

place at or beyond the exit plane of the cone than when the expansion process is completed within the cone itself, giving rise to a normal shock at this point. Such a condition would arise at higher than design evaporator temperature and at start-up if no shortening of the cone took place. Incidentally, complete elimination of the divergent cone so that only a convergent nozzle remains will only reduce ejector performance by about 5 percent.⁷

It is imperative that the nozzle axis coincide with the main ejector axis and it is usual for the nozzle to be so mounted that its axial position relative to the inlet of the mixing section at Station X of Fig. 3 can be varied to achieve optimum performance. It can be shown theoretically (and this has been supported by test data) that the use of superheated motive steam causes a slight decrease in ejector efficiency.

While saturated steam is the most efficient motive steam condition for ejector operation, such a steam condition is difficult to control due to heat losses. Since drops of wet steam can cause serious erosion of both nozzle and main ejector parts due to the very high velocities encountered, a few degrees of superheat is often used to effectively eliminate the erosion hazard from ejectors. In actual installations, erosion is virtually nonexistent. Good thermal insulation on the steam lines and good steam trapping practices also contribute to this situation.

Ejector Suction Opening and Suction Chamber

The ejector suction opening is generally sized to give an average vapor velocity of about 250 fps. The suction chamber is a volume in which the entering water vapor is turned 90 deg and accelerated to some optimum velocity at Station X (Fig. 3). It can be shown both theoretically⁸ and experimentally that there is an optimum velocity at Station X of the vapor entering the mixing section for every ejector design condition. Typically, the vapor velocity increases as the design pressure ratio of the ejector (condenser to evaporator pressure) decreases. For steam-jet refrigeration units employed on comfort air conditioning applications the vapor velocity may be, typically 400 to 600 fps at Station X (based on evaporator state conditions).

Mixing Section

The mixing section is conical in shape. Most ejector manufacturers use a single cone angle between Station X and 1, while a few employ a double cone in series arrangement with the larger included angle cone starting at Station X and the smaller cone ending at Station 1. The length of the mixing section is often expressed in terms of throat diameters at Station 1 to 2. For steam-jet refrigeration work, the mixing section will be 6 to 8 throat diameters long, with an average value of 7. It can be shown theoretically⁹ and by test that the optimum mixing section cone angle will decrease with decreasing ejector design pressure ratio. For a design pressure ratio of one, the cone angle becomes zero and a cylindrical constant area mixing section results. For the typical steam-jet ejector on refrigeration service, the mixing section included cone angle is about 6 deg. A larger angle will result in a loss of ejector efficiency and a smaller angle will result in the ejector being unable to compress design vapor flow to the design condensing pressure.

Constant Area Section

The constant area section is the supersonic shock diffuser section (Stations 1-2). In an ideal one-dimensional analysis, the pressure rise in this section might be expected to take place across a normal shock of essentially zero axial length.

This is not the real situation. Because of a thick boundary layer and a very peaked velocity profile (as opposed to the ideal case of no boundary layer and a uniform velocity profile) the shock is not fully normal but includes complex oblique shock patterns¹⁰ as well, and in practice several throat diameters of axial length are required to complete this pressure recovery process.¹¹ During load variation, this complex and lengthy shock pattern shifts axially. For a broad and efficient ejector operating characteristic this constant area throat section is typically 3 to 5 throat diameters long (with 5 diameters a preferred length) to accommodate the shock pattern and its axial movement under load. The constant area section diameter is critical for a specific design and although the literature gives several methods for computing this dimension, none are precisely accurate.^{12,13} Only ejector manufacturers possess data with which to accurately predict the throat diameter for ejectors of all sizes and operating requirements.

Subsonic Diffuser

The subsonic diffuser is always conical in shape with an included angle range of 6 to 12 deg, although 8 to 10 deg is most common. An axial length of 4 to 12 throat diameters is found in practice with a 5 diameter length most common. A small included angle of 6 to 7 deg would be more efficient if sufficient axial length were available to decelerate the flow to the typical average leaving velocity of 250 fps at Station 3 (Fig. 3). However, ejector manufacturers have attempted to standardize ejector sizes and usually employ cone angles greater than 7 deg in order to keep within length limitations (which are often quite arbitrary).

Typical Ejector Function

To better understand how a typical ejector functions, a brief description of its operation will be given on a Mollier Diagram (see Fig. 4). The saturated motive steam enters the ejector at a total pressure (165 psia) corresponding to state i and expands to a static pressure (0.10 psia) at state A. The steam velocity will be about 4400 fps and contain 24.5 percent moisture. Saturated vapor (at 45 F evaporator temperature) enters the ejector at a total pressure (0.147 psia) corresponding to state 0 and expands to the static pressure P_2 at state B. The vapor velocity will be about 1000 fps here. The two streams start to mix at Station X in the ejector. Mixing is assumed to be completed at Station 1, and this corresponds to state 1 where the mixture velocity is about 3400 fps. Mixing

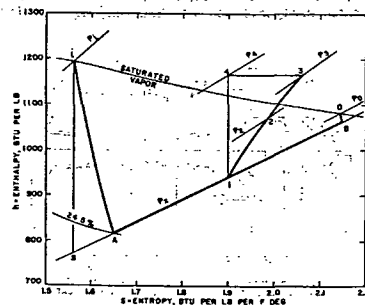


Fig. 4 Mollier Diagram for Typical Steam Ejector

Steam-Jet Refrigeration Equipment

is assumed to take place at constant static pressure. An ejector discharge pressure of 5.15 psia would be theoretically possible (in this example) if the available kinetic energy at state 1 could be transformed into potential energy at state 4. Due to losses, a pressure rise to only 1.10 psia (the condenser pressure corresponding to 105 F condensing temperature) is actually realized. In theory and in practice this diffusion process can be assumed to take place in two parts. The first is supersonic shock diffusion from state 1 to state 2 (between Stations 1 and 2) and the second is subsonic diffusion from state 2 to state 3 (between Stations 2 and 3). Since the ejector exit velocity at Station 3 is finite and not zero, a small loss in available energy from ideal must occur. This loss is too small to show on the diagram, however. Static pressure P_3 corresponds to conditions at Station 3 and is fixed by the condensing temperature (105 F).

Note that virtually the entire ejector process occurs within the wet region of the Mollier Diagram below the saturation line. Both motive steam and vapor enter the ejector at essentially saturated conditions. The motive steam often expands into a region of 20 percent moisture (24.5 percent moisture in the previous example). Because of this, the specific heat ratio for steam will be found to be 1.10 to 1.15, and not 1.3 as on the saturation line. The difference in enthalpy between state 1 and state A is the mixing loss. A measure of the efficiency of compression of an ejector is

$$\frac{(h_2 - h_1)(G_0 + G_1)}{G_2(h_2 - h_1)}$$

where

h = enthalpy, Btu per pound.

G_0 = mass flow rate of steam, pounds per hour.

G_1 = mass flow rate of vapor, pounds per hour.

The numerator of this ratio is the actual compression energy recovered, and the denominator is the theoretical energy available in the motive steam. This ratio is about 0.27 for the example given. The steam rate in the example would be 27.5 lb per (hr) (ton of refrigeration).

TWO-FLUID SYSTEMS

One possible method to theoretically improve ejector performance is the use of a two-fluid system. By utilizing one fluid (of higher molecular weight) as the motive fluid and the other (of lower molecular weight) for a refrigerant, an improvement in ejector performance is possible. During the 1930's a mercury water system¹⁴ was marketed in limited quantities for cooling household refrigerators. However, the problems and potential hazards outweighed the small improvement realized. Some more recent efforts in developing a two-fluid system can be noted occasionally in the literature¹⁵ and in patents¹⁶ granted. However, all of these systems, while having superior ejector efficiency require higher compression ratios. This increased work of compression is necessary to separate the constituents of the mixture in the condenser. Therefore, two-fluid systems seldom offer a performance advantage over a conventional single-fluid system.

PERFORMANCE

The efforts that have been made in recent years to improve the performance of steam-jet refrigeration have often gone unpublished. In this field, lack of literature on this subject is not necessarily a measure of the development work being conducted by ejector manufacturers. However, the actual performance of steam-jet refrigeration units over the past thirty years has improved about 10 percent. This assumes that

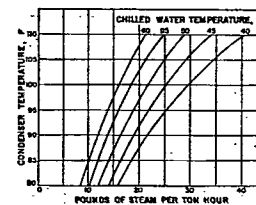


Fig. 5 Steam Demand for Typical Steam Jet Refrigeration Systems

the system design conditions were identical in making this comparison.

The steam rate required to produce a ton of refrigeration varies widely for different steam-jet systems. If the components have been properly designed, the steam rate is a function of steam pressure, condenser temperature, and evaporator temperature. Fig. 5 shows the effect of both condenser temperature and evaporator temperature at a constant steam pressure of 100 psig. In industrial practice, actual steam consumption will vary within about 5 percent of the curves shown. For example, with 100 F condensing temperature and 45 F chilled water temperature, a steam consumption of 25.5 lb per ton-hour is typical of comfort cooling applications when using 100 psig saturated motive steam. The amount of steam required to produce one ton of cooling effect decreases with decreasing condenser temperature, and with increasing chilled water temperature. The steam rate is low, therefore, when the ejector pressure ratio or temperature lift is low.

Figs. 6, 7, and 8 show the effect of steam pressure on the steam requirements at 90, 100, and 110 F condenser temperatures respectively. These curves show that it is more economical to operate at the highest steam pressure available, although pressures above 100 psig have a diminishing effect on the steam requirements. The influence of steam pressure is significantly more pronounced with chilled water temperatures below 45 F than at higher chilled water temperature. Pressures below 30 psig increase steam requirements so rapidly that even when the low-pressure steam is cheaply available it is not always economical to use this steam for jet refrigeration cooling.

Steam-jet units have a high overload capacity without much increase in condenser water temperature because as the load increases the chilled water temperature increases, and for the same motive steam flow the pounds of vapor (hence refrigeration) decreases.

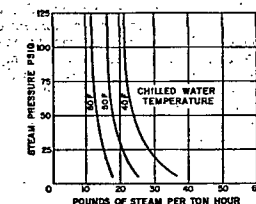


Fig. 6 Effect of Steam Pressure on Steam Demand at 90 F Condenser Temperature