picking up STEAM

Jim Lines, Graham Corp., USA, outlines how a solid understanding of best practices for specifying, procuring, and operating steam surface condensers, as well as good maintenance and performance evaluation practices, can help fertilizer plants to achieve their operating objectives.

Ammonia/urea and nitrogen fertilizer production plants have various centrifugal services that require on-specification performance to achieve plant production capacity. Owing to the ammonia process internal heat generation and heat recovery integration, steam is generated within the synthesis process, providing an energy source to economically drive steam turbines that in turn, power compressors or large pumps. Steam surface condensers support turbine drivers by establishing subatmospheric pressure at the turbine discharge, in order to maximise energy recovery from the steam internally generated within the ammonia synthesis process. When surface condenser performance is below specification,

Figure 1. Steam surface condenser in a 2300 tpd KBR nitrogen plant on the US Gulf Coast.

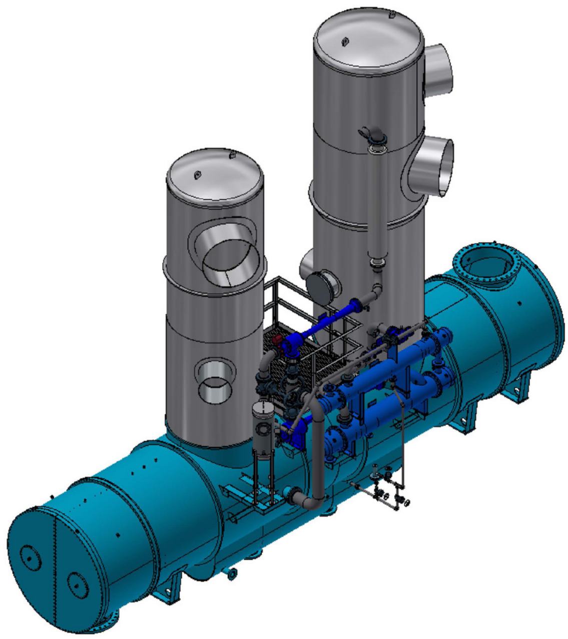

Figure 2. Ammonia plant steam surface condenser with plenum chambers receiving multiple steam turbine exhausts.

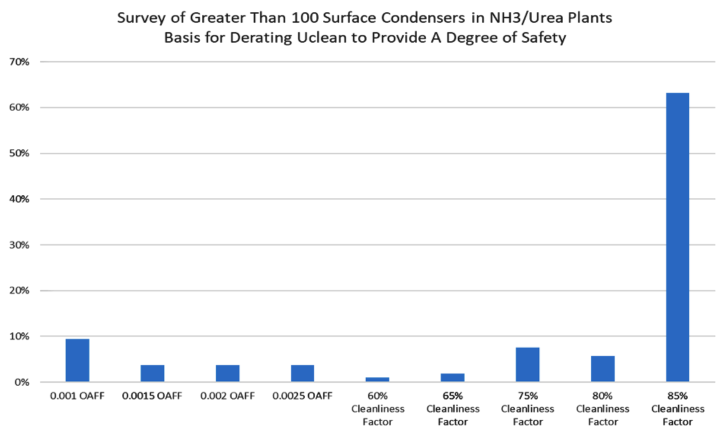

Figure 3. Common approaches to steam surface condenser design safety factor.

specifically elevated turbine discharge pressure, this reduces the power generated by the steam turbine and consequently lowers the capacity or discharge capability of a compressor linked to the condensing turbine.[1]

Steam surface condensers are an accessory to the compressor-turbine driver unit, however, they are critical to their reliable and on-specification performance. Having a solid understanding of best practices for specifying, procuring, and operating steam surface condensers along with good maintenance and performance evaluation practices after installation will aid in fertilizer plants achieving operating objectives.

Compressor and pump services with steam turbine drivers

There are various compressor and pump services within an ammonia/urea complex driven by condensing turbines. They are all critical services. In an ammonia/nitrogen plant, common services include:

- SYNGAS compressor.
- Process air compressor.
- Ammonia refrigerant compressor.
- Cooling/boiler feed water pump.

In a urea plant, common service includes:

- A CO_2 compressor.

Equipment layout can vary from plant to plant. There are plants where a common condenser services multiple steam turbines. Other plants have a condenser for each steam turbine. Figure 1 illustrates a 2300 tpd ammonia plant, in which a large single steam surface condenser receives exhaust from five different turbines – the SYNGAS, CO_2, process air, refrigeration, and boiler feedwater pump turbine drivers. Figure 2 is a 3D CAD drawing illustrating a common steam surface condenser with multiple turbine exhausts entering plenum chambers on the condenser. Also noted is a two-stage ejector package that evacuates air and other non-condensables from the steam surface condenser.

Design and excess surface area

Steam surface condensers are typically designed to Heat Exchange Institute (HEI) standards. The turbine exhaust is steam with a degree of wetness, however, it is relatively clean, flowing on the outside of the tubes, and not subject to fouling. Typically, cooling water is inside the steam surface condenser tubes. The water may be circulated cooling tower water, seawater, river water, secondary loop water or brackish water, depending upon site location and source of cooling water for the steam surface condenser. The water side can be susceptible to fouling.

There are a couple of specific ways that surface condensers are designed in order to provide a measure of safety against overall heat transfer rate

Figure 4. Examples of tubeside fouling.

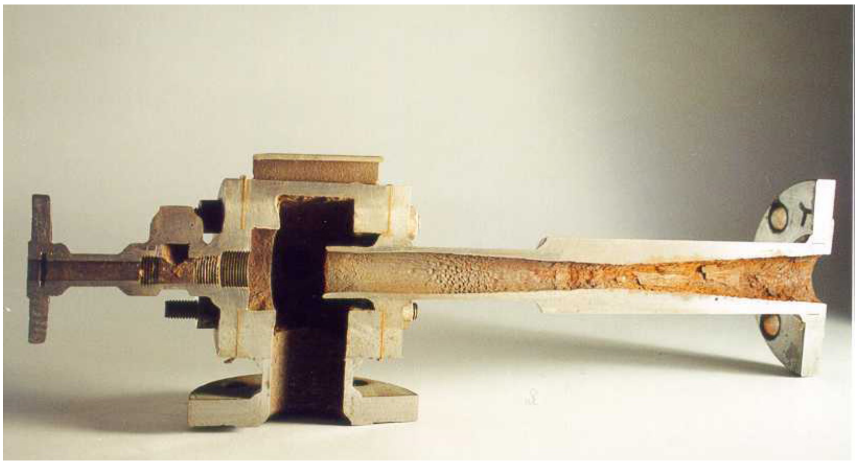

Figure 5. Pitting from wet steam that erodes or pits the inlet diffuser section of an ejector.

degradation from fouling. The most common is to design with a cleanliness factor. For example, if an 85% cleanliness factor is applied, the heat transfer rate is reduced by 15% or is multiplied by 85%. The condenser is designed for a surface area based on 85% of the clean overall heat transfer coefficient as estimated by HEI methods.

U_{design} = U_{clean} x cleanliness factor, or for example, U_{design} = 85% of U_{clean}.

The amount of excess surface area provided by a cleanliness factor is (1/cleanliness factor -1) or 17.6% applying an 85% cleanliness factor.

Alternatively, and less commonly, an overall fouling factor (OAFF) can be specified to derate the clean heat transfer coefficient. If for example, a 0.001 hr ft^2 °F/Btu OAFF is specified:

$$Udesign = \frac{1}{\frac{1}{Uclean} + OAFF}$$

If U_{clean} = 600 (U_{design} = 375), this will correspond to a 62.5% cleanliness factor or 60% excess surface area.

A recent survey of more than 100 steam surface condensers installed in ammonia/urea production plants determined that an 85% cleanliness factor is most common, although other levels of derating U_{clean} are applied. Figure 3 illustrates that greater than 60% of the surveyed installations based their design on an 85% cleanliness factor.

The relationship between cleanliness factor, excess surface, and OAFF varies based on the cooling water velocity. There are vastly varied differences based on tubeside velocity. Table 1 illustrates how cleanliness factor, excess area, and OAFF correlate as tubeside velocity varies. 0.75 in. diameter x 20BWG 304SS tubing is used as a basis for this analysis (Table 1) and HEI methods are applied to determine U_{clean}.

Tubeside fouling

Fouling is an insidious problem, somewhat mitigated by maximising tubeside velocity. The greater the tubeside velocity, the greater the fluid shear stress is at the tube wall. It has been shown that a key variable to reduce actual fouling is high shear stress, mitigating the potential of fouling deposit build up on the tube wall.[2] Fouling can be severe and well beyond design derating assumptions, requiring careful consideration of the water source, potential for debris, and the consequences of getting it wrong.

An 85% cleanliness factor is perfectly acceptable and has achieved outstanding long-term performance, however, there are instances where performance degrades quickly as Figure 4 illustrates.

Consequences of fouling

Fouling impedes heat transfer as deposit builds on the inside of the tube wall. Tube holes may become blocked in certain cases. If actual fouling exceeds the basis used to establish the surface area for a steam surface condenser, then operating pressure will rise. This increase in operating pressure by the surface condenser reduces the energy recovered from the steam by the turbine, thereby lowering the capability of the compressor, which is an undesirable consequence.

The reason that pressure rises as fouling exceeds the design basis for cleanliness, is that the working overall heat transfer rate falls below the U_{design}. For all intents and purposes, the turbine exhaust thermal duty to be rejected is unchanged, the surface area of a condenser is unchanged, and the thermal duty equation must balance Q = $U_{working}$ * Area * LMTD, where 'Area' is the condenser heat exchange surface area.

The following example illustrates why pressure must rise.

$$U_{working} = \frac{Q}{Area * LMTD}$$

If Q and Area are constant, and if $U_{working}$ is reduced, ($U_{working} < U_{design}$), then LMTD must increase. In the end, a condenser will always condense the turbine exhaust; the key variable is the pressure at which this will occur.

For example:

- Turbine exhaust flow rate = 250 000 #/hr (113 400 kg/hr).
- Turbine exhaust enthalpy = 975 Btu/lb (541.7 kcal/hr).
- Thermal duty = 250 000*975 = 243.8 MMBtu/hr.

- Condenser operating pressure for design = 4 in. Hg Abs (steam saturation temperature is 125.4°F) (13.5 kPa/51.9°C).
- Cooling water is flowing at 8 ft/sec, using 0.75 in. x 20 BWG 304 SS tubing (2.44 m/sec).
- Water inlet is 88°F and outlet is 102°F (31.1/38.9°C).
- Applying 85% cleanliness factor.

From Table 1, U_{clean} = 658.8 Btu/hr ft^2°F (3216 kcal/hr m^2°C), U_{design} = 560 Btu/hr ft^2°F (2734 kcal/hr m^2°C), and LMTD = 29.8°F (16.6°C).

The surface condenser has 14 582 ft^2 (1355 m^2) of effective surface area.

Take a case where fouling inside the tubes of the condenser has resulted in the actual working heat transfer coefficient being lowered to 475 Btu/hr ft^2°F (2319 kcal/hr m^2°C) rather than the design basis of 560 Btu/hr ft^2°F. To balance the thermal duty equation, again assuming duty is unchanged and, of course, surface area is unchanged, the LMTD must rise ~18% (560/475-1). Condenser pressure rises from 4 to 4.6 in. Hg Abs (13.5/15.6 kPa) which represents condensing temperatures increasing approximately 5°F (2.8°C) from 125.4°F to 130.6°F (51.9/59.8°C) so as to balance the thermal duty equation with LMTD, correspondingly increasing from 29.8°F to 35.1°F (16.55/19.5°C).

Layout

When a steam surface condenser is close-coupled to a turbine, there is minimal pressure loss between the two pieces of equipment. When, however, there is a long length of piping between the turbine exhaust and surface condenser, as is often the case when multiple turbines exhaust to a common condenser, then hydraulic losses (pressure drop) must be considered. There will be pressure loss between the turbine and condenser, with the condenser at a lower pressure. This lower pressure results in a lower LMTD and, therefore, a large condenser surface area is needed. Table 2 provides a comparison in which a close-coupled arrangement has an LMTD of 29.8°F (16.55°C) while the LMTD is progressively lower, and velocity along the piping between turbine and condenser increases. Piping losses are not trivial in such cases with respect to surface condenser area. An 11% lower LMTD requires 11%

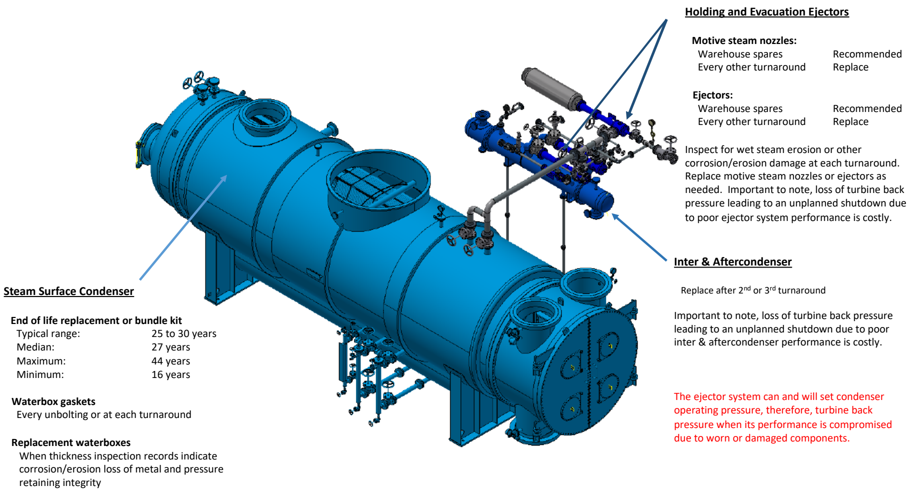

Figure 6. Condenser system components and replacement parts recommendation.

Table 1. Illustration of how cleanliness factor, excess area and OAFF correlate as tubeside velocity varies.

		85% Clean design basis				0.001 hr ft^2°F/BTU OAFF design basis			
Tube velocity	U_{clean}	U_{design}	OAFF	Excess area	Cleanliness factor	U_{design}	OAFF	Excess area	Cleanliness factor
3	403.5	343.0	0.00044	17.6%	85.0%	287.5	0.001	40.3%	71.3%
3.5	435.8	370.4	0.00040	17.6%	85.0%	303.5	0.001	43.6%	69.6%
4	465.9	396.0	0.00038	17.6%	85.0%	317.8	0.001	46.6%	68.2%
4.5	494.1	420.0	0.00036	17.6%	85.0%	330.7	0.001	49.4%	66.9%
5	520.8	442.7	0.00034	17.6%	85.0%	342.5	0.001	52.1%	65.8%
5.5	546.3	464.3	0.00032	17.6%	85.0%	353.3	0.001	54.6%	64.7%
6	570.5	485.0	0.00031	17.6%	85.0%	363.3	0.001	57.1%	63.7%
6.5	593.8	504.8	0.00030	17.6%	85.0%	372.6	0.001	59.4%	62.7%
7	616.2	523.8	0.00029	17.6%	85.0%	381.3	0.001	61.6%	61.9%
7.5	637.9	542.2	0.00028	17.6%	85.0%	389.5	0.001	63.8%	61.1%
8	658.8	560.0	0.00027	17.6%	85.0%	397.2	0.001	65.9%	60.3%
8.5	676.5	575.1	0.00026	17.6%	85.0%	403.5	0.001	67.7%	59.6%
9	693.8	589.7	0.00025	17.6%	85.0%	409.6	0.001	69.4%	59.0%
9.5	710.2	603.7	0.00025	17.6%	85.0%	415.3	0.001	71.0%	58.5%
10	725.7	616.9	0.00024	17.6%	85.0%	420.5	0.001	72.6%	57.9%
ft/sec	BTU/hr ft^2 °F	BTU/hr ft^2°F	hr ft^2°F/BTU	NA	NA	BTU/hr ft^2°F	hr ft^2°F/BTU	NA	NA

Table 2. A comparison in which a close-coupled arrangement has an LMTD of 29.8°F (16.55°C) while the LMTD is progressively lower, and velocity along the piping between turbine and condenser increases.

	Turbine exhaust pressure		Condenser pressure		Condensing temperature		LMTD		
Velocity in piping	Hg Abs	KPa Abs	Hg Abs	KPa Abs	°F	°C	°F	°C	LMTD reduction
300	4	13.5	3.83	13.0	123.8	51.00	28.2	15.7	-5.5%
350	4	13.5	3.78	12.8	123.3	50.72	27.7	15.4	-7.2%
400	4	13.5	3.72	12.6	122.8	50.44	27.2	15.1	-8.9%
450	4	13.5	3.65	12.35	122.1	50.05	26.5	14.7	-11.2%
Close coupled without piping	4	13.5	4	13.5	125.4	51.89	29.85	16.59	0.0%

more surface area. For this case, cooling water measured 88°F (31.1°C) in and 102°F (38.9°C) out. The analysis in Table 2 assumes 100 ft of 36 in. diameter piping between turbine and condenser that includes two 90° elbows.

Maintenance, spare parts, and end of life replacement

Steam surface condensers are static equipment, however, while having long overall lives, they do require maintenance and performance evaluation to ensure an unplanned shutdown is prevented.

A recent analysis of replacement part history and field performance evaluation by Graham Corp. highlighted interesting best practices by the industry. More than 100 surface condensers sold between 1970 and 1995 into the compressor-turbine driver service were evaluated to analyse the spare parts history, the type of components being replaced, and the durations of operation before complete units needed to be replaced or underwent some form of major refurbishment, such as replacement bundle kits.

Most steam surface condensers will utilise two stage ejector systems for evacuating noncondensable gases, which is necessary in order to maintain the desired vacuum level at the turbine discharge. Owing to their operation, and being subjected to supersonic steam velocity with a degree of moisture, ejectors are prone to erosion and/or corrosion. Consequently, ejector replacement motive steam nozzles are frequently procured by the user, as are replacement ejectors. Users may source replacement motive steam nozzles at the time of initial startup to have these critical wear parts at the ready. It was not observed that all users will do this, however, those keen to ensure high on-stream tend to do so. Otherwise, it is observed that ahead of the first or second planned turnaround, replacement motive steam nozzles are procured and installed. If steam wetness is expected, the user will purchase replacement ejectors as well. Figure 5 highlights pitting from wet steam that erodes or pits the inlet diffuser section of an ejector.

The inter and aftercondenser that is part of the ejector system is also prone to wear and there is need for replacement after a period of operation, typically around the second or third turnaround or after 8 – 12 years of operation.

While every operating plant may be different from one another with regard to steam quality, maintenance practices, metallurgy and its resistance to corrosion, or erosion, and overall process unit operation, some commonalty is found regarding equipment replacement trends. Figure 6 highlights the findings and recommendations of Graham Corp.

It is important to note that only about one third of the sample evaluated had a major end of life replacement. There was not data for the other two thirds confirming that such an investment had taken place. It was not possible to confirm all units were still in commercial operation. Also, bearing in mind that surface condensers installed in the 1990s are just coming up to 25 years of operation, they may not have reached a point in time where replacement is warranted.

Performance survey/evaluation

It is recommended that a performance evaluation of a steam surface condenser and its ejector package should be conducted about 12 to 15 months before a planned turnaround. Ideally, conducting a performance evaluation during summer months is best, as this is when cooling water is the warmest and the condenser operates under more extreme conditions. It is desirable to compare this to a baseline or benchmark evaluation that is conducted shortly after full commercial operation. This permits comparing current performance to that benchmark. The original surface condenser supplier may have a technical services or performance improvement team able to conduct each performance survey. Graham Corp. has a performance improvement organisation to aid in the troubleshooting and benchmarking of performance.

Common findings that are root causes for elevated pressure and reduction in compressor capability include:

- Excessive air leakage – the turbine exhaust and steam surface condenser operate under vacuum conditions, thus loose flanges, weld cracks, or open valves, for example, will admit air into the system, causing system pressure to rise and compressor performance to decline.
- Warm cooling water – during the summer months or as a consequence of added heat load to the cooling water system, inlet temperature to a surface condenser may be warmer than anctipated. This will cause an increase in operating pressure and compressor performance to decline.
- Poor ejector system performance – worn steam nozzles and worn ejectors in suboptimal condition will cause system pressure to rise and compromise compressor performance.

- Fouling – tubeside fouling or blockage of tubeside flow will lower the working overall heat transfer rate, causing operating pressure to rise with compressor performance impacted negatively.
- Water circulation – cooling water flowrate to a surface condenser may be less than designed. This increases the temperature of the cooling water, effectively lowering the LMTD, and also lowering the overall heat transfer coefficient, which causes condenser pressure to rise. Again, compressor performance is impacted negatively.
- Excess turbine exhaust flow – as plants attempt to push higher plant operating capacity, steam flow to a turbine may be elevated above design specifications. This increases the thermal duty to a condenser and can cause the operating pressure to rise in response.
- Non-OEM replacement equipment – when shell and tube or aftermarket service companies replace main condensers or the inter and aftercondenser, it has been common to find system performance is poor, owing to excessive pressure drop or inability to properly evacuate noncondensable gases from a condenser. This leads to elevated operating pressure and losses in compressor capability.

In the above examples, the thermal duty equation, $Q = U_{working} * Area * LMTD$ is at play. If $U_{working}$ lowers due to inadequate cooling water flow, pressure rises to increase LMTD in order to balance the thermal duty equation. The same occurs as a result of tubeside fouling. Likewise, if turbine exhaust flow is above design specifications, the thermal duty increases. The condenser's response is that pressure must rise in order to elevate LMTD and balance the thermal duty equation.

Conclusion

Steam surface condensers are important accessories to the compressor-turbine driver. Often, they are classified as critical equipment, as their performance is imperative to proper process plant performance. Much attention and focus is placed around compressor and steam turbine selection. The same care and detailed evaluation are needed for steam surface condensers that support a steam turbine driver. Failure to get correct surface condenser integration into the compression system will lead to loss of plant throughput capacity or a decline in product quality. For optimal on-stream performance, surface condenser performance should be evaluated routinely against a baseline, and the original equipment supplier should assess its performance.

Keeping the surface condenser system – main condenser, ejectors, and inter/aftercondenser – in good health with routine maintenance, via replacement of typical wear parts, is vital. An unplanned shutdown to address worn ejectors or a poorly performing inter/aftercondenser is an expensive proposition. The cost of effective health and well-being of a surface condenser and venting system is a drop in the ocean compared to cost of unplanned shutdown or outage caused by the need to replace worn steam nozzles, ejectors or inter/aftercondensers. WF

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