Refrigeration

4:multi-pressure Refrigeration Systems

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INTRODUCTION

A **multi-pressure** system is a refrigeration system that has two or more low-side pressures. The low-side pressure is the pressure of the refrigerant between the expansion valve and the intake of the compressor.

A multi-pressure system is **distinguished** from the single-pressure system, which has one low-side pressure.

A multi-pressure system may be found, for example, in a dairy where one evaporator operates at -35°C to harden ice cream while another evaporator operates at 2°C to cool milk. Another typical application might be in a process industry where a two- or three-stage compression arrangement serves an evaporator operating at a low temperature of -20°C or lower.

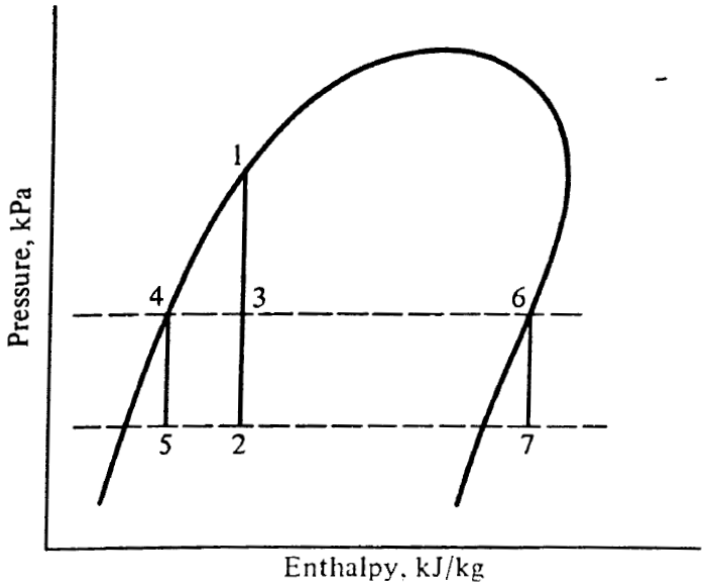
Two functions often integral to multi-pressure systems are the **removal of flash gas** and **intercooling**. They will be discussed first. Then several combinations of multiple evaporators and compressors will be analyzed.

Removal of flash gas

A saving in the power requirement of a refrigeration system results if the flash gas that develops in the throttling process between the condenser and evaporator is removed and recompressed before complete expansion. When saturated liquid expands through an expansion valve, the fraction of vapor or flash gas progressively increases. The expansion process shown on the pressure-enthalpy diagram in Fig. 16-1 takes place from 1 to 2. The state point, as the expansion proceeds, moves into a region of a greater fraction of vapor.

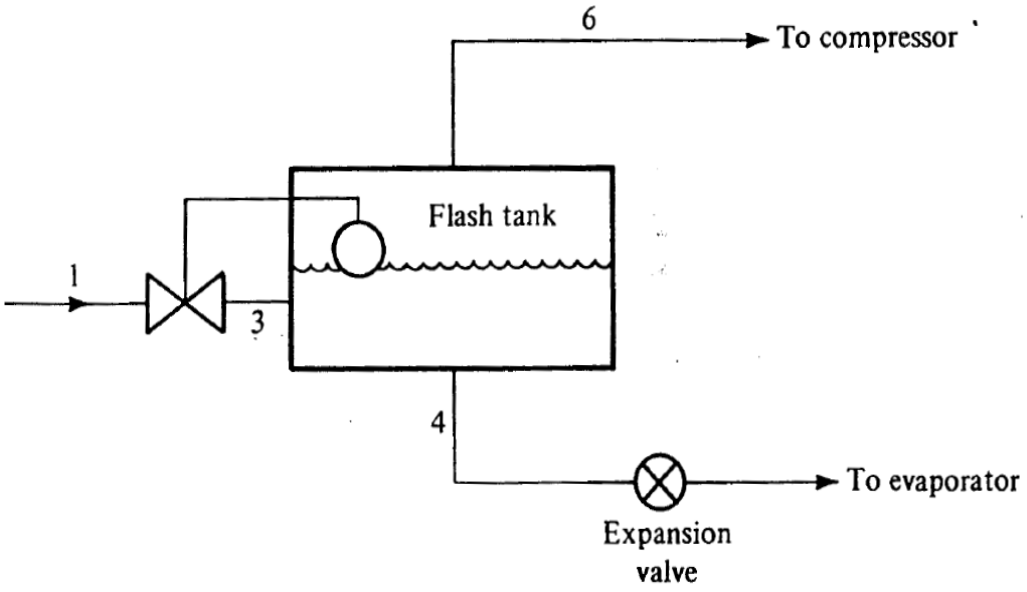
The end point of the expansion, 2, could have been achieved by interrupting the expansion at 3 and separating the liquid and vapor phases, which are 4 and 6, respectively.

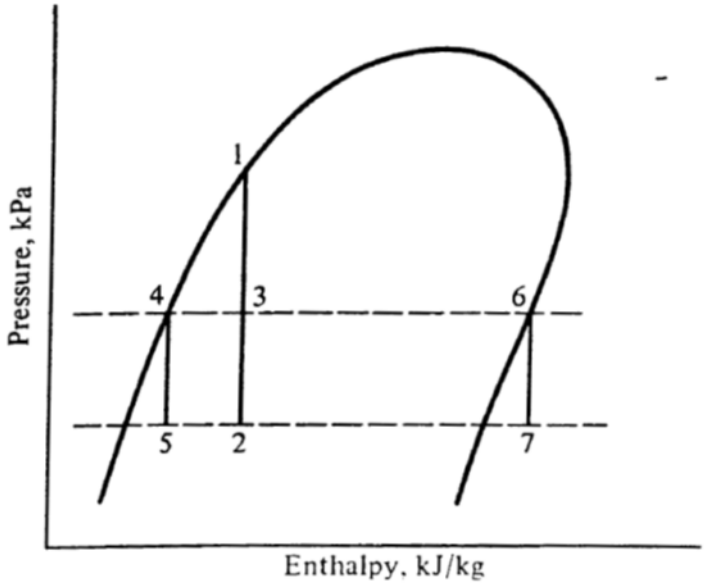
The expansion could then continue by expanding the liquid at 4 and the vapor at 6 to the final pressure, giving 5 and 7, respectively. The combination of refrigerant at states 5 and 7 gives point 2.



Removal of flash gas

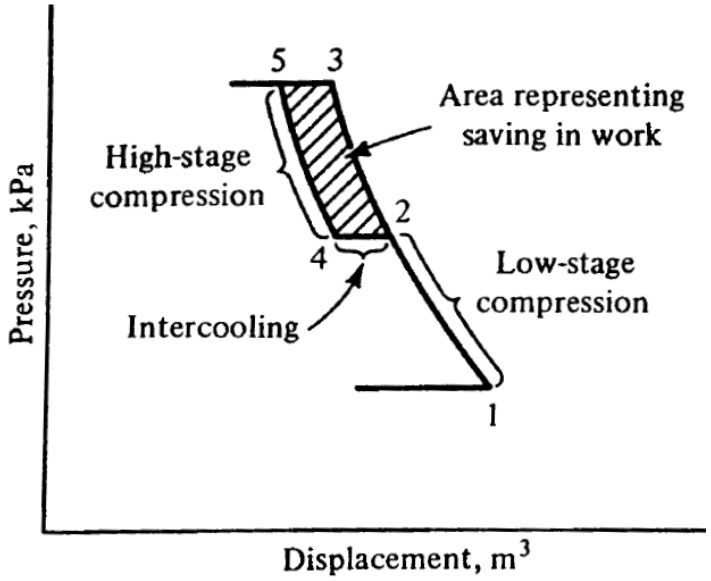
Inspection of the expansion from 6 to 7 confirms that it is wasteful. In the first place, the refrigerant at 7 can do **no refrigerating**; in the second place, **work will be required** to compress the vapor back to the pressure it had at 6. Why not perform part of the expansion, separate the liquid from the vapor, continue expanding the liquid, and recompress the vapor without further expansion? The equipment to achieve this separation is called a **flash tank** (see Fig. 16-2). The expansion from 1 to 3 takes place through a float valve, which serves the further purpose of maintaining a constant level in the flash tank. To recompress the vapor at 6, a compressor must be available with a suction pressure of 6. Thus, **two compressors** are needed in the system.





Intercooling

Intercooling: Intercooling between two stages of compression reduces the work of compression per kilogram of vapor. In two-stage compression of air, for example, an intercooling from point 2 to 4 on the pressure-displacement diagram of Fig. 16-3 saves some work. If the processes are reversible, the saving is represented by the shaded area in Fig.16-3 .



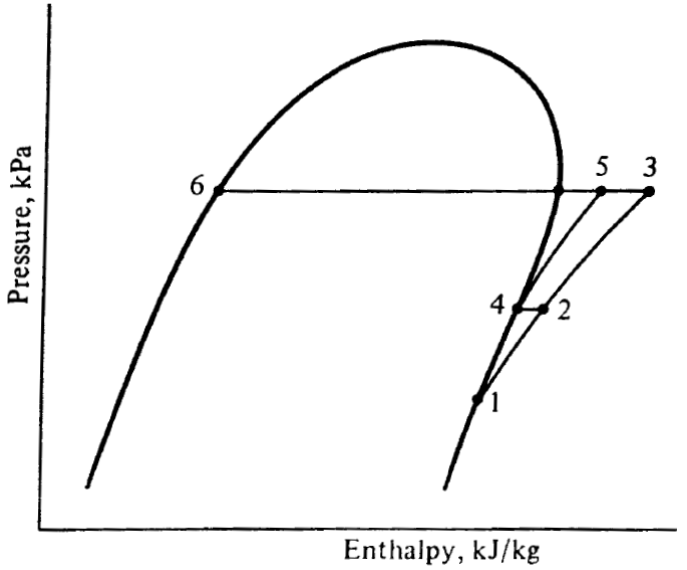
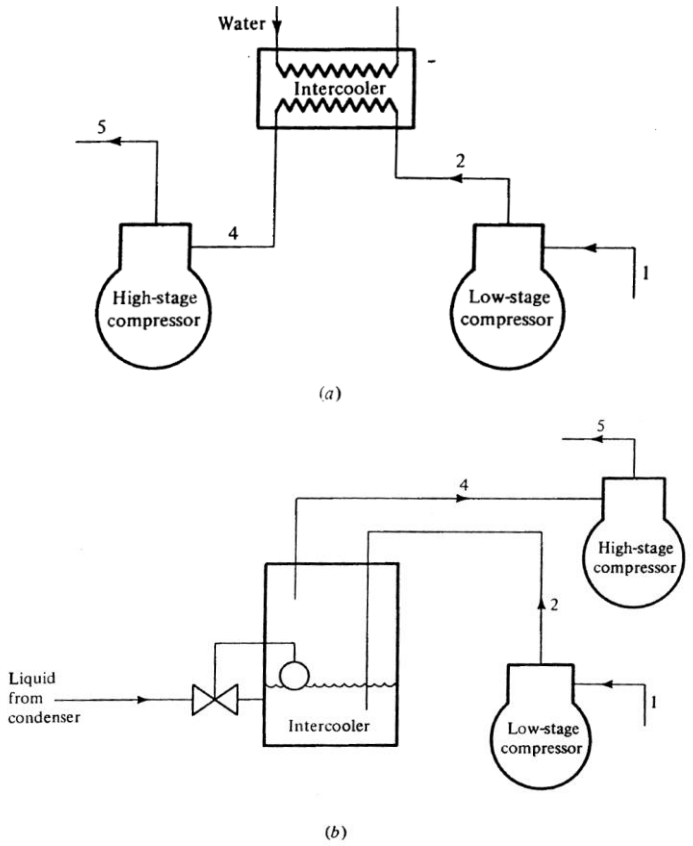
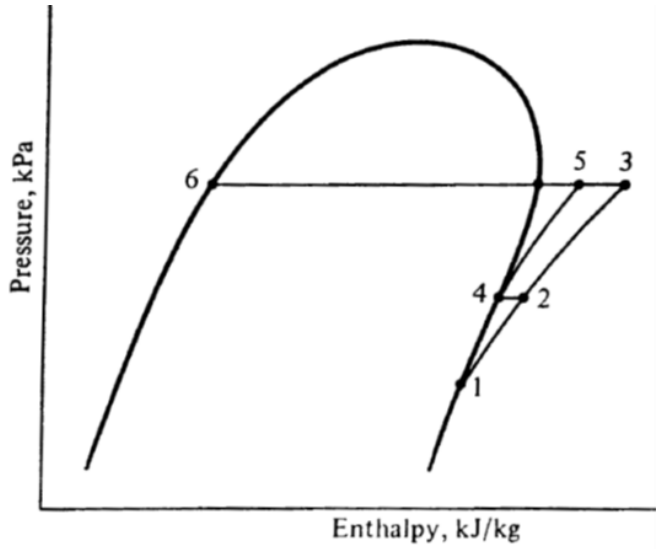


Figure 16-4 shows how compression with intercooling appears on the pressure enthalpy diagram of a refrigerant. Processes 1-2-3 and 4-5 are on lines of constant entropy, but process 2-3 falls on a flatter curve than process 4-5. Between the same two pressures, therefore, process 4-5 shows a smaller increase in enthalpy, which indicates that less work is required than in 2-3.

Intercooling

Intercooling in a refrigeration system can be accomplished with a water-cooled heat exchanger or by using refrigerant. (Fig. 16-5 a and b). The water-cooled intercooler may be satisfactory for two-stage air compression, but for refrigerant compression the water is usually not cold enough. The alternate method of Fig. 16-5 b uses liquid refrigerant from the condenser to do the intercooling. Discharge gas from the low-stage compressor bubbles through the liquid in the intercooler. Refrigerant leaves the intercooler at 4 as saturated vapor.





Intercooling

Example 16-1: Calculate the power needed to compress 1.2 kg/s of ammonia from saturated vapor at 80 kPa to 1000 kPa **(a)** by single-stage compression and **(b)** by two-stage compression with intercooling by liquid refrigerant at 300 kPa.

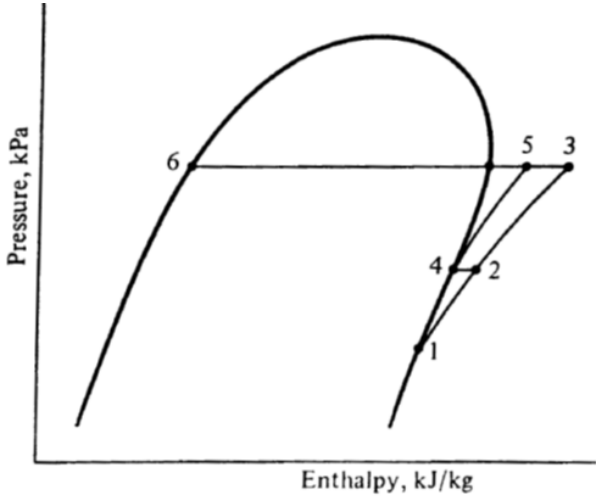
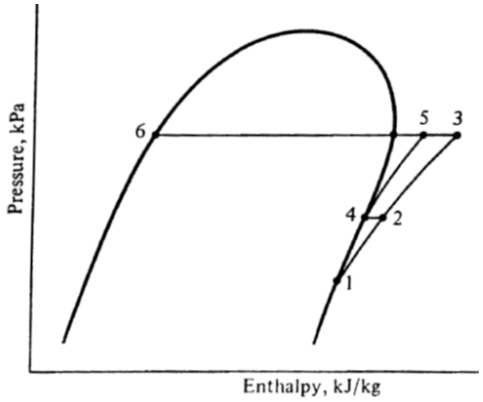


Table 16-1 Comparison of ammonia compression with and without intercooling

	Without intercooling, processes 1–2 and 2–3	With intercooling, processes 1–2, 2–4, and 4–5
$h_2 - h_1$, kJ/kg	1588 – 1410	1588 – 1410
$h_3 - h_2$, kJ/kg	1800 – 1588	
$h_5 - h_4$, kJ/kg	.....	1628 – 1450
Flow rate, kg/s, 1 to 2	1.2	1.2
2 to 3	1.2	
4 to 5	.....	1.346
Power required, kW, 1 to 2	213.6	213.6
2 to 3	254.4	
4 to 5	.....	239.6
Total power, kW	468.0	453.2

Intercooling

The high-stage compressor in the intercooled system must compress 1.2 kg/s plus the flow rate of refrigerant that evaporates to desuperheat the gas at 2. The flow rate of ammonia compressed in the high stage can be calculated by making a heat and a mass balance about the intercooler, as shown in Fig. 16-6. Heat balance:



$$w_6(316\ \text{kJ/kg}) + (1.2\ \text{kg/s})\,(1588\ \text{kJ/kg}) = w_4(1450\ \text{kJ/kg})$$

Mass balance:

$$w_6 + 1.2 = w_4$$

Solving gives

$$w_4 = 1.346\ \text{kg/s}$$

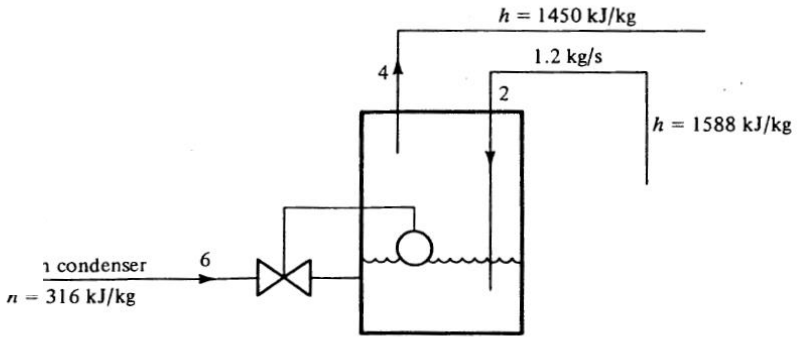


Figure 16-6 Heat and mass balance around intercooler in Example 16-1.

Intercooling the ammonia with liquid refrigerant reduced the power requirement from 468 to 453.2 kW.

A further benefit of intercooling the ammonia is that the discharge temperature from the high-stage compressor will be reduced from a value of 146°C, the temperature at 3 based on isentropic compression, to 77°C. The lower discharge temperature permits better **lubrication** and results in **longer life** of the compressor.

Intercooling

Example 16-2: Compare a compression of 3.5 kg/s of refrigerant 22 from saturated vapor at 100 kPa to a condensing pressure of 1000 kPa **(a)** by single-stage compression and **(b)** by two-stage compression with intercooling at 300 kPa, using liquid refrigerant.

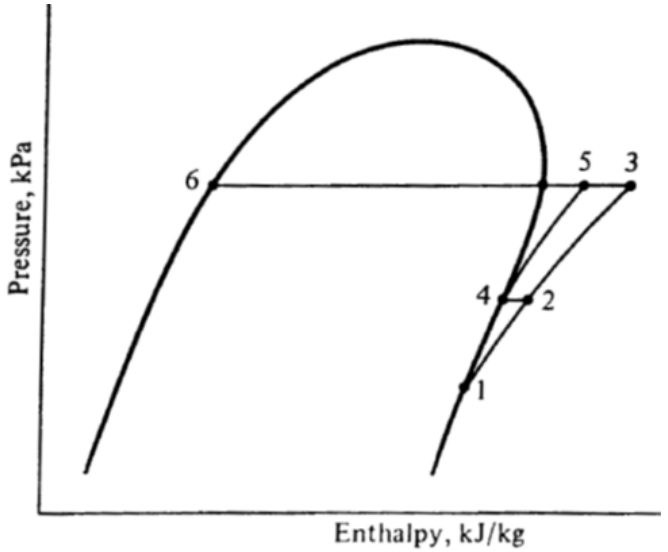


Table 16-2 Comparison of refrigerant 22 compression with and without intercooling

	Without intercooling, processes 1–2 and 2–3	With intercooling, processes 1–2, 2–4, and 4–5
$h_2 - h_1$, kJ/kg	416 – 387	416 – 387
$h_3 - h_2$, kJ/kg	449 – 416	
$h_5 - h_4$, kJ/kg	.....	430 – 399
Flow rate, kg/s, 1 to 2	3.5	3.5
2 to 3	3.5	
4 to 5	.....	3.74
Power required, kW, 1 to 2	101.5	101.5
2 to 3	115.5	
4 to 5	.....	115.9
Total power, kW	217.0	217.4

Intercooling

For refrigerant 22 **intercooling** with liquid refrigerant midway during the compression is **ineffective**. The dissimilarity during intercooling of ammonia and refrigerant 22 systems results from the difference of their properties. The lines of constant entropy for ammonia become **flatter** in the superheat region than those of refrigerant 22. In a refrigerant 22 system, therefore, the saving in work per kilogram by performing the compression close to the saturated-vapor line does not compensate for the increased flow rate which must be pumped by the high-stage compressor.

There is an **optimum pressure** at which the intercooling should take place in an ammonia system. In the compression of air, where the intercooling is achieved by rejecting heat to the ambient or to cooling water, that intermediate pressure for minimum total power is

$$P_i = \sqrt{p_s\, p_d}$$

where P_i = intercooler pressure, kPa

P_s = suction pressure of low-stage compressor, kPa

p_d= discharge pressure of high-stage compressor, kPa

The development of the equation does not consider the additional refrigerant compressed by the high-stage compressor, but it does provide an approximate guideline for the optimal intermediate pressure.

One evaporator and one compressor

The flash tank and intercooler appear in most multi pressure systems and will now be examined in various compressor evaporator combinations.

With one compressor and one evaporator the flash tank may function as shown schematically in Fig. 16-7. A pressure-reducing valve throttles the flash gas from the intermediate pressure to the evaporator pressure.

The throttling is necessary because there is no compressor available with a high suction pressure. Calculations would show that the **flash tank does not improve the performance of the system**. The only reason for using the flash tank would be to **keep** the **flash gas** in **the machine** room rather than sending it to the evaporator. The flash gas in the evaporator tubes and long suction line **does** no **refrigeration** but does **increase** the **pressure drop**. This system is used **infrequently**.

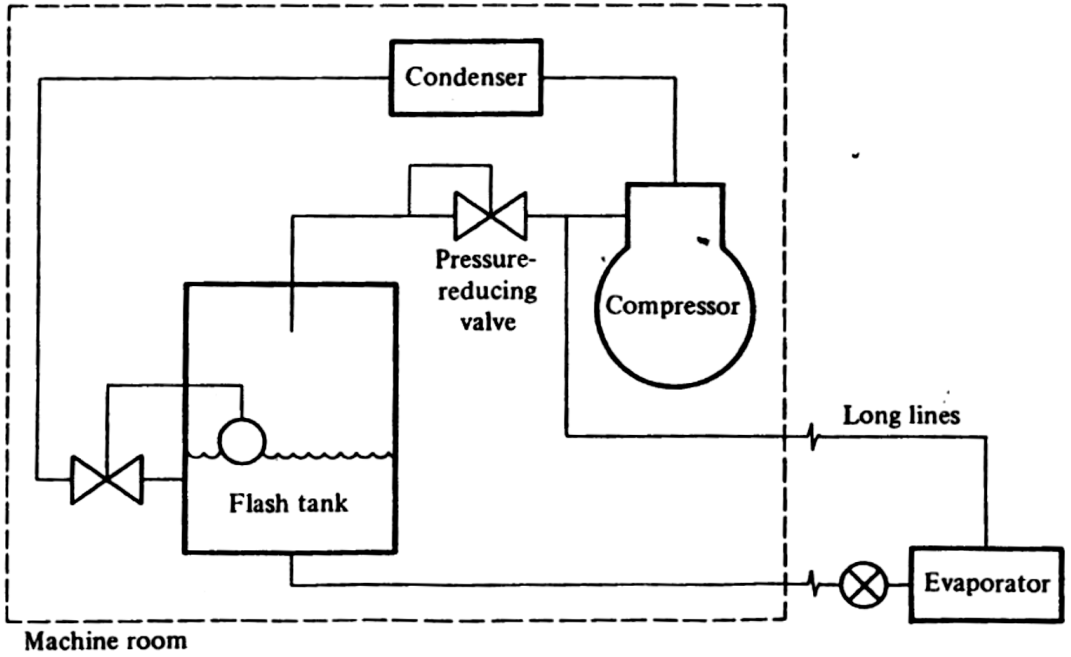
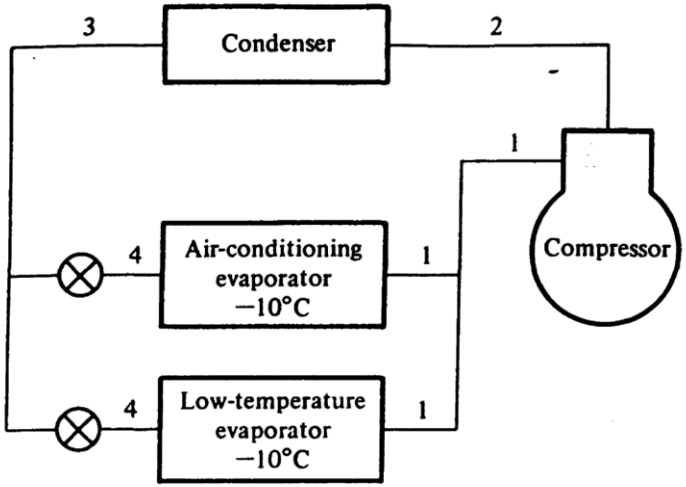


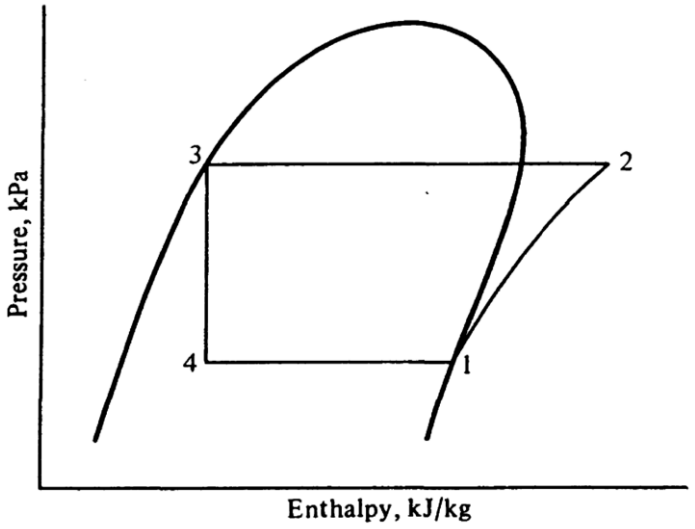
Figure 16-7 System with one compressor and one evaporator using a flash tank.

Two evaporators and one compressor

In many situations one compressor serves two evaporators having different temperature requirements. An example is an industry which needs low-temperature refrigeration for a process and which must also provide air conditioning for some offices. Figure 16-8a shows one method of arranging this system, and Fig. 16-8b shows the corresponding pressure-enthalpy diagram. In Fig. 16-8a the air-conditioning evaporator operates at -10°C even though a higher temperature in this evaporator would cool the air sufficiently. Furthermore, difficulties may arise when an evaporator operates at an **unnecessarily low temperature**: an evaporator which cools air for air conditioning **may collect frost**, which blocks the flow of air; an evaporator which chills a liquid may freeze the liquid; and an evaporator which cools air for a room where meat or produce is stored may dehumidify the air so much that the products will be dehydrated.



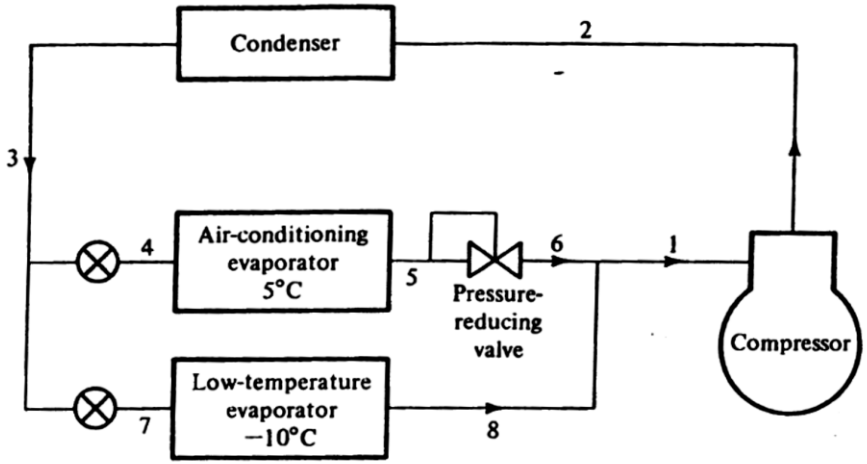
(a)



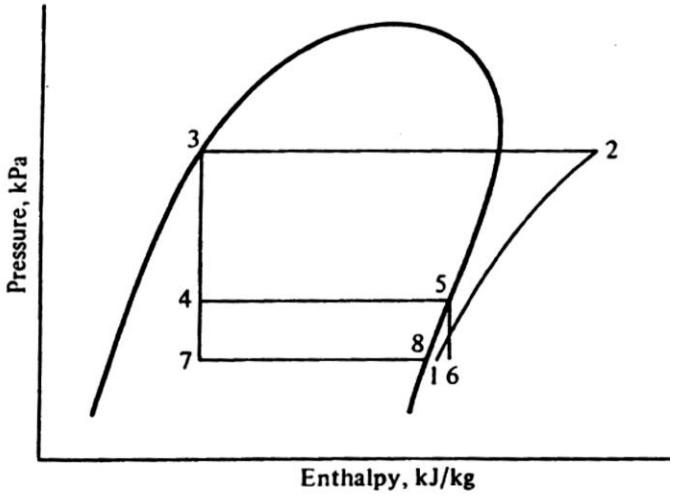
(b)

Two evaporators and one compressor

To overcome the drawbacks of the system in Fig. 16-8a, a revision may be made as shown in Fig. 16-9a. A pressure-reducing valve installed after the high-temperature evaporator regulates the pressure and maintains a temperature in the air-conditioning evaporator of 5°C, for example. Figure 16-9b shows the corresponding pressure enthalpy diagram. Differences in performance between the systems in Figs. 16-8a and 16-9a are as follows. In the system of Fig. 16-9a, the **refrigerating** effect in the **high-temperature evaporator** is **greater** than it is in the system of Fig. 16-8a. This is an advantage for the system of Fig. 16-9a. To counter balance this advantage, the compression in Fig. 16-9b occurs farther out in the superheat region than in Fig. 16-8b. The system of Fig. 16-9a therefore demands more work per kilogram of refrigerant. From a power standpoint, the systems are practically a standoff, but for proper operation of the high-temperature evaporator the system of Fig. 16-9a is preferred.



(a)



(b)

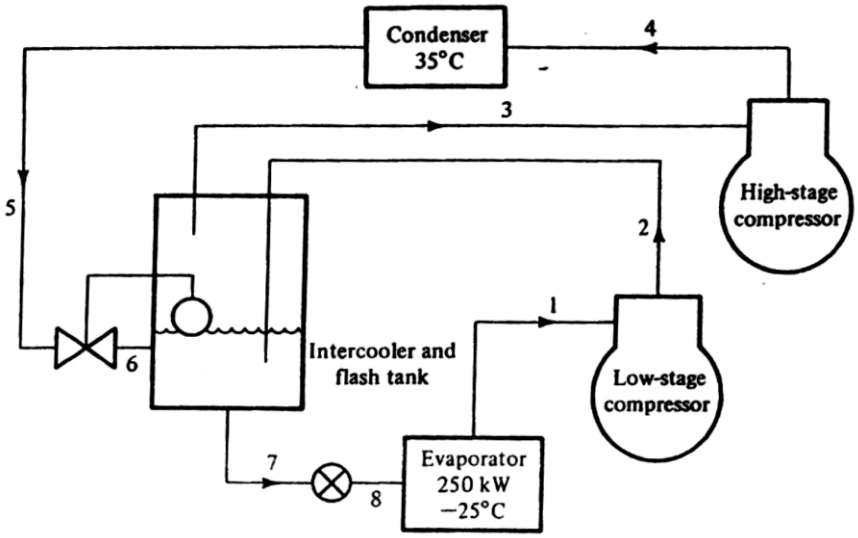
Two compressors and one evaporator

Two compressors and one evaporator

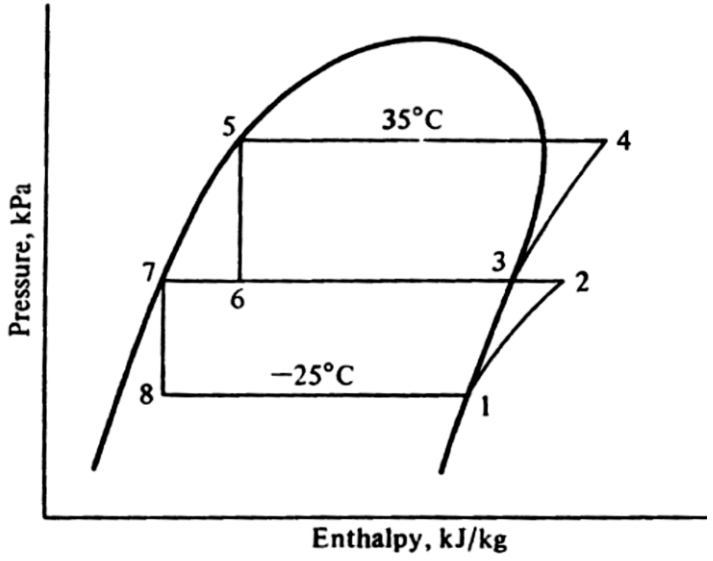
Two-stage compression with intercooling and removal of flash gas is often the ideal way to serve one low-temperature evaporator. This system requires less power than with a single compressor, and often the saving in power will justify the cost of the extra equipment.

Example 16-3 Calculate the power required by the two compressors in an ammonia system which serves a 250-kW evaporator at -25°C. The system uses two stage compression with intercooling and removal of flash gas. The condensing temperature is 35°C.

Solution: First sketch the schematic diagram of the system (Fig. 16-10a) and the corresponding pressure-enthalpy diagram (Fig. 16-10b*).* The functions of the inter cooler and flash tank are combined in one vessel.



(a)



(b)

Two compressors and one evaporator

The intermediate pressure for optimum economy can be calculated from Eq. (16-1):

p_s = saturation pressure at –25°C = 152 kPa
p_d = saturation pressure at 35°C = 1352 kPa
$p_i = \sqrt{152(1352)} = 453$ kPa

The enthalpies at all points can now be determined from Table A-3 and Fig. A-1:

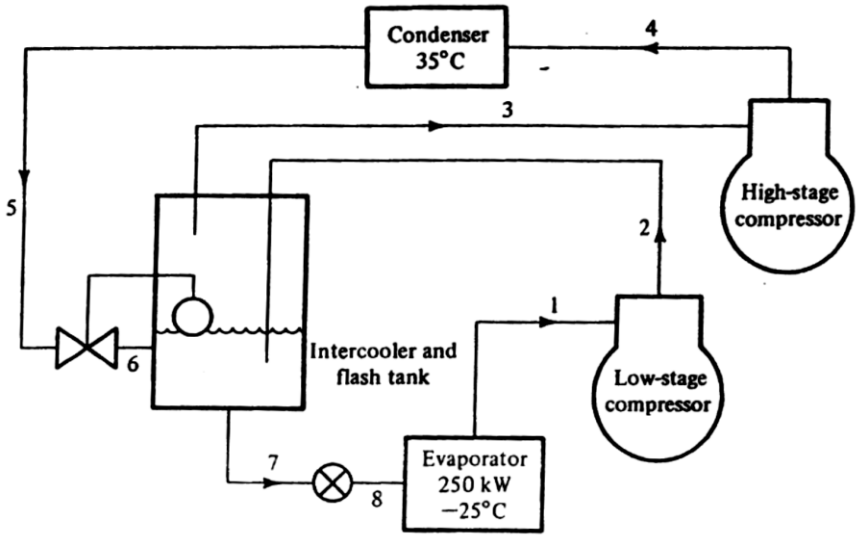
$h_1 = h_g$ at –25°C = 1430 kJ/kg
$h_2 = h$ at 453 kPa after isentropic compression = 1573
$h_3 = h_g$ at 453 kPa = 1463
$h_4 = h$ at 1352 kPa after isentropic compression = 1620
$h_5 = h_f$ at 35°C = 366 $\quad h_6 = h_5 = 366$
$h_7 = h_f$ at 453 kPa = 202 $\quad h_8 = h_7 = 202$

Next, the mass rates of flow through the compressors can be calculated by means of heat and mass balances.

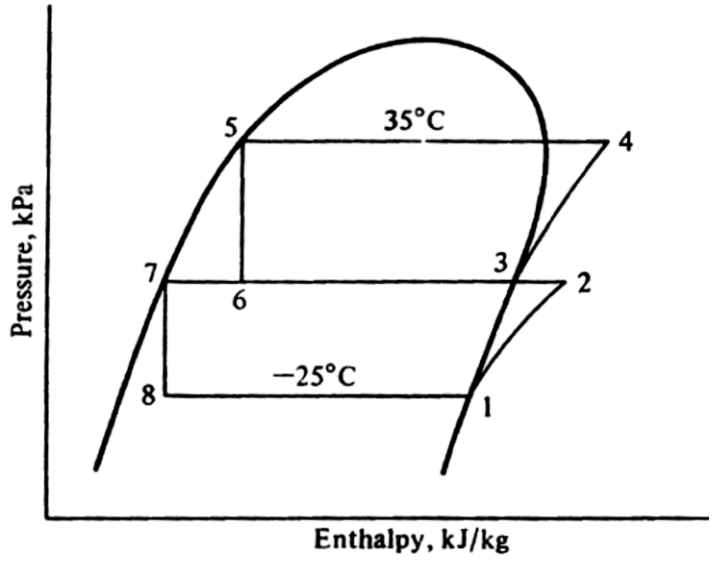
Heat balance about the evaporator:

$$w_1 = \frac{250 \text{ kW}}{1430 - 202} = 0.204 \text{ kg/s}$$

$$w_1 = w_2 = w_7 = w_8 = 0.204 \text{ kg/s}$$



(a)



(b)

Two compressors and one evaporator

Heat and mass balance about the intercooler:

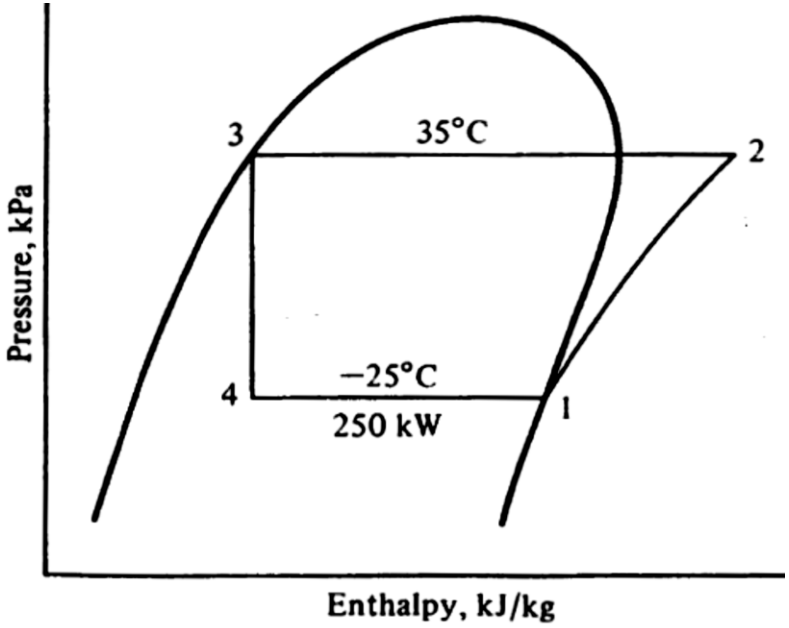
$w_2h_2 + w_6h_6 = w_7h_7 + w_3h_3$

$w_6 = w_3$ and $w_7 = w_2$

$0.204(1573) + w_3(366) = 0.204(202) + w_3(1463)$

$w_3 = 0.255$ kg/s

Low-stage power: (0.204 kg/s) (1573 – 1430 kJ/kg) = 29.2 kW
High-stage power: (0.255 kg/s) (1620 – 1463 kJ/kg) = 40.0 kW
Total power: 29.2 + 40.0 = 69.2 kW



This power requirement can be compared with that of a single-compressor system developing 250 kW of refrigeration at -25°C with a condensing temperature of 35°C. The pressure-enthalpy diagram is shown in Fig. 16-11. The .enthalpies are:

$h_1 = 1430$ kJ/kg $\quad h_2 = 1765 \quad h_3 = h_4 = 366$

$$w_1 = \frac{250 \text{ kW}}{1430 - 366} = 0.235 \text{ kg/s}$$

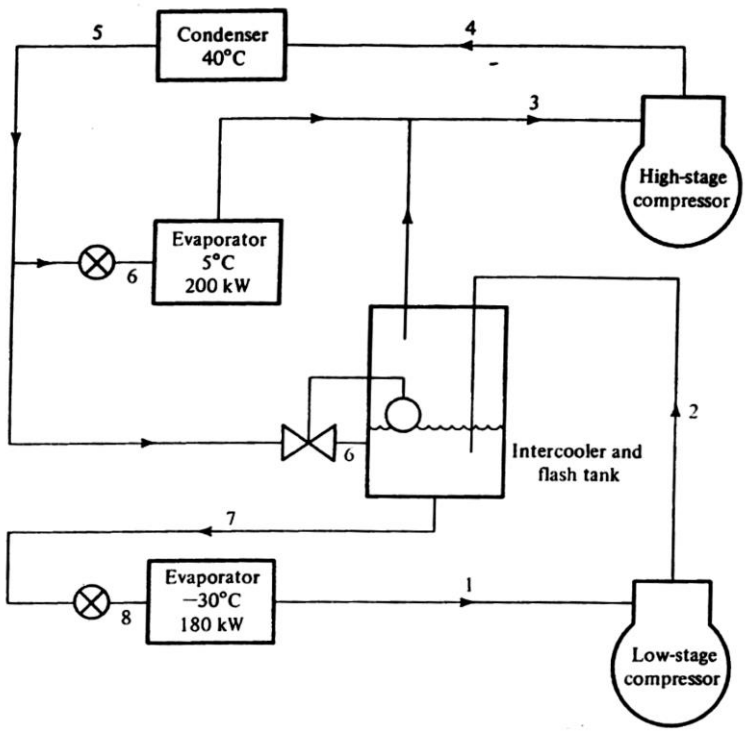
Power = 0.235(1765 – 1430) = 78.7 kW

The two-stage compressor system requires 69.2 kW, or 12 percent less power than the single-compressor system.

Two compressors and two evaporators

The system which has two evaporators operating at different temperatures is common in industrial refrigeration. A dairy cooling milk and manufacturing ice cream has been mentioned. A frozen-food plant may require two evaporators at different temperatures, one at -40°C to quick-freeze the food and the other at -25°C to hold the food after it is frozen. Process and chemical industries often require different temperatures of refrigeration in various sections of the plant. Evaporators at two different temperatures can be handled efficiently by a two-stage system which employs intercooling and removal of flash gas.

Example 16-4: In an ammonia system one evaporator is to provide 180 kW of refrigeration at -30°C and another evaporator is to provide 200 kW at 5°C. The system uses two-stage compression with intercooling and is arranged as in Fig.
16-12a. The condensing temperature is 40°C. Calculate the power required by the compressors.



(a)

Two compressors and two evaporators

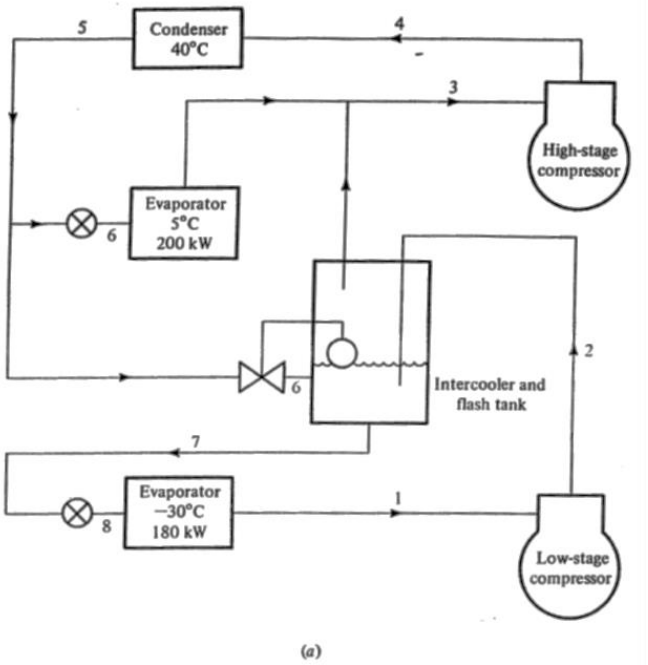
Solution Sketch the pressure-enthalpy diagram of the cycle as in Fig. 16-12b. The discharge pressure of the low-stage compressor and the suction pressure of the high-stage compressor are the same as the pressure in the 5°C evaporator. Next determine the enthalpies at the state points.

$h_1 = h_g$ at -30°C = 1423 kJ/kg
$h_2 = h$ at 517 kPa after isentropic compression = 1630
$h_3 = h_g$ at 5°C = 1467
$h_4 = h$ at 1557 kPa after isentropic compression = 1625
$h_5 = h_f$ at 40°C = 390.6 $\quad h_6 = h_5 = 390.6$
$h_7 = h_f$ at 5°C = 223 $\quad h_8 = h_7 = 223$

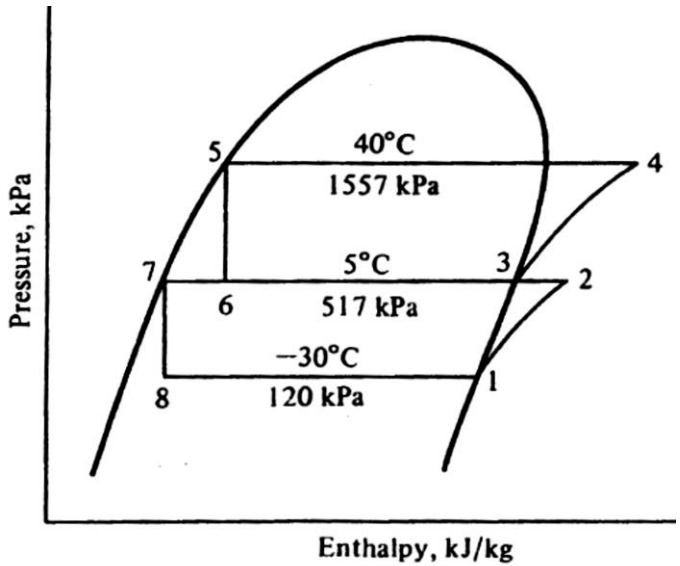
The mass rates of flow are

$$w_1 = \frac{180 \text{ kW}}{1423 - 223} = 0.150 \text{ kg/s}$$

$$w_7 = w_8 = w_2 = w_1 = 0.150 \text{ kg/s}$$

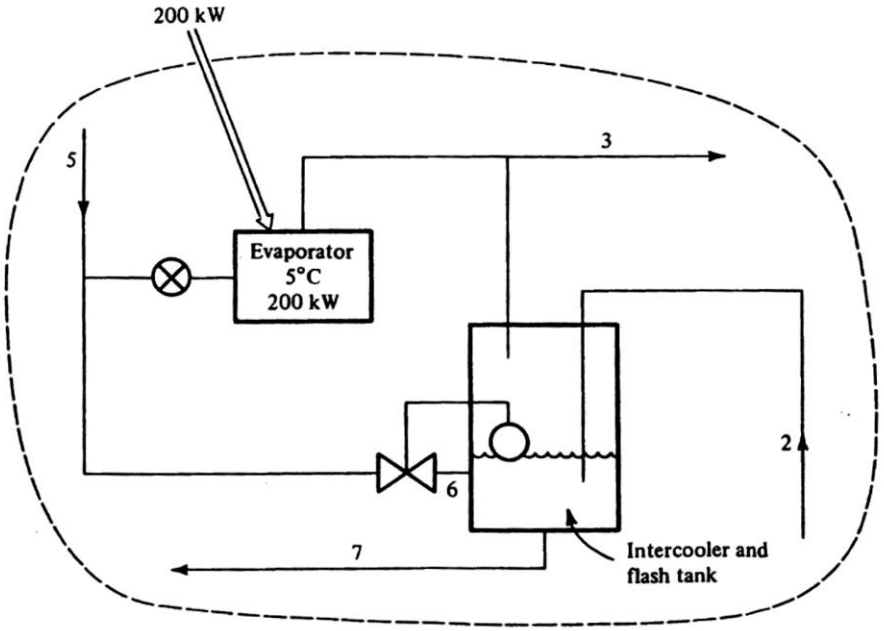


(a)



Two compressors and two evaporators

Probably the simplest way to calculate the mass rate .of flow handled by the high-stage compressor is to make a heat and mass balance about both the high temperature evaporator and the intercooler, as shown in Fig. 16-13.



Heat balance:

$$w_5 h_5 + 200\ \text{kW} + \bar{w}_2 h_2 = w_3 h_3 + w_7 h_7$$

Mass balance:

$$w_2 = w_7 = 0.150\ \text{kg/s}$$

Therefore

$$w_5 = w_3$$

Combining gives

$$390.6 w_3 + 200 + 0.150(1630) = 1467 w_3 + 0.150(223)$$

Solving leads to

$$w_3 = 0.382\ \text{kg/s}$$

The power required by the compressors can now be calculated:

Low-stage power: $0.150(1630 - 1423) = 31.1$ kW
High-stage power: $0.382(1625 - 1467) = 60.4$
Total 91.5 kW

Two compressors and two evaporators

If one compressor served each evaporator in single-stage compression, the power requirements of the two compressors would have been as follows:
Flow through low-temperature evaporator:

$$\frac{180 \text{ kW}}{1423 - 390.6} = 0.174 \text{ kg/s}$$

Flow through high-temperature evaporator:

$$\frac{200 \text{ kW}}{1467 - 390.6} = 0.186 \text{ kg/s}$$

Power for low-temperature system: $0.174(1815 - 1423) = 68.2$ kW
Power for high-temperature system: $0.186(1625 - 1467) = 29.4$
Total 97.6 kW

The combined power for the compressors serving the evaporators individually is greater than with the combined system in Fig. 16-12. The power required for the high temperature evaporator is the same in both cases, so all of the saving is attributable to flash-gas removal and intercooling of refrigerant serving the low-temperature evaporator.

Next: Multistage Refrigeration cycles

Thank you