

Experimental Study on Temperature Distribution of Two-Stage Cascade Vapor-Compression Cycle

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Abstract: This paper presents the experimental study on temperature distribution of two-stage cascade vapor-compression system for achieving super-low temperature. On this study, two different types of Halocarbon refrigerants are utilized, HCFC-22 and HFC-23. Four successive trials were conducted for different outlet pressure of low stage compressor. By changing mass of refrigerant, low-stage compressor outlet is adjusted from 6 bar to 21 bar. The lowest temperature achieved by this arrangement is down to -85°C at 16 bar of low stage compressor outlet. It is also found that at the range of 6 to 16 bars, evaporation temperature decrease as compressor outlet-pressure increase.

Keywords: Cascade vapor-compression, high stage, low-stage

NOMENCLATURE

Symbol	Description [unit]
COP	Coefficient of Performance
h	Enthalpy
HS/LS	High Stage/low stage Cycle
T	Temperature [°C]
P	Pressure [bar]
q	Heat transfer rate [kJ/kg]
<i>Subscript</i>	
amb	Ambient
coro	Cold room
Cond	Condenser
disc	Discharge line
Evap	Evaporator
SC	Sub-Cooling
SH	Super-heating
suct	Suction line
1	HS Compressor Inlet
2	HS Compressor Outlet
3	HS Condenser inlet
4	HS Condenser Outlet
5	HS Expansion valve inlet
6	HS Expansion valve inlet
7	HS Evaporator Inlet
8	HS amid Evaporator (upstream)
9	HS amid Evaporator Inlet (downstream)
10	HS Evaporator Outlet
11	LS Compressor Inlet
12	LS Compressor Outlet
13	LS Condenser inlet
14	LS Condenser Outlet
15	LS Expansion valve inlet
16	LS Expansion valve inlet
17	LS Evaporator Inlet
18	LS amid Evaporator (upstream)
19	LS amid Evaporator Inlet (downstream)
20	LS Evaporator Outlet

1. INTRODUCTION

For particular application in industrial process for instance: chemical processing, natural gas liquefaction, automobiles manufacturing etc, where super-low temperature condition is required, single stage refrigeration cycle become impractical^[1], since compression ratio of the compressor will be very high which imply higher work and low refrigeration effect. Therefore, more sophisticate system should be utilized for super-low temperature application.

Cascade refrigeration system is sort of multi-stage refrigeration system that connected each other in the cascade heat exchanger^[2]. This two or more cycle is working on the different evaporation temperature, condensing temperature and different refrigerant.

Both evaporator and condenser of low stage cycle operate at low temperature environment. Base on the second law of thermodynamics, heat cannot be rejected from low to the higher temperature body

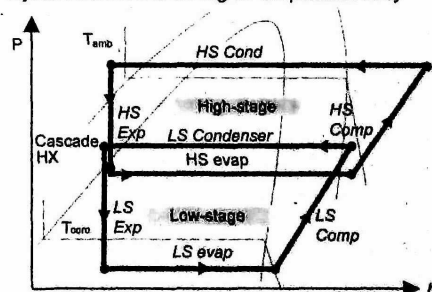


Figure 1: Illustration of Ideal P-h diagram of two-stage cascade refrigeration cycle

Therefore, for rejecting heat from the low stage, high stage evaporator serves as low temperature body for low stage condenser. Subsequently, this heat will be rejected by high temperature condenser (relative to T_{amb}) of high-stage cycle. These arrangements enable the heat rejecting process from cold room of low stage evaporator to environment.

Figure 1 illustrate basic P-h diagram of two-stage cascade refrigeration cycle. Process taking place in these cycles are:

- High-stage cycle: HS compression → HS condensation → HS expansion → HS evaporation
- Low-stage cycle: LS compression → LS condensation → LS expansion → LS evaporation

The main advantage of a cascade system is that different refrigerants, equipment, and oils can be used for the higher and the lower systems. This is especially helpful when the evaporating temperature required in the lower system is less than 73°C. The specific volume of the suction vapor is extremely important in such temperature applications. A greater always requires a large compressor and more space. One disadvantage of a cascade system is the overlap of the condensing temperature of the lower system and the evaporating temperature of the higher system for heat transfer in the condenser. The overlap results in higher energy consumption. In addition, a cascade system is more complicated than a compound system.^[3]

Several researches on two-stage cascade vapor-compression system have been carried out. Kilcarslan^[1] investigate the COP of each the single system as well as the overall cascade system using R-134 as the refrigerant. The result state that the single cycle of the cascade system has lower COP compare then if it work in the two-stage cascade refrigeration system. Prior to this, Zubair and Khan^[3] had experimented the same refrigerant (R134) for loses investigation in the system. Nicola^[5] investigates COP of cascade refrigeration cycle using blend of carbon dioxide (R744) and hydrocarbon as working fluid to study the possibility of carbon dioxide application in the cascade system. The result shows that the blend is an attractive option for low temperature circuit of cascade system.

Most of the studies highlighting the performance of the system so called COP in the cascade refrigeration system. COP of refrigeration system is strongly influenced by the evaporator as well as compressor performance. Yet, another important parameter, that is transient characteristics of refrigeration system, is also importance to investigate the reliability of the system.

Therefore, this experiment is aimed to study the effect of low stage and high stage inlet pressure to temperature distribution characteristics of cascade refrigeration system, without constant cooling load at low stage evaporator. This experiment use the well know refrigerant, HCFC-22 in the high-pressure stage and HFC-23 in the low-pressure stage.

Experiment conducted at four consecutive pressures at the low stage compressor outlet. First experiments at 6 bars, second at 11 bars, third at 16 bars and the last experiment at 21 bars. This pressure was achieved by adjusting the mass of

HFC-23 refrigerant in the low stage refrigerator system. This mass was 0.0134 kg, 0.035 kg, 0.1399, 0.2189 kg for obtaining 6, 11, 16, and 21 bars of low stage cycle successively.

2. TEST SETUP

Figure 2 shows the test facility to conduct the experiment. It consists of two refrigeration circuits. Copper pipe with 9.5 mm of outer diameter is utilized for connecting each device.

Low stage and high stage refrigerant is pumped to the main cycle by hermetic compressors. High stage cycle refrigerant from condenser inlet is cooled down by air-fin forced convection type condenser at T_{Cond} (refer to table I) to sub-cool temperature (T_4).

Expansion valve decreases the pressure as well as the temperature of high stage refrigerant (HCFC-22) to the evaporating temperature (T_{Evap}) at cascade heat exchanger (T_7).

Heat rejected by the low stage causing the HCFC-22 in the higher circuit evaporates until reaching super-heat gas condition before entering high stage compressor. This heat exchanging process is occurring in the counter-flow direction of concentric spiral 14 m cascade heat exchanger with 16 mm and 6 mm outside diameter of external and internal pipe respectively. External pipe of cascade heat exchanger is covered with adiabatic material to prevent the heat rejection to the environment.

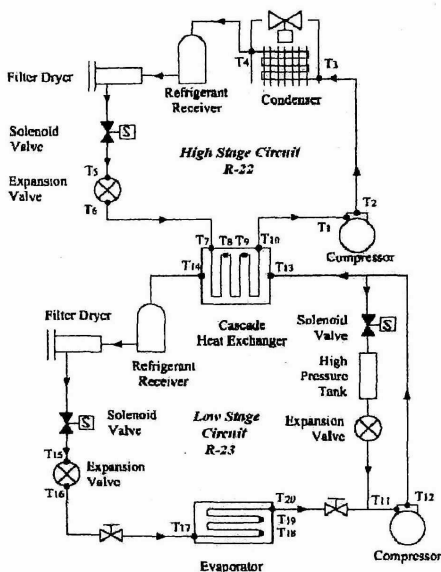


Figure 2: Temperature measurement location of cascade refrigeration system.

Table 1 Operating Condition

Parameters	
High stage $T_{\text{Cond}} / P_{\text{Cond}}$	20-30 °C/10-15 bar
High stage $T_{\text{Evap}} / P_{\text{Evap}}$	-30 °C/1.635 bar
High stage T_{SC}	0-10 °C
Low stage refrigerant mass	0.0216-0.079 kg
Low stage refrigerant mass	0.0134-0.2189 kg
Ambient temperature	20 °C

Equivalent processes occur in low stage circuit, with exception of heat exchanger type utilized. In case of low stage circuit, condensation occurs in the external stream of the cascade heat exchanger. Base on second law of thermodynamics, condensing R-23 at low-stage circuit obviously should have higher temperature than evaporating R-22 at high stage evaporator ($T_{\text{LS, Cond}} > T_{\text{HS, Evap}}$) to enable heat transfer process from hotter fluid of R23 low stage condenser to cold fluid of R-22. Leaving LS expansion valve, Super-low temperature refrigerant passes through LS evaporator inside a cold chamber. This cold chamber is a thermally isolated chamber with 300 mm inside diameter of 10 mm acrylic covered with 20 mm adiabatic materials to ensure the no heat absorbed from environment. Notice that this experiment is intentionally investigate cascade system without steady cooling load. Characteristics of system will be described as time-line graph function during 50 minutes of experiment. To ensure the HP evaporator has already reach desired evaporating temperature, the LP compressor is turned on after five minutes.

Table 1 presents operational condition of experiment. High stage cycle is maintained at specific range. Meanwhile low pressure is liberated.

Figure 2 shows measurement points in which 20 thermocouples are located to measure temperature of each locus. For this experiment, T-type thermocouples are employed since this thermocouple is relatively steady for low temperature application - 200°C to -300°C and more suitable for oxidation atmosphere environment. Uncertainty of this thermocouple is 0.5°C or special tolerance is 0.4%. These thermocouples are attached on the copper tube surface. Temperature distribution data was recorded using data logger every 10 second.

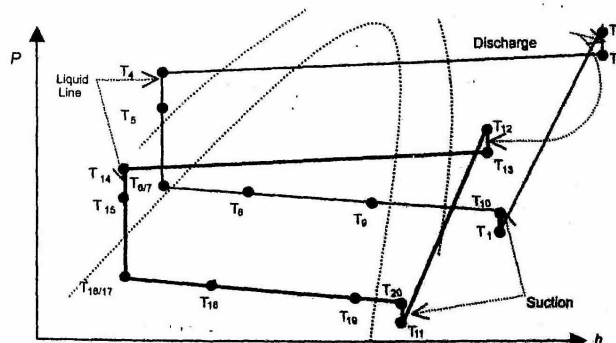

Figure3: Thermocouple position corresponding to theoretical P-h diagram of 2-stage vapor compression cycle

Table 2 Thermophysical Property of Refrigerants

Parameters	R22	R23
Molecular Mass	86.47	70.01
Normal Boiling Point [°C]	-40.8	-82
Critical temperature (T_c) [°C]	96.2	25.9
Critical Pressure (P_c) [Bar]	49.9	48.4
Threshold Limit value [ppm]	1000	1000
Heat of Combustion [MJ/Kg]	2.2	-12.5

For pressure sensing, Bourdon type pressure gage with 1% middle half of scale accuracy was employed at compressor's suction and discharge line of each cycle.

Since this experiment is dedicated to investigate the cascade vapor compression cycle temperature distribution instead of the refrigerant comparison study. Therefore, during refrigerant selection, well known refrigerant, R-22 and R-23 are selected base on its wide utilization on present-day, and availability of supporting equipment, regardless its effect to environment.

In the table 2, thermophysical property of R-22 and R-23 are presented. This data is taken from ANSI/ASHRAE standard 34^[6] and Refrigerant Data summary^[7].

Normal boiling point indicates that lowest evaporating temperature that can be obtained by each cycle at normal ambient pressure (1 Atm). Evaporating temperature can be more low then this if outlet pressure of expansion valve can reach vacuum pressure (<1 bar).

3. THEORETICAL PREDICTION

Consider a thermally isolated cascade refrigeration system, with evaporators working at normal boiling point of refrigerant (T_7 and T_{17}). This system is also working between ambient temperature (T_{amb}) and cold room (T_{cold}) temperature with arbitrary degree of superheat and subcool.

Pressure drop in the pipeline (suction, discharge line, and liquid line) as well as in heat exchanger causes energy losses in the cycle; this pressure drop also causes decreasing of temperature.

Consequently, in gas phase, adiabatic pressure drop means temperature drop, thus:

$$T_{2/12} > T_{3/13} \text{ And } T_{10/11} > T_{20/11}$$

In addition, assume a throttling process in expansion valve occurs in adiabatic condition. Therefore

$$h_{5/15} = h_{6/16} \text{ with } P_{5/15} > P_{6/16}$$

$$\text{Thus: } T_{5/15} > T_{6/16}$$

In the meantime, expansion valve outlet is made very close to the evaporator inlet thus no pressure drop occurs (therefore point 6=7 and 16=17).

Temperatures sequence of each cycle can be summarized as follow (refer to figure 3):

- HS: $T_2 \rangle T_3 \rangle T_5 \rangle T_4 \rangle T_1 \rangle T_{10} \rangle T_7 \geq T_6 \rangle T_8 \rangle T_9$
- LS: $T_{12} \rangle T_{13} \rangle T_{15} \rangle T_{14} \rangle T_{11} \rangle T_{20} \rangle T_{17} \geq T_{16} \rangle T_{18} \rangle T_{19}$

$T_{4/5}$ and $T_{9/10}$ as well as $T_{14/15}$ and $T_{19/20}$ can be altered each other depend on the degree of superheat and subcool. In addition, overall temperature sequence is:

$$T_2 \rangle T_3 \rangle T_{12} \rangle T_{13} \rangle T_5 \rangle T_4 \rangle T_1 \rangle T_{10} \rangle T_{15} \rangle T_{14} \rangle T_7 \geq T_6 \rangle T_8 \rangle T_9 \rangle T_{11} \rangle T_{20} \rangle T_{17} \geq T_{16} \rangle T_{18} \rangle T_{19}$$

Position $[T_1, T_{10}]$, $[T_{15}, T_{14}]$ as well as $[T_5, T_4]$, $[T_{12}, T_{13}]$ can also be altered depend on the degree of superheat and subcool of each cycle.

4. RESULT AND DISCUSSION

Figure 4 to 11 shows experiment progress during 50 minutes for 6, 11, 16, and 21 bars of low stage compressor outlet. At the $t=0$, high stage compressors has been activated for 10 minutes and LS compressor was started. During the experiment, outside temperature as well as initial temperature was 20 °C.

Starting low stage circuit, give refrigerant load to high stage circuit. Therefore, during first five minutes, temperature entering high stage compressors (T_1) increased, causing temperature increments in high compressor outlet (T_2).

At High stage compressor (figure 4, 5, 6), refrigerant enters compressor (T_1) with low superheat temperature. Little heat absorbs from low stage condenser, caused by low refrigerant mass and condensing temperature of low stage circuit. Meanwhile at fourth trial of 21 bar low stage compressor outlet, superheat temperature of high compressor inlet gas is extremely high, it is observed to be 29 °C at final time (refer to T_1 of figure 7).

For 6, 11 bar trial; temperatures distribution of high stage is relatively steady at specific range. In the contrary, at 16 and 21 bars, temperature distribution seem to be fluctuating, especially in the area between downstream of evaporator (T_9) to compressor outlet (T_2).

Additionally, compressor outlet temperature continues increasing until final moment. Temperature of HS condenser inlet (T_3) as well as compressor inlet (T_1) and outlet (T_2) are increasing as LS outlet pressure increases. This is because heat transfer rate from low stage increases due to the pressure increment [9], it can be verified by analyzing temperature increment at low stage compressor outlet.

Temperature of high stage liquid line seems to be constant and stable during pressure alteration. These also occur in low stage circuit.

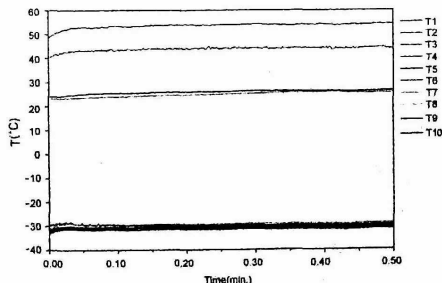


Figure 4: Temperature distribution of high stage at compressor outlet pressure of low stage: 6 Bars.

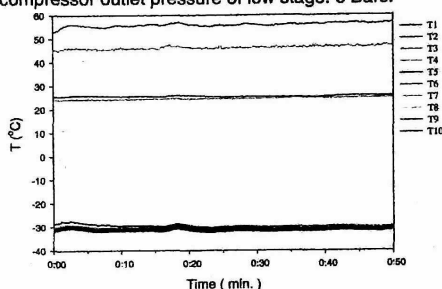


Figure 5: Temperature distribution of high stage at compressor outlet pressure of low stage: 11 bars

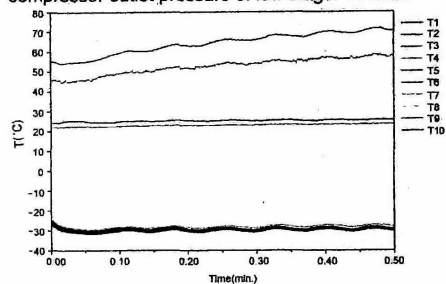


Figure 6: Temperature distribution of high stage at compressor outlet pressure of low stage: 16 Bars.

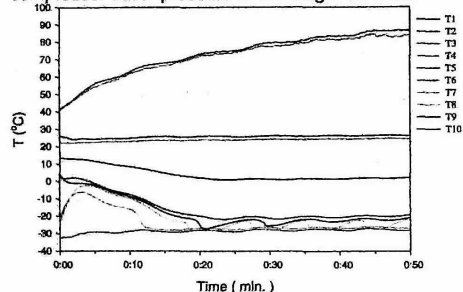


Figure 7: Temperature distribution of high stage at compressor outlet pressure of low stage: 21 bars

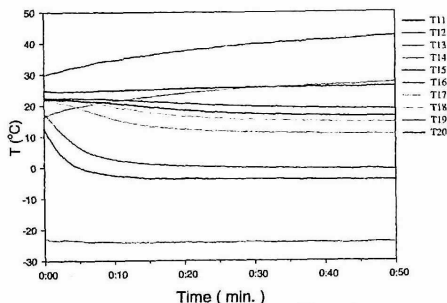


Figure 8: Temperature distribution of low stage at compressor outlet pressure of low stage: 6 Bars.

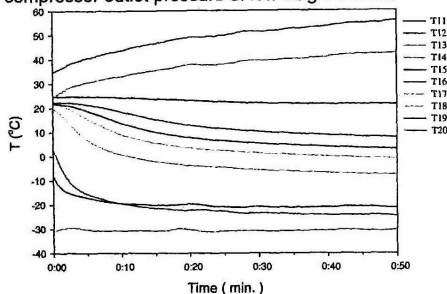


Figure 9: Temperature distribution of low stage at compressor outlet pressure of low stage: 11 Bars.

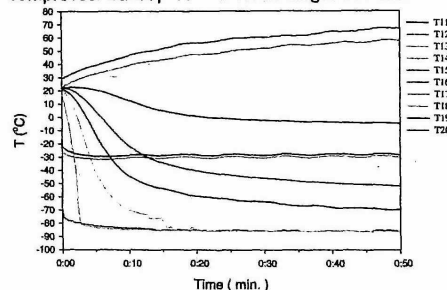


Figure 10: Temperature distribution of low stage at compressor outlet pressure of low stage: 16 bars

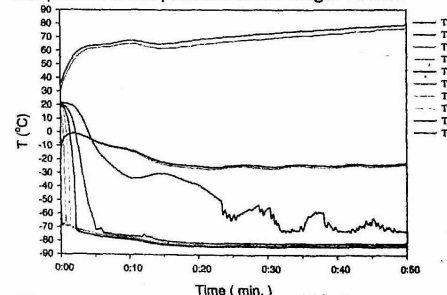


Figure 11: Temperature distribution of low stage at compressor outlet pressure of low stage: 21 bars

Temperature of evaporator outlet ($T_{10/20}$) and compressor inlet ($T_{1/11}$) do not follow adiabatic path, because copper wall was not fully thermally isolated wall, and wall temperature is very low. Some heat is extracted from ambient air by low temperature refrigerant $q_{suct} = h_{1/11} - h_{10/20}$

In the same time, in the discharge line, small amount of heat rejected through copper pipe to the environment, causing more decreasing temperature at condenser inlet ($T_{3/13}$). $q_{disc} = h_{2/12} - h_{3/13}$

At first and second trial, because of low compression pressure, Compressor inlet and outlet pressure of low circuit are very close. As a result, evaporation temperature lay near by condensing temperature. Therefore, evaporating temperature detected does not lie on super low temperature region. As pressure of compressor outlet increase, evaporating temperature increases. Yet, turning point of decrement occurs between 16 bars to 21 bars. It decreases from 0 °C at 6 bars, -20 °C at 11 bars, -86 °C at 16 bars and increase to -82 °C at 21 bars.

Expansion valve does not working optimally at low compressor outlet pressure. As can be seen in figure 8 and 9, temperature difference of expansion inlet and outlet ($T_{6/16}$) is under of 4 °C. It scales up as compressor inlet pressure reach 16 bars and 21 bars. Because expansion valve outlet ($T_{6/16}$) and evaporator inlet ($T_{7/17}$) location are very close, this circumstance effects to evaporator inlet. Therefore, change in the expansion valve outlet causes change in evaporation temperature and pressure.

Best results for achieving lowest evaporating temperature are between 16 bars to 21 bars when vacuum condition is occur at low stage evaporator outlet. Yet, at 21 bars, high fluctuation occurs in compressor inlet. Condensing temperatures of low stage circuit do not change too much by compressor outlet changing. It remains constant at -30~-20 °C during pressure change and experiment progress.

Degree of superheat at low stage compressor inlet decreases during the time span, because refrigerating load is decreasing. As mentioned previously, cooling chamber is thermally isolated ($\Delta h=0$), meanwhile heat is absorbed continuously from cooling chamber ($\Delta q_{cool} = q_{evap} = h_{10/20} - h_{7/17}$). Therefore, temperature of as well as cooling chamber enthalpy decreases.

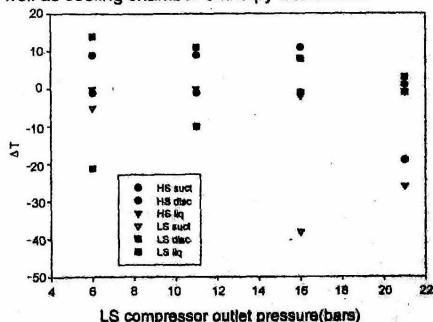


Figure 12: Temperature difference of suction, discharge, and liquid line of Low and high stage circuit.

Average temperature difference at suction, discharge, and liquid line is given by Figure 12. Temperature difference at suction line (both in low and high cycle) decreases when low stage compressor outlet increases. In the contrary, in discharge and suction line, it decreases

Figure 13 and 14 summarize the temperature distribution of two-stage cascade vapor-compression cycle at the end of the experiment

At high stage circuit, temperature distribution follows theoretical sequence except for T_8 and T_9 , where these temperatures are in the reverse order

Meanwhile, at lower stage circuit, this order seems to be disregard, particularly at low compressor-outlet pressure area.

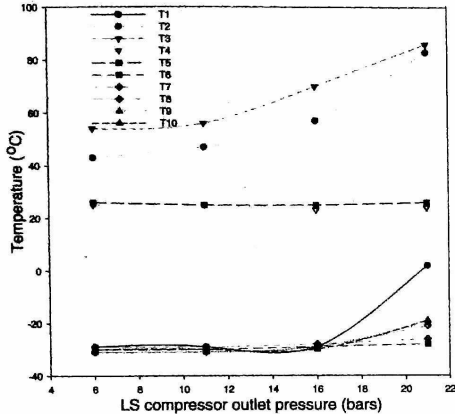


Figure 13: Temperature distribution of high stage sensing point at 6, 11, 16, and 21 bars of LS Compressor outlet (data of 50 min).

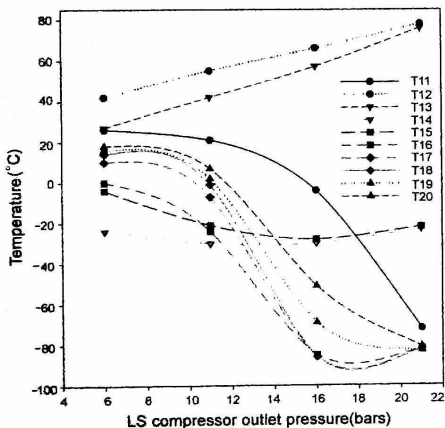


Figure 14: Temperature distribution of low stage sensing point at 6, 11, 16, and 21 bars of LS Compressor outlet (data of 50 min)

5. CONCLUSION

High stage temperature distribution, excluding compressor area (T_1 , T_2 , T_3), tends to be unchanged by altering low stage compressor. While, temperature distribution of low stage cycle changes continuously, yet, condenser outlet (T_{14}) remains constant. Most significant change at low stage side is expansion valve outlet and evaporating temperature. The lowest evaporating temperature of two-stage cascade vapor-compression cycle lay between 16 bars to 21 bars.

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