	The overall heat transfer Coefficient (kJ/(m <sup>2</sup> -K-s))	$d_s$	-	Specific diameter
$A$	-	The contact area for each fluid side (m <sup>2</sup> )	$n_s$	-	Specific speed
$r_p$	-	pressure ratio	<b>Subscripts</b>		
$\dot{m}$	-	Mass flow rate (kg/s)	gen	-	generation.
$LN_2$	-	liquid nitrogen	o	-	Reference state
$\dot{Q}$	-	Rate of heat transfer (kW)	j	-	Any point
$T$	-	Temperature (K)	cv	-	Control volume
$s$	-	Specific entropy (kJ/kg-K)	dest	-	destruction
$V$	-	Velocity (m/s)	EXP	-	Expander
$P$	-	Pressure (kPa)	comp	-	Compressor
			in	-	entry point
			out	-	exit point

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