For those who prefer to deal with isentropic analysis, an efficiency can be defined as the ratio of the isentropic energy rise to the actual energy rise between suction conditions and the discharge (or condensing) pressure. If the suction and discharge temperatures and pressures are known, it is then possible to determine the isentropic work and efficiency as follows:

$$H_s' = h_d' - h_s$$
$$\eta_s = H_s'/H_s \qquad (17)$$

*where*

$H_s'$ = isentropic energy rise, Btu per pound.
$h_d'$ = isentropic enthalpy at discharge pressure and suction entropy, Btu per pound.

Fig. 18 shows a typical single-stage characteristic with polytropic head, $H$, plotted against volume flow $Q$, with lines of constant efficiency, all at various compressor speeds.

A direct plot of head vs inlet volume flow can only be drawn for a given refrigerant at a fixed suction condition. A more universal plot, usable for the common halocarbon refrigerants under any conditions, can be developed by relating flow, head, and impeller tip speed to the inlet sonic velocity $a_o$ of the gas being handled. The shape of the performance plot related to inlet sonic velocity is similar to that shown in Fig. 18, with the parameters being: $Q/a_o$, $H/a_o^2$, and $U_2/a_o$.

For optimum performance, the selection should be made in the high efficiency region of the compressor. Maximum stability for part load operation will be obtained when the selection can be made as far as possible from the surge line. For this reason, a selection as shown at point $A$ on Fig. 18 is preferable to that at point $B$, although either flow will give the same efficiency.

Compressor selections for two-, three-, or four-stage machines are performed in a manner similar to that outlined above. The actual inlet conditions to each stage are calculated in these cases, based on performance of the previous stage. For economizer cycles, such as shown in Fig. 10, it is necessary to calculate weight flows and inlet conditions separately for each stage by mixing the appropriate flows.

The input gas horsepower for each stage is then calculated by:

$$P_g = \frac{W_f H_s}{33{,}000} \qquad (18)$$

*where*

$P_g$ = input gas horsepower.
$W_f$ = weight flow, pounds per minute.

The total heat rejection to the condenser is therefore:

$$q_c = q + 0.212\, P_g \qquad (19)$$

*where*

$q_c$ = condenser load, tons.
$q$ = evaporator (cooler), load, tons.

It may be necessary to perform the calculations for more than one refrigerant, in order to select one with the most suitable characteristics. After a refrigerant has been selected by the approximate methods outlined, it is possible to design or adapt the heat exchangers, using the information given in Chapters 42 and 43. In standard water chillers using cooling towers, both evaporators and condensers are usually designed for approaches from 6 to 10 F deg, with a nominal fouling factor of 0.0005. Actual selection of the design approach is governed by the requirements of the particular application and will depend on the relative importance of first cost and operating cost, and also upon the equipment available.

As noted earlier in this chapter, one of the more effective and commonly used forms of capacity control to meet the normal system requirement is the variable inlet guide vane. To obtain full effectiveness from such guide vanes, they should be designed to operate efficiently over a wide range of angles. If they are mounted in an axial flow duct, they are very similar to the blade row of an axial flow compressor and may be designed in accordance with axial flow compressor design methods. In order to minimize starting torque, prewhirls should close reasonably tight, but should maintain sufficient clearance to admit enough gas to prevent overheating of the compressor.

In the case of standard water chilling machines, manufacturers' selection tables have been developed, which make possible the immediate selection of a combination of heat exchangers and compressors to meet the requirements. In such tables, the net rated capacities are usually tabulated for several leaving condenser water temperatures to cover the most common range of design conditions. Interpolation may then be used for intermediate values.

For each listed condenser leaving water temperature, the net rated capacities are tabulated for various combinations of pass arrangements in both evaporator and condenser and for the most common range of leaving chilled-water temperature. Interpolation may again be used for the leaving chilled-water temperature falling within the range given. Opposite each evaporator and condenser pass arrangement are usually listed the allowable limits of water flow, both minimum and maximum, for any given selected pass arrangement. The specified water flow rates must fall within the tabulated ranges for the pass arrangements selected. Curves are also usually provided to give the water pressure drops as functions of the number of passes and the water flow. The tabulated net capacity ratings are usually based on rated motor brake horsepower and kilowatt input, unless otherwise indicated. A nominal fouling factor of 0.0005 is usually applied throughout. A number of selections using different combinations will usually be required in order to obtain the most economical selection for the specified requirement.

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CHAPTER 39

# STEAM-JET REFRIGERATION EQUIPMENT

*Basic Concepts, System Components, Steam Ejector Design Considerations, Two-Fluid Systems, Performance*

THE steam-jet refrigeration cycle has been used for over fifty years, yet it is one of the least commonly employed refrigeration cycles in use today. The lack of popularity of the steam-jet cycle stems from some inherent performance limitations despite the basic simplicity of the cycle itself. Improved equipment design is gradually broadening the application of this cycle and its future popularity will depend upon the improvements that manufacturers can incorporate into the equipment.

## BASIC CONCEPTS

The steam-jet refrigeration cycle is quite similar to more conventional refrigeration cycles with an evaporator, a compression device, a condenser, and a refrigerant as the basic system components. Instead of a mechanical compression device, the system characteristically employs a steam ejector or booster to compress the refrigerant to the condenser pressure level.

Water is used as the refrigerant and the cooling effect is produced in the steam-jet refrigeration cycle by the continuous vaporization of a part of the water in the evaporator at a low absolute pressure level. When chilling water from 55 F to 45 F, about 1 percent of the water flowing through the evaporator must be vaporized. The pressure in an evaporator operating at 45 F will be 0.30 in. of mercury absolute and, due to the high specific volume of steam at this low pressure, about 400 cfm of water vapor per ton of refrigeration must be compressed to the condenser pressure level. This should be compared to a Refrigerant 12 system in which the suction pressure will be about 40 psig and the corresponding flow rate about 4 cfm per ton of refrigeration. Steam-jet ejectors are ideally suited to handle such large volume flows, although they must typically operate over a pressure ratio (absolute condenser pressure to absolute evaporator pressure) of 6 to 8 which is rather high for efficient ejector performance. Since the motive steam energizing the ejector must be condensed along with the water vapor drawn from the evaporator, it is necessary to reject 2 to 3 times the amount of heat in the condenser that a conventional mechanical refrigeration cycle would require. Thus, a larger condenser and a relatively higher cooling water flow must be employed in steam-jet units.

In order to efficiently maintain a low absolute pressure in the condenser (for example, 2.24 in. of mercury absolute at 105 F condensing temperature) it is common to employ a small two-stage ejector package which removes air and other noncondensable gases from the condenser and pumps them up to atmospheric pressure where they are discharged.

Despite the advantages of simplicity, ruggedness of design, freedom from vibration, high reliability, low maintenance, and low cost, the steam-jet refrigeration cycle has not enjoyed wide application in the field of comfort air conditioning. This apparent paradox can be understood through knowledge of the basic characteristics of the cycle.

In certain process cooling applications, the steam-jet refrigeration unit has proven to have outstanding characteristics. In those applications where direct vaporization is used for concentration or drying of foods and chemicals, the use of heat exchangers is eliminated. This is of particular advantage in processing heat-sensitive foods or chemicals where the cooling produced by evaporation reduces the processing temperature and prevents product deterioration. The concentration of fruit juices, the freeze-drying process of preserving food, the dehydration of pharmaceutical products, such as antibiotics, the crystallization of many chemicals and foods, and the chilling of leafy vegetables are typical examples of steam-jet system cooling applications in which mechanical refrigeration is often either not suitable or not competitive.

In common practice, steam-jet refrigeration systems produce chilled water at temperatures between 35 to 70 F, which would include the ranges used for comfort air conditioning (35 to 50 F is used for this purpose). However, in process work where the processed substance (which may not be water) is directly evaporated, vaporization at higher temperatures is common.

Since water is the refrigerant in a steam-jet refrigeration unit, it cannot be operated at evaporator temperatures below 32 F due to freezing, and in comfort cooling applications evaporator temperatures above 50 F are impractical due to the inability to adequately control humidity in the conditioned space. Thus, with water as a refrigerant, the useful range of evaporator temperature is thermodynamically restricted.

Steam-jet refrigeration systems are available commercially in 30 to 500 ton capacities, although they have been built in both smaller or larger sizes.

## SYSTEM COMPONENTS

A steam-jet refrigeration system consists of an evaporator (or flash tank), a steam-jet ejector (or booster ejector), a condenser, and a two-stage ejector noncondensable pump. Auxiliary components include a chilled water circulating pump, a cooling water circulating pump, a condenser condensate pump, and necessary valves and control elements. The pumps and valves are the only system components which contain moving parts. Figs. 1 and 2 show two of the most common basic systems and their component arrangements.

### Evaporators

The evaporator is usually a large volume vessel which must provide a large water surface area for efficient evaporative cooling action. Water sprays (Fig. 1) or cascading water in sheets (Fig. 2) are two common means of maximizing water surface.

The general responsibility for this chapter is assigned to TC 5.6, Absorption Machines.