

# Specifying Steam Surface Condensers

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Factors affecting condenser design and selection.  
A typical air conditioning surface condenser specification is included.

THE FORMULA for determining the amount of surface required in a surface condenser is as follows:

$$Q = UA (LMTD) (1)$$

where

Q = Heat to be absorbed, Btuh = pounds per hour of steam times 950 Btu per pound;

U = Overall heat transfer rate, Btu per (hr) (sq ft) (F);

A = Surface required, sq ft;

LMTD = Log<sub>e</sub> mean temperature difference between condensing temperature and cooling water.

No attempt will be made to go through a formal condenser selection. The procedure for condenser rating is amply covered in the publication of Heat Exchange Institute entitled, "Standards for Steam Surface Condensers."\* Rather we shall attempt to discuss qualitatively the factors affecting selections and their effects on size and price.

## Turbine Steam Rate

Turbine steam rate is usually expressed in pounds per hour of steam per brake horsepower and is commonly called the water rate. This water rate is a function of motive steam pressure and temperature, condensing temperature, or equivalent vacuum, and turbine efficiency. Specific information for a given turbine application can be obtained

\* This Standard is currently out of print, but a new edition is in preparation and will be available in the future from Heat Exchange Institute, 122 E. 42nd St., New York, N.Y. 10017.

from the turbine manufacturer.

Generally speaking, on systems where the motive steam pressure is approximately 150 lb and the turbine is discharging to a vacuum of 26.inches of mercury (4 in. Hg absolute), a steam rate of 14-16 lb per hr per bhp is economically feasible. Although greater efficiency is available at increased cost, utility costs for this type of installation seldom justify reducing the steam consumption by means of higher efficiency turbines or lower condensing pressures. When special conditions are present, the specifying engineer may consider either of the above modifications to improve performance.

## Condenser Vacuum

The vacuum produced in a steam condenser is that pressure corresponding to the temperature at which the steam condenses. This steam condensing temperature, in turn, is dependent upon the temperature of

cooling water entering the condenser and the quantity of water available. The steam condensing temperature can get down to as low as 5F above the entering cooling water temperature, but in most air conditioning installations the differential is 20F, sometimes 10F, but seldom smaller, as the closer approach would require a costly amount of condenser surface.

In practice, cooling tower water is first used in the refrigerant condenser, where it rises about 10F and is then available for steam condensing. The great bulk of steam-driven centrifugal refrigeration systems operate with a steam condenser vacuum of approximately 4 in. Hg abs., which is equivalent to a condensing temperature of 125.4F. Thus, referring to Fig 1 and presuming that 85F cooling water is supplied to the system from the cooling towers, water temperature into the steam condenser is approximately 95F. Assuming a turbine steam rate of 14-16 lb per hr per bhp (bhp per ton of refrigeration is approximately 1) and a condensing water circulation rate of 3 gpm per ton of refrigeration, water rise on the steam condenser is also about 10F. Thus, the leaving water temperature approach is 20F(125F minus 105F).

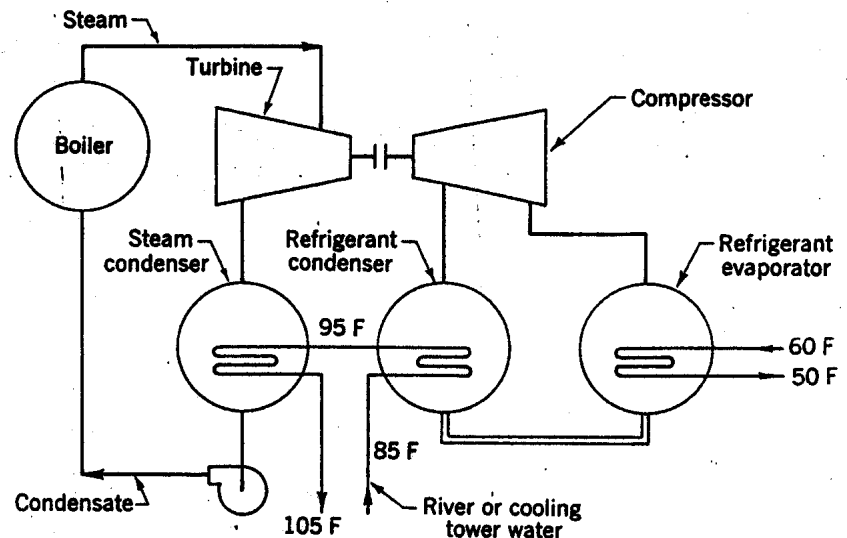


Fig. 1. Centrifugal refrigeration machine (condensing turbine driven)

If, on the other hand, we designed the steam condenser for 3 in. Hg. abs., which is equivalent to a condensing temperature of 115F for the same approximate water temperature rise, the approach would be 10F (115F minus 105F). Thus from Eq (1), because the log mean temperature difference is greater, we see that the condenser designed for 4 in. Hg. abs. requires less surface than the one designed for 3 in. Hg. abs.

The foregoing example is an oversimplification. In practice, the steam rate of a turbine discharging at 4 in. Hg. abs. is somewhat higher than one discharging at 3 in. Hg. abs. Choice of a lower absolute pressure might permit using a turbine with one less wheel, which would result in reduced turbine cost. This reduced turbine cost might or might not be offset by the increased condenser cost. (See "Is There a Steam Turbine in Your Future," ACHV, April, 1968, pp 43-47.) Similarly, condenser cost could be reduced if the condenser were operated at 5 in. Hg. abs. (133.8F condensing). However, turbine cost might increase more rapidly if the steam rate had to be held to a previously set value, which, as indicated, is generally in the range of 14-16 lb per hr per bhp.

Thus, the specification should simply state the number of pounds per hour of steam to be used and the maximum temperature and quantity of available cooling water, omitting the precise operating vacuum to be maintained by the surface condenser. This permits the manufacturer to offer the best combination of condenser, turbine and refrigeration unit to comply with the limitations on steam consumption specified.

Other considerations, such as length limitations and turbine vapor exhaust size, very often have a determining affect on condenser cost.

### Water Temperature

As cooling water temperature determines the vacuum that can be produced, it is imperative that the maximum anticipated water temperature

be specified. This temperature is determined by the water source and varies with the location of the installation. Water flow rate to the condenser is usually determined by the refrigeration manufacturer and is generally set at approximately 3 gpm per ton of refrigeration when cooling towers are used. However, when river water or water at relatively low temperature is available, water flow rate varies and specifications should take these variations into account.

A recent trend in large multi-story office buildings has been to use 2 gpm per ton of refrigeration. Although this affects condenser, and turbine adversely due to higher condensing temperatures, there is a saving in condenser water piping.

Too often, specifications state that the surface condenser shall be capable of handling turbine exhaust while maintaining the desired vacuum with a 5% increase in normal steam load. Compliance with this requirement is almost impossible. Specifications seldom provide the 5% greater water quantity needed to balance this 5% loading factor. Further, usually no provision is made for the extra load on the cooling tower and the change in steam rate of the turbine when overload hand valves are used. Furthermore, the refrigeration machine is operating at a different point on its curve and may be rejecting more heat which, in turn, means hotter water to the steam condenser. The net result under the 5% excess load condition is that the steam condenser receives cooling water that is more than 5% above the initial design temperature and the resulting excess heat imparted to the water overloads the cooling tower, further aggravating the situation.

If it has been decided that the surface condenser should have additional surface in it, the most practical approach would be to state that the unit shall have 5% excess surface or 10% or whatever is needed. In general, a surface condenser will handle the entire excess load at a slight increase in back pressure. If this is permissible (and it must happen unless all

accessory items have built-in excess capacity), no statement of excess capacity need be made.

### Water Pressure Drop

Water side pressure drop is a function of tube velocity, condenser length and the number of passes the water has to make in a given condenser. Air conditioning practice permits velocities as high as 10 fps in the tube side of refrigerant condensers and, in recent years, this has been extended to apply to steam surface condensers for air conditioning installations.

It has been common practice to limit water pressure drop to 10 ft. However, if this limitation is rigidly enforced, velocities must be reduced to comply with this pressure drop requirement, resulting in lower heat transfer rates and higher condenser costs. Generally, water pressure drops of approximately 15 ft can be attained in most single-pass condensers operating at 4 in. Hg. abs. Most two-pass condensers, usually those running at 3 in. Hg. abs., require pressure drops as high as 20-30 ft.

Specification of water velocity and materials of construction for tubes and tube sheets where river water or brackish water is involved must be carefully evaluated. Information should be obtained on the experience of other users in the area having similar design conditions before a specific material is selected. Generally speaking, for brackish water applications, velocities in excess of 6.5 fps can seriously reduce tube life, but for cooling tower applications a 10 fps velocity is acceptable.

### Fouling Factor

Most surface condensers for air conditioning and power plant applications are designed with a cleanliness factor of 85%. This means that the heat transfer rate used in designing the condenser is 85% of the clean heat transfer coefficient. For those installations where cooling water sources produce rapid fouling of

the tubes, a higher factor must be used. This holds true for both the refrigerant condenser and the steam condenser.

For most applications using clean cooling tower water, refrigeration condensers should be specified with a fouling factor of 0.0005 and steam surface condensers designed for an 85% clean tube coefficient. If the refrigerant fouling factor is increased, the surface condenser cleanliness factor should be increased by the same percentage. Thus, if the refrigerant condenser is specified as 0.001 vs. 0.0005, the corresponding cleanliness factor for the surface condenser is 70% clean vs. 85 % clean.

### Steam Inlet Velocity

Velocity at the exhaust of the steam turbine is limited by National Electrical Manufacturers Association to 450 fps. Should the exhaust line from turbine to condenser be long, there can be excessive pressure drop in the vacuum line, which means that pressure at the condenser inlet is penalized by the pressure drop between turbine and condenser. Condenser manufacturers usually guarantee to maintain the specified vacuum at the inlet to the steam condenser. However, pressure drop between turbine exhaust and steam condenser must be calculated by the design engineer and compensated for by specifying that the condenser shall maintain a vacuum slightly below the vacuum at the exhaust of the turbine. A specification that calls upon the supplier to supply a specified steam consumption obviates this problem. Where a given velocity is used to determine the pressure drop, this should be stated in the specification so that it is not exceeded. In any event, velocities in excess of 450 fps should not be permitted. Impingement plates should be provided below the condenser steam inlet to protect tubes from the effect of locally high inlet velocities and impingement of condensate. Steam inlet velocities should not exceed 400 fps and provision should be made based upon design experience in this

field for proper vapor velocities within the shell.

Calculation of condenser heat load is based on the fact that steam entering the condenser has a value of approximately 950 Btu per lb. More precise calculations can be made if the specific turbine characteristics are available, but this figure is sufficiently accurate for the sizing of the cooling tower and condenser.

### Space Limitations

Space considerations often influence the cost of a condenser. For example, if the surface condenser is limited in tube length, a multipass design may be required in place of a single-pass design. This results in a short, stubby unit, which is generally more expensive than a long, thin unit.

Specifications of head room limitations and permissible condenser tube length should take into consideration the distance beyond the tube sheets of the condenser required to allow removing tubes for replacement. Condensers must be mounted so that liquid level on the hotwell is sufficiently high to permit removal by condensate pumps. For air conditioning installations, the liquid level should be at least 5 ft above the inlet to the condensate pump and preferably 1 or 2 ft higher. This permits an economical pump selection.

### Air Leakage

As both the exhaust of the steam turbine and the condenser are operating under a substantial vacuum, air is bound to leak into the system. This leakage occurs through the gland seals on the steam turbine and through minute holes in the piping connections associated with the surface condenser itself. Over a period of years, Heat Exchange Institute has determined the normal quantity of air that should leak through properly designed turbines and piping systems, and these are specified in their "Standards for Surface Condensers." Manufacturers have similarly standardized their ejector sets so that several standard sizes are available for specific air

leakage quantities based on various team quantities entering the condenser. Specification that the ejector shall be designed for the quantities of air leakage as designated in HEI Standards is sufficient.

### Codes

Steam surface condensers operate under a vacuum and are, therefore, not considered pressure vessels. The ASME Code is a pressure vessel code and is not, strictly speaking, applicable to surface condensers operating under a vacuum. However, the tube side of a surface condenser is considered a pressure vessel, as it is subjected to the full water pressure. When necessary, this side of the condenser can be designed and constructed to ASME Code requirements. Most surface condensers are designed and constructed in accordance with HEI Standards. They can be built to ASME Code requirements and so stamped by a qualified inspector. This type of construction, which requires specific methods of welding, specially qualified welders and keeping of material records with certified copies of these records to be supplied, results in a more expensive surface condenser. However, if the unit is to be built to ASME Code requirements, it should be stated that both the shell and tube sides, or simply the tube side, shall be designed, constructed and stamped in accordance with ASME Code and shall be accompanied by certifications signed by a qualified insurance inspector. Do not be misled by the statement, "Condenser is constructed in accordance with applicable ASME Codes." It is common for some manufacturers to indicate that, as ASME is a pressure vessel code, it is not applicable to vacuum condensers and therefore their equipment complies with applicable ASME Codes.

## Materials of Construction

Materials used in construction of steam surface condensers are given in Table 1.

Use of copper alloy tube sheets in steam surface condensers with the accompanying requirement that tube sheets be bolted to the shell by means of collar bolts, is a carryover from marine practice. There is no reason why a steel tube sheet cannot be used, in view of the fact that the refrigeration condenser just upstream of the steam condenser uses this type of construction. When a steel tube sheet is used, the specification should indicate that it may be welded to the shell.

## Specification Considerations

When tube alloys that differ from the standard admiralty material are used, ratings must conform to HEI Standards. These standards reduce the heat transfer rate to conform with the characteristic performance of various alloys. Similarly, when tube gages must be increased above the normal 18 BWG, heat transfer rates should be corrected in accordance with HEI Standards to compensate for increased tube wall resistance.

It is not uncommon in power plant practice to use a divided water box construction on surface condensers, but this type of construction is unusual in air conditioning practice, as air conditioning equipment is seldom critical. However, where cooling water is particularly dirty, fouling of the tubes may be rapid and, under these circumstances, a divided water box condenser might have some advantage.

Corrosive effects on water boxes should be compensated for by use of alloy cladding, fiber glass reinforced polyester lining or similar procedure.

## Auxiliary Equipment

Selection of auxiliary equipment to be used in conjunction with the surface condenser can be very important. Failure of the ejector set or condensate pump is critical to the operation of the entire system. The condensate pumps,

which are the only moving parts in a surface condenser system should be supplied in duplicate. Power plant practice usually requires twin air ejector sets, one a standby, but for air conditioning installations, a single set is sufficient.

## Ejectors

For condensing pressures of 5 in. Hg abs. (25 in. of vacuum) and lower, a two-stage ejector system is recommended because, as the condenser water temperature drops, the two-stage ejector permits operating at reduced condenser pressure that, in turn, reduces overall steam consumption of the turbine. Also, as these systems are designed on the basis of maximum summer temperatures, a two-stage ejector provides operating economies during those days when maximum water temperature levels are not reached.

It is important in specifying steam ejectors that the minimum anticipated steam pressure be stated. Allowance should be made for pressure drop between the source of steam and the point at which steam enters the ejectors. If an ejector is designed for 100 lb steam and lower pressure steam is delivered, the ejector may not function properly; if the pressure is more than 10% below design level it will fail to operate altogether.

Specifications should state that design air leakage for the ejector system be in accordance with HEI Standards. When more than one turbine discharges into a common condenser, the ejector must be designed for 1 1/2 times the normal air leakage for the total steam quantity. For more than two turbines, other correction factors are applicable.

The cooling medium for the inter and aftercondensers of the two-stage ejector system can be cooling water from the cooling tower or condensate from the hotwell of the surface condenser. As intercondenser condensing temperature is about 145F and aftercondenser condensing temperature is about 212F, fouling of the inter and aftercondensers will be more rapid than that of the main condenser.

**Table 1. Steam Surface Condenser Construction Materials**

Waterboxes and Waterbox Covers
Cast iron—ASTM Spec. A-48, Class 30. Steel—ASTM Spec. A-285, Grade C, flange quality or ASTM Spec. A-283. Aluminum bronze—ASTM Spec. B-169, Alloy D. 90-10 Copper nickel—ASTM Spec. B-171. Ductile cast iron.
Shell Plate
Steel—ASTM Spec. A-285, Grade C, flange quality or ASTM Spec. A-283.
Tube Sheets
Muntz Metal—ASTM Spec. B-171. Aluminum bronze—ASTM Spec. B-169, Alloy D. Silicon bronze (copper silicon alloy)—ASTM Spec. B-96, Alloy A. 90-10 Copper nickel—ASTM Spec. B-171. Steel—ASTM Spec. A-285, Grade C, flange quality.
Tube Support Plates
Steel—ASTM Spec. A-285, Grade C, flange quality. ASTM Spec. A-36. ASTM Spec. A-7. ASTM Spec. A-283.
Tubes
Admiralty—ASTM Spec. B-111-62, type A, B, C or D.
Specifications taken from Heat Exchange Institute Standards.

It is therefore recommended that condensate be used as a cooling medium for the inter and aftercondensers, as condensate is practically pure water and contains little or no solids. A typical schematic control diagram using condensate over the inter and aftercondensers is shown in Fig. 2.

Condensate from the intercondenser must be drained back to the hotwell, as the intercondenser is under a vacuum. This can be done by means of a condensate trap or, where headroom permits, by means of a loop seal, which must be at least 8 ft high. Condensate from the aftercondenser can be drained back to the hotwell through a trap or may simply be drained to an open sight drain. The writer prefers the second method

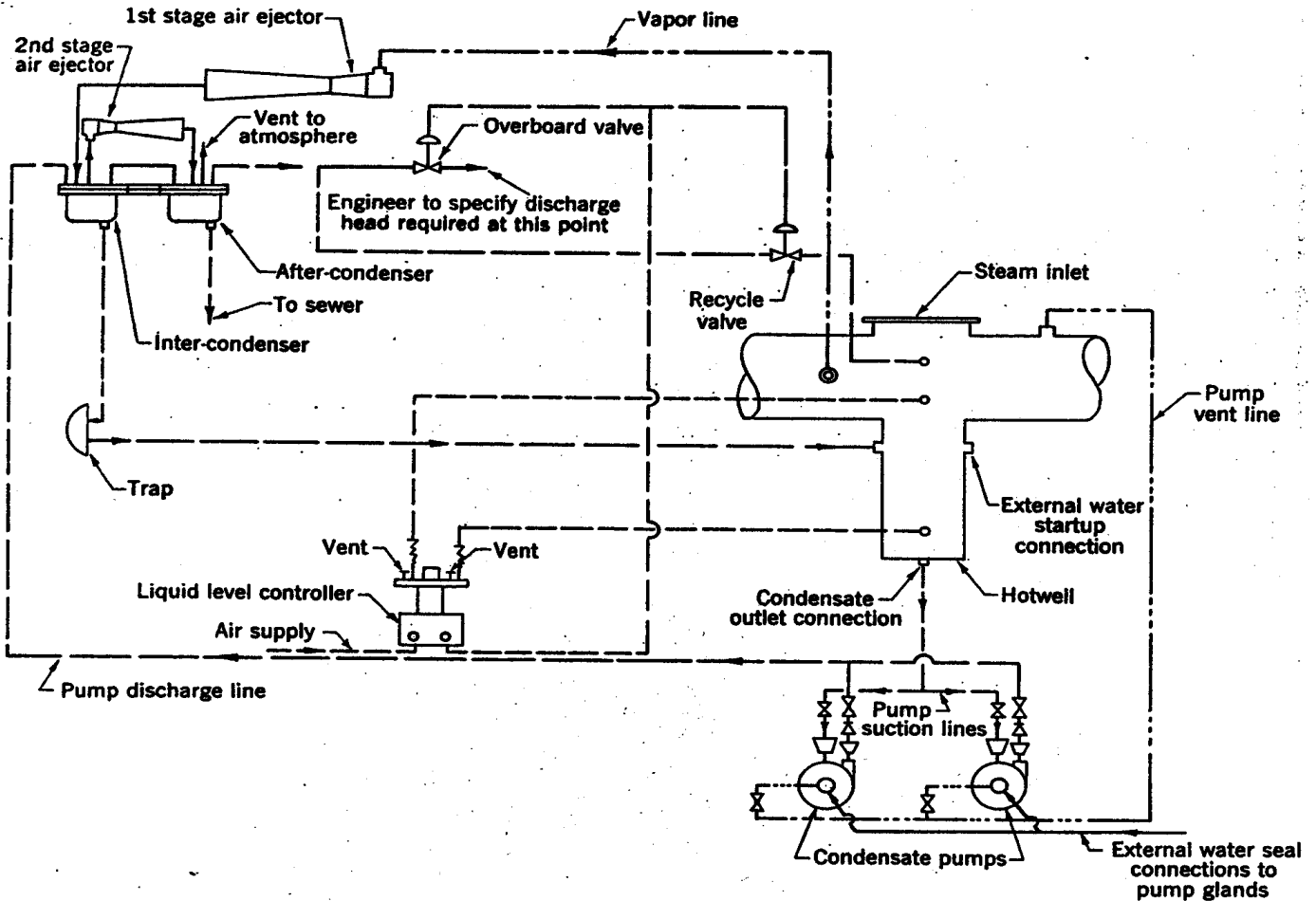


Figure 2. Condensate flow diagram

because it eliminates the possibility of a malfunction of this trap, which could bleed air back to the hotwell.

#### Pumps

Condensate pumps for air conditioning are customarily of the horizontal type, designed for approximately 4 1/2 - 5 ft net positive suction head (NPSH) and total dynamic heads of from 80-150 ft. Actual NPSH available for these pumps depends upon the physical location of the condenser. The discharge head is affected by where the condensate is pumped after it leaves the condensate pump; that is, to an open condensate tank or back to the cooling tower, etc. The importance of providing sufficient NPSH for proper operation of the condensate pumps cannot be overemphasized. The condenser should be located sufficiently high off the floor to put as much head on the inlet of the condensate pumps as is physically

possible. The greater the available NPSH, the easier job the pump has and the less danger there is of cavitation. Condensate pumps are generally of the horizontal close coupled type, 3450 rph and with open drip-proof motors.

#### Relief Valves

Surface condensers most commonly use atmospheric relief valves of the water-sealed or O-ring type, although bursting discs are occasionally applied. Size and rating characteristics for these valves are given in HEI Standards. In operation, a small amount of water is kept above the relief disc to prevent the inleakage air. Water in the valve must be supplemented periodically, as a certain amount will evaporate. A sight glass is provided, but because relief valves are usually located at a considerable height and the sight glasses are frequently obscured by sediment, they are difficult to read. This author prefers to supply a small constant drip of water to the valve, permitting the

overflow to drop into an open sight drain. Thus, as long as there is an overflow, we know that the valve is supplied with water.

#### Liquid Level Control

The method of control used depends upon whether or not condensate is used as the cooling medium for the inter and aftercondensers of the ejectors. When condensate is used as the cooling medium, a scheme such as that shown in Fig. 2 should be used. If condensate is simply dumped overboard, then a simple mechanically operated level control on the discharge of the condensate pump is sufficient. The arrangement shown in Fig. 2 requires that all of the condensate be pumped over the inter and aftercondensers before diverting a portion back to the hotwell, or overboard, depending upon load conditions.

**Minimum Simplified Specification for  
Steam Surface Condensers in  
Air Conditioning Practice**

Vacuum at inlet to steam condenser \_\_\_\_\_  
 Steam flow to condenser, lb per hr \_\_\_\_\_  
 Condenser water inlet temperature, F \_\_\_\_\_  
 Condenser water gpm \_\_\_\_\_  
 Permissible condenser water pressure drop, ft \_\_\_\_\_  
 Number of water passes \_\_\_\_\_  
 (Single-pass for 26 in. of vacuum and two-pass  
 for .27 in. Should be left up to supplier) .  
 Maximum water velocity, fps \_\_\_\_\_  
 10 fps for cooling tower applications)  
 Tube diameter and gage \_\_\_\_\_  
 (3/4 in. X 18 BWG)

Cleanliness factor \_\_\_\_\_  
 (85% clean for cooling tower applications)  
 Waterbox design pressure \_\_\_\_\_  
 Water test pressure \_\_\_\_\_  
 (1.5 times design pressure except for cast iron  
 which is 5 Lb above design pressure) \_\_\_\_\_  
 Shell side design pressure \_\_\_\_\_  
 (Full vacuum test to 20 Lb)  
 Steam pressure and temperature at ejector inlet \_\_\_\_\_  
 Condensate pump capacity, gpm \_\_\_\_\_  
 Condensate pump net positive suction head \_\_\_\_\_  
 Condensate pump discharge  
 head \_\_\_\_\_  
 (Add pressure drop through inter- and  
 aftercondensers and overboard valve to total  
 system loss.)  
 Electrical characteristics of pump motors \_\_\_\_\_

1. The steam condenser shall be of the shell and tube type, single or multipass, designed and constructed in accordance with Heat Exchange Institute (HEI) Standards, except as noted herein, and shall be suitable for the design conditions noted above. Condenser manufacturer shall demonstrate successful installations of similar equipment that has been in operation for at least \_\_\_\_\_ (5 or 10) years.
2. Condenser shall have cast iron or steel waterboxes, steel tube sheets, steel tube supports, admiralty tubes and a steel hotwell complete with screen, strainer and hand-hole clean-out. Hotwell shall be of the vertical tank type, suitable for one-minute storage capacity.
3. Provision for tube expansion shall be made either by means of bowed tubes or shell side expansion joint. Covers of the waterboxes shall be removable without disturbing water piping. Connections shall be provided in the hotwell for ejector drain, condensate outlet, liquid level control bypass, fresh water startup, gage glass and high level water alarm. Connections shall also be supplied in the condenser shell for pump vent, vacuum gage, pressure gages to be located on waterboxes, an atmospheric relief valve and steam inlet from the turbine drive. Waterboxes shall be supplied with a plugged vent and drain and shall have flanged water connections. All vapor side flanges shall be 125 lb FF. Water side flanges shall be \_\_\_\_\_ lb (raised or flat face, depending upon pressure).
4. A single-element, two-stage ejector assembly, complete with internal interconnecting water and steam piping with surface inter- and after-condenser shall be provided. Ejector shall be cast iron. Ejector motive steam nozzles shall be stainless steel. The two-stage ejector shall be a self-contained package, complete with steam valves, interconnecting steam piping, steam strainer and interconnecting vapor piping, and shall be provided with mounting bracket so that ejector can be mounted directly on the surface condenser.
5. Condensate pumps shall be supplied in accordance with design conditions and shall be close coupled, single suction, enclosed impeller type with cast iron casings, bronze or cast iron impellers, stainless steel shafts, shaft seals suitable for the service, grease lubricated bearings and open drip-proof motors. Pump motor starters will be provided by others. Electrical characteristics will be as specified under \_\_\_\_\_ herein.
6. A hotwell gage glass shall be provided.
7. A steam inlet expansion joint shall be provided, sized to conform with the diameter of the turbine exhaust with corrugations. Expansion joint shall have copper or stainless steel bellows and shall be provided with a liner if steam velocity exceeds 300 fps.
8. An atmospheric relief valve shall be furnished, sized for protection in accordance with HEI Standards.

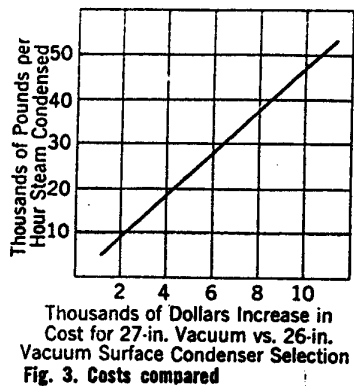


Fig. 3. Costs compared

When condensate quantity exceeds that required by the inter and aftercondensers, a suitable bypass should be supplied.

Selection of air-operated level control valves is dependent upon information supplied by the consulting engineer; namely, the discharge head beyond the overboard level control valve and the total dynamic head the pump must develop. Specifications should indicate whether the total dynamic head stated for the pump has taken into account the condensate pressure drop through the inter and aftercondensers and the overboard valve. Generally, pressure drop through the inter and aftercondensers is less than 15 ft and that on the overboard valve is 10-30% of the available discharge

head beyond the overboard control valve.

#### Gages and Thermometers

A combination pressure and vacuum gage of the dial type should be supplied to give a rough indication of condenser vacuum. Exact vacuum can be properly determined only by use of a mercury manometer, which can be specified. Pressure gages should be installed on the inlet and outlet water boxes. It is also common practice to install pressure gages at the condensate pump discharge, as well as at the inlet to the condensate pump, so the operating engineer can easily check the performance of pump and condenser.

In addition, thermometers should be placed so that inlet and outlet water temperatures, as well as temperature of the condensate in the hotwell of the surface condenser, may be read.

#### Air Leakage Meter

Most operating engineers of air conditioning installations judge excessive air leakage by the amount of steam and droplets of condensate issuing from the aftercondenser vent. However, an air leakage meter placed on the vent of the aftercondenser is more reliable and, as it represents an investment of approximately \$150, is well worth the expense.

#### Expansion Joint

A single or multi-corrugation expansion joint should be placed between the exhaust of the turbine and the inlet to the surface condenser. These expansion joints usually have copper or stainless steel bellows and a steel flange. However, as the cost of copper increases, stainless steel joints are replacing them. The number of corrugations is determined by the amount of expansion that must be accommodated by the joint and should be determined by the engineer prior to completing the specification. When the number of corrugations is not stated, a single-corrugation joint is supplied, which may or may not be sufficient for the job.

When steam velocities through the expansion joint exceed 300 fps, it is advisable to install a stainless steel liner to prevent noise and excess vibration of the joint.

It is possible to obtain reduced steam consumption simply by designing the surface condenser for a higher vacuum. Relative costs of condensers designed for 26 and 27 in. of vacuum are given in Fig. 3.