

# Full Steam Ahead



**Alicia Johnson, Graham Manufacturing, USA,** looks at the variables that affect capital cost and performance reliability in steam surface condensers.

Steam surface condensers are critical equipment in a fertilizer plant, and can directly impact plant throughput and product quality. Design choices in the front-end engineering and design (FEED) phase directly impact capital cost and performance reliability. A thorough understanding of what influences steam surface condenser cost and performance is important for ensuring proper syngas compressor, CO<sub>2</sub> compressor and main air blower performance. This article reviews key design decisions that affect capital cost and outlines operating variables that can negatively impact performance reliability.

## What affects capital costs?

### Cleanliness factor versus fouling factor

Steam turbine exhaust surface condensers are most commonly designed for cleanliness factors, as they are in relatively clean service. The cleanliness factor is the factor applied to the clean heat transfer coefficient as an allowance for fouling. The common cleanliness factor used for turbine exhaust surface condensers is 85% clean, which indicates 17.6% excess surface area. The fouling factor is another way to account for fouling and is typically used in process type service where shell and tube sides are dirty or tend to foul. The example shown on page 30 compares the typical turbine exhaust surface condenser cleanliness factor versus a typical process condenser fouling factor and a more common surface condenser fouling factor. A fouling factor may be applied in a clean application if the factor is specified correctly. As shown in the example, if a fouling factor is specified similar to a process condenser, it will result in an excessive amount of surface area to the surface condenser design. Additional surface area increases the capital cost. In addition,







the larger condenser increases the footprint required along with the support structures, cooling water pressure loss, and even transportation costs.

Example:

Clean overall heat transfer rate ( $U_c$ ) is 750 btu/hr ft<sup>2</sup>°F

Cleanliness factor design heat transfer rate  $U_d$  = cleanliness factor  $\times U_c$

$$U_d = 0.85 \times 750 = 637.5 \text{ btu/hr ft}^2\text{°F}$$

Fouling factor design heat transfer rate

$$U_d = \frac{1}{\frac{1}{U_c} + \text{fouling factor}} = \frac{1}{\frac{1}{750} + .002} = 300 \text{ btu/hr ft}^2\text{°F}$$

Using the process type fouling factor of 0.002 hr ft<sup>2</sup>°F/Btu equals a clean percentage of 40% (300/750 = 0.40). Converting the value to excess area equals 2.50 (1/0.4), or 150% excess area.

For comparison, using a .0003 fouling factor equals a clean percentage of 82% (612.2/750 = 0.82). Converting the value to excess area equals 1.23 (1/0.82), or 23% excess area.

### Materials of construction and thermal design considerations

Material selection of the tubes and tubesheets is important to the design of the surface condenser and one of the main items that impacts capital cost on the condenser itself. The shell and waterbox materials of condensers are typically specified as carbon steel. The tubes and tubesheet are based on the site requirements, as it is dependent on the water quality at the site. Once the tube material is known, different grades and options are available to select from that will change the design of the condenser, affecting the capital cost.

HEI Standard for Steam Surface Condensers Eleventh Edition Table 8 shows typical materials for everything but tubes.

Additional materials not shown for the tubesheets are types of duplex and the option of clad. The option of clad tubesheets is ideal for larger condenser sizes over approximately 60 in. in diameter with higher cost material – for example, naval rolled brass (NRB) or titanium – to save capital cost while not impacting performance. In addition, this would allow the tubesheet to be welded to the shell with similar materials along with eliminating the body flanges and reducing the weight.

HEI Appendix J & K show tube material except some duplex similar to the tubesheets. As shown on the tables and appendixes, there are many material options to select from to ensure the correct materials are selected. Since the tube material is based on the water quality, the tube diameter, tube gauge, and velocity may be altered. Changing the tube diameter, tube gauge, or tube velocity will affect the heat transfer rate. In turn, it will also affect the capital cost.

For example, with 18 BWG (0.049 in. thickness) admiralty tubes with 8 ft/sec. tube velocity, assuming fixed cooling water conditions versus 18 BWG 304 stainless steel tubes with the same tube velocity of 8 ft/sec., the heat transfer rate is approximately 24% lower for the stainless steel option to the admiralty option. This would result in a longer condenser for the stainless steel tubes as more surface area would be required. The capital cost of the condenser would be similar to each other based on current material cost, but the footprint may impact the overall capital cost of the project depending on what space is available.

Stainless steel, duplex, and titanium tube gauge can be thinner (approximately 22 BWG, 0.28 in. thick) than the typical thickness of a copper base due to strength. This is also the case with the tube velocity as industry standards show a higher velocity (approximately 10 ft/sec.) is acceptable for stainless steels, duplex, and titanium materials. Per HEI, the heat transfer coefficient ( $U$ ) is calculated per the below equation.  $U_i$  equals the uncorrected heat transfer coefficient, which is based on the tube diameter and tube velocity.  $F_w$  is the factor based on the inlet cooling water temperature. In the example, 85°F is used.  $F_M$  is the factor based on the tube material and tube gauge.  $F_C$  is the cleanliness factor as discussed above. In the example, 85% clean is used. Going back to the example and changing the stainless steel tubes to a thinner gauge (22 BWG versus the 18 BWG), the stainless steel heat transfer coefficient increases by approximately 14%. In addition to changing the tube gauge, an increase in tube velocity increases the heat transfer coefficient by another 10%. If the tube temperature were to increase from the example at 0.75 in. to 1 in., the heat transfer rate would decrease by approximately 2%.

$$U = U_i \times F_w \times F_M \times F_C$$

In addition to the tube characteristics, the pressure drop across the tubes shall be considered. As shown in Table 1, as the tube velocity increases with a constant cooling water flowrate, the pressure drop increases. The change in tube passes also increases the pressure drop. This is another factor that needs to be evaluated based on the capabilities of the site and may impact capital costs if restricted in pressure drop.

Consequently, care shall be taken in the selection of material and all the characteristics as they affect the heat transfer rate and capital cost of the surface condenser. Per Table 1, there is minimal cost difference between the admiralty option and the 304SS option, with the same tube characteristics in capital cost of the condenser. However, the 304SS has a greater surface area and is longer in length, taking up more footprint. Changing the stainless steel tube gauge to 22 BWG decreases the cost by approximately

**Table 1. Tube selection and velocity affects condenser length and diameter**

Tubes	Tube gauge (BWG)	Tube diameter (in.)	Velocity ft/sec.	Tubesheet	Number of passes	Pressure drop (psi)	Tube length	Shell diameter	$U_d$ (Btu/hr ft <sup>2</sup> °F)
Admiralty	18	0.75	8	NRB	1	7.0	Base	Base	679
304SS	18	0.75	8	304LSS	1	9.0	Longer	Similar	513
304SS	22	0.75	8	304LSS	1	9.5	Longer	Similar	586
304SS	22	0.75	10	304LSS	1	13.5	Very long	Similar	646
304SS	22	0.75	10	304LSS	2	14.25	Shorter	Larger	646
304SS	22	1.0	10	304LSS	2	13.75	Similar	Larger	636

10%. Furthering the reduction in cost by approximately 2% occurs when the tube velocity is increased to 10 ft/sec. Changing the tube characteristics of the stainless steel option brings the heat transfer rate closer to the admiralty option.

As shown in Table 1, the option with SS tubes, 22 BWG, 10 ft/sec., and one pass is very long. The long and narrow condenser may not be practical, due to space constraints. Therefore, approximately a 6% increase in condenser cost for a two pass may outweigh the cost of the larger foundation, special transportation charges, and any additional plot area impacts. With current tube material prices, the 304SS two pass option with the same tube diameter is approximately 7% less than the admiralty option. Comparing the tube diameter for SS with the more practical two pass selection, there is an increase of approximately 10% in cost to go to 1 in. diameter tubes versus 0.75 in. diameter. In the end, every item will need to be carefully reviewed starting with the site requirements and moving to the items that allow the condenser to be optimised for design and capital cost.

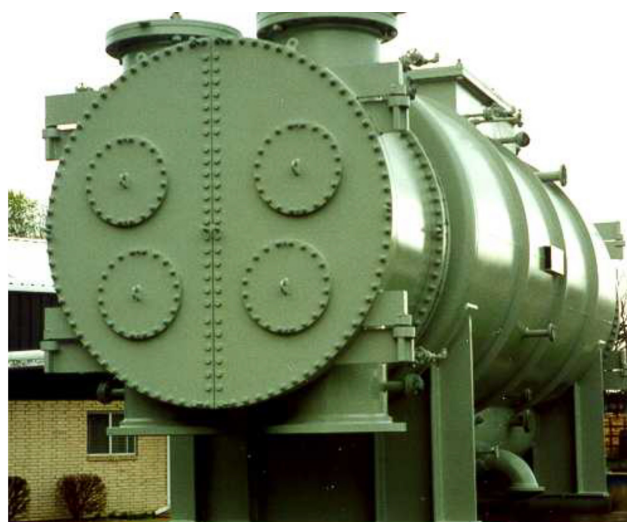
### Optional specification items

Divided waterboxes versus non-divided waterboxes is a question that may help with maintenance, but will add to the capital cost of the surface condenser. The divided waterbox allows for cleaning of dirty tubes or plugging of leaking tubes in one half of the unit while the other half continues to operate with the surface condenser at a higher turbine backpressure. For a typical condenser that is approximately 60 in. diameter, it is only approximately a 7% increase in cost to add divided waterboxes. About 3% of the 7% increase is to change one waterbox from bonnet to channel types. To change both sides of the waterboxes, it is approximately a 6% increase, which is more common in divided waterboxes to allow for access. Therefore, the total to increase the waterboxes to divide with channel covers on both sides is approximately a 10% increase. Piping cost is another factor that may add to the capital cost as a non-divided waterbox will require a larger single connection for the inlet and one for the outlet, while the divided waterbox will require two separate cooling water inlets and outlets, each at a slightly smaller diameter. In addition, in order to isolate the divided waterbox, isolation valves are required. Figures 1 – 3 show the different types of waterboxes with divided and non-divided options.

Codes and standards are another specification requirement to consider the added benefits versus the capital costs. The surface condenser is at least typically supplied per HEI standards with construction and stamp per ASME Sec. VIII Div. 1. Including ASME code ensures the unit will be constructed in accordance with the provisions of a consensus standard recognised worldwide. The function of ASME as excerpted from the forward of the code "is to establish rules of safety governing the design, fabrication and inspection during construction of boilers and pressure vessels." Although HEI is a highly reputable organisation, its construction standards are incomplete and leave areas open to the discretion of the manufacturer, in lieu of analytical design procedures. Another common standard in this industry is the TEMA standard. This standard has three different variances: TEMA 'C', 'B', and – the most stringent – 'R'. TEMA 'C' is a moderate addition to the ASME code for commercial and general process applications. TEMA 'B' is for chemical process applications, and TEMA 'R' is generally severe requirements of petroleum applications per TEMA standard definitions. Each code or standard increases the requirements and capital cost to the surface condenser. Some of the major items include additional corrosion allowance, special flatness (TEMA 'R'),



**Figure 1.** Divided waterbox with a bonnet.



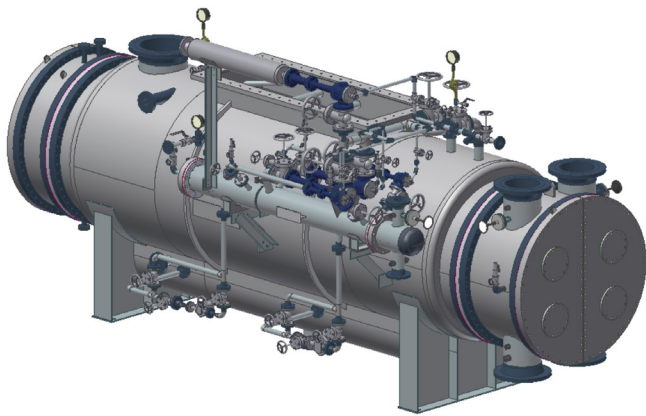
**Figure 2.** Divided waterbox with channel covers.



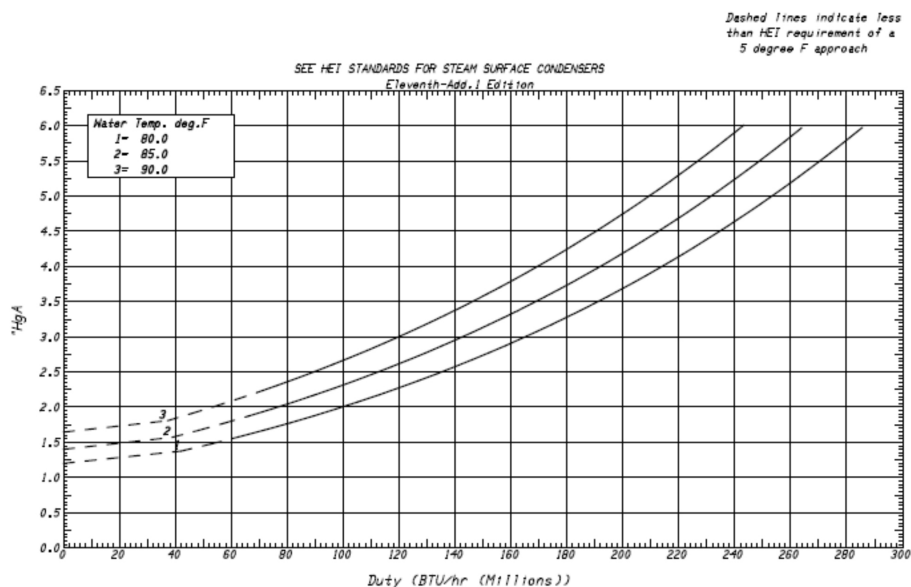
**Figure 3.** Non-divided waterbox with bonnets.



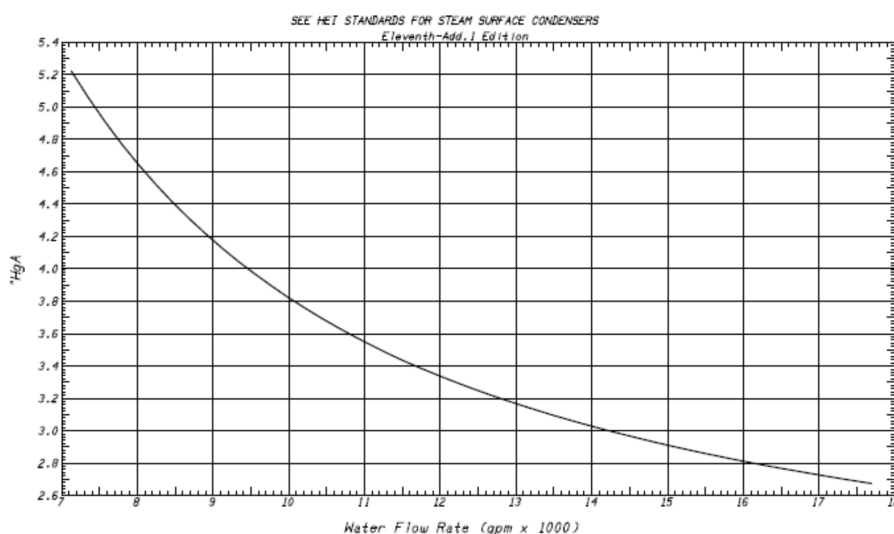
**Figure 4.** Surface condenser with multiple steam inlets with plenum shown in bright green.



**Figure 5.** 3D model showing ejector system mounted on a surface condenser.



**Figure 6.** Condenser inlet pressure versus duty with varying cooling water inlet temperature.



**Figure 7.** Condenser inlet pressure versus cooling water flowrate.

confined joint (TEMA 'R'), tube hole grooves, tube projection and expansion requirements, clad tubesheet requirements, minimum thicknesses, pass partition groove requirements, pipe tap connection requirements, instrument connection requirements, and minimum bolt sizes.

Some caution when adding requirements to the surface condenser is the material selected. For thinner gauge tubes, the typical tube hole grooves are not available. In addition, applying the API 660 standard adds additional requirements that may not be necessary. Since the steam surface condenser application is in fairly clean service vacuum-operated, API 660 is not applicable as noted in section 1 of the standard. If API 660 is required, the standard may be applied, but some requirements may not be available. For example, API 660 requires special machining for non-welded tubesheet material. API 660 also adds additional capital costs compared to TEMA requirements.

A common option to reduce the number of surface

condensers is to combine multiple turbines discharging into a single condenser (see Figure 4). This option varies from just two turbines entering one condenser to five or more turbines entering one surface condenser. Multiple turbines entering one surface condenser increases the size of the condenser, which, as mentioned above, increases capital cost. In this case, it may be more economical up front on the capital cost for the surface condenser as there is only one condenser with a larger size in lieu of two or more condensers, each at a smaller size. With the condenser having multiple steam inlets, a plenum is utilised to reduce the connections on the surface condenser shell (see Figure 4). The comparison for this option will come down to the installation cost analysis as the steam inlet to turbine exhaust piping, depending on the site layout, may become costly. In addition, thought should be given to the amount of turbines entering one condenser as the condenser then becomes the device controlling the plant. Any fouling in the condenser results in worse operating pressures on all the turbines connected. This requires all turbines to be in operation or all shut down if maintenance is required for one turbine, unless large isolation valves are included on all of the steam inlets to turbine exhaust connections, which are costly to install. If one turbine is out of operation and not isolated, the air leakage rate would increase and more often than not overload the venting equipment.

## Air removal equipment



Surface condensers are normally vented by an ejector system or liquid ring vacuum pump. The type and configuration of this critical piece of equipment has a direct impact on the capital cost. Given the critical nature of the venting equipment, a 200% air removal system for redundancy is often selected in this application. While this adds to the capital cost of the system, it is generally insignificant compared against the cost of a shutdown. Liquid ring vacuum pumps have higher capital and maintenance costs when compared to ejector systems. The site requirements and local utility costs will highly influence which type of venting system is selected. In addition to the equipment, the option of mounting the ejector package onto the condenser has an added benefit and reduced installation costs. However, specifying a complete package does add to the capital cost. Figure 5 shows a 3D model showing an ejector system mounted on a surface condenser.

## What affects reliable performance?

### Fouling or plugging of the tubes

When the surface condenser begins to foul and the tubes are to be plugged, the condenser thermal design needs to be re-evaluated along with the other conditions on the unit. As mentioned above, condensers are typically designed with an 85% cleanliness factor that allows for some fouling. As the condenser begins fouling and uses the extra area, the condenser performance is affected. The tube velocity will increase if the same cooling water flowrate is used. The increase in tube velocity will need to be verified based on the material being used to ensure it is acceptable. The pressure drop across the cooling water side will increase. Using the same examples in Table 1 for the two pass 0.75 in. SS tubes with 5% tubes plugged, the cooling water pressure drop increases to 15.75 psi, and  $U_d$  increases to 679, resulting in an 87% cleanliness factor based on 10.5 ft/sec. tube velocity. Changing the example to 10% tubes plugged, the pressure drop increases to 17.25 psi, and  $U_d$  increases to 716, resulting in a 90% cleanliness factor based on 11.1 ft/sec. tube velocity. If this were to continue, even at the 10% tube plugged example, a reduced cooling water flowrate should be considered depending on what the site utilities may handle and recommended tube velocities.

### Warmer than design cooling water temperatures

In the design of the surface condenser, the warmest cooling water inlet temperature the surface condenser will see at the design steam flow should be known and accounted for. Shown in the equation earlier,  $F_w$  is the factor based on the inlet cooling water temperature in calculating the heat transfer coefficient. When the cooling water inlet temperature is changed, the inlet pressure of the condenser is affected. Figure 6 shows the effect as the cooling water temperature increases, the inlet pressure increases to maintain the condenser duty. In order to maintain the same pressure, the condenser will have to run at a reduced load. For example, at 3 in. HgA inlet pressure, at 85°F the duty is 142.5 million Btu/hr. When the cooling water temperature is increased to 90°F, to maintain the 3 in. HgA, the duty decreases to 120 million Btu/hr. This greatly impacts the condenser performance and should be carefully looked at in the field and upfront design.

### Low cooling water flowrates

As the cooling water flowrate decreases, the pressure of the surface condenser increases, as shown in Figure 7. For example, the cooling water decreases from 14 250 gal./min. at 3 in. HgA to 12 750 gal./min., then the inlet pressure increases to approximately 3.2 in. HgA. In addition to the inlet pressure of the surface condenser being affected, the tube velocity decreases, which may cause additional fouling if the velocity falls outside the recommended velocity for the tube material. The pressure drop decreases, which may be considered a benefit, but the cooling water temperature rises. If the cooling water outlet temperature increases enough, it may affect downstream equipment. When the cooling water temperature increases, the log mean temperature difference (LMTD) is also affected, which directly affects the area required, which is calculated as  $Q$  divided by ( $U_d$  times LMTD).

### High turbine exhaust flowrates

The basic equation for the duty of the surface condenser is steam flow times the change in latent heat. When the steam condensed increases, the duty will increase. As the duty of the surface condenser increases, the inlet pressure of the condenser increases, as shown in Figure 6. In addition, the velocity of the steam coming into the condenser increases and will need to be verified to ensure excessive velocities will not affect the condenser.

### Air removal equipment

Surface condenser performance is hindered by non-condensable build-up inside of a condenser. In sub-atmospheric service, air will inevitably leak into a condenser through flanges, connections, turbine seals, and with the incoming steam. Even though these flows are normally small, they are cumulative and build up over time. This air negatively impacts the condenser performance by blanketing the tubes and displacing surface area inside of the condenser. The air removal system continually vents the air leaking into the equipment. If the venting equipment is unable to vent the air at the operation pressure of the condenser, the condenser pressure will degrade until the air can be removed. The venting system does not control the main condenser pressure, but can limit it if the air leakage is higher than design or if the venting equipment is not functioning properly. A few major items that will impact the performance of the venting equipment are air leaks, off design motive conditions, and wear on the equipment degrading the performance.

## Conclusion

The capital costs for turbine exhaust steam surface condensers are impacted in many ways at the initial design phase of a project. The surface condenser additional surface area, material requirements, specific design factors, optional specifications, and air removal package design will need to be carefully evaluated to determine the benefits of each item versus the associated capital costs. Once the equipment is in service, understanding the impacts of performance is important to maintain ideal operation. Fouling or plugging of tubes, cooling water temperatures and flowrates, turbine exhaust flowrates, and the air removal equipment are the main sources impacting performance. Ultimately, careful review of the equipment is essential in all phases of the project and operation to optimise performance and a good manufacturer of this equipment will have the resources available and the understanding to discuss these items at any time. **WF**