Characteristics of the Steam-Jet Vacuum Pump

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This paper is a discussion and presentation of the theoretical and practical limitations, attainable efficiencies, general performance characteristics, and present-day regions of operation of single and various multi-stage steam-jet vacuum pumps working at pressures from atmosphere to 10^{-3} torr.

Rapid or high-capacity vacuum pumping to 10^{-3} torr of contaminating gases and vapors is generally most economically performed by steam-jet vacuum pumps. Many applications involving clean and non-contaminating gases and vapors can also be performed economically by steam jets.

The steam-jet vacuum pump has no rotating or reciprocating parts, no lubrication or oil problems, nor extremely close tolerances. It is mechanically the simplest of all of the present-day types of vacuum pumps and compressors. However, the thermodynamics and fluid-dynamics phenomena producing the pumping action within the simple unobstructed straight-through flow structure is as complex and technically sophisticated as those encountered in the propulsion of some of our present-day aircraft and missiles.

Operation

The steam-jet pump is basically comprised of an expanding motive steam nozzle producing high supersonic velocities, a combining chamber for bringing the suction fluid into contiguous or mating contact with the high-velocity motive steam, and a diffuser for compressing both the suction and motive fluids up to the discharge pressure as illustrated in Fig. 1. The expansion through the nozzle to the design suction pressure is not isentropic because of friction and thermal losses in the nozzle. The expansion ratio is extremely high when compared to other types of work-producing

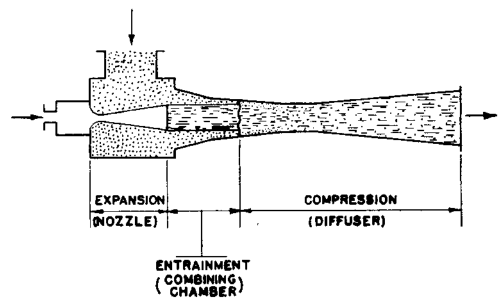

Fig. 1 Fundamental steam jet-vacuum pump parts

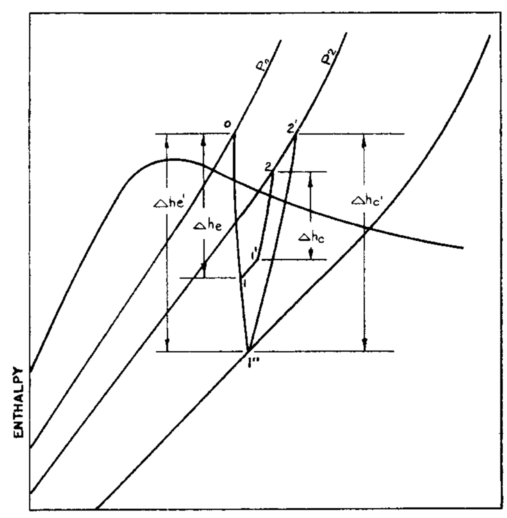

Fig. 2 Typical thermodynamic cycle for steam jet-vacuum pump operating with a suction load (0–1–2), and at zero suction load (shutoff) (0–1"–2')

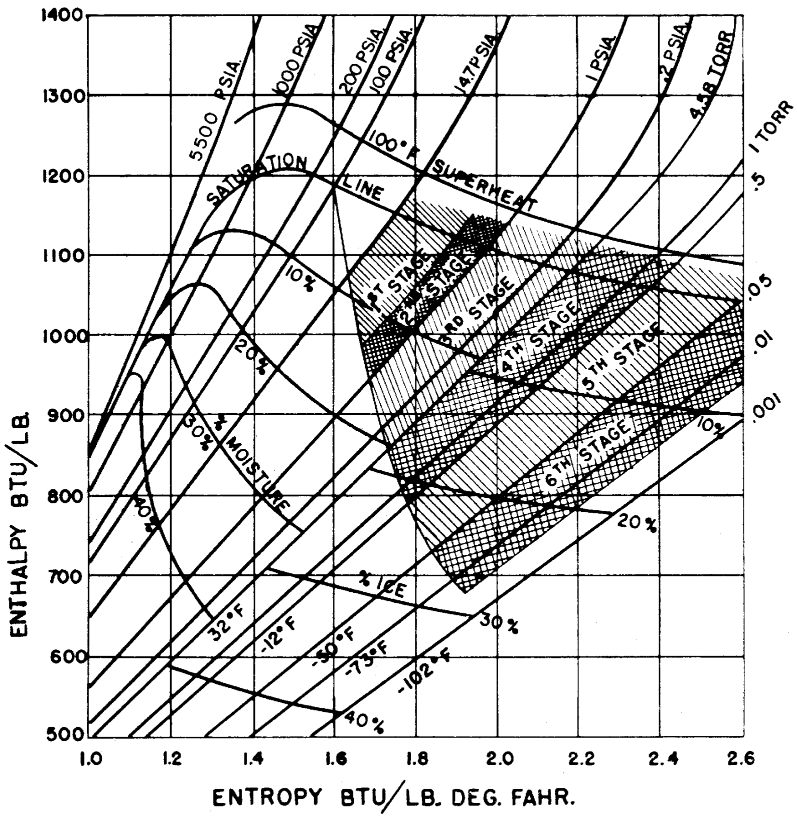

Fig. 3 Mollier diagram to 10^{-3} torr showing operating regions of steam jet-vacuum pump stages

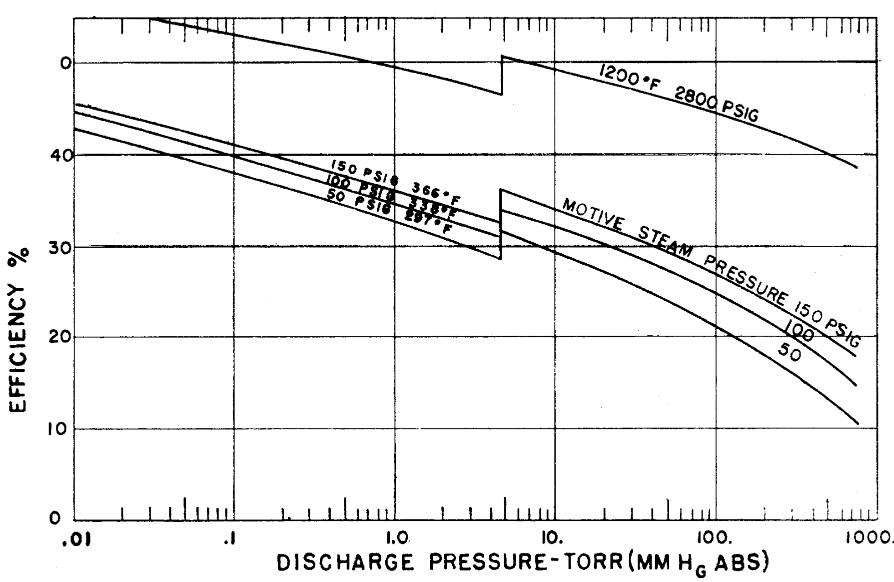

Fig. 4 Rankine cycle efficiency, steam jet-vacuum pump single stage

thermodynamic devices. It is in the range of 100 to 1 for units operating near atmosphere up to 1,000,000 to 1 for booster stages operating near 10^{-3} torr. In the combining chamber, an entrainment and diffusion action of the molecules of both streams occurs at the mating-surface boundaries. This action causes the main body of the suction stream to be accelerated from the inlet-pipeline velocity to a value higher than sonic. At the same time, owing to the transfer of energy to the suction stream, the outer regions of the motive stream in contact with the suction fluid decrease in velocity but not lower than sonic. The location of the sonic barrier of the suction stream varies with operating conditions, and is the imaginary dividing line between the combining chamber and the diffuser. After the sonic barrier is reached, diffusion and entrainment between the two streams still continues, but the process of compressing both streams up to the discharge pressure commences. This compressing action is the process of reconverting the velocity energy into pressure, and is also not isentropic owing to various friction and thermal losses.

The enthalpy-entropy path of the motive steam passing through a single stage is illustrated in Fig. 2 by the points 0-1-1'-2. The amount of work or energy available for compressing the suction fluid is the difference between the motive fluid-velocity energy $h_0 - h_1$ and the energy required to compress the motive to compress the motive stream to the discharge pressure, $h_{2'} - h_{1''}$ in Fig. 2 is equal to the actual velocity energy in the motive stream, $h_0 - h_{1''}$. This is the shutoff pressure and zero-usction-load point of the jet pump. This shutoff point may be in the range of 1/20 to 1/50 of the absolute discharge pressure.

To achieve pumping to various other levels down to 10^{-3} torr, multistage arrangements of as many as seven jets in series are required. The approximate regions of operation of the various stage units as presently applied in this country are illustrated on an extended version of the Mollier diagram, shown in Fig. 3. The normal version of the Mollier diagram stops at 0.2 psia. The graphical extension to 10^{-3} torr was generated by using the solid-vapor saturation tables for water vapor.[1,2] The diagram reveals several interesting thermodynamic characteristics peculiar to steam jets. (a) The 4th, 5th, 6th, 7th-stage units operate in a region where the expanded motive fluid is composed of gas and solid particles. (b) The mass of gas available for pumping is continually decreasing as the suction pressure gets lower. In the region of 10^{-3} and 10^{-2} torr, assuming expansion occurred at thermodynamically stable conditions, the mass available for pumping is approximately 20 to 30 percent less than that entering the nozzle. Since the entrainment is a function of the gas mass of the motive fluid, this characteristic would have a tendency to cause the actual efficiency of the 10^{-3} and 10^{-2} torr stages to be poorer than those operating at 1 torr and above.

Efficiencies

All steam-jet pumps have long been known to have low thermal efficiencies in the range of 1 to 5 percent. The thermal efficiency expresses the actual work versus the total quantity of heat supplied. This value contains large percentages of heat or energy that are supplied but are unavailable for use in producing work. The Rankine-cycle efficiency when used in conjunction with the thermal efficiency gives a better indication of the performance of the steam-jet pump. The Rankine-cycle efficiency defines the maximum amount of energy that is available for use as work, under the ideal isentropic expansion and compression conditions. In the Rankine-cycle process, all of the heat is added at the higher constant supply pressure level and all of the heat remaining after performing work is rejected from the cycle at the lower constant discharge pressure level. This is expressed in the simplest manner by the following well-known equation:

$$E_{Rankine} = (Q_{supplied} - Q_{rejected}) / Q_{supplied}$$

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[1] J. H. Keenan and F. G. Keyes, *Thermodynamic Properties of Steam,* First Edition, John Wiley & Sons, Inc., New York, NY, April, 1957, p. 76

[2] *Standard for Steam Jet Ejectors,* Third Edition, Heat Exchange Institute, New York, NY, 1956.

The Rankine-cycle efficiencies for the steam-jet pump operating at various discharge pressures and motive pressures are shown in Fig. 4. These values represent the theoretical maximum that can be obtained or is available for use as work. It is interesting to note that when the motive steam is supplied at substantially higher temperatures, major improvements occur in the theoretical maximum efficiency. The numerous types of internal-combustion engines which are known to have higher thermal efficiencies in the range of 20 to 40 percent, are working under conditions where the heat is supplied at temperatures of 1800 to 3000 F, which are considerably higher than those shown in Fig. 4. When considering all of the cost factors involved, operating at substantially higher steam temperatures to obtain the higher efficiencies may not be as economically attractive in steam jets as it is in other types of equipment.

The ratio of the actual work obtained to the theoretical maximum as determined by the Rankine cycle is a better indication of the operating efficiency of a jet pump than thermal efficiency. It can be expressed in several ways:

$$E_{jet} = E_{thermal} / E_{rankine} = \text{actual work} / (Q_{supplied} - Q_{rejected})$$

The general range of attainable jet efficiencies of a single-stage steam jet pumping air at various compression ratios and operating with steam rates of 400 lb. per hr and above are shown in Fig, 5. One point along a given ejector pumping curve will have a peak efficiency that reaches the upper curve and the other useful pumping points along the same curve will have efficiencies that fall in the cross-sectional region between the upper and lower curves. Owing to the mixing action of the motive and suction streams it is extremely difficult to determine the actual or true work of compression performed on the suction fluid. For the purposes of the graph, shown in Fig. 5, the compression work performed on a suction fluid of air was assumed to occur at adiabatic conditions. Although the values shown are not thermodynamically precise, the information plotted, clearly shows the major decrease in efficiency occurring at the higher compression ratios above 10:1, and also the maximum efficiency occurring in the range of 4:1 compression ratio.

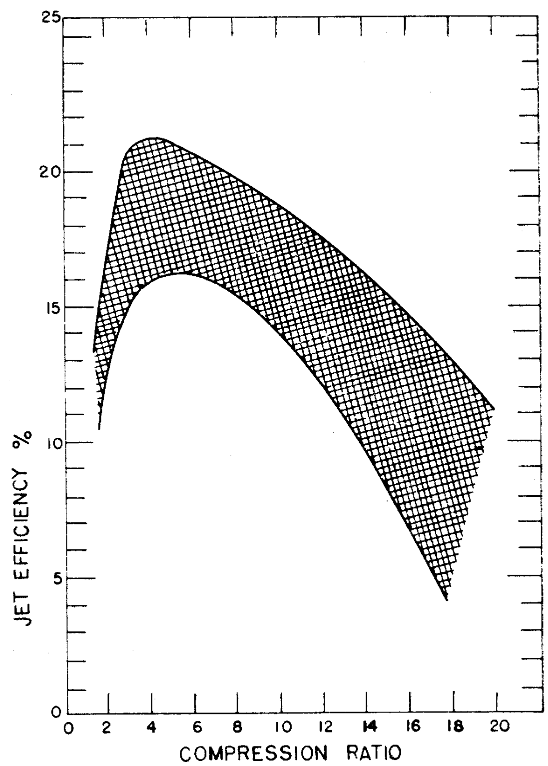

Fig. 5 Range of attainable jet efficiencies for steam-jet pumping air

Stage Pumping Characteristics

The pumping characteristic for any given size unit can be varied considerably by varying the position of the expanding nozzle with respect to the diffuser and also by using different motive steam rates. A typical pumping curve for a given nozzle position and a given steam rate is illustrated by the corresponding curves labeled 5 in Fig. 6. The upper curve 5 in Fig. 6 is the discharge pressure to which the jet will compress for a corresponding air capacity-suction pressure point on the lower curve. Corresponding families of curves 1 to 6 show the various other pumping curves obtainable by putting the nozzle in different positions with respect to the diffuser. Curve 1 is closest to the diffuser and 6 is farthest. In general, the curves reveal that as the nozzle position is moved farther away from the diffuser, the air-handling capacity at any given suction pressure increases while the discharge pressure and compression ratio decrease. When the nozzle position is moved too far into the diffuser, decreases in air-handling capacity occur without any increase in discharge pressure or compression ratio as illustrated on the corresponding curves 1 and 2 in Fig. 6. At the other extreme, when the nozzle is too far from the diffuser, the jet becomes unstable at pressures near shutoff, as indicated by curve 6 in Fig. 6. For any one set of conditions there would be only one theoretically best nozzle position. The family of curves shown produces an envelope of attainable performance for all nozzle positions at a given steam rate.

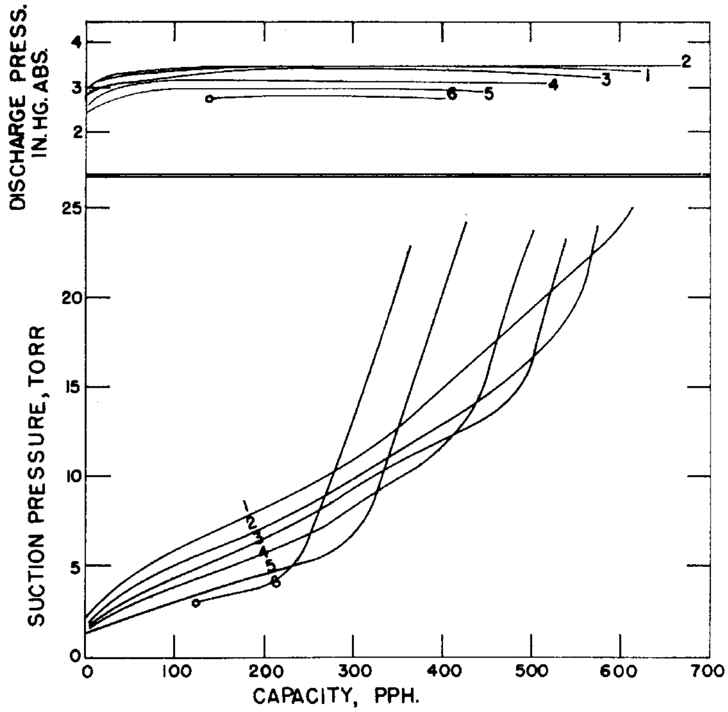

Fig. 6 Typical effect of nozzle location on stage performance

By varying the steam rate, the individual performance curve can also be varied. At another steam rate, changing the nozzle position will produce another family of pumping performances similar to but not the same as that shown in Fig. 6. The effect of changing the steam rate upon the envelope of the family of nozzle-position curves is shown in Fig.7. In general, it has been observed that increasing the steam rate at any given nozzle position increases compression ratio and also increases the shutoff pressure. As illustrated in Fig. 7, depending upon the suction pressure, the air-handling capacity may decrease or increase. Near the shutoff point the air handling decreases as the steam rate is increased, whereas at higher suction pressures, the air-handling capacity increases as the steam rate increases.

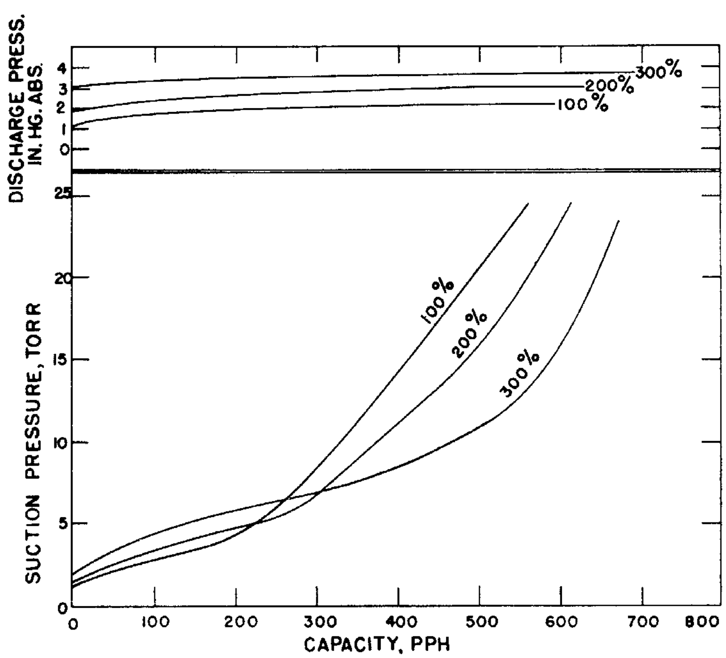

Fig. 7 Typical effect of steam rates on stage performance

The corresponding jet efficiency curve, Fig. 8, for the various nozzle-position curves, shown in Fig. 6, illustrate that maximum efficiency is achieved at only one spot, near the knee in the pumping curve. In addition, it also reveals that the jet efficiency is best at some nozzle position midway between the extremities of operation. The information in Figs. 6, 7, and 8, point out that although the air-capacity, suction and discharge-pressure characteristics can be obtained by numerous combinations of nozzle position and steam rate, there is going to be only one combination that will produce the desired characteristic for the minimum amount of steam. In any given size unit more air-handling capacity can usually be obtained at the expense of greater steam consumption. This characteristic is similar to that of many other types of pumps whereby the first cost is lowered by using a smaller size at the expense of increasing the operating cost.

Some typical capacities for single-stage units handling air can be determined approximately for suction and discharge pressures from 5 torr to atmospheric by using the following equations developed by Blatchley[3].

@ 1.6 lb air per lb steam $P_2 = 1.45\ (P_s)^{0.955}$

@ 1.0 lb air per lb steam $P_2 = 3.7\ (P_s)^{0.897}$

@ 0.8 lb air per lb steam $P_2 = 5.0\ (P_s)^{0.870}$

@ 0.5 lb air per lb steam $P_2 = 10.1\ (P_s)^{0.803}$

@ 0.4 lb air per lb steam $P_2 = 13.85\ (P_s)^{0.771}$

P_s = suction pressure, torr
P_2 = discharge pressure, torr
Motive steam is at 100 psig

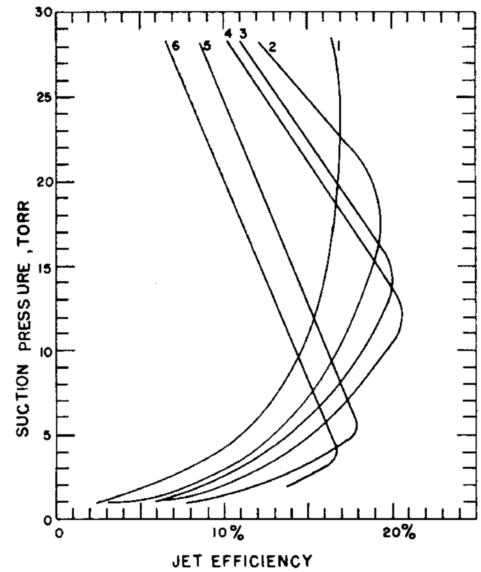

Fig. 8 Typical effect of nozzle position upon jet efficiency of one stage

Smallest Practical Size

The jet vacuum-pump size is usually specified by the diameter of the suction opening, which in turn is determined by a range of permissible suction pipeline velocities, of approximately 100 to 300 fps. As the desired suction pressure becomes lower, the weight of the suction fluid flowing decreases greatly even though the volume remains about the same. Since the entrainment phenomenon is a direct function of the weight, the weight of steam required also decreases greatly. As the suction pressure is decreased, a point is reached in the design of the motive steam nozzle, where the critical orifice becomes so small that the friction losses in the nozzle become excessive, and do not permit the jet to function satisfactorily. The minimum practical stage sizes resulting from this phenomenon for units operating below 1 torr are shown in Fig. 9. Applications requiring the use of stages in or near the minimum practical size range will have very poor jet efficiencies and high steam-to-air rates. Major improvements in the jet efficiencies and steam rates occur when the units are two to three times larger than the minimum practical sizes shown.

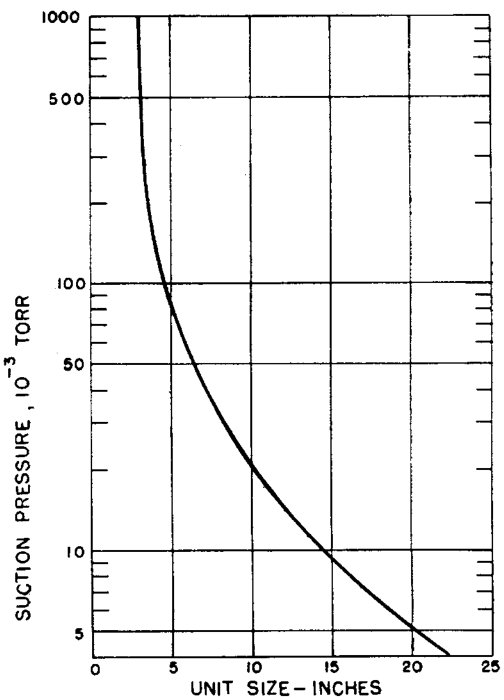

Fig. 9 Smallest practical stage size

Combine Stages

When it is desired to pump from pressures lower than 50 torr, multistages with as many as seven steam jets in series are required. The number of stages required is dependent upon the vacuum desired, and the user's interest either in low equipment cost or low operating cost. When using multistages, the supporting stages have to be large enough in capacity to handle the initial suction load plus the motive steam from the preceding stages. In a multistage arrangement, the steam consumption and size of the last two supporting jets can be reduced substantially by using condensers between the stages. The condensers decrease the suction load plus the motive steam from the preceding stages. In a multistage arrangement, the steam consumption and size of the

[3] C. G. Blatchley, "Selection of Air Ejectors," *Chemical Engineering Progress,* vol. 57, 1961, p. 115.

last two supporting jets can be reduced substantially by using condensers between the stages. The condensers decrease the suction load to the supporting jets by removing most of the condensable vapors. Usually they are interstaged between the last three jets near atmosphere, at various interstage pressures from 30 mm Hg abs. up to atmospheric. The interstaged condensers can be barometric or low-level, direct-contact units, air and/or evaporative cooled units, or shell-and-tube units. Many different arrangements of multistage jets and from one to three condensers are used.[4] The arrangement best-suited for any particular application is dependent upon the user's economic conditions, his mode of operation, and the evaluation factors used.

When combining stages the supporting units can be designed so that the system is completely stable over the entire primary-stage pumping curve as illustrated by lines B and D in Fig. 10. The suction pressure-capacity pumping characteristics, D, of the supporting stage are rarely identical to the discharge-pressure characteristics B' of the primary stage, B. As a result, at the design point a supporting stage for a completely supported and stable system will usually provide considerably more support than is necessary, as indicated by the area between lines B' and D in Fig. 10. When it is desired to obtain minimum steam consumption and lower equipment cost, the steam consumption and size of the supporting stages can be reduced by supporting the primary stage in the region of the design point only as illustrated by lines A and C. This produces a system which is stable at the design point, and has a steady and reproducible suction pressure at conditions other than the design point. Further reductions in systems steam consumption are sometimes possible by using stages which have unstable shutoff points as previously illustrated by curve 6 in Fig. 6. Such systems will be stable at the design point, but unstable and unsteady at the shutoff point.

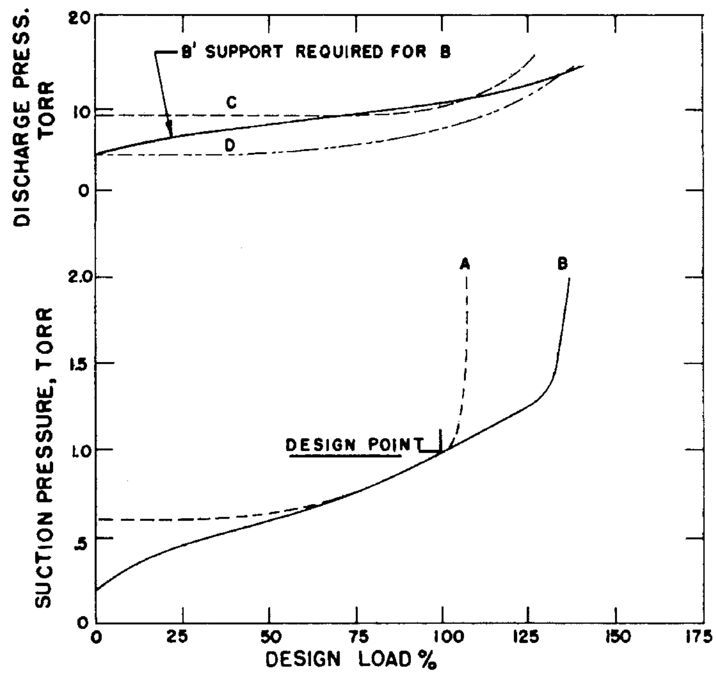

Fig. 10 Typical combination of two noncondensing stages

At the expense of steam consumption, unit size, number of stages required, and equipment cost, systems can be designed to meet special requirements. Such as (a) having two design load points at different suction pressures; (b) having pumping capacity at 150 percent or greater than the design point; (c) being capable of evacuating chambers to given pressures within specified time limitations.

All of the aforementioned systems can be optimum design systems with the individual jets sized and arranged to operate at or near the maximum efficiency points.

The general trend in this country is to use compression ratios up to approximately 10:1 for the primary and supporting non-condensing stages. As indicated in Fig. 5 higher stage efficiencies and consequently lower system steam consumption can be obtained by using stage compression ratios in range of 4:1 to 6:1. In sections of the world where water is scarce, such as in Europe, the users find it more economical to use the lower compression ratios. This requires using one or two more stages than is general practice in this country.

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[4] Ibid., pp. 144 – 145.

System Pumping Capacities

A composite picture of the typical pumping capacities and corresponding steam consumptions of various multistage condensing steam-jet-vacuum pump systems with up to seven stages in series and operating in the range from 50 to 10^{-3} torr is shown in Fig. 11. The values shown are for optimum systems that are stable in the general region of the design point and steady down to the shutoff point. The two-stage system steam consumption, W_s, in pounds per hour can be estimated for air capacities, W_a, lower than shown in Fig. 11 down to 40 lb. of air per hr by using the following equations, which are for air at 70°F, water at 85°F, and using 100 psig steam:

For P_s = 25 torr….$W_s = 7 \times W_a$

For P_s = 50 torr….$W_s = 4.5 \times W_a$

Owing to the shapes and characteristics of the individual stage-performance curves, and having to construct units to standard sizes, there are some sets of conditions of capacity and suction pressure which cannot be accomplished at the rates shown by the generalized curves and equations. There are also sets of conditions where steam-consumption rates better than shown are possible. The corresponding water required for all the multistage systems, except the two-stage arrangements, can be estimated roughly in GPM by multiplying the steam rate shown in Fig. 11 by factor of 0.15.

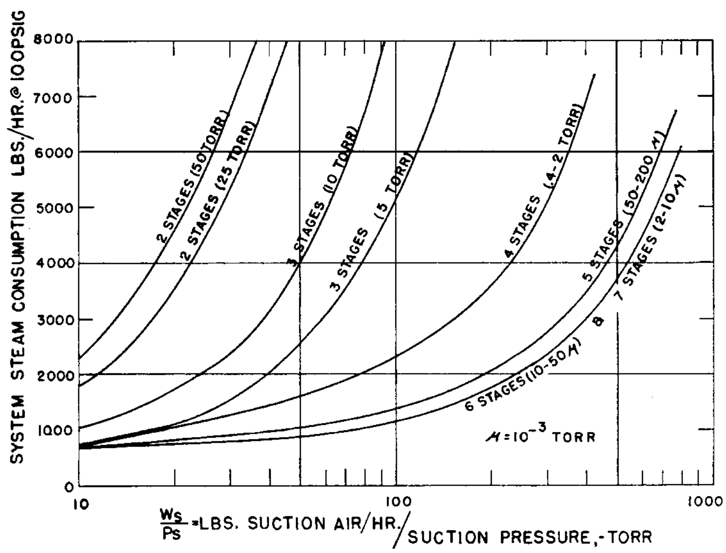

Fig. 11 Multistage, barometric condensing steam-jet vacuum pumps. Pumping and steam consumption characteristics at 100 psig steam, 85 F water, for air.

In the capacity regions where the curves are not approaching the minimum practical size nor minimum system steam consumption, the use of one additional stage could possibly reduce the system optimum steam and water consumptions by approximately 10 to 20 percent.