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Industrial Refrigeration Energy Conservation -- Application Data

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INDUSTRIAL REFRIGERATION ENERGY CONSERVATION-

APPLICATION DATA

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ABSTRACT

Refrigeration system efficiency can be improved by reducing the system head, minimizing flows, and increasing component efficiences. This paper discusses application data used for conserving energy and includes actual application.

INTRODUCTION

There are many methods of reducing energy in refrigeration systems which are essentially attempts to reduce compressor and pumping power. Power is related to head, flow, and efficiency which can be optimized by considering individual equipment as well as the overall system design.

Design considerations include the type of equipment, heat transfer surface, fluid temperatures and pressures, thermodynamic cycle, tube fouling, part load requirements, and selection of a secondary refrigerant (brine). This paper discusses each of these and provides application data derived from actual system design and operating experience.

Reduced Head

System head can be reduced by continuously operating cooling tower fans to maintain minimum allowable cooling water supply temperatures, increasing chilled liquid supply temperatures, providing additional heat transfer surface in the condenser and evaporator, and using evaporative condensers. The energy savings depends on evaporator temperature and is about 1 - 1.5% HP/^{o}F for water chilling systems. Figure No. 1 shows the power savings for each ^{o}F reduction in head and is based on manufacturer's performance data. In general, high efficiency refrigerants such as R-11 tend to give lower power savings than low efficiency refrigerants such as R-114 (high HP/Ton) for equal head reductions.

Figure No. 2 is a part load curve for a 2000 ton centrifugal water chilling system. The effect of reducing cooling water supply temperature from 85^{o}F to 65^{o}F is a power savings of 22% or 1.1% HP/^{o}F. This system includes intercooling and additional tube surface which tend to reduce the power savings for lower head operation.

Reduced Flow

Increasing the chilled liquid temperature differential reduces both pump and compressor power. Increasing the cooling water temperature differential reduces pump power but also increases compresser power and is not usually economical. Since chilled water temperature differential(ΔT_e) and flow (GPM) are related by the expression Gpm/Ton = 24/ΔT_e an increase in ΔT_e will reduce the required flow for a given capacity reducing pumping energy. For example, a large central chilled water system uses a 12^{o}F ΔT_e which saves 800 gpm and about 30 HP at full load. Compressor input energy is also reduced by 1.1% HP/^{o}F or about 20 HP.

Condenser cooling water flow (GPM) and temperature differential (ΔT_c) are related by the expression: Gpm/Ton = 30/ΔT_c. If ΔT_c is increased from 10^{o}F to 15^{o}F the pumping flow decreases by 33%. However, this increases compressor HP which reduces or even cancels the pump energy savings.

Cooling water systems with low pumping heads favor a low ΔT_C whereas high head pumping systems usually justify an increased ΔT_C. For example, Figure No. 3 compares compressor plus pump HP for 55 ft and 220 ft pumping heads. The standard 10^{o}F ΔT_C is optimum for 55 ft while it is more economical to use a 16^{o}F T_C for a 220 ft head. Chiller efficiency is also a factor and for this example an 18% increase in efficiency increases the optimum ΔT_C to 12^{o}F and 18^{o}F.

Intercooling

An intercooling cycle can be used with two or more stages of compression. Savings vary from 8% power for water chilling systems to over 15% power for low temperature refrigeration systems.

Figure No. 4 shows the intercooling cycle on a pressure-enthalpy diagram. Condensed liquid refrigerant (d) is throttled to an intermediate pressure in the intercooler (e) where flashed vapor returns to the compressor (h) and liquid (f) flows to the evaporator (g) where it is vaporized and returned to the compressor suction (a). Flashed intercooler vapor enters the compressor between stages (h) and cools hot compressed vapor (b) to a mixture temperature (m). The second stage compresses superheated vapor from (m) to (c) where it enters the condenser. Cooling water removes the heat of condensation producing liquid (d).

Figure No. 5 compares power savings at various evaporator temperatures with one and two stages of intercooling. For water chilling duty, 8% HP (one stage) and 10% HP (two stages) is saved although only one stage is normally used. For -30^{o}F refrigeration duty, 16% and 19% power is saved. The first stage of intercooling is most beneficial and the economics of installing a second stage must be evaluated for each application.

Because ammonia has such a high latent heat only about half of the energy savings obtained for halocarbons can be realized.

Tube Fouling

A major cause of excessive energy consumption is tube fouling. Maintaining clean heat transfer surfaces is important and can save over 10% in compressor power.

Fouling predominately occurs in condensers where dirty and corrosive cooling water from cooling towers or rivers is circulated through the tubes. Evaporators are normally operated in a closed circulating liquid system and, with proper chemical treatment, experience very little fouling.

In a refrigeration condenser, fouling increases the temperature differential between the shell side refrigerant and the tube side water resulting in poor heat transfer, high head pressure and reduced capacity. Refrigeration unit ratings are based on a fouling factor of .0005 hr. ft.^{2} ^{o}F/BTU although factory clean tube surfaces can have factors of .00025 or lower during initial startup. After a brief period of operation, scaling often forms and increases the fouling factor to .001 or higher depending on cooling water quality.

In water chilling systems the power penalty is about 1% HP for each increase of .0001 in fouling (approximately .002" scale) which makes periodic shutdowns necessary for tube cleaning.

One method of maintaining clean tube surfaces is to install an on line brush cleaning system consisting of one nylon bristle brush inserted in each condenser tube and plastic baskets installed at the end of each tube, as shown in Figure No. 6. A 4-way valve is installed in the cooling water supply and return piping which reverses the condenser cooling water flow and propells the brushes through the tubes. The brush, with bristles slightly larger than the tube diameter, cleans loose deposits form the tube wall which are carried away by the water. Energy savings of 20% HP have been obtained with dirty, corrosive cooling water although a 10% HP savings is more common.

During chiller operation the degree of fouling can be estimated by recording condensing liquid temperature (or pressure), cooling water flow and cooling water temperature in/out and using the following equation:

$$R_{f_i} = \frac{\dfrac{A \times LMTD}{W \times \Delta T} - \left[\dfrac{x}{k} + R_{f_o} + \dfrac{1}{h_o}\right]}{(A_o/A_i)} - \frac{1}{hi}$$

where:

R_{f_i} - Internal Tube Fouling Factor - $\frac{hr.ft^2 {}^oF}{Btu}$

h_i - Internal Tube Film Coefficient - $\frac{Btu}{hr.ft^2 {}^oF}$

A_o - External Tube Area - ft^2

A_i - Internal Tube Area - ft^2

W - Cooling Water Flow - lb/hr

ΔT-Cooling Water Temperature Difference-^{o}F

LMTD-Log Mean Temperature Difference - ^{o}F

x - Tube Wall Thickness - ft

k - Tube Material Thermal Conductivity $\frac{Btu \cdot Ft}{hr\ ft^2\ {}^oF}$

R_{f_o} - External Tube Fouling Factor - $\frac{hr\ ft^2 {}^oF}{Btu}$

h_o - External Tube Film Coefficient - $\frac{Btu}{hr.\ ft^2 {}^oF}$

$$\frac{1}{U_o} = \frac{A \times LMTD}{W \times \Delta T}$$

For example, a centrifugal water chilling system with 6000 gpm of cooling water (85^o/94.5^{o}F) was designed for a condensing liquid temperature of 103.8^{o}F with a tube side fouling factor of .001. If the liquid temperature increases to 110^{o}F (other data remaining constant) the calculated R_{f_i} increases to .0021.

Multiple Compressors

Multiple small capacity refrigeration units improve part load compressor efficiency. This not only saves energy at low loads but improves system flexibility and reliability.

Figure No. 7 is a typical compressor part load curve although actual performance can vary $\pm 10\%$ and even more with low temperature suction. At less than 50% capacity the HP/Ton increases rapidly resulting in wasted energy and high operating cost.

For example, if one 600 ton unit were installed and operated at 150 tons (25% capacity) there would be a 10% power penalty relative to the proportional % power/% capacity line. However, if two 300 ton compressors were installed, one could be shut down while the other unit operated at 50% capacity with no power penalty. This would save 60 HP, increase flexibility when system demand varies, and provide a back-up compressor.

Thermosiphon Cooling

Thermosiphon cooling systems provide chilled water during cold weather operation while the compressor is shut down. Considerable energy can be saved and capacities over 60% of design can be achieved.

For the past six winters a thermosiphon system has been used with two 2000 ton centrifugal chillers to supply chilled water for building cooling. Whenever the outside wet bulb temperature drops to 45°F or below, the compressors are shut down and cold cooling tower water is used to chill water. This system has operated an average of 3000 hours each winter and has saved over $45,000 per year in electrical costs.

Figure No. 8 is a schematic of this system. Cold cooling water flows through the condenser and condenses R-12 vapor. Cold liquid refrigerant then flows to the evaporator by gravity where it is boiled by warm (50° to 55°F) chilled water and returns to the condenser as vapor where the cycle is repeated. As refrigerant boils, heat is removed from the chilled water which increases the refrigerant temperature and pressure. This produces a small pressure differential between evaporator and condenser causing the refrigerant to flow.

Because of increased system demand a third 2000 ton chiller was installed which included a "free cooling" system with high flux evaporator tubes. This system approximately doubles the old thermosiphon capacity and reduces compressor power by 17%. Figure No. 9 compares the performance of Free Cooling and Thermosiphon systems.

Brine Application

Selection of the proper secondary refrigerant fluid is important for low pumping power and efficient heat transfer. A more efficient design is a direct refrigeration system which eliminates the secondary refrigerant and reduces pumping power.

The use of brine or anti-freeze solutions for low temperature refrigeration applications and ambient freeze protection usually increases pumping power and decreases capacity. Ethylene glycol is commonly used in water chilling systems and reduces capacity as shown in Figure No. 10. A 50% solution of ethylene glycol/water provides anti-freeze protection to -32°F but reduces capacity by about 28%.

In low temperature refrigeration applications, brine selection has an even greater effect on capacity. For example, an ammonia evaporator has operated for several years providing 256 tons of process cooling with a 30% solution of calcium chloride/water at -13^{o}F. Then the calcium chloride was replaced with a less corrosive solution of 50% ethylene glycol/water. This caused a sharp decrease in capacity to 102 tons. In an effort to increase capacity, several alternate brines were considered and compared in Figure No. 11 (constant gpm).

Direct Refrigeration

A more efficient method of process cooling is to eliminate the secondary refrigerant (brine) and circulate primary refrigerant directly to the process exchangers. Figure No. 12 shows a typical brine system and Figure No. 13 is a simplified direct expansion system schematic.

A direct refrigeration system (direct expansion or liquid overfeed) eliminates the refrigeration evaporator, chilled liquid pump and the chilled liquid. Warm high pressure liquid refrigerant flows from the condenser through distribution piping to the process where it is expanded (throttled) to a lower pressure and temperature in a heat exchanger. Process fluid enters the exchanger vaporizing the cold liquid refrigerant which returns through a pipe header to the compressor.

In most applications at least 10% HP can be saved, however, in one specific application 54% HP was saved. Three separate cooling circuits were used to cool a process chemical and consisted of 1) an ammonia refrigeration cycle, 2) a 30% ethylene glycol brine circuit, and 3) an R-11 thermosiphon loop. This system was replaced with an R-114 direct expansion system which improved the cycle efficiency by using two stages of intercooling, eliminating two shell and tube heat exchangers and brine circulating pumps, and increasing compressor suction temperature from 10^{o}F to 16^{o}F.

SUMMARY

Several methods of achieving energy conservation in refrigeration systems have been presented and application data have been provided for system design and operation. Because of the many variables to consider in each system, it is only practical to consider the data in this paper as general guidelines rather than specific data.

It should be emphasized that total system energy savings is not necessarily obtained by adding the savings of each method employed. A system with intercooling would have a lower % HP savings per ^{o}F head reduction compared to a simple cycle. Therefore, a total energy analysis should be made including all methods used.

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Biography

Thomas L. White is a Principal Engineering Specialist in the Energy Systems Design Section of Monsanto's Corporate Engineering Department. His duties include process design of refrigeration, heating, and compressed air systems and refrigeration consulting.

Prior to joining Monsanto in 1974 he worked for York Division of Borg Warner Corp. in design, development, and testing of centrifugal refrigeration compressors. His experience also includes the design of screw compressors, mechanical transmissions, and compressor noise reduction. He was graduated from the University of Kansas with a BSME degree in 1961 and is a member of ASHRAE and ASME and is a Registered Professional Engineer in Pennsylvania.

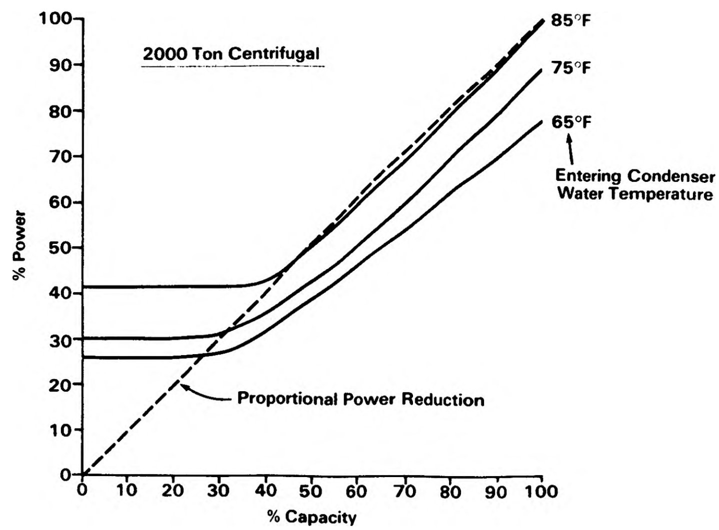

Figure No. 2—Centrifugal Part Load

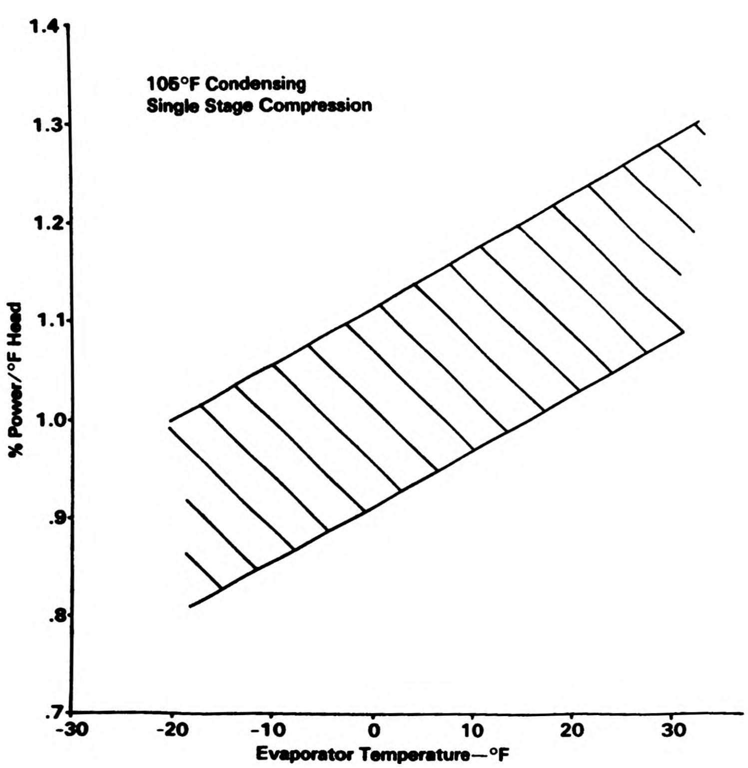

Figure No. 1—Reduced Head Power Savings

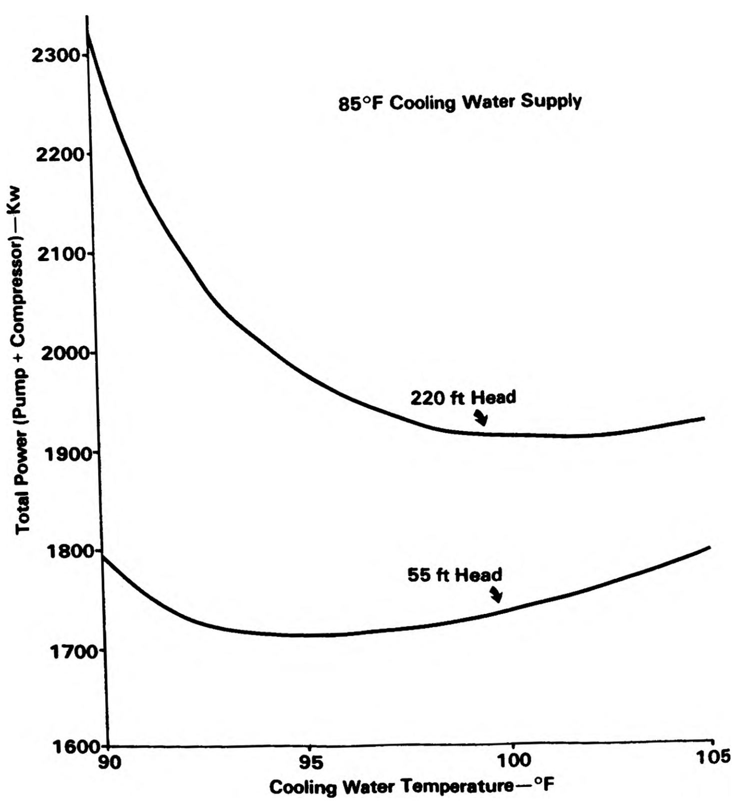

Figure No. 3—Optimum Condenser Water ΔT—°F

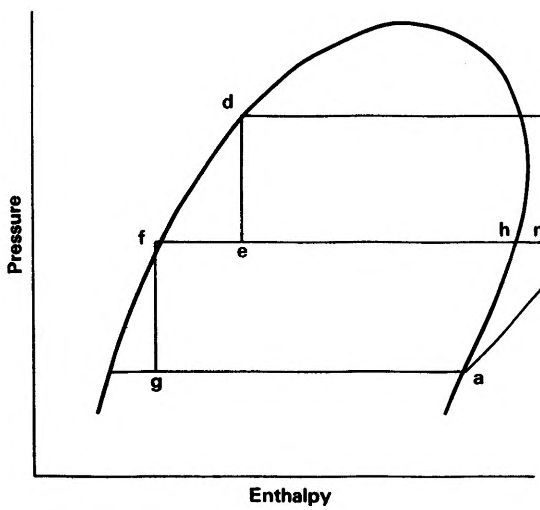

Figure No. 4—Intercooling Cycle

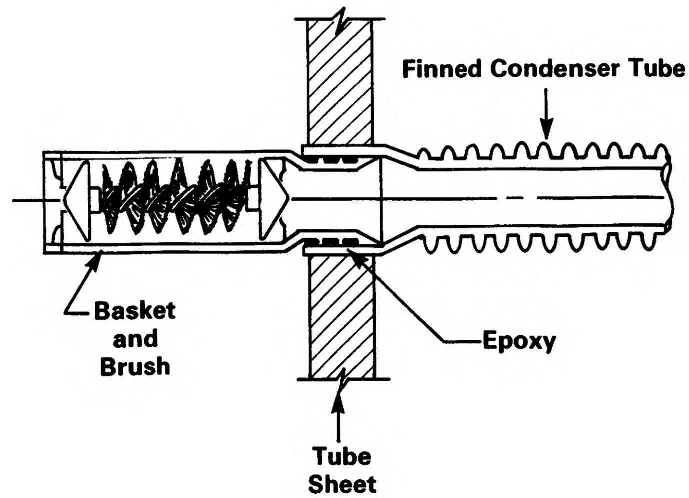

Figure No. 6—Brush Cleaning System

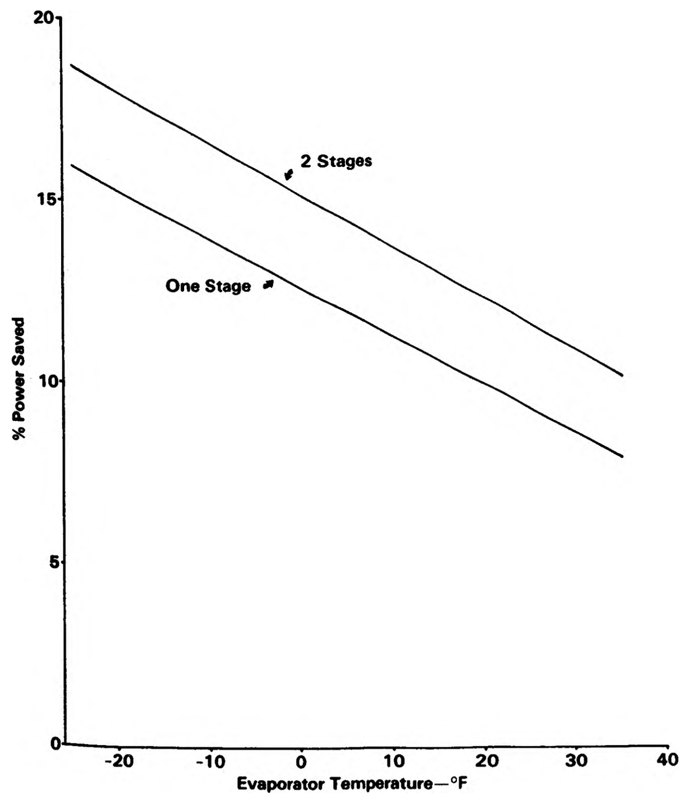

Figure No. 5-Intercooler Power Savings

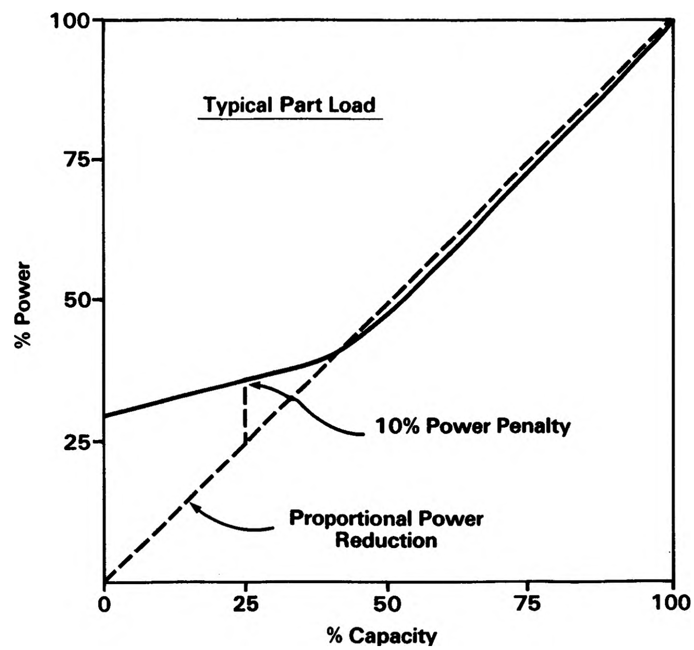

Figure No. 7—Part Load

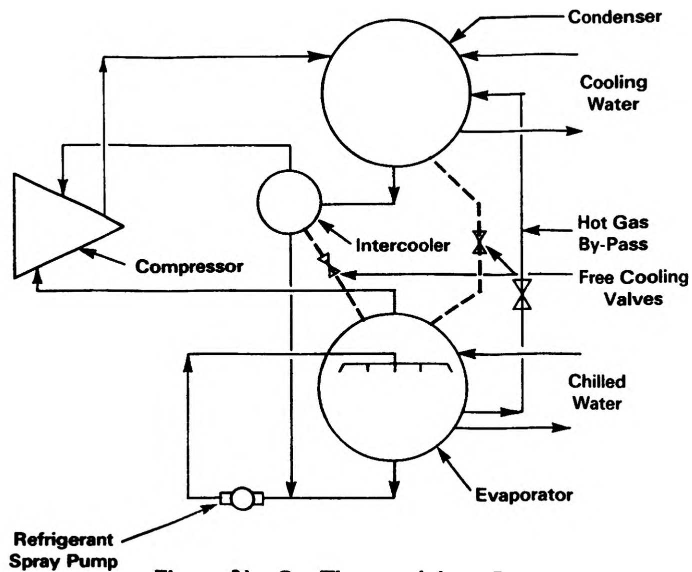

Figure No. 8—Thermosiphon System

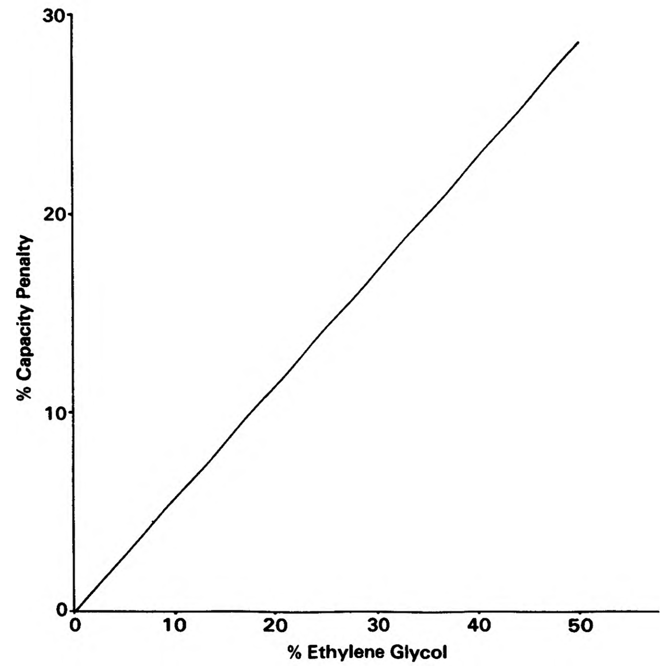

Figure No. 10—Ethylene Glycol Penalty

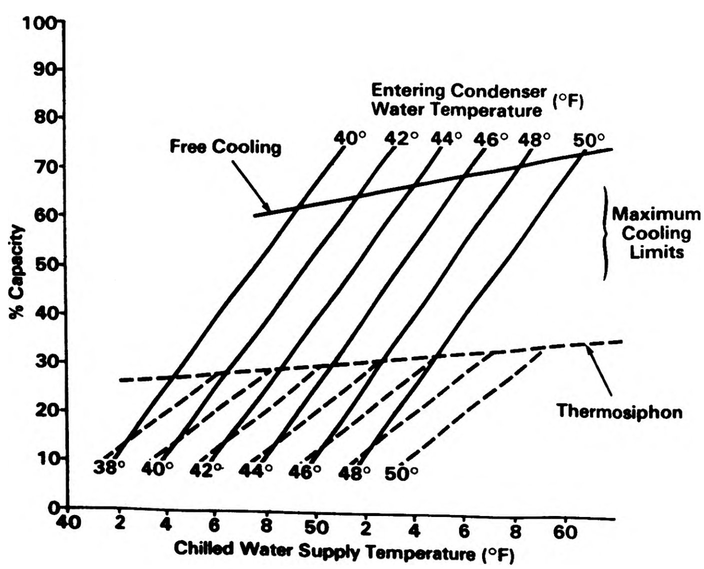

Figure No. 9—Thermosiphon Performance

Brine	h_i	Uo	ΔP (psi)	Tons
$CaCl_2$ (30%)	189	90	36	256
Ethylene Glycol (50%)	51	36	33	102
Diethylbenzene	207	94	15	265
Methanol	232	107	15	302
Methylene Chloride	414	141	18	398
Trichlorethylene	332	128	20	361
R-11	283	118	20	334

Figure No. 11—Brine Capacity

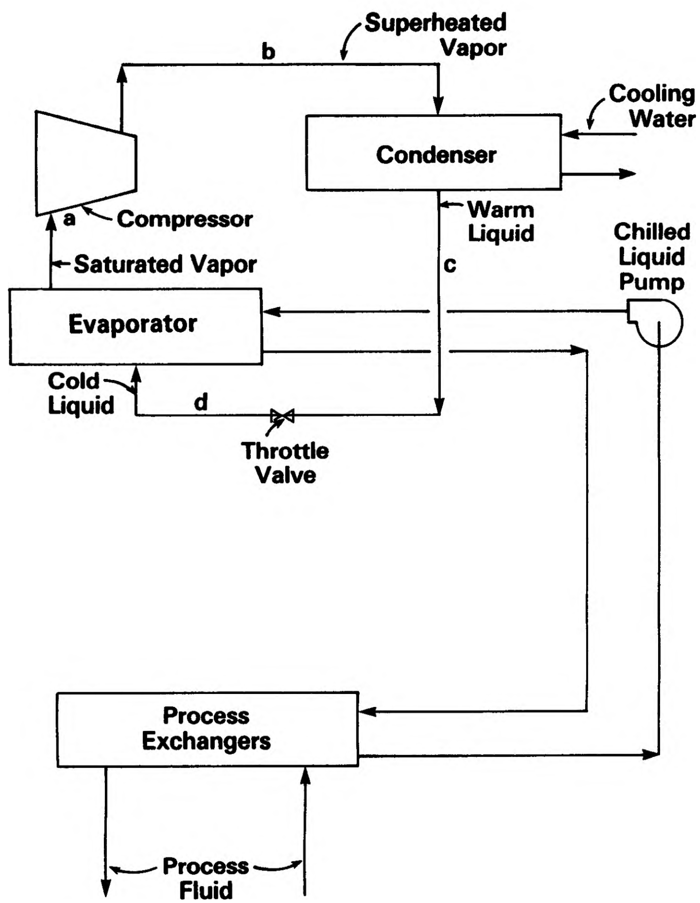

Figure No. 12—Indirect System

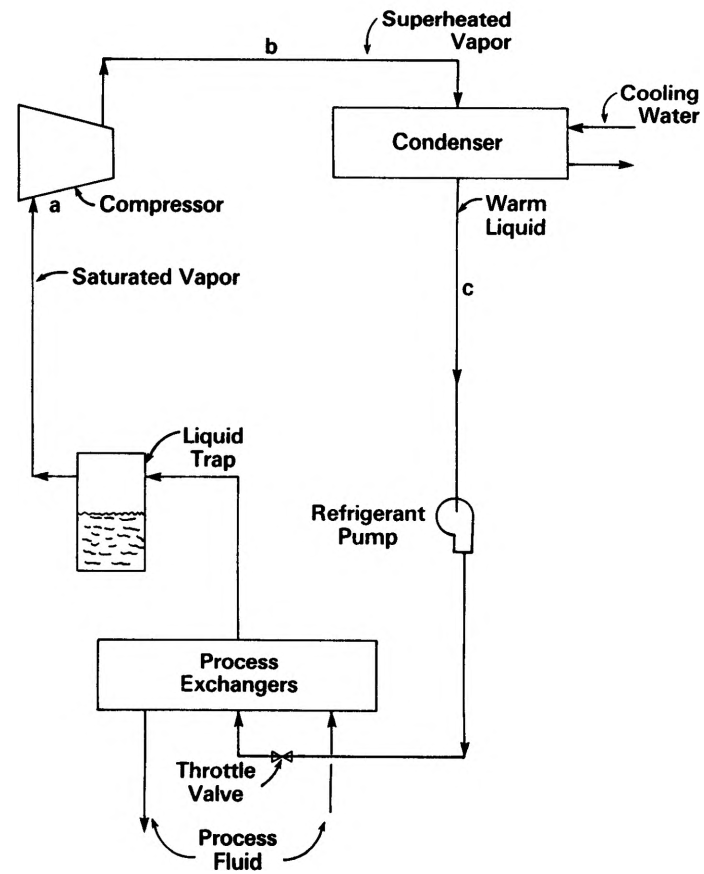

Figure No. 13—Direct Expansion System