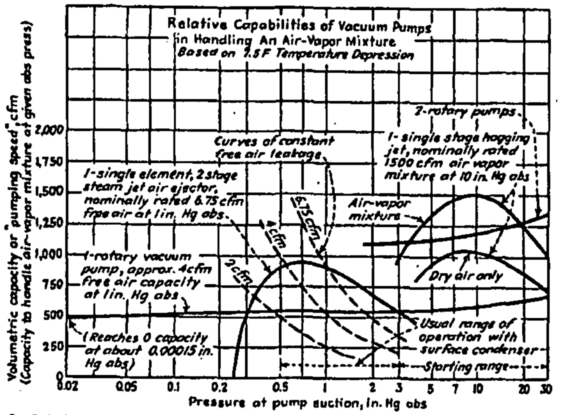

4 Relative capabilities of vacuum pumps depend on type and individual characteristics; the rotary-pump capacity remains relatively constant at all pressures

In view of the small steam and power requirements this has been disregarded.

Making Decision. An ammonia contamination problem may dictate selection of the rotary pump. Since an air ejector can readily furnish greater air-removal capacity than a rotary pump this may lead to its selection in other cases. Use of a rotary pump necessitates keeping air leakage within limits of best practice to avoid loss in vacuum. At lower loads, where air pumps rather than the condenser control vacuum, the air ejector may produce an average of 0.1 in. Hg better vacuum.

Combination Unit. An interesting arrangement, in which both high vacuum capability under normal operation and high pumping capacity when starting are combined in one set of equipment, occurs in the primary steam jet—secondary reciprocating vacuum pump unit. This has been in successful operation in one modern station for a number of years. The secondary pumping unit that operates down to about 4-in. Hg abs is used to evacuate the condenser when starting, and then continues to run as the second stage of a steam jet, which operates at about 1 in. Hg abs. The primary jet can be operated with low-pressure extraction steam, thereby improving the economy of this combination unit.

AIR-REMOVAL CAPABILITY

Generally speaking, the air-removal capability of a rotary vacuum pump decreases with (1) higher vacuums, (2) a decrease in temperature depression (below the temperature corresponding to the condenser vacuum) of the air-vapor mixture as it passes over the air-cooler tubes, and with increased air leakage.

Air leakage in a tight surface condenser, with the turbine glands in good condition, should not normally exceed 3 to 4 cfm. Some condensers perform consistently with only 1.5- to 2-cfm leakage. Others may run as high as 5 to 6 cfm, but greater leakage is unusual with a condenser in good condition.

Condenser manufacturers base their air ejectors on the requirements of the Heat Exchange Institute. The following representative figures are from the Condenser Section of its Standards:

Maximum Steam Condensed, lb per hr	Free Air Leakage cfm at 70 F
100,001-150,000	6.5
150,001-250,000	8.5
250,001-350,000	10.0
350,001-450,000	11.5
450,001-600,000	13.5
600,001 and up	16.0

As applied to large surface condensers, this table would bear revision, since it requires designing for much higher air-leakage rates than are normally experienced.

As an example, if an 80,000-kw turbine is selected for which the full-load condenser flow is 495,000 lb per hr, the air leakage for which air ejectors are supplied under the above regulation is 13.5 cfm. This requirement is excessive. But furnishing two 2-stage steam jets, each rated 6.75 cfm and operating only one under normal conditions, would be a reasonable interpretation of the Heat Exchange Institute requirements. Unless otherwise requested, a condenser manufacturer may furnish double this capacity. If this is supplied in two jets, the previously mentioned annual operating costs would be doubled.

When operating in conjunction with a surface condenser a vacuum pump handles an air-water vapor mixture that has been subcooled several degrees below the saturation temperature by passing over the tubes in the air-cooler section of the condenser. Under Dalton's law of partial pressures, each component of this mixture (air and water vapor) occupies the same total volume as that of the entire mixture, but each component is present at its own partial pressure. Sum of the two partial pressures makes up the total pressure at the air offtake. This partial pressure of the air at condenser-air off-take has a profound effect on capacity of air-removal apparatus.

In all computations involving partial pressures, temperature depression of the air-vapor mixture below the saturation temperature of the vapor must be either measured or assumed. (If saturation temperature were used, the mixture would be 100% vapor.) The Heat Exchange Institute specifies a 7.5-F temperature depression. This is conservative, and good condensers should do better. Another criterion being considered in the industry assumes the air-vapor offtake at definite elevation in temperature above that of the inlet circulating water. A figure of 5 F has been mentioned, but it requires a good air-cooling section to accomplish this. Such an assumption implies a greater subcooling of the air-vapor mixture at high loads than 7.5 F below saturation temperature, this being more favorable to either type of air removal equipment.

Under the pressure and temperature conditions in a condenser, both air and water vapor obey the perfect gas law and Dalton's law with sufficient accuracy. Thus for a given weight of air at constant temperature

$$V_2 = V_1 \frac{P}{P_a} \qquad (1)$$

where

V_2 = air volume under reduced pressure
V_1 = free air volume, or "air leakage"
P = atmospheric pressure, or 30 in. Hg abs
P_a = partial pressure of the air at air offtake, in. Hg abs

For example, if the volumetric capacity or pumping speed of a rotary pump is 550 cfm at 29-in.-Hg vacuum, (from Fig. 4) what corresponding rate of leakage of free atmospheric air can it handle and still maintain this vacuum? Assume 7.5-F temperature depression of the air-vapor mixture.

Condenser pressure, in. Hg abs	1.0
Saturation temperature, F	79.0
Temperature depression, F	7.5
Temperature at air offtake, F	71.5
Partial pressure of vapor at 71.5 F, in. Hg abs	0.78
Partial pressure of air, in. Hg abs	0.22

From equation (1) and neglecting effect of temperature change on volume:

$$550 \times \frac{0.22}{30} = 4.0 \text{ cfm}$$

Thus a rotary pump can handle an air-leakage rate of 4 cfm of free atmospheric air and maintain a vacuum of 29 in. Hg.

Since the action of a steam-jet air ejector is based on the transformation of the jet kinetic energy into pressure energy of the air-vapor mixture in the diffuser, such jets are always rated in terms of their weight-rate capacity to handle an air-vapor mixture. A representative "6.75-cfm" jet with capacity in lb-per-hr plotted against vacuum is shown on the left-hand curve of Fig. 3. If the temperature depression of the air-vapor mixture as it enters the jet suction is known or assumed, the capacity of the jet in equivalent volume of free air may be determined as shown in the right-hand curve of Fig. 3. For this transformation use the following:

$$W_m = W_a + \frac{0.62 W_a P_v}{P_a} \qquad (2)$$

where

W_m = total weight of air-vapor mixture, lb
W_a = weight of air in the air-vapor mixture, lb
P_v = partial pressure of vapor
P_a = partial pressure of air

Fig. 4 shows the volumetric capacity of both types of pumps in terms of their capabilities under actual vacuum conditions rather than in terms of free-air handling ability. The rotary pump exhibits an almost constant, although restricted, volumetric capacity through the usual operating range of vacuum in surface condensers, extending from atmospheric pressure into a range of low absolute pressures never realized in power-plant practice. The 2-stage steam-jet air ejector, on the other hand, exhibits its characteristics of high capacity over a more limited range of pressure with rapid reduction in capacity when passing into pressure regions, both above and below the operating range.

Table I shows the effect of vacuum variation on air-pump performance, and Table II the effect of variation in temperature at the condenser air offtake. These tables were calculated using equation (1) and show the effect of 4-cfm and 6.75-cfm free air leakage.

The volume of the air-vapor mixture to be handled should be compared with the capacities of the two types of air pumps. The two pumps in Fig. 4, although of different capabilities, have been compared because the rotary pump is, to the author's knowledge, the largest made at present and the one usually used for this service. The "6.75-cfm" air ejector could be supplied for this condenser, although it is larger than the rotary pump, but it is smaller than those often furnished in the past.

Fig. 4 includes curves of constant free air leakage. Intersections of these curves with the characteristic of either vacuum pump indicates the limiting vacuum obtainable. The figure also indicates the "starting range" of pump suction pressure and the comparative performance of a large hogging jet, rated 1350 lb per hr or 1500-cfm air-vapor mixture at 10 in. Hg abs with one and two rotary pumps. Note that these performances are comparative only; air ejectors and hogging jets are available both larger and smaller in capacity, at prices and with operating costs in proportion.

The higher the vacuum and the lower the temperature depression at the air offtake, the less favorable are conditions for either type of air pump. Under the assumed air leakage, when the volume to be handled exceeds the air-pump capacity at a given vacuum (assuming the condenser capable of producing that vacuum) the air-removal equipment will be unable to hold the vacuum and it will fall off until the smaller air-vapor specific volume comes into equilibrium with the air-pump capability. If desired, air-removal capacity can be doubled by operating two units. Remember that some of the more extreme conditions in the tables and those in which the air pump capacity is apparently exceeded may be beyond the reach of the condenser or uneconomical to attain. Large turbines, unless designed for unusually high vacuum, reach a point where an increase in vacuum produces very little additional power output with constant throttle flow.

Fig. 2 demonstrates the relative abilities of the condenser and its air pumps to "control" the vacuum over the load range, using an 80,000-kw steam turbine and a 75,000-sq-ft 70,000-gpm condenser unit as an example. In this case, the temperature of the air-vapor mixture has been taken 5 F above that of the circulating-water inlet, instead of 7.5 F below saturation temperature.

Solid lines show the condenser characteristic at 65- and 80-F inlet circulating-water temperatures corresponding to average and summer conditions. Dashed and dotted lines show air-pump performance with 4-cfm free air leakage for the various pump combinations designated. For the assumed rate of air leakage, at all points to the right of the intersection of the two sets of curves, the air pump can handle the air leakage, and has no effect on condenser performance. In other words,

(Continued on page 146)

Table I—Effect of Vacuum on Air-Vapor Mixture Volume

Condenser vacuum, in. Hg	27.0	28.0	29.0	29.5
Condenser pressure, in. Hg abs	3.0	2.0	1.0	0.5
Volume of air-vapor mixture, cfm, to be handled with:				
4-cfm free air leakage	203	285	550	1000
6.75-cfm free air leakage	340	480	920	1700

Table II—Effect of Vacuum and Temperature on Air-Vapor Mixture Volume

Temperature depression of air-vapor mixture below saturation temperature, F	5	7.5	10
Volume of air-vapor mixture, cfm, to be handled with 6.75-cfm free air leakage for condenser pressures of:			
1 in. Hg abs	1350	920	700
2 in. Hg abs	700	480	360