5-24-1988

Thermodynamics Analysis of Refrigeration

Joe Lam
Utah State University

JOE LAM

HONORS 390 / SENIOR PROJECT

Thermodynamics Analysis of Refrigeration

For Honors 390

Grade A , [illegible] , 5/24/88

In this term paper,I will look indepth at refrigeration. Refrigeration is a rather long and complex topic. Therefore, I will only discuss it as detail as conditions permit . As for the parts where minute details required, I will refer you to the selected references. In discussing refrigeration, I will first look at the Carnot cycle as a fundamental issue for refrigeration.

Carnot Cycle: The Carnot cycle is the fundemental of the heat engine . It is a very complex topic. I will only dicuss the most fundamental issue so that I can talk about refrigeration. Hence it is by no means is the complete discussion of the Carnot cycle.

Sadi Carnot was the first to state the essential elements required for highest efficiency.

I will first look at the process of reversibility and irreversibility before going any further into the Carnot cycle.

All processes described in thermodynamics are either reversible or irreversible. A process is said to be reversible when it satisfies the following conditions:

1) When the direction of the processes is reversed, the system taking part in the process can assume in the reverse order the states traversed in the direct process.
2) The external actions are the same for the direct and reversed in processes or differ by an infinitesimal amount only.

3) Not only the system undergoing the change,but all connected systems can be restored to initial conditions.

Any process which fails to satisfy any of the above conditions in any way is called irreversible process.

The reversible cycle are quasistatic and can be plotted in any plane that one chooses, say, the p-V plane. All friction, both that internal to the working substance and that between moving surfaces, is neglected. Such engines have to operate infinitely slow,receive heat reversibly, reject all heat reversibly, etc.. Since engines must be operate at high speed, they are subject to turbulence , pressure drops, dynamic effects due to accelerations and decelerations, etc., and a host of other irreversible effects.

Recognizing that reversible cycles are better than nonreversible cycles, Carnot came up with a criterion for heat reservoirs, namely, two. Accordingly, heat can be absorbed only along the higher isotherm, and none may be rejected except along the lower isotherm.

Carnot states his famous principle: "Given an engine that is reversible and that operates between two fixed temperatures. Then no other engine operating in between these temperatures can exceed this engine in efficiency."

In general, the Carnot engine operates so that a medium first expands reversible at constant temperature. The constant temperature reversible expansion is follow by a rever-

sible adiabatic expansion to the temperature of the receiver. The fluid is then compressed by a reversible isothermal process to a predetermined state, after which compressed adiabatically and reversible to the initial state.

Refrigeration: Refrigeration is a process by which energy is removed from a body at such rate that the temperature of the body is maintained lower than the temperatures of the surroundings. Refrigeration may be accomplish under certain conditions by means of ice, rapid evaporation of liquids, and mechanical devices.

First of all, I will talk about some of the fundamental units of refrigeration. The refrigeration capacity is related to the latent heat of fusion of ice. A ton of refrigeration is the cooling effect or heat exchange equivalent to that obtained by melting 1 ton of ice at 32 F. Or precisely, by definition:

$$1 \text{ standard ton refrigeration} = (2000 \text{ lbs.})(144 \text{ Btu/lb.}) = 288{,}000 \text{ Btu.}$$

More often the ton of refrigeration is considered to be a rate

$$1 \text{ standard commercial ton refrigeration} = 288{,}000 \text{ Btu/ 24 Hrs.} = 200 \text{ Btu/min.}$$

To obtain the cooling effect called refrigeration, work must be expended. The coefficient of performance CP is defined as the ratio of the refrigeration to the work supplied:

$$CP = \left.\frac{\text{refrigeration}}{\text{work added}}\right]_{cycle} = \left.\frac{\Delta Q_A}{-\sum \Delta W_A}\right]_{cycle}$$

The refrigeration ΔQ_A is the heat added to the working substance in the cycle, and $\sum \Delta W_A$ is the work or the available energy used to drive the apparatus. The value of CP can be greater or less than unity. Another gauge of the performance of the refrigerating machine is the horsepower per ton of refrigeration:

$$Hp/ton = \frac{200}{CP(42.5)} = \frac{4.71}{CP}$$

There are many processes which refrigeration can be attained. I will discuss a few frequent used processes here.

The Carnot Refrigeration Cycle: The Carnot cycle, being completely reversible, is a model of perfection for a refrigeration cycle. Two important concepts involving reversible cycles are: 1) no refrigerating cycle may have a coefficient of performance higher than that for a reversible cycle which would be operated between the same temperatures of source and receiver, 2) all reversible cycle when operted at the same temperatures would have the same coefficient of performance. The Carnot engine (heat pump) with its working substance, or refrigerant, confined to saturated and two-phase states is illustrated in diagram below, Fig. 1.

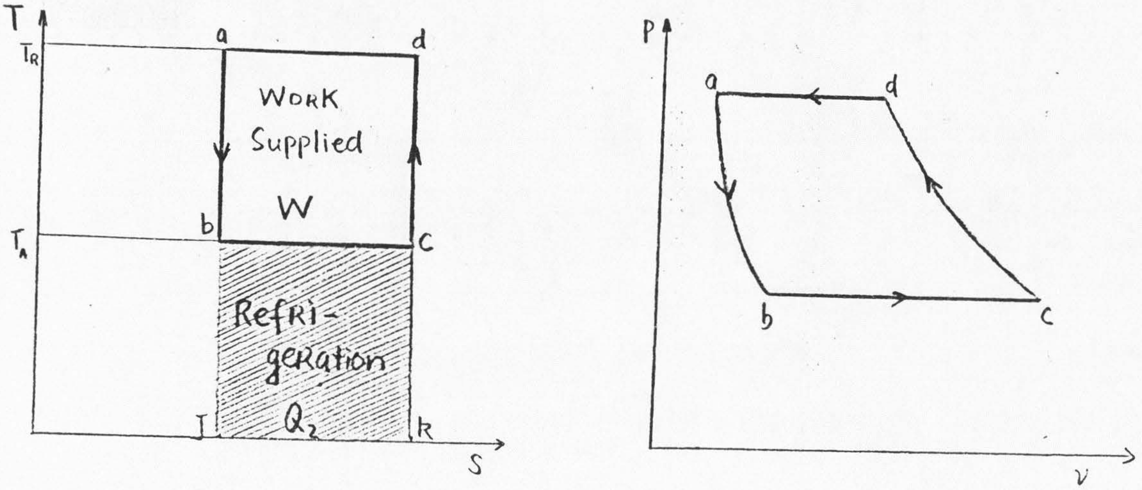

Fig. 1 - Carnot refrigeration cycle.

The cycle consists of the following processes:

ab - work produced by isentropic expansion to the lower temperature T_A .

bc - heat added in the evaporator at the lower temperature T_A .

cd - work added in the isentropic compression to the higher temperaturre T_R .

da - heat rejected in the condenser at the higher temperature T_R .

The refrigeration is proportional to the area jbck, the net work supplied is proportional to area abcd, and the heat rejected is proportional to area jakd. For this cycle,

$$Q_1 = T_R (s_d - s_a)$$

$$Q_2 = T_A (s_d - s_a)$$

$$W = Q_1 - Q_2$$

$$CP_{CARNOT} = \frac{\Delta Q_A}{\sum \Delta W} = \frac{\Delta Q_{bc}}{-(\Delta Q_{bc} + \Delta Q_{da})} = \frac{T_A}{T_R - T_A}$$

It has been demonstrated that, between fixed temper-

ature limits, all reversible engines have the same thermal efficiency, the thermal efficiency of the Carnot cycle. Thus, the work of the Carnot cycle is the maximum work that can be obtained from transformation of heat energy in a heat-engine cycle. But if the Carnot cycle or any reversible cycle dilivers the maximum work as a heat engine, it must therefore require the least work for the reversed operations as a heat pump. Thus the coefficient of performance of a reversible engine is the optimum performance for the refrigeration machine. Moreover, all reversible heat pumps operating between the same temperature limits will have the same coefficient of performance no matter what fluid is used as the refrigerant and dependent only upon the temperatures of source and sink. These conclusions are valid only for the reversible cycle because the irreversibilities and the temperature limits of the real cycle will be affected in greater or less degree by the properties of the refrigerant.

A relative efficiency can be defined by comparing the work of the real refrigeration cycle to that of the Carnot:

$$\eta_c C = \left.\frac{\Delta W_{\text{CARNOT CYCLE}}}{\Delta W_{\text{REFRIGERATION CYCLE}}}\right|_{Q_A = C} = \frac{CP_{\text{REFRIGERATION CYCLE}}}{CP_{\text{CARNOT PUMP}}}$$

The relative efficiency is always less than unity. When this efficiency is computed, the work of the Carnot cycle should be based upon the temperatures of the refrigerated space and cooling medium. If, instead, the relative effi-

ciency is computed from the temperatures of heat addition and rejection within the cycle, then the system is not penalized for the irreversibility of the temperature differences that actually are present.

The Vapor-Compression Refrigeration Cycle: for most commercial purposes, the temperature range demmanded of the refrigeration cycle is small, and therefore the work produced in the expansion process is negligible relative to the compression work. Thus, the refrigeration machine can be simplified by substituting a throttling valve for the expansion turbine, with consequent savings in initial cost and maintenance. The elements of the ideal vapor-compression cycle are illustrated below, Fig. 2.

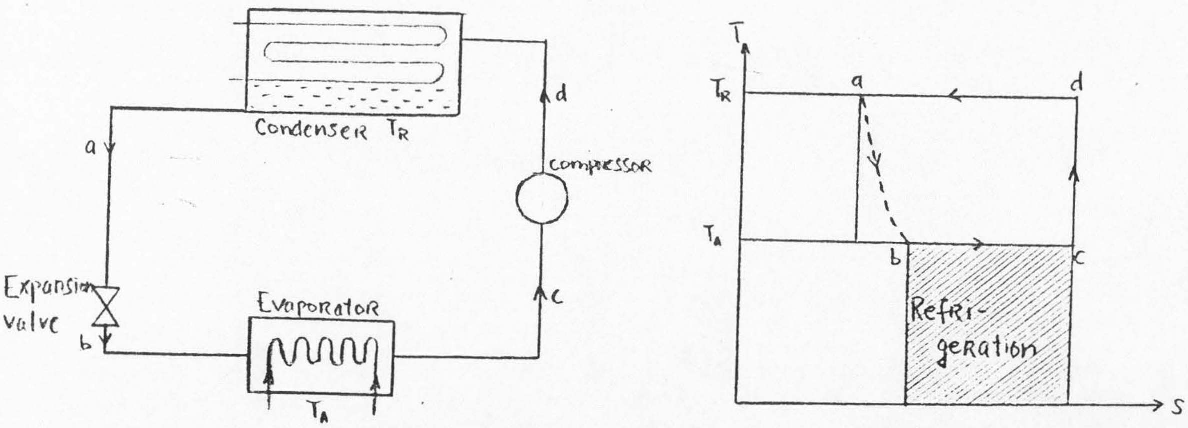

Fig. 2 - Ideal vapor-compression refrigeration cycle.

ab - throttling at constant enthalpy to the lower temperature T_A.

bc - heat added in the evaporator at the lower temperature T_A.

cd - work added in the isentropic compression to

the higher temperature T_R .

da - heat rejected in the condenser at the higher temperature T_R .

Note that in the throttling process the temperature of the refrigerant falls to the saturation temperature corresponding to the pressure maintained in the evaporator by the suction of the compressor.

In the real system, all processes are irreversible and temperature differences will be present. Diagram below, Fig. 3, will illustrate some of these effects.

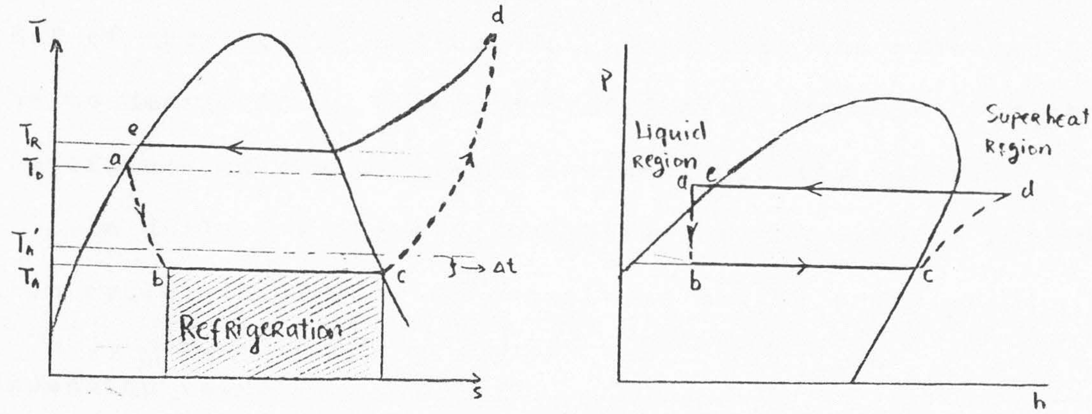

Fig. 3 - Irreversibilities in the vapor-compression refrigeration cycle.

The temperature T_A at the end of the throttling process ab must be lower than the temperature T_A' of the room or space being refrigerated. The rate of vaporization of the refrigerant in the evaporator depends on this temperature difference and on the surface offered by the evaporator. The refrigerant entering the compressor may be wet or dry vapor, the state being controlled by the location of the

vapor take-off on the evaporator as well as by the mass-flow rate, which is governed by a control mechanism on the expansion valve. Dry compression is usually preferred because a greater refrigerating effect can be abtained per unit of mass flow and because the presence of large drops of liquid in the compressor may cause damage. Consequently, the final state d lies, most often, in the superheat region at a temperature above T_R . The gas is cooled and condensed in the condenser because of a temperature difference between gas and coolant. The design of the condenser is to facilitate subcooling ea to a temperature quite close to that of the cooling medium, T_D. Note that the pressure in the condenser for a selected refrigerant depends upon the temperature of the available coolant, usually water.

In analyzing the flow processes of the vapor - compression cycle, the following quantities may be calculated:

Expansion valve:

$$\Delta q - \Delta w = \Delta h = 0$$

$$h_a = h_b = (h_f + x h_{fg})_b$$

Evaporator:

$$\Delta q - \Delta w = \Delta h \quad \text{and} \quad \Delta w = 0$$

$$\Delta q_A = h_c - h_b$$

Compressor:

$$\Delta q - \Delta w = \Delta h$$

$$\Delta w_{cd} = -(h_d - h_c) + \Delta q_{cd}$$

(Δq rejected is negative, by convention)

If compression is isentropic,

$$\Delta w_{cd,rev} = -(h_d - h_c)_{s=C}$$

Condenser:

$$\Delta q - \Delta w = \Delta h \qquad \text{and} \qquad \Delta w = 0$$

$$\Delta q_R = h_a - h_d$$

Coefficient of performance:

$$CP = \frac{\Delta Q_A}{-\Delta W_A} = \frac{h_c - h_b}{(h_d - h_c) - \Delta q_{cd}}$$

Mass-flow rate of refrigerant per ton of refrigeration:

$$m_f = lb_m / (min)(ton) = \frac{200\ Btu/(min)(ton)}{\Delta q_A\ Btu/lb_m} = \frac{200}{h_c - h_b}$$

Compressor capacity per ton of refrigeration:

$$C(cfm/ton) = m_f[lb\ /(min)(ton)]v(ft^3/lb_m) = m_f v$$

v = specific volume of refrigerant at compressor inlet

Horsepower required per ton of refrigeration:

$$hp/ton = \frac{12000\ Btu/(hr)(ton)}{(2544\ Btu/hp\text{-}hr)\ CP} = \frac{4.71}{CP}$$

The real cycle contains an irreversible throttling processes and also varying degrees of superheat. Since these variables are governed by the characteristics of the fluid and not directly by the evaporator or condenser pressure, the CP of different refrigerant will not be the same.

Properties of Refrigerants: It is desirable from practical as well as theoretical considerations that the refrigerant should exhibit certain characteristics. The properties of the ideal refrigerant would show the following qualities:

1 - The latent heat of vaporization should be large, and the heat capacity of the liquid should be

small because then the mass-flow rate would be low. Note that, the smaller the heat capacity of the liquid, the less will be the vaporization during throttling and therefore the greater the amount of heat that can be abstracted from the cold source.

2 - The critical point should be above the highest operating temperature, for then the fluid after compression is close to the two-phase region where condensation can take place at constant temperature; not oly are heat-transfer rates better in the two-phase region, but also the irreversibility of a temperature difference is reduced by the constancy of the temperature.

3 - The vapor pressure in the condenser should not be high. High pressures increase design costs and maintenance.

4 - The vapor pressure in the evaporator should be higher than atmospheric pressure. This would prevent air from leaking into the system and so increasing the amount of work that must be supplied to the compressor for a definite amount of refrigeration. The humidity in the air is especially troublesome because the water tends to freeze in the smallest section of the system, the expansion valve.

5 - The entropy of the saturated vapor should not

change markedly with pressure, or else it should increase slightly as the pressure increases because then the refrigerant can enter the condenser as a wet or saturated vapor..

6 - The properties of the fluid should be condusive to high rates of heat transfer in order that both surface areas and temperature differences can be small in the heat exchangers.

7 - The refrigerant should be cheap in cost, stable, nonexplosive, noncorrosive under all conditions of operation, and nonpoisonous for safety purposes.

No refrigerant is known that have all these properties, but certain fluids have qualities that are particularly suited for special applications.

Vacuum Refrigeration: Water without a doubt, the safest as well as the cheapest vapor refrigerant although the cycle temperatures must ordinarily be above 32° F. For certain processes, notably air conditioning, low temperatures are not needed and water can be used as the refrigerant, an ejector is generally used although a centrifugal pump is a possible substitute.

In Fig. 4 below, relative warm water is sprayed into a flask chamber that is maintained at a low pressure by an ejector or pump. A small portion of the water flashes into steam, and the latent heat of vaporization so demanded is supplied by the water, the flash chamber is insulated to

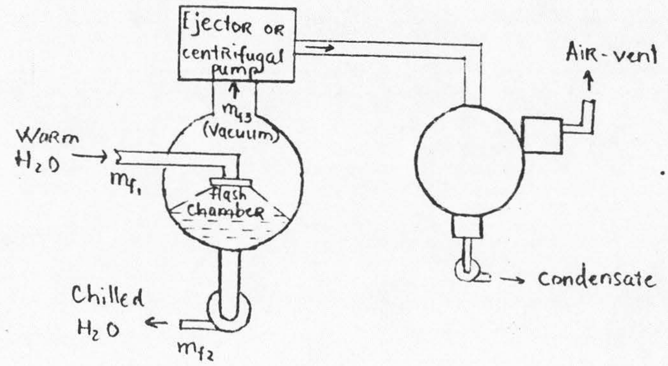

Fig. 4 - Vacuum refrigeration system.

reduced heat transfer from the surroundings. Thus, the water is cooled to the saturation temperature dictated by the pressure. The chilled water is then pumped to the point where it is required, and warmed water returned to the flash chamber for cooling.

The vapor withdrawn from the flash chamber by the ejector or pump is compressed and delivered to the steam condenser. Here the pressure, as in the flash chamber, is far below atmospheric, the particular value being determined by the temperature of the available cooling water.

The vacuum system, using steam-jet ejectors, has few moving parts because a mechanical compressor is eliminated. This simplification, along with the cheapness and nontoxicity of water, makes up for the inefficiency of the ejector. And if waste steam is available, at pressures above 5 psia, or preferably much higher, a water-vapor system becomes highly desirable.

An energy balance can be made on the flash chamber to

show the refrigeration:

$$m_{f_1} h_1 = m_{f_2} h_2 + m_{f_3} h_3 \qquad (\Delta Q, \Delta W = 0)$$

Since $m_{f_2} (h_1 - h_2) = m_{f_3} (h_3 - h_1)$.

then $m_{f_2} (h_1 - h_2) = m_{f_3} (h_3 - h_1)$

and the refrigeration is,

$$\Delta Q_A = m_{f_2} (h_1 - h_2) = m_{f_3} (h_3 - h_1)$$

Absorption Refrigeration: It has already been remarked that the work necessary to compress a liquid is but a small fraction of that required to compress a gas. Thus, the work supplied to the refrigeration system could be reduced if the refrigerant were pumped to the condenser pressure as a liquid rather than as a gas. A means of achieving this objective is offered by the absorption system in Fig. 5.

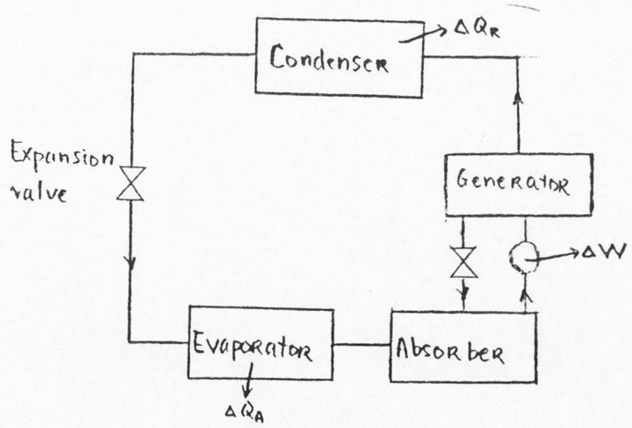

Fig. 5 - Elements of the absorption refrigeration cycle.

Here as in the compression system the refrigerant passes from the condenser to expansion valve to the evaporator. But, unlike the compression system, the vapor issuing from the evaporator is dissolved in a cold solvent in the absorber;

this liquid solution is then pumped into the high-pressure generator where the solution is heated.

The refrigerant is thus liberated from the solution and passes to the condenser while the solvent returns to the absorber. Of course, the solvent must be able to hold in the cold solution a greater amount of refrigerant than in the hot solution.

An aqua-ammonia solution is generally used (ammonia as the refrigerant, water as the solvent) in the absorption system. The ammonia vapor entering the absorber is dissolved in relatively cold water, about 80° F. Heat is liberated in the process, and cooling coils are necessary to maintain the low temperature. The cold solution of water and ammonia (strong aqua) is pumped to the generator where it is heated to about 200° F to 300° F. The hot solution can not hold as much ammonia as the cold solution; hence, ammonia vapor is liberated and passes to the condenser. The hot and therefore weak solution of ammonia and water or the weak aqua is then throttled back to the absorber to be cooled and strengthened.

The simple system of Fig. 5 would deliver not only ammonia but also a large amount of water to the condenser. To improve the performance, a more complicated system must be used.

Liquifying and Solidifying Processes: Suppose that the problem is to produce a liquid or a solid phase of a sub-

stance. One of the following processes, Fig. 6, could be selected:

1 - Change of phase by cooling at essentially constant pressure (1-5).

2 - Change of phase by essentially adiabadic expansion in either a piston-type or a turbo expander (2-5).

3 - Change of phase by compression and cooling (3-5).

4 - Change of phase by throttling (4-5).

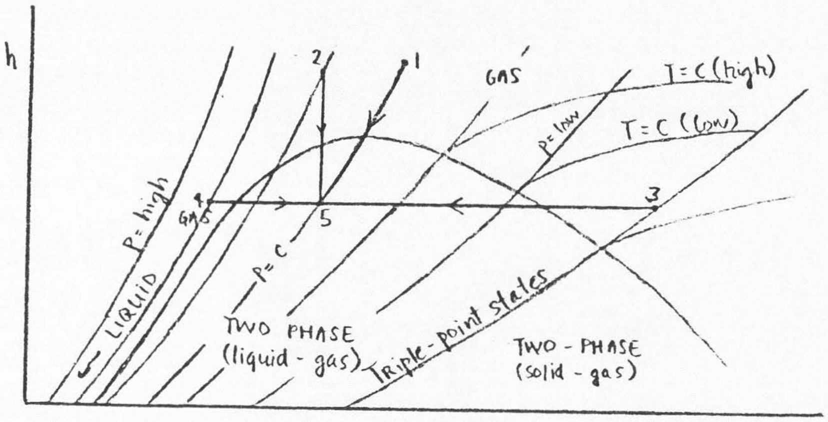

Fig. 6 - Cooling processes.

The throttling process has the merit of simplicity, it has no moving parts for attaining low temperatures where lubrication may be a problem.

In the Linde process for liquefying gases such as oxygen or air, the apparatus appears as shown below, in Fig.7. Gas enters at 1 and is throttled at x to a lower pressure, with consequent fall in temperature, and the lower-pressure and lower-temperature gas counterflows to the exit, 2, thus cooling the incoming flow. Eventually, a steady state is

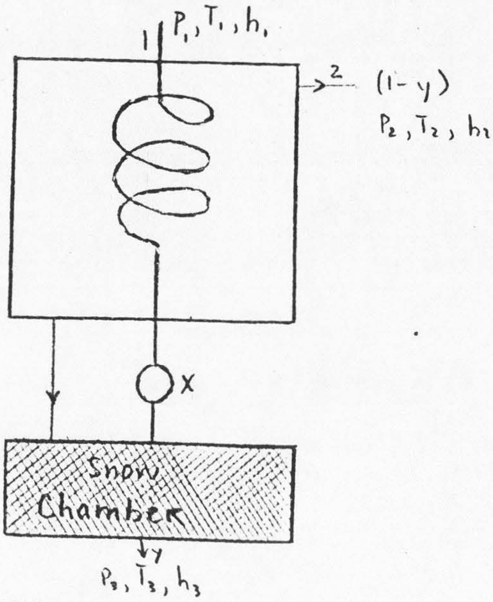

Fig. 7 - Linde process.

reached, and a fraction y of the incoming flow is liquified or solidified. For an adiabatic steady flow process without work,

$$\sum m_f \Delta h = 0$$

$$y h_3 + (1 - y) h_2 = h_1$$

And the solid or liquid fractions y equals

$$y = \frac{h_2 - h_1}{h_2 - h_3}$$

The enthalpy h_3 is set by the exit pressure p_3, the enthalpy h_2 is determined by the exit pressure and the pressure drop in the heat exchanger, while the initial enthalpy h_1 is set by the values of p_1 and t_1. Of all these variables, p_1 is most readily varied; hence, for maximum yield, the foregoing equation shows that h_1 should be small.

The Refrigeration Cycle as a Heat Pump: The heating of buildings is a recurring engineering problem. A building can be

heated by burning fuels or by dissipating work, for example, when an electric current passes through a resistance heater. Electric-resistance heating, while convenient, is the ultimate degradation of energy. Consider that in the power plant a fuel is burned and work is obtained in amount seldom as much as 25 percent of the heat of combustion of the fuel. Then, the work must have value at least four times that of the fuel used. Because of this fact, the average building can be heated more cheaply by direct firing of an expensive fuel in an inefficient furnace than by irreversibly using electrical energy that was produced by burning an inexpensive fuel in an efficient furnace.

Even when the electrical energy is produced by water power, the use of such energy for an irreversible heating purpose may be more expensive than direct firing of fuel. The cost of electrical energy includes not only the cost of any fuel used but also the fixed costs of the installation and the distribution costs.

The remedy is to replace the highly irreversible electric-resistance heating process with a process that can at least approach reversibility. Since work in the form of electrical energy can be derived from a heat engine cycle, then the cycle can be reversed and heat obtained by supplying work. By this means the ratio of performance is reversed and the amount of the heat received can be many times the amount of work added. Consider the familiar Carnot cycle of Fig. 8.

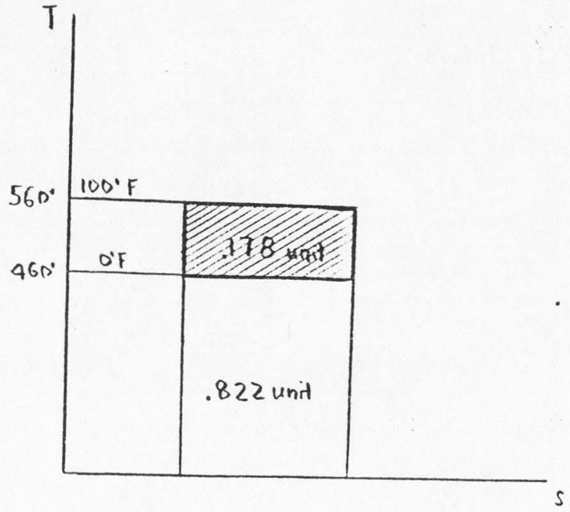

Fig. 8 - The coefficients of performance of the reversed Carnot cycle.

Here for every unit of heat added to the cycle, work is obtained of amount equal to

$$\Delta W = \eta_t \Delta Q_A = \frac{560 - 460}{560}(1) = .178$$

But this cycle can be reversed to act as a heat pump. Work of amount 0.178 unit can be supplied, and 1 unit of heat will be received at the higher temperature. Thus, for the heat pump, the coefficient of performance is defined as

$$CP = \frac{\text{heat delivered}}{\text{work supplied}} = \frac{\Delta Q_R}{\sum \Delta W}$$

Although the heat pump is a refrigeration cycle, the coefficients of performance differ. Comparision shows that

$$CP_{\text{heat pump}} = CP_{\text{Refrigeration}} + 1$$

It is unfortunate that a heat pump can have two different coefficients of performance.

The reversed heat engine cycle is called a refrigera-

tor, when the evaporator is used for cooling puposes, as shown in Fig. 9a; the same cycle is called a heat pump.

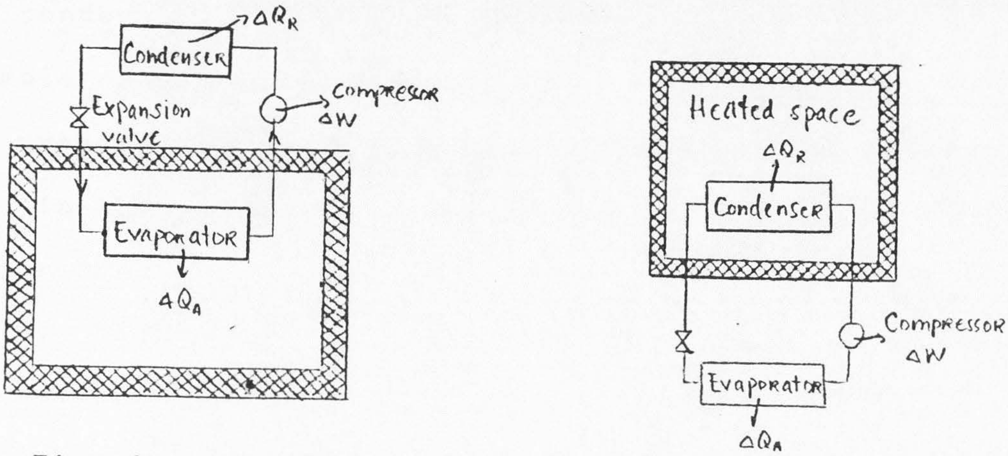

Fig. 9 - The vapor cycle as a refrigeration cycle and as a heat pump cycle. a) Refrigeratiing; b) Heating.

When the condenser is used for heating puposes, as shown in Fig. 9b. There is a combined system that serves as a heat pump in the winter and a cooling system in the summer. Here the system consists of a heat exchanger A, a condenser B, an expansion valve C, an evaporator D, and a compressor. The refrigerant is circulated through compressor, condenser, expansion valve, and evaporator. In the heating cycle the cooling water from the condenser passes through the heat exchanger alone and warms the supply air, which is also humidified by the spray humidifier. The evaporator is supplied with water from a deep well, and therefore the temperature of this water is higher than that of the outside winter air. This high temperature water increases the coefficient of performance. In the cooling cycle the well water is pumped through the evaporator and cooled to a low temperature; it

then enters the heat exchanger, which is now a cooling section. The water leaving the heat exchanger passes through the condenser before returning to the ground. This is an example of a water-to-water design; water is used to heat the evaporator and also to cool the condenser.

In the process just decribed, water passed over the evaporator of the heat pump and so transferred heat to the cycle. The temperature of the water is frequently above the temperature of the atmosphere because the temperature of the earth does not markedly change with changes in climatic conditions. Where well water is not available, or where withdrawal of water with consequent lowering of the water table is to be avoided, a heat exchanger can be buried in the earth. The heat exchanger can be a vertical U tube running several hundred feet below the surface of the earth. A small quantity of liquid can be circulated through the heat exchanger and the evaporator, the water being heated in the heat exchanger by the relatively warm earth and cooled in the evaporator by the cold refrigerant.

In the winter the surface temperature of the earth decreases in pace with the air temperature as the winter progresses, but the temperature below the surface will lag the air temperature because of the heat capacity and thermal resistance of the earth. Thus, the lowest temperature reached at a depth of 20 ft may occur more than a month after the lowest air temperature of the winter has been experienced. Under strong sunlight, the ground surface temperature may be

much higher than the air temperature because of the absorption of radiant energy. Thus, heat exchangers buried at shallow depths are better.