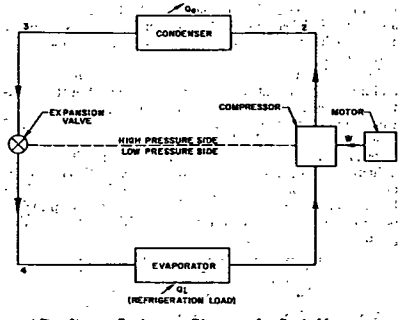


Fig. 1 .... Equipment Diagram for Basic Vapor Compression Cycle

absorbs the heat, except during the desuperheating process in the condenser.

Performance Factors for the Compression Refrigeration Cycle

The coefficient of performance (CP) of the vapor compression refrigeration cycle is the ratio of the refrigeration load (Q_{in} at the evaporator) and the shaft work input W to the compressor. Within limitations of economic factors, the design engineer should obtain the highest value of (CP), i.e., for a given refrigeration load, the engineer designs toward the lowest economical work input. Factors that affect the value of this performance ratio for the actual cycle are: (1) the temperatures of the low temperature region containing the evaporator and the coolant of the condenser, (2) the temperature differences required for the heat transfers in the evaporator and the condenser, (3) the pressure losses in the evaporator, condenser, connecting pipes, and compressor valves, (4) the efficiency of the compressor, (5) heat transfer in the connecting pipes, which is other than the heat flows in the evaporator and condenser, (6) the inherent irreversibility of the expansion-valve process, and (7) the refrigerant.

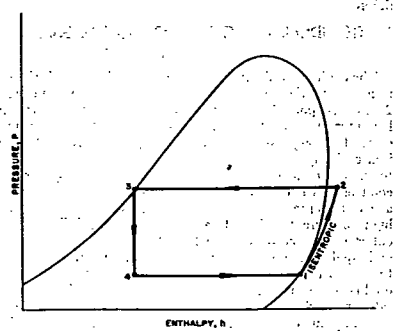


Fig. 2 .... Pressure-Enthalpy Diagram for Basic Vapor Compression Cycle

THE REVERSED CARNOT CYCLE

For given temperatures of the refrigerated space and the atmosphere, (usually the ambient air or economically available water), and for negligible temperature difference during heat transfer to or from the refrigerant, the reversed Carnot cycle possesses the highest available (CP).

As shown on the T-S diagram of Fig. 4, this cycle is composed of four reversible processes: (1) an isothermal expansion, 4-1, during which heat (the refrigeration load) flows from the refrigerated space to the refrigerant, (2) an adiabatic compression, 1-2, (3) an isothermal compression, 2-3, in which heat flows from the refrigerant to the atmosphere, (4) an adiabatic expansion, 3-4. Area (b) represents the refrigeration load, area (a) the net work input to the cycle, and area (a + b) the heat rejection to the atmosphere. The (CP) is represented by the ratio of the areas (b/a). Area (a + b) also represents the unavailable heat flow both to and from the cycle, while the negative of area (a) represents the available portion of the refrigeration load.

The (CP) of the reversed Carnot cycle shown on Fig. 4 is a function of the absolute temperatures of the refrigerated space T_R and the atmosphere T_a.

(CP) = refrigeration load/net work input

$$= Q_i/(Q_o - Q_i) = Q_i/W = T_R\Delta S/(T_a\Delta S - T_R\Delta S)$$

$$= 1/\left(\frac{T_a}{T_R} - 1\right)$$

Departures from the idealities of negligible temperature difference during heat transfer and the reversible processes of the reversed Carnot cycle require that the net work input to the refrigeration cycle be greater than the ideal minimum. Such departures degrade available energy and cause an increase in the entropy of the universe. The increased net work input is equal to the sum of all the available energies degraded, and is also equal to the product of the absolute temperature of the atmosphere and the increase in the entropy of the universe, $T_a\Delta S_{univ}$. Ideal conditions are illustrated in *Example 1*, and departures from ideality are shown in *Examples 2* to *9*.

Example 1: Determine the entropy change, work and coefficient of performance, for the cycle shown in Fig. 4. The temperature of the refrigerated space (T_R) is 400 R and that of the atmosphere (T_a) is 500 R. The refrigeration load is 200 Btu.

Solution:

$$\Delta S = S_1 - S_4 = Q_i/T_R = 200/400 = 0.5 \text{ Btu/R deg.}$$
$$W = \Delta S(T_a - T_R) = 0.5\ (500 - 400) = 50 \text{ Btu}$$
$$(CP) = Q_i/(Q_o - Q_i) = Q_i/W = 200/50 = 4.$$

The flow of energy, available energy (E_A), unavailable energy (E_U), and their area representations on Fig. 4 are:

Energy	Btu	Area
Q_i	200	b
Q_o	250	a+b
W	50	a
$(E_A)_i$	−50	−a
$(E_U)_i$	250	a+b
$(E_A)_o$	0	
$(E_U)_o$	250	a+b

The Principle of $T_a\Delta S_{univ}$ or ΣE_{AA} as the Increased Work Input to the Refrigeration Cycle

The change in the entropy of the universe as a result of the refrigeration cycle of *Example 1* is zero. The net change of entropy of any refrigerant in any cycle is always zero. The change in entropy of the refrigerated space in *Example 1* is $\Delta S_R = -200/400 = -0.5$ Btu/R deg, and that of the atmosphere is $\Delta S_a = 250/500 = +0.5$, giving a change in entropy of the universe of $\Delta S_{univ} = \Delta S_R + \Delta S_a = 0$.

In *Example 1*, there was no degradation of available energy, as all processes in the cycle were assumed reversible and all heat flows occurred with negligible temperature difference.

In actual cycles, the process of heat transfer occurs with finite temperature differences and all processes of the refrigerant involve friction. These processes are irreversible and cause degradation of available energy, an increase in the entropy of the universe, and an augmented work input to the cycle.

These concepts may be expressed as:

$$\Delta W = W - W_{id}$$
$$W = Q_o - Q_i$$
$$W_{id} = (T_a - T_R)(-\Delta S_R)$$
$$Q_o = T_a\Delta S_a$$
$$Q_i = T_R(-\Delta S_R)$$

Therefore:

$$\Delta W = T_a\Delta S_a - T_R(-\Delta S_R) = (T_a - T_R)(-\Delta S_R)$$
$$= T_a(\Delta S_a + \Delta S_R) = T_a\Delta S_{univ}$$

Also:

$$Q_o = (E_A)_o + (E_U)_o$$
$$(E_U)_o = (E_U)_R + (\Sigma E_{AA})_i + (E_{AA})_i$$
$$(E_U)_R = T_a(-\Delta S_R)$$

Therefore:

$$W = (E_A)_o + T_a(-\Delta S_R) + (\Sigma E_{AA})_i + (E_{AA})_i - T_R(-\Delta S_R)$$
$$- (T_a - T_R)(-\Delta S_R)$$
$$= (E_A)_o + (\Sigma_{AA})_i + (E_{AA})_i$$
$$= \Sigma E_{AA}$$

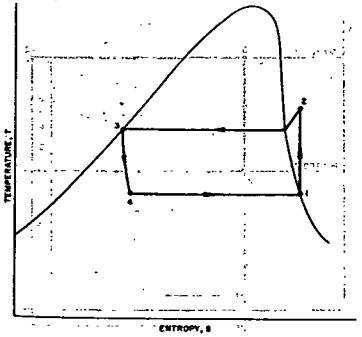


Fig. 3 .... Temperature-Entropy Diagram for Basic Vapor Compression Cycle

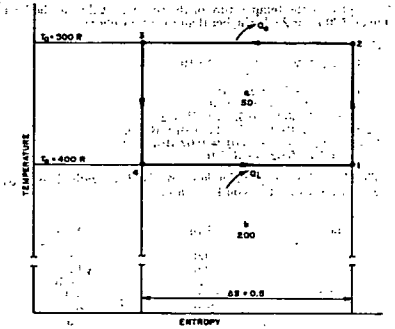


Fig. 4 .... Temperature-Entropy Diagram for Ideal Reversed Carnot Cycle of *Example 1*

where

$(E_U)_R$ = the unavailable portion of the energy removed from the refrigerated space at temperature T_R.

$(\Sigma E_{AA})_i$ = the sum of all the available energy which has been degraded by the refrigerant in its cyclical operation. This will be illustrated in later examples.

$(E_A)_o$ = the available energy outflow from the refrigerant during its process of heat rejection. $(E_A)_o$ is rendered unavailable when it is absorbed by the atmosphere (see *Example 2*).

$(E_{AA})_i$ = the available energy which is degraded by the heat transfer from the refrigerated space to the refrigerant when the process is accomplished through a finite temperature difference (see *Example 5*).

Example 2: For the cycle illustrated in Fig. 5, determine the additional net work input. The data are the same as for *Example*

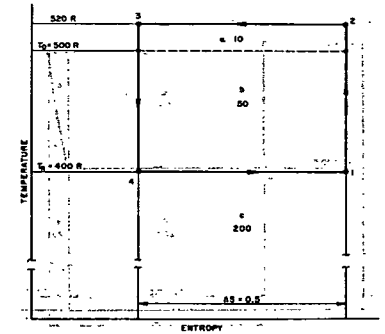


Fig. 5 .... Temperature-Entropy Diagram for Reversed Carnot Cycle of *Example 2*