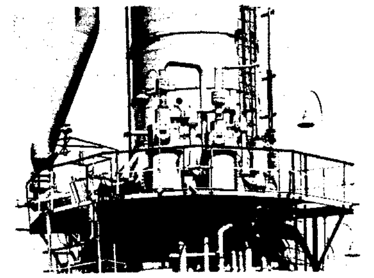
Schutte & Koerting



Selection of Air Ejectors

Part I. Basic design, operating information, and operation limitations of air ejectors.

By: C.G. Blatchley
Schutte & Koerting

An ejector is a pump which uses the jet action of one fluid to entrain and compress another fluid. The high velocity primary stream is produced by expanding a high pressure fluid through a nozzle, or nozzles, of various configurations to the low pressure of the secondary stream. The high velocity jet entrains the low pressure stream and imparts energy to it to produce a mixed stream moving at an intermediate velocity. This mixed stream is slowed down in the diffuser converting the velocity energy to pressure to discharge at an intermediate pressure.

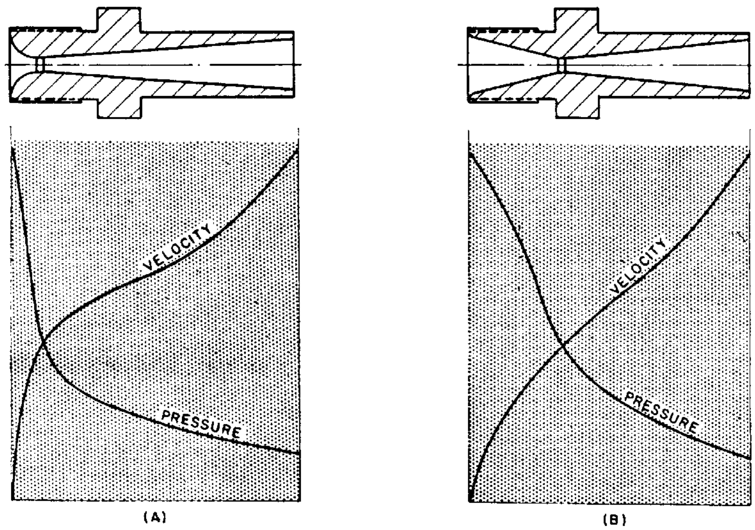


Figure 1. The steam nozzle is of the converging-diverging type.

Ejectors are operated with either gases or liquids as motive fluid entraining gases or liquids (1). Th early boiler injector utilized a motive gas (steam) which was condensable in the suction liquid (water). A syphon operates on the same principle. (Educators are liquid jet liquid pumps.) Ejectors for pumping gases are operated by liquids and gases; Water Jet Exhausters pumping air, Steam Jet Compressors pumping steam, Steam Jet Exhausters handling air. The ejectors most generally used to produce a vacuum are steam operated gas pumps—Steam Jet Ejectors.

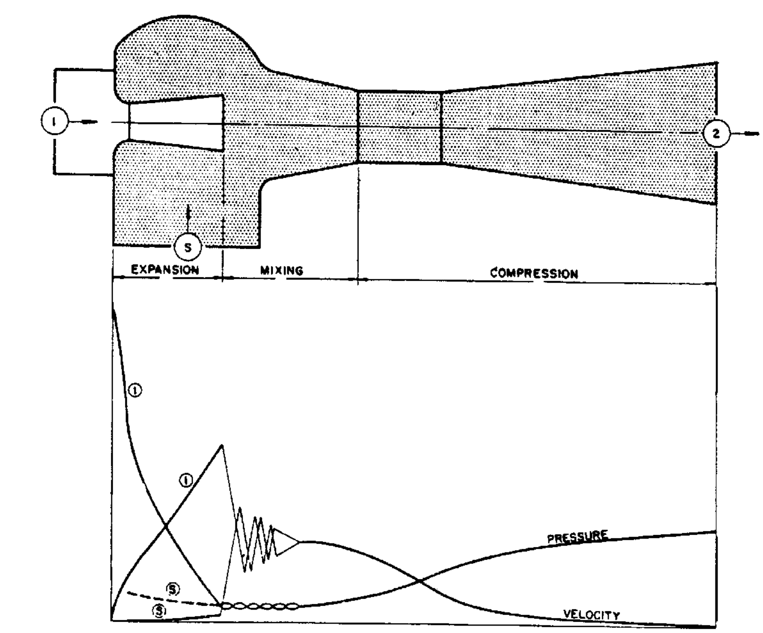


Figure 2. A plot of the ejector process.

Sub-atmospheric pressures applied to prime mover condensers improve efficiency by allowing expansion to lower temperatures. Evaporation at lowered pressures reduces boiling temperature and permits the processing of foods without destroying color and taste, and distillation and crystallization at controlled temperatures and pressures. Improved filtering rates; low temperature and freeze drying of foods and pharmaceuticals;

degassing of water, plastics and metals; and simulated high altitudes for aeronautical research have resulted. The Steam Jet Ejector makes up a major part of the total vacuum producing capacity.

BASIC DESIGN AND OPERATION

Although the jet ejector is simplicity itself the many possible arrangements and designs make the specification and selection a mystery to many users. An ejector consists of three basic parts; the nozzle designed to produce a jet of high velocity steam in a controlled pattern, the body or mixing chamber acting as a conduit for the suction fluid and positioning the nozzle and diffuser in the proper relationship, and the diffuser which converts the kinetic energy of the mixed fluids into pressure at the discharge.

The steam nozzle is of the converging-diverging type (Figure 1). Nozzles are generally of the two configurations shown, depending upon manufacturer and space requirements. Steam flow calculations are described in various codes (2, 3). The Standards also include steam flow charts for critical flow at pressures up to 620 lbs./sq. in. and temperatures to 1,000°F. A well-rounded orifice as in Figure 1-A will have a capacity coefficient of approximately 97%. Coefficients of nozzles having inlet tapers are lower (92-95%), wider entrance angles give lower coefficients.

The ejector process is partially described in Figures 2 and 3. Pipe line velocities are negligible and are shown as approaching zero. Nozzle exit velocity for the single-stage ejector shown is approximately 3,500 ft./sec. The suction fluid is accelerated slightly in the suction chamber and the two fluids are mixed at pressures approximately the suction pressure. The resultant mixture velocity is below the nozzle velocity. As the fluid flows through the diffuser the velocity decreases and the pressure increases.

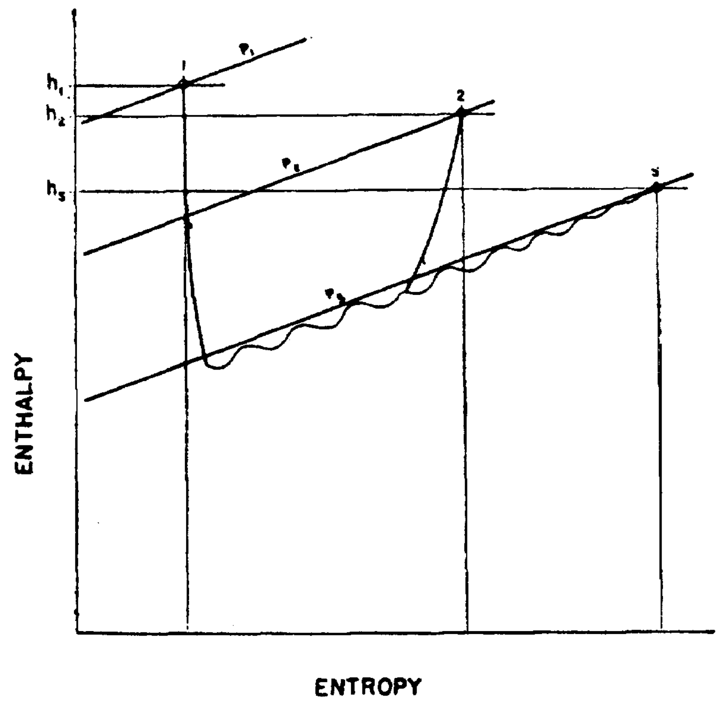


Figure 3. Enthalpy-entropy chart showing one assumption as to thermodynamics of the ejector process.

The enthalpy-entropy chart Figure 3 shows one assumption as to the thermodynamics of the ejector process. (For sake of simplicity a single fluid ejector is considered). The nozzle expansion involves losses in the range of 2 to 3% as indicated by the reheat shown. The diffuser loss is greater - in the range of 10 to 15%. The greatest loss occurs in the mixing region. The process of transmitting energy from t he high velocity fluid to the low velocity fluid has not been adequately described theoretically. It appears to be a combination direct impact, friction and induction pressure difference. Since the ejector process is essentially adiabatic the enthalpy at the discharge (2) is the sum of the enthalpies at the inlets (1) and (S). The losses appear as heat, raising the outlet temperature.

OPERATING LIMITATIONS

An ejector stage has operating limitations as to the compression attainable and will operate efficiently only up to a definite maximum ratio of compression (4). The ratio of compression is the absolute discharge pressure divided by the absolute suction pressure. The majority of steam jet vacuum pumps operate with compression ratios greater than three to one. For such compression ratios the velocity of the fluid mixture entering the diffuser must be greater than critical (sonic) and the diffuser capacity and suction pressure relationship becomes fixed for a given design. The maximum economical compression ratios vary from 6 to 1 to 10 to 1 depending upon the attainable ratio of nozzle velocity to diffuser velocity. This is a function of the available motive steam pressure and temperature and the relative suction pressure. Since the nozzle velocity must always be higher than the mixture velocity in the diffuser the compression ratio economically attainable is higher for higher nozzle velocities. Compression ratios for ejectors discharging to atmosphere do not economically exceed 6 or 7 to 1 and steam consumption begins to rise rapidly as design operating pressure decreases below 90 lbs./sq. in. Ejectors with suction pressures below 1mm. Hg. absolute may be operated with steam at pressures as low as atmospheric.

The steam rate for any ejector is a function of several interrelated variables. (1, 5). It is determined primarily by the amount of fluid to be pumped and the compression ratio through which it is pumped. The percentage of water vapor, and other gases condensable in interstage coolers, also has a major effect on the quantity of motive steam required. Secondary factors are the operating steam pressure and temperature and the cooling water temperature. Third order factors are the type of condensers used, direct contact or surface, and the selection of interstage pressures in multi-stage units.

The motive steam requirement for individual stages is a function of compression. For suction pressures from 5 mm. Hg. absolute to atmosphere the following equations will give approximate results for steam jet ejector stages handling air.

For 1.6 lb. air per lb. steam

	$P_2 = 1.45 \quad (P_S)$
1.0	$P_2 = 3.7 \quad (P_S)$
0.8	$P_2 = 5.0 \quad (P_S)$
0.5	$P_2 = 10.1 \quad (P_S)$
0.4	$P_2 = 13.85 \quad (P_S)$

P_S = suction pressure mm Hg. abs.
P_2 = discharge pressure mm Hg. abs. when motive steam is at 115 lbs./sq. in.

For the greater compression ratios than can be attained in a single ejector stage, two or more stages can be arranged to operate in series. This assembly constitutes a multi-stage ejector.

For estimating steam consumption of multi-stage ejectors, Westbrook (6) has developed an equation from empirical data. It appears to be applicable for two-stage condensing units between 15 and 100 mm Hg. absolute suction pressure. Rearranging the terms:

$$W_s = \left(\frac{P_s}{49}^{0.6} \right) \left(\frac{P_n - 0.38 P_w}{P_s - P_w} \right)^{0.52}$$

W_S = weight of fluid entrained
W_1 = weight of motive steam
P_S = suction pressure mm Hg. abs.
P_W = partial pressure of water vapor in suction fluid

This equation covers only the system air and water vapor.

Part 2. Advantages and disadvantages of various multi-stage arrangements are outlined for reference in plant design.

An ejector is inherently a constant capacity device. To obtain variation in capacity, two or more ejectors, either single or multi-stage, can be arranged to operate in parallel, each series constituting an element of a multiple element ejector. This arrangement permits the operation of the number of elements needed for the required capacity, each element capable of completely compressing a portion of the total capacity.

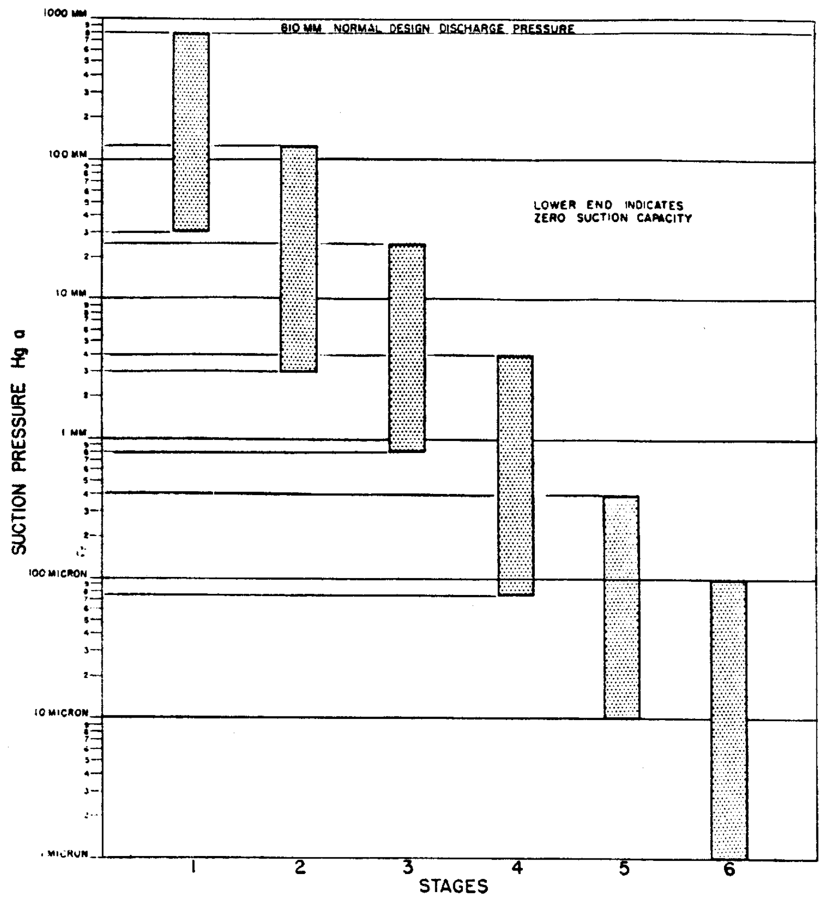


Figure 1. Range of suction pressure of various multiple stages.

For a given total compression an increased number of stages (each with lower compression ratio) will use less total motive steam. If the suction pressure is to be varied over a range of pressure; fewer, higher compression stages are more flexible and usually lower in first cost. Figure 1 shows the range of suction pressure of various multiple stages. The minimum pressure shown in each case is the shut-off, or no load point. The overlap indicates the possible use of more than one type depending on the criteria for selecting the most advantageous unit for the given application.

The most economical unit for any given use may be judged by many criteria. A unit designed for a continuous process may require steam economy or water economy depending upon the relative costs of the operating fluids. If the process is to be in operation for only a short life, low first cost may be the governing factor. For batch processes the time lost evacuating (pumping down) to the process pressure may be more costly than the use of more steam in a larger unit for faster evacuation. The choice depends upon the relation of processing time at vacuum to the total processing time; and the volume to be evacuated relative to the processing load. In large installations a separate single-stage “hogging” ejector is used to reduce evacuation time. If the process pressure varies, or if the system is used for a variety of products, a flexible unit capable of meeting all the operating points will be more economical than a unit designed for minimum utilities with a narrow operating range.

MULTI-STAGE ARRANGEMENTS

The various methods of arranging multiple-stage ejectors (Figure 2) allow the units to be designed to suit the requirements of the individual application.

A single-stage ejector may be installed in any location. When operating noise is objectionable a muffler is used on the discharge. An after-condenser will prevent discharge of steam and also muffle noise.

The simplest and lowest cost two-stage is non-condensing (2a). It is installed and used like a single-stage. Since the second stage must handle the steam from the first stage, as well as the load, the units are not economical to operate. Their use in indicated for small loads where the smallest ejector that will give trouble-free operation is larger than needed. For intermittent use such as evacuation, or where barometric installation is impractical, they give satisfactory service.

The most common ejector is the two-stage unit with direct contact barometric inter-condenser (2b). The inter-condenser reduces the load on the second stage, reducing its size and steam consumption. The evacuating capacity (for the same pumping rate) is less than the non-condensing due to the smaller second stage.

Three-stage non-condensing units are occasionally used (3a). They have advantages in installation, but the disadvantages of high steam consumption is more acute. Since the third stage must handle the steam from both preceding stages it is comparably much larger.

Three-stage condensing units may have one or two condensers. When cooling water temperatures are high tow-stages are need to compress to the point where condenser may be used (3b). This occurs on units where the load is largely non-condensable which raises the effective pressure of the condenser. With the usual 80-90°F cooling water, condenser pressures are limited to 44-50 mm Hg. absolute. In unusual cases 60°F water will allow condenser pressures as low as 38 mm Hg. absolute. With cooler water the condenser can follow the first stage. If the load is largely condensable a two-stage non-condensing is used after the intercondenser (3c). For non-condensable loads and for maximum economy two condensers are used (3d).

The discharge pressure of the first stage of a four-stage unit is too low to use a condenser. Two non-condensing stages are required before the condenser. For small units one condenser is economical (4a). Two condensers improve steam economy and are particularly indicated with large units (4b).

Five stage units are most frequently made with three non-condensing stages. Small units are made with one condenser (5a), larger units with two (5b).

Six-stage ejectors require four non-condensing stages with all normal condensing water temperatures. Although the volumetric capacity of these units is very large, the small weight capacity of non-condensables usually requires only a non-condensing two-stage.

Air ejectors

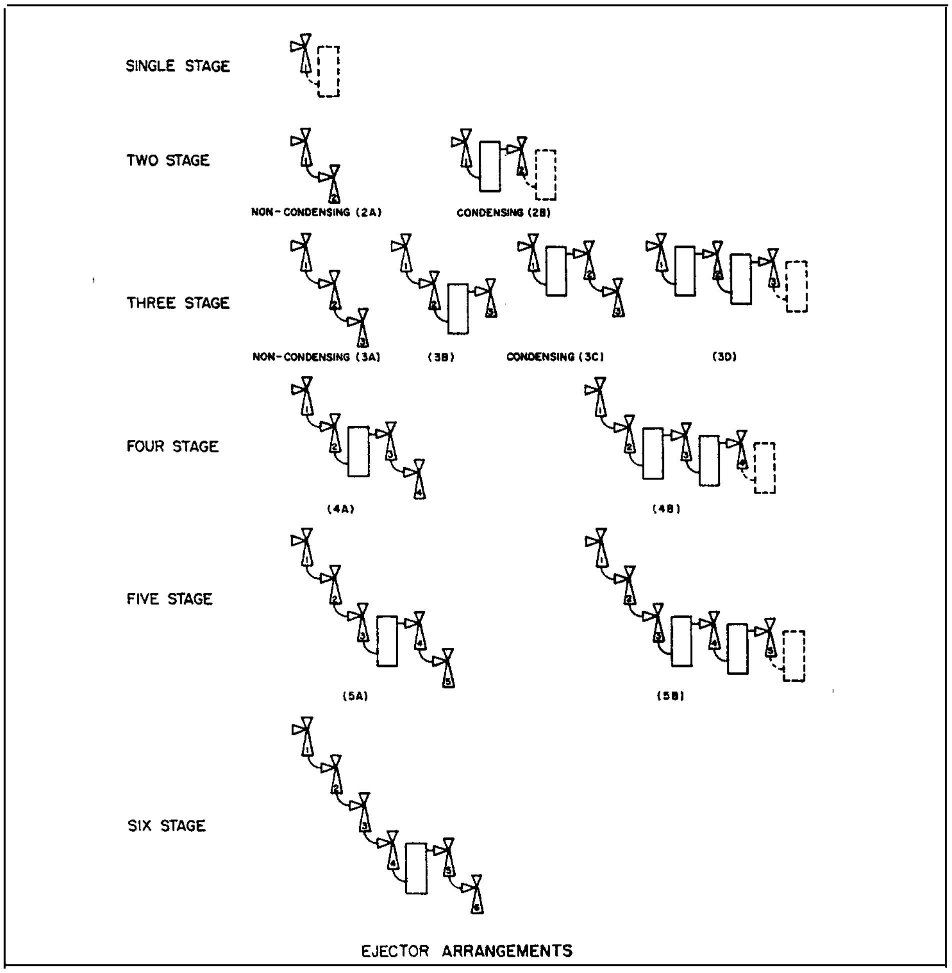


Figure 2. Multiple-stage ejectors can be arranged to suit requirements of individual application.

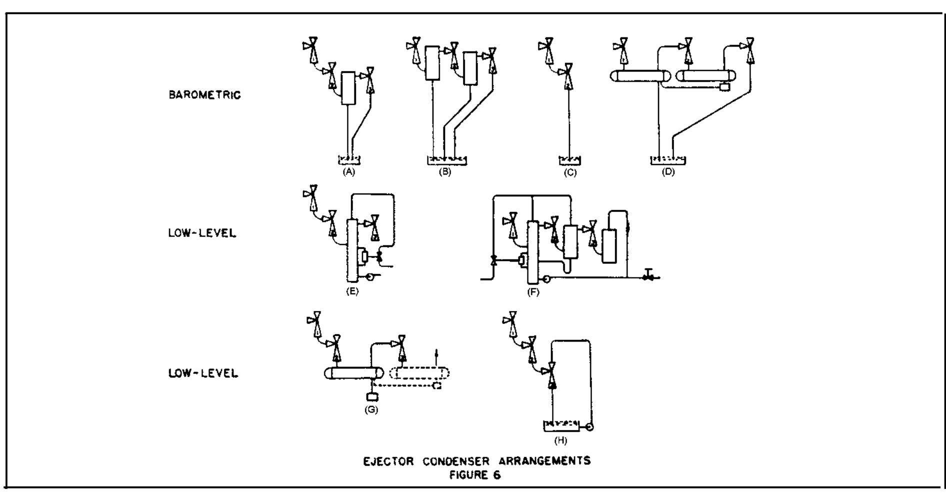


Figure 3. The type of inter-stage cooler-condenser used may be varied with the application requirements.

COOLER-CONDENSER TYPES

The type of inter-stage cooler-condenser used may be varied with the application requirements (Figure 3). The simplest and most efficient form is the barometric counter-current condenser. This type of unit operates with a minimum terminal difference (condensing temperature to outlet water temperature) and the minimum air-vapor approach (non-condensable outlet temperature to inlet water temperature) thus giving the maximum cooling with the minimum amount of water. Another advantage is the minimum pressure drop between vapor inlet and outlet and long life without reduction in effectiveness caused by fouling.

The terminal difference varies with the amount of non-condensables in the load. With 1% N.C. or less terminal difference may be as low as 3°F. Air-vapor approach may be as low as 3°F but normally will not exceed 10°F, even with high non-condensable loading. Pressure drop is less than 2.5 mm Hg.

Multiple barometric condensers may discharge into a common hotwell (6b). the discharge of the atmospheric stage may also enter the hotwell provided it is not submerged over six inches, which would impose excess back pressure. Sealing this stage prevents blow-back of atmospheric air when the unit is shut down or if steam fails. The system pressure cannot rise beyond the vapor pressure of the cooling water. When barometric installations are not possible this type of condenser may be located at low-level by use of a closed hotwell and removal pump (6e). To protect against flooding of the process a level controller is required, limiting the inlet water flow by hotwell level. Two condensers can be installed with one hotwell, removal pump and level control by use of a drop-leg to control the flow of water from the high pressure condenser to the lower pressure unit (6f). An after-condenser can be supplied from the discharge of the condensate pump. Such units can be assembled advantageously as a package. Steam consumption is similar to barometric units and pump horsepower comparable.

When non-condensable loads are small a multi-nozzle jet ejector condenser may be used as the atmospheric stage. Although the air handling capacity of such units is low, the condensing capacity is large. Installation may be barometric (6c) or low level (6h). Small low level units are particularly advantageous for laboratory use when built as a package complete with hotwell and circulating pump.

Where disposal of contaminated water is a problem, or where it is desirable to reclaim a condensed product, surface condensers are used. The overall economy of the unit will decrease due to the higher terminal difference (10-15°F) and the reduction of heat transfer rates by progressive tube fouling. The pressure drop on the vapor side is higher 6-7 mm Hg. for any reasonable shell size, and the air-vapor approach may be as high as 25°F in mixed flow units. When multiple condensers are used they may be constructed in one shell, however, maintenance of such units is nearly impossible and they are therefore rarely used except in power plants where condensate is used as cooling water. Barometric installations are not required (6d) although they greatly simplify condensate removal. When two condensers are used, individual traps or condensate pumps are required, unless the higher pressure condenser is vacuum trapped or has a drop-leg to the low pressure condenser (6g).

The growing shortage of water is increasing the interest in air-cooled condensers. The attainable condensing temperatures are high for economical use with units of more than two stages. Evaporative condensers (the air side is wetted by just enough water to maintain the wet bulb temperature) are used to obtain the minimum condensing temperature possible. It is possible to package such units with inter- and after-condensers for compact installation.

Steam jet vacuum pumps may be effectively and economically applied to many applications. The selection of the proper type depends on many variables. In order to permit the supplier to make proper recommendations all of the controlling variables should be known. An adequate specification (Figure 4) will define the load conditions as completely as possible and state available utilities. The range of suction pressures involved, the evacuation rate if required, the operating time at design pressure and the amortized cost of steam and water allows the supplier to evaluate the net cost to the user to determine the most economical unit.

LOAD

Design for Steady Condition

1. Suction capacity dry air . . . . PPH
   Non-condensables—mol. wt. . . . . PPH
   Condensables—mol. wt. . . . . PPH
2. Suction temperature . . . . F.
3. Vapor pressure—condensables . . . . . mm Hg. a. at . . . . F
   . . . . . mm Hg. a. at . . . . F
   Latent heat—condensables . . . . Btu/lb. at . . . . F
   Solubility—condensables in water . . . . % at . . . . F
   Solubility—water in condensables . . . . % at . . . . F
4. Suction pressure . . . . mm Hg. a.

Design for Evacuation

1. System volume . . . . cu. ft.
   System temperature . . . . F.
2. Mol. wt. gases . . . .
3. Evacuation time . . . . minutes
4. Air leakage rate . . . . PPH

UTILITIES

1. Minimum steam pressure . . . . . . psi . . . . . . F.T.T.
2. Maximum condensing water temperature . . . . . F. at . . . . psi

DISCHARGE

1. Pressure . . . . mm Hg. a.
2. Barometric . . . . mm Hg. a.
   or elevation at installation . . . . ft. above sea level.

STABLE OPERATING RANGE

1. Unit to be stable from shut-off to 1 . . . . % of design load or
2. Unit to be stable from shut-off to . . . . . . mm Hg. a.

CONSTRUCTION

1. Ejectors
   Body . . . .
   Nozzle . . . .
   Diffuser . . . .
   Steam Head . . . .
2. Condensers (Dircet Contact)
   Body . . . .
   Nozzle . . . .
3. Condensers (Surface)
   Tubes . . . .
   Tube Sheets . . . .
   Shell . . . .
   Channels . . . .

Installation vertical—horizontal
Fouling factor—shell side 0.00 . . . .
tube side 0.00 . . . .

EVALUATION CRITERIA

1. Steam—An amount of $ . . . . per PPH steam in excess of lowest steam consumption quoted will be added to bid price for evaluation purposes.
2. Water—An amount of $ . . . . per GPM water in excess of lowest water rate quoted will be added to bid price for evaluation purposes.

APPLICATIONS

1. Continuous operation . . . . hours per week.
2. Batch operation . . . . hours on, ... hours off.

Figure 4. Adequate specifications will define load conditions and state available utilities.

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