Effect Of Pressure Ratios Across Compressors On The Performance Of The Transcritical Carbon Dioxide Cycle With Two-State Compression And Intercooling

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EFFECT OF PRESSURE RATIOS ACROSS COMPRESSORS ON THE PERFORMANCE OF THE TRANSCRITICAL CARBON DIOXIDE CYCLE WITH TWO-STAGE COMPRESSION AND INTERCOOLING

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ABSTRACT

A computer model was developed to perform a thermodynamic analysis of the transcritical carbon dioxide cycle with two-stage compression and intercooling. In typical two-stage compression with intercooling applications, the intercooler serves the purpose to cool the fluid to the lowest possible temperature before it enters the 2nd-stage compressor. Ideally, the fluid temperature at the inlet to the 2nd-stage compressor is the same as the fluid temperature at the inlet to the 1st-stage compressor. In this case, the minimum compression work and thus, the highest system efficiency, is achieved by using the same pressure ratio across both compressors. However, this is not the case for the transcritical carbon dioxide cycle. Due to the supercritical heat rejection of the transcritical cycle and the slopes of the isotherms in the supercritical region, the highest system efficiency may be achieved at pressure ratios of the 1st and 2nd-stage compressors that are significantly different from each other depending on operating conditions. This paper presents the results of the system analysis of the transcritical carbon dioxide cycle with two-stage compression and intercooling and identifies the pressure ratios that provide maximum system efficiency.

INTRODUCTION

The transcritical cycle technology using carbon dioxide as the refrigerant has recently received increased attention as a possible replacement for the vapor compression cycle technology using fluorocarbon-based refrigerants. Carbon dioxide (CO_2) has zero ozone depletion potential and negligible global warming potential as a refrigerant and is also nontoxic and nonflammable.

Figure 1 shows the schematic of the basic transcritical CO_2 cycle and Figure 2 illustrates the cycle on a temperature-entropy (T-s) diagram. As shown in Figure 1, the basic transcritical CO_2 cycle consists of a compressor, a gas cooler, an expansion valve and an evaporator. The cycle is composed of four basic processes; compression (1-2), heat rejection (2-3), expansion (3-4h) and heat absorption (4-1) as shown in Figure 2. In the expansion process, the paths 3-4s, 3-4h and 3-4w represent isentropic expansion, isenthalpic expansion, and expansion through a work producing expansion device, which is referred to as ED-WOW (Expansion Device With Output Work), respectively. In the compression process, the paths 1-2s and 1-2 stand for the isentropic and actual processes, respectively.

The thermodynamic performance of a transcritical carbon dioxide cycle is typically less than the one of a HFC or HCFC based vapor compression cycle. However, since the volumetric heat capacity of CO_2 is up to five times higher than that of current refrigerants, the transcritical carbon dioxide cycle is receiving strong consideration in reduced weight and volume applications, such as automotive air conditioning and military environmental control units. In either application, system improvements may be necessary to meet the goal of the reduction of weight and volume while still maintaining the same or achieving a higher system efficiency than the HFC based vapor compression cycle. One of several methods to increase the system efficiency is the use of two-stage compression with intercooling (Intercooler Cycle).

Figures 3 shows the schematic of the Intercooler Cycle. As shown in this figure, the Intercooler Cycle comprises two compressors, a gas cooler, an expansion valve, an evaporator, and an intercooler. The CO_2 exiting the 1^{st}-stage compressor is cooled down in the intercooler by rejecting heat to the environment. Afterwards it enters the 2^{nd}-stage compressor. Figure 4 shows the T-s diagram of the Intercooler Cycle.

In typical two-stage compression applications, in which the working fluid enters the 2^{nd}-stage compressor at the same temperature as it enters the 1^{st} stage compressor, the minimum compression work is achieved by using the same pressure ratio across both compressors [Wark, 1988]. The intermediate pressure can be identified as follows:

$$p_{inter} = \sqrt{p_{low} \times p_{high}} \tag{1}$$

where, p_{low} and p_{high} stand for low-side and high-side pressures, respectively. As mentioned above, it is desired that the temperature of the CO_2 at the outlet of the 1^{st}-stage compressor ($T_{I1'}$) is cooled down to the temperature at the inlet to the 1^{st}-stage compressor (T_{I1}) by rejecting heat through the intercooler.

$$T_{I2'} = T_{I1} \tag{2}$$

However, the CO_2 intercooler outlet temperature is limited by the heat exchanger effectiveness and the environmental temperature. In the case of an ideal intercooler, the CO_2 intercooler outlet temperature is equal to the environmental temperature. During warm weather periods, the environmental temperature may be larger than the critical temperature of carbon dioxide (30.82°C) [ASHRAE, 1997]. Due to the nature of the transcritical cycle and the slopes of the isotherms in the supercritical region, significantly different pressure ratios for each of the two compressors will be required to achieve the minimum compression work for certain operating conditions. Therefore, this paper studies the variations of the combinations of pressure ratios across the compressors to give the minimum compression work and the resultant maximum system performance depending on operating condition.

COP CALCULATIONS

A computer model was developed based on first law thermodynamic relations to predict the performance of the Intercooler Cycle. The assumptions made in the analysis are listed below:

1. Steady state and steady flow.
2. Compression processes are isentropic and the expansion process is isenthalpic.
3. There are no pressure losses in the heat exchangers and piping.
4. The CO_2 temperature at the outlet of the gas cooler and the outlet of the intercooler is equal to the heat sink/environmental temperature, which is 35°C.
5. The CO_2 enters the 1^{st} stage compressor superheated at a temperature of 20°C.
6. The high-side pressure was set to 10 MPa, which is on average the pressure that provides maximum performance of the transcritical carbon dioxide cycle (Robinson and Groll 1998)

During the analysis the low-side and intermediate pressures were varied. For each low-side pressure, the intermediate pressure varied from the low to the high-side pressure. Table 1 shows the operating conditions for the analysis.

RESULTS AND DISCUSSIONS

Effect of pressure ratio on COP with variation of low side pressure

Table 2 presents the calculated COPs of the Intercooler Cycle in case of equal pressure ratios across the two compressors, and also the maximum COP that can be achieved if different pressure ratios across the compressors are considered as a function of operating conditions. As shown in this table, for the given operating conditions the maximum COP of the Intercooler Cycle with different pressure ratios is larger than the COP with equal pressure ratios across the compressors. The table also illustrates the pressure ratios at which the maximum COP occurs and the percent increase of the maximum COP with different pressure ratios compared to the COP with equal pressure ratios. It can also be observed from Table 2 that the improvement of the maximum COP with different pressure ratios compared to that with equal pressure ratios increases with increasing low-side pressure. These trends are confirmed by the graph that is shown in Figure 5. Figure 5 presents the COP of the Intercooler Cycle for five different low-side pressures as a function of the pressure ratio across the 1st-stage compressor. Note that as the pressure ratio across the 1st-stage compressor varies, the pressure ratio across the 2nd-stage compressor has to follow suit to obtain the total pressure increase from the evaporation to the gas cooler pressure. Instead of pressure ratio across the 1st-stage compressor, the intermediate pressure could serve as the abscissa in Figure 5 and the same result would be witnessed. It can be seen from Figure 5 that the COP curves of the transcritical Intercooler Cycle are different from the "typical bell curve behaviors" that are observed when plotting the COP of a two-stage compression cycle versus the intermediate pressure. As the pressure ratio across the 1st-stage compressor increases, the COP curve initially follows the typical bell curve behavior. However, instead of decreasing after the intermediate pressure given by Equation (1) has been reached, the COP first levels out and then shows a sudden spike, where the maximum in COP occurs. After this spike, the COP sharply decreases with a further increase in pressure ratio across the 1st-stage compressor. As illustrated in Figure 5, the maximum COP consistently occurs at pressure ratios of the 1st-stage compressor that are significantly larger than the pressure ratios of the 2nd-stage compressor. It can also be seen that the sharp spike in the COP is more explicit as the low-side pressure increases from 2.5 MPa to 4.5 MPa.

Table 3 shows the required work inputs for the 1st-stage and 2nd-stage compressors and the heat removal capacity of the evaporator for the cases when the maximum COP has been reached with different pressure ratios. As shown in this table, the total compression work and the evaporator capacity decrease as the low-side pressure increases. However, the decrease of the evaporator capacity is smaller than the reduction of the total compression work and thus the COP increases as the low side pressure increases.

Figure 6 illustrates the COP as a function of pressure ratio across the 1st-stage compressor in more detail for the specific case where the low side pressure p_{low} = 3.5 MPa (Run No. 3). It can be seen from this figure when equal pressure ratios are applied across both compressors, a COP of 3.774 is calculated at the pressure ratio of 1.69. For different first and second stage pressure ratios of 2.32 and 1.23, respectively, a maximum COP of 3.941 is reached, which represents an increase of 4.43% compared to the maximum COP at equal pressure ratios.

As shown in the Figure 5 and Figure 6, the slope of the COP curve of the Intercooler Cycle is zero for pressure ratios from 1.02 to 1.25. The COP value for that range of the pressure ratio is 3.35 and is the same as the COP of the single-stage basic transcritical CO_2 cycle (Figure 1). This is due to the fact that when the CO_2 temperature at the outlet of the 1st-stage compressor, $T_{II'}$, is lower than the ambient air temperature, T_{air}, there is no heat transfer from the intercooler to the environment and the intercooler was omitted. This resulted in a COP that is equal to the COP of the single-stage transcritical cycle. It can be seen from Figure 6 that $T_{II'}$ begins to be larger than T_{air} for pressure ratios larger than 1.25.

Furthermore, it can be seen from Figure 6 that there is sudden increase in COP at a pressure ratio of 2.25 and the COP reaches its maximum point at a pressure ratio of 2.32. This behavior in COP is due to the sudden decrease of total compression work at these pressure ratios. Figure 7 shows a comparison of the required compression work for the case of Run No. 3 as a function of the pressure ratio across the 1st-stage compressor. This figure presents the work required by the 1st-stage and 2nd-stage compressors, and the net work required for both stages. The compression work of the 2nd-stage compressor decreases sharply at a pressure ratio of 2.25 and the total compression work becomes a minimum at the pressure ratio of 2.32 at which the maximum COP occurs. At this point, the pressure ratio across the 2nd-stage compressor is 1.23.

The above behavior of the required compression work is due to the thermodynamic characteristics of the transcritical carbon dioxide cycle. Figure 8 shows the processes of the Intercooler Cycle with equal pressure ratios and the pressure ratios at which the maximum COP occurs on the pressure-enthalpy diagram for the case of Run No. 3. When equal pressure ratios of 1.69 are applied, the intermediate pressure of the intercooler becomes 5.92 MPa. In this case, the paths of the refrigeration processes are I1-I1'$_f$-I2'$_f$-I2$_f$-I3-I4-I1. The subscript "f" represents the case of equal pressure ratio. If different pressure ratios of 2.32 for the 1st-stage compressor and 1.23 for the 2nd-stage compressor are considered, the intermediate pressure is 8.12 MPa. Hence the paths of the processes become I1-I1'$_d$-I2'$_d$-I2$_d$-I3-I4-I1. The subscript "d" represents the case of different pressure ratios. The compression works of the cases of equal pressure ratio and different pressure ratio are given in Equations (3) and (4), respectively:

$$w_s = \left(h_{I1'_s} - h_{I1}\right) + \left(h_{I2_s} - h_{I2'_s}\right) \tag{3}$$

$$w_d = \left(h_{I1'_d} - h_{I1}\right) + \left(h_{I2_d} - h_{I2'_d}\right). \tag{4}$$

The first and second terms on the right hand sides of Equations (3) and (4) stand for the compression works of the 1st-stage and 2nd-stage compressors, respectively. As discussed above, the 2nd-stage compression work decreases suddenly at a pressure ratio of 2.32 across the 1st-stage compressor. It can be seen from Figure 8 that as the intermediate pressure increases above the critical pressure, the 2nd-stage compression process moves quickly into the supercritical region where the slopes of the isentropic lines are steeper that in the conventional superheated region. In fact the inlet state to the 2nd-stage compression process moves to lower enthalpies than the critical enthalpy due to the almost zero slope of the isotherms in the supercritical region. As a result, the enthalpy increase through the 2nd-stage compressor decreases sharply as the intermediate pressure rises across the critical pressure. Just before the slope of the isotherms changes again towards lower enthalpies, the total compression work reaches a minimum and thus, the COP of the Intercooler Cycle becomes a maximum. It has to be noted that in the case when the Intercooler Cycle operates at a maximum COP, the enthalpy difference across the intercooler at the intermediate pressure of 8.12 MPa is significantly larger than the enthalpy difference across the gas cooler at the pressure of 10 MPa and thus, the intercooler is the main heat rejection heat exchanger for the overall cycle.

Consideration of a work producing expansion device to run the 2nd-stage compressor

In the transcritical carbon dioxide cycle, the carbon dioxide usually expands from high pressure to low pressure through an expansion valve as shown in Figure 1. Since this process starts in the supercritical region and ends at 40 to 50% quality in the two-phase region, the 2nd largest amount (after compression) of the cycle irreversibilities occur during this expansion process [Robinson and Groll, 1998]. If the expansion valve is replaced with a work extracting expansion device, the expansion process's contribution to the total cycle irreversibility can be significantly reduced. In addition, if the work extracted from the work output expansion device is used to reduce the compression work, the cycle performance can be increased by up to 34% [Robinson and Groll, 1998]. Significantly, the work that is needed to run the 2nd-stage compressor of the Intercooler Cycle, which operates at maximum COP with different pressure ratios across the compressors, is approximately equal to the work generated by a work output expansion device that has a 50% isentropic efficiency. This fortuitous match could significantly simplify direct coupling of the expander to the compressor in an actual system.

Table 3 presents the improvement in COP if a 50%-efficient expansion device with output work (ED-WOW) is used to drive the 2nd-stage compressor of the Intercooler Cycle. It can be seen that depending on operating conditions, improvements of 11 to 29% compared to the Intercooler Cycles with equal pressure ratios across both compressors can be achieved. In particular, in case of Run No. 3, the COP of the Intercooler Cycle with different pressure ratios across the compressors and an ED-WOW that drives the 2nd-stage compressor is 33% greater than the COP of the basic single-stage transcritical cycle. The use of an ED-WOW in a transcritical CO_2 cycle not only results in the potential to use the expansion work a reduce the compression work, but also in a decrease of the CO_2 enthalpy at the inlet to the evaporator. This results in increase of the evaporator capacity. A comparison of the last column in Table 2 and the second column in Table 3 shows that the heat removal capacity of the evaporator increased through the use of an ED-WOW in all cases.

CONCLUSIONS

A thermodynamic computer model was developed to analyze the system performance of the transcritical carbon dioxide cycle with two-stage compression and intercooling (Intercooler Cycle). In particular, the effect of variation in pressure ratios across the compressors on COP was investigated. A total of 5 cases as a function of low-side pressure are presented. The following conclusions can be drawn:

- The maximum COP of the Intercooler Cycle occurs at a pressure ratio across the 1st-stage compressor that is significantly larger than the pressure ratio across the 2nd-stage compressors due to the characteristics of the transcritical cycle.
- The COP of the Intercooler Cycle with different pressure ratios across the compressors is up to 9% larger than the COP of the Intercooler Cycle with equal pressure ratios across both compressors.
- The work required to drive the 2nd-stage compressor of the Intercooler Cycle operating at maximum COP (different pressure ratios across both compressors) is approximately equal to the work extracted using a work extraction device instead of the conventional expansion valve. This fortuitous match could significantly simplify direct coupling of the expander to the compressor in an actual system.
- The COP of the Intercooler Cycle with different pressure ratios across the compressors and an ED-WOW that drives the 2nd-stage compressor is up to 29% larger than the COP of the Intercooler Cycle with equal pressure ratios across both compressors and without the ED-WOW, and up to 44% greater than the COP of the basic single-stage transcritical cycle.

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Table 1: Operating conditions for COP calculations of the Intercooler Cycle.

Run No.	p_{low} (MPa)	p_{high} (MPa)	T_{in} (°C)	$T_{gc,out} = T_{int,out}$ (°C)
1	2.5	10	20	35
2	3.0	10	20	35
3	3.5	10	20	35
4	4.0	10	20	35
5	4.5	10	20	35

Table 2: COP in case of equal and different pressure ratios across compressors for given operating conditions.

Run No.	COP in case of equal pressure ratios	Pressure ratio (equal) for each compressor	COP in case of different pressure ratios	Increase of COP (%)	Pressure ratio of 1st-stage compressor	Pressure ratio of 2nd-stage compressor	Heat removal capacity of evaporator (kJ/kg)
1	2.765	2.00	2.777(1)	0.43	3.24	1.23	185.612
2	3.241	1.826	3.319	2.41	2.70	1.23	178.935
3	3.774	1.690	3.941	4.43	2.32	1.23	171.623
4	4.381	1.581	4.665	6.48	2.02	1.24	163.465
5	5.081	1.491	5.525	8.74	1.80	1.23	154.108

Note: (1)The maximum COP of 2.811 occurred at pressure ratios of 2.5 for the 1st-stage compressor and 1.6 for the 2nd-stage compressor. However, the second maximum COP of 2.777 which occurred just before the COP decreased suddenly as shown in Figure 5 was considered here in order to run the 2nd-stage compressor by an ED-WOW as discussed in Table 3.

Table 3: COP improvement with an ED-WOW(2) for given operating conditions.

Run No.	Heat removal capacity of evaporator (kJ/kg)	Work of 1st-stage compressor (kJ/kg)	Work of 2nd-stage compressor (kJ/kg)	Work produced through the ED-WOW (kJ/kg)	COP with different pressure ratios and ED-WOW	Increase of COP(3) (%)
1	194.556	63.153	3.698	8.944	3.081	11.43
2	186.478	50.211	3.698	7.542	3.714	14.59
3	178.033	39.969	3.576	6.410	4.454	18.02
4	168.940	31.202	3.835	5.475	5.414	23.58
5	158.799	24.197	3.698	4.691	6.563	29.17

Note: (2)The isentropic efficiency of the ED-WOW was assumed to be 50%.
(3)The increase of the COP was calculated compared to the COP in case of equal pressure ratios.

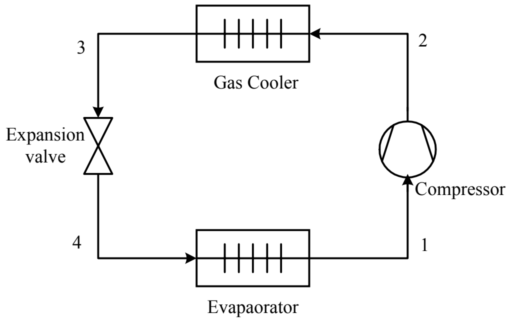

Figure 1: Schematic of basic transcritical carbon dioxide cycle.

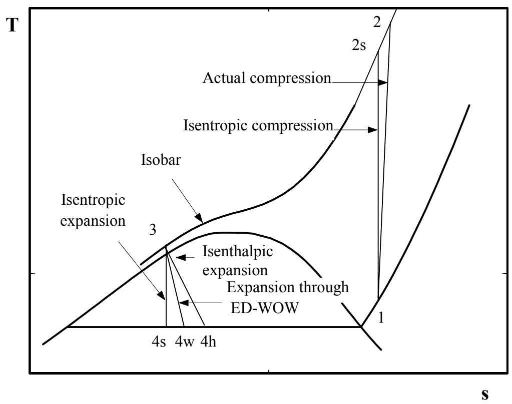

Figure 2: Temperature-entropy diagram of a transcritical carbon dioxide cycle.

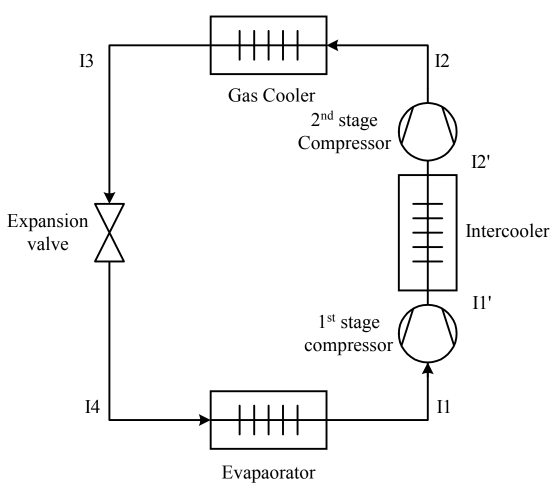

Figure 3: Schematic of a transcritical CO_2 cycle with two-stage compression and intercooling (Intercooler Cycle).

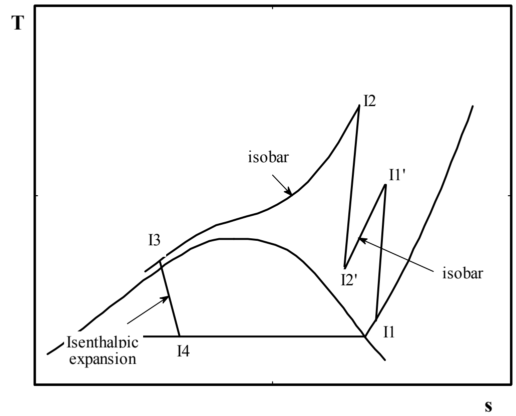

Figure 4: Temperature-entropy diagram of a transcritical CO_2 cycle with two-stage compression and intercooling (Intercooler Cycle).

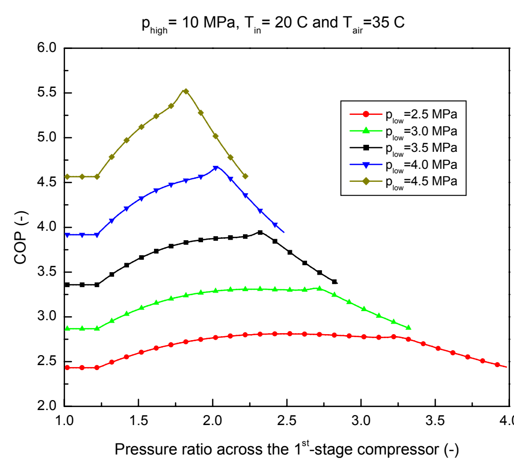

Figure 5: COP of Intercooler Cycle versus pressure ratio across 1st-stage compressor for varying low-side pressures.

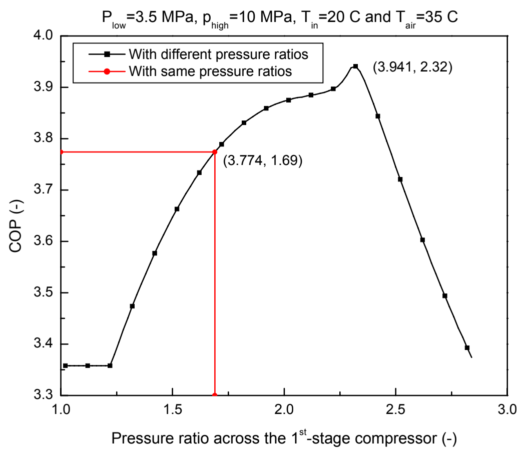

Figure 6: COP of Intercooler cycle versus pressure ratio across 1st-stage compressor for Run No. 3.

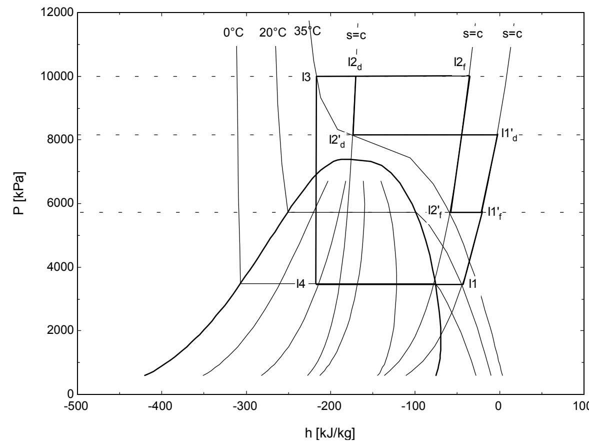

Figure 8: Processes of Intercooler cycle on pressure-enthalpy diagram

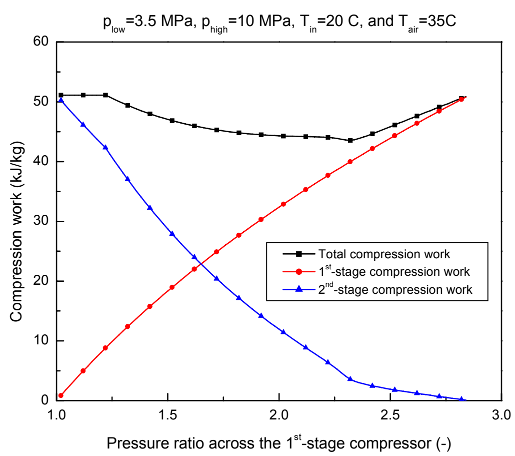

Figure 7: Total, 1st-stage, and 2nd-stage compression work versus pressure ratio across 1st-stage compressor for Run No.3.