1, except that the temperature of the refrigerant during the heat efflux is 520 R, or 20 deg higher than the atmosphere.

Solution:

$$
\begin{aligned}
W &= 0.5(520 - 400) = 60 \text{ Btu}\\
(CP) &= 200/60 = 3.33\\
\Delta W &= 60 - 50 = 10 \text{ Btu}\\
Q_o &= 200 + 60 = 260 \text{ Btu}\\
\Delta S_o &= 260/500 = 0.52 \text{ Btu/R deg}\\
\Delta S_R &= -200/400 = -0.50 \text{ Btu/R deg}\\
\Delta S_{univ} &= 0.52 + (-0.50) = 0.02 \text{ Btu/R deg}\\
T_o \Delta S_{univ} &= 500\,(0.02) = 10 \text{ Btu}
\end{aligned}
$$

The flow of energy, available energy and unavailable energy, and their representations on Fig. 5 are:

Energy	Btu	Area
Q_i	200	c
Q_o	260	a+b+c
W	60	a+b
$(E_A)_i$	−50	−b
$(E_U)_i$	250	b+c
$(E_A)_o$	10	a
$(E_U)_o$	250	b+c

The heat efflux from the refrigerant (260 Btu) is composed of its available portion (10 Btu) and its unavailable portion (250 Btu). When this heat efflux is absorbed by the atmosphere, the above available energy (10 Btu) is degraded by the irreversible process of heat transfer through the 20 deg finite temperature difference, and the heat thus absorbed by the atmosphere is rendered wholly unavailable. It is this degradation of 10 Btu of available energy that has caused the change in the entropy of the universe to be 0.02 Btu per R deg, and $T_o\Delta S_{univ}$, as well as the additional net work input, to be 10 Btu.

Example 3: The data are the same as in *Example 1*, except that the reversed Carnot cycle has been modified so that the adiabatic-compression process is irreversible, due to the presence of friction, and undergoes an entropy increase of 0.05 Btu/R deg. This cycle is depicted in Fig. 6.

Solution:

$$
\begin{aligned}
Q_o &= 500 \times 0.55 = 275 \text{ Btu}\\
W &= 275 - 200 = 75 \text{ Btu}\\
(CP) &= 200/75 = 2.67\\
\Delta W &= 75 - 50 = 25 \text{ Btu}\\
\Delta S_o &= 0.55 \text{ Btu/R deg}\\
\Delta S_R &= -0.50 \text{ Btu/R deg}\\
\Delta S_{univ} &= 0.55 - 0.50 = 0.05 \text{ Btu/R deg}\\
T_o \Delta S_{univ} &= 500 \times 0.05 = 25 \text{ Btu}
\end{aligned}
$$

Energy	Btu	Area
Q_i	200	d
Q_o	275	(a→e)
W	75	a+b+c+e
$(E_A)_i$	−50	−a
$(E_U)_i$	250	a+d
$(E_A)_o$	0	
$(E_U)_o$	275	(a→e)
$(E_{AA})_r$	25	b+c+e

The irreversibility of the adiabatic process has caused: (1) a degradation of available energy [$(E_{AA})_r$] of 25 Btu, (2) an increase in the entropy of the universe (ΔS_{univ}) of 0.05 Btu/R deg, (3) an increase in net work input (ΔW) of 25 Btu, and (4) the value of $T_o\Delta S_{univ}$ to be 25 Btu. As there were no heat transfers through finite temperature differences, there was no degradation of energy from that source.

Example 4: The data are the same as in *Example 1*, except that the reversed Carnot cycle has been modified so that the refrigerant, during the process of heat influx, undergoes an irreversible process and has an additional entropy increase, due to friction, of 0.05 Btu/R deg. The cycle is presented in Fig. 7.

The evaluation of this cycle is similar to that in *Example 3*, with the following differences. In *Example 3*, $S_1 - S_4 = 0.5$ Btu/R deg, and was due solely to the heat influx. In *Example 4*, $S_1 - S_4 = 0.55$ Btu/R deg, which is partly due to the influx of heat (0.5 Btu/R deg), and the remainder (0.05 Btu/R deg) is caused by friction. In *Example 3*, ΔW, $T_o\Delta S_{univ}$ and E_{AA} (each equal to 25 Btu) were due to the irreversibility of the adiabatic-compression process, while in *Example 4* their value is the result of the fluid friction during the process of heat influx.

The tabulated values in *Example 3* apply exactly to *Example 4*.

Example 5: In this case, the departures from the idealities of the reversed Carnot cycle of *Example 1* are a combination of those in *Examples 2, 3*, and *4*. All four processes are irreversible and have a change in entropy ΔS_f, which is due to friction alone and is equal to 0.05 Btu/R deg. During the efflux of heat, the temperature of the refrigerant is 520 F (*Example 2*), and during the heat influx, the refrigerant temperature is 380 R (20 deg lower than the refrigerated space). The refrigeration load and the temperatures of the atmosphere and the refrigerated space remain the same. The cycle is shown in Fig. 8.

Solution:

$$
\begin{aligned}
\Delta S_{Q_i} &= 200/380 = 0.52632 \text{ Btu/R deg}\\
\Delta S_{Q_o} &= (S_3 - S_2) - \Delta S_f\\
&= -0.67632 - 0.05 = -0.72632 \text{ Btu/R deg}\\
Q_o &= 520 \times 0.72632 = 377.69 \text{ Btu}\\
W &= 377.69 - 200 = 177.69 \text{ Btu}
\end{aligned}
$$

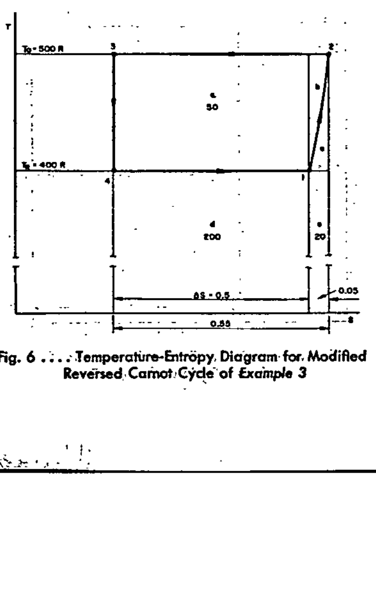

Fig. 6 . . . . Temperature-Entropy Diagram for Modified Reversed Carnot Cycle of *Example 3*

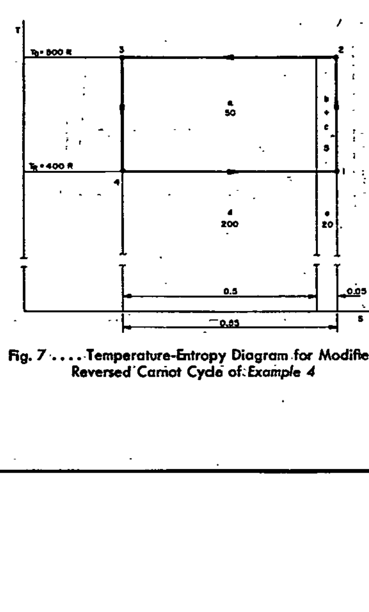

Fig. 7 . . . . Temperature-Entropy Diagram for Modified Reversed Carnot Cycle of *Example 4*

$$
\begin{aligned}
\Delta W &= 177.69 - 50 = 127.69 \text{ Btu}\\
(CP) &= 200/177.69 = 1.1256\\
\Delta S_R &= -200/400 = -0.5 \text{ Btu/R deg}\\
\Delta S_o &= 377.69/500 = 0.75538 \text{ Btu/R deg}\\
\Delta S_{univ} &= 0.75538 - 0.5 = 0.25538 \text{ Btu/R deg}\\
T_o \Delta S_{univ} &= 500 \times 0.25538 = 127.69 \text{ Btu}
\end{aligned}
$$

The energy flows to and from the refrigerant, and their area representations on Fig. 8 are:

Energy	Btu	Area
Q_i	200	o
Q_o	377.69	(a→p)
W	177.69	(a→n) + p
$(E_A)_i$	−63.16	− k
$(E_U)_i$	263.16	k + o
$(E_A)_o$	14.53	(a → g)
$(E_U)_o$	363.16	(h → p)
$(E_{AA})_r$	100	(h) + (i + j) + (l+p) + (m+n)

The equations for the various energy balances on the refrigerant are given below, and under each term is its numerical value.

$$
\begin{aligned}
Q_i + W &= Q_o\\
200 + 177.69 &= 377.69\\
Q_i &= (E_A)_i + (E_U)_i\\
200 &= -63.16 + 263.16\\
Q_o &= (E_A)_o + (E_U)_o\\
377.69 &= 14.53 + 363.16\\
W + (E_A)_i &= (E_A)_o + (E_{AA})_r\\
177.69 - 63.16 &= 14.53 + 100\\
114.53 &= 114.53\\
(E_U)_i + (E_{AA})_r &= (E_U)_o\\
263.16 + 100 &= 363.16
\end{aligned}
$$

The heat flow from the refrigerated space to the refrigerant has occurred through a temperature drop of 20 deg. This in itself has caused a degradation of available energy equal to 13.16 Btu, which is the decrease in availability of the heat flow (from −50 Btu in *Example 1* to −63.16 Btu in this example). Also, the process of heat flow from the refrigerant to the atmosphere has been accomplished by a 20 deg temperature drop. The available energy degraded here is 14.53 Btu, which is the value of the available energy rejected by the refrigerant, and which was rendered unavailable when the efflux heat flow was absorbed by the atmosphere.

The available energy degraded by the refrigerant in its cyclical operation is 100 Btu. Thus, the total degradation for the combination of refrigerant, refrigerated space, and atmosphere is 127.69 Btu, which equals the additional net work input to the cycle.

THE ACTUAL BASIC VAPOR COMPRESSION REFRIGERATION CYCLE

Although the ideal basic vapor compression refrigeration cycle shown in Figs. 2 and 3 has been termed *ideal*, the cycle, in its interaction with the atmosphere, entails two irreversibilities: (1) the inherent irreversibility in the expansion valve process, and (2) the process of heat transfer through a finite temperature difference during the desuperheating of the vapor in the condenser. To remove the first irreversibility, a reversible adiabatic expansion engine would be substituted for the expansion valve, and the work output from this engine would supply part of the work input to the cycle. The process for this engine is shown as 3–4′ in Fig. 9. To remove the second irreversibility, the compression process 1–2 of Fig. 9 would be replaced by a reversible two part compression, the isentropic process 1−2′ and the isothermal process 2′−2″ of Fig. 9. These improvements would produce a reversed Carnot cycle which is equivalent in all respects to the one in Fig. 4, except that, due to the increased entropy change during the heat influx to the refrigerant, the refrigeration load, net work input, and heat rejection would all be increased 4.52 percent.

The complexity of the equipment needed for these improvements precludes its application, and the practical arrangement is that shown in Fig. 1.

In the following three demonstrations, *Example 6* will present a Second-Law analysis of an ideal vapor cycle as depicted on Fig. 3 and *Examples 7 and 8* will analyze vapor cycles that represent the actual irreversible conditions of friction and heat transfer found in practice.

Example 6: The data are those of *Example 1*. The vapor is Refrigerant 12. All processes for the vapor are reversible, except that through the expansion valve. Heat transfers are accomplished with negligible temperature differences, except for the desuperheating process in the condenser. The cycle is shown in Fig. 9.

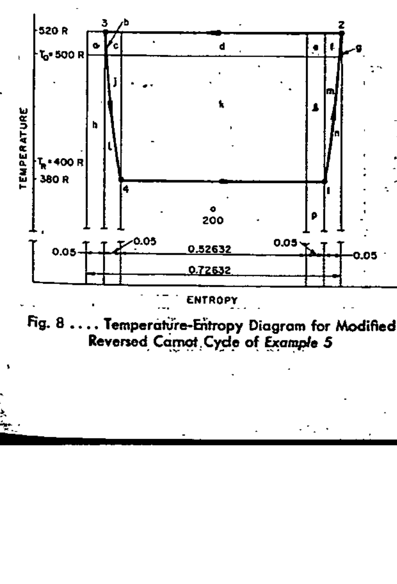

Fig. 8 . . . . Temperature-Entropy Diagram for Modified Reversed Carnot Cycle of *Example 5*

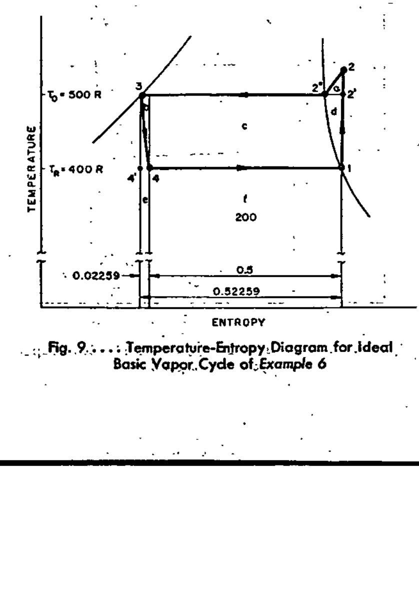

Fig. 9 . . . . Temperature-Entropy Diagram for Ideal Basic Vapor Cycle of *Example 6*