THE MEASUREMENT OF CONDENSER LOSSES DUE TO FOULING AND THOSE DUE TO AIR INGRESS

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Abstract

The performance of a steam surface condenser is negatively affected not only by tube fouling but also by the presence of excessive amounts of air within the shell side of the condenser. However, because they have similar effects, it has been the custom to lump their costs together. Clearly, if the contribution of each to operating cost could be estimated with some confidence, then maintenance decisions would become focused on whether it is more important to clean the condenser or to remove the source of an air in-leakage.

There are two general methods for distinguishing between these two sources of condenser performance degradation: (a) measure the resistance due to fouling and subtract this from the total increase in tube thermal resistance to obtain the increase due to air ingress and (b) estimate the change in tube thermal resistance due to air ingress and subtract this from the total increase in thermal resistance to obtain the increase due to fouling.

For both methods, the frame of reference is an estimate of the *total* apparent increase in single-tube heat transfer coefficient resulting from these two effects. This can be obtained by calculating the present single-tube heat transfer coefficient using the standard Fourier equation, and comparing it with the single-tube U-coefficient for a clean condenser operating under the same load and cooling water conditions. In both cases, the condenser Performance Factor needs to be applied.

One approach to measurement method (a) is outlined in the new ASME Power Test Code PTC.12.2-1998. A variation of this method is embodied in an EPRI/Bridger Scientific report in which the flow through one of the tubes in each pair is also measured.

Measurement method (b) involves calibrating the degradation in performance due to air ingress by injecting known quantities of air or nitrogen and also noting the reading of the flow meter measuring the air removal rate. Subsequently the flow meter can be used to infer the degradation due to air ingress based on the change in air removal rate. Interference with the precision of these methods from “air binding” and “zones of stagnation” are also discussed.

Introduction

Tube fouling as well as the presence of excessive amounts of air within the shell side of the condenser both negatively affect the performance of a steam surface condenser. The thermal resistance due to fouling reduces the overall tube heat transfer coefficient. Similarly, the presence of air either reduces the shell side film heat transfer coefficient or, by completely blanketing tubes in certain regions of the tube bundle, reduces the heat transfer area as suggested by Harpster(10). Unfortunately, because they have similar effects, their cost, measured as the increase in duty over that of a clean condenser, have normally been lumped together(10). Clearly, if the contribution of each to operating cost could be estimated with some confidence, maintenance decisions would become focused on whether it is more important to clean the condenser or to remove the source of an air in-leakage, or both.

To distinguish between the two sources of condenser performance degradation there are two general methods: (a) measure the resistance due to fouling and subtract this from the total increase in tube thermal resistance to obtain the increase due to air ingress and (b) estimate the change in tube thermal resistance due to air ingress and subtract this from the total increase in thermal resistance to obtain the increase due to fouling. Unfortunately, the task has been hampered until recently by a lack of suitable instrumentation. The equipment associated with these two methods, and now available, will be described; as well as the way the data is processed in order to quantify separately the effect of each of these causes of condenser performance degradation.

Condenser Performance Monitoring

Approaches (a) and (b) both require that the current increase in the single-tube thermal resistance due to the combined effects of fouling and air ingress be known. This may be obtained by first calculating the *clean* single-tube heat transfer coefficient U_{ref}, which is a function of the sum of the wall thermal resistance and the water and steam side film resistances, these being based on the current operating conditions of cooling water inlet and outlet temperatures, tube water velocity and back pressure. This becomes the reference. The principles involved were included in both recent editions of the ASME Power Test Codes for Condensers(1,2).

To obtain the equivalent single-tube value for the *fouled* conditions, the current value of the effective heat transfer coefficient U_{ref} is calculated using the well-known Fourier equation, as outlined below, this then being modified by the value of the Performance Factor(3) that corresponds to the present value of generated power. The increase in single-tube thermal resistance may then be obtained by subtracting the fouled value from the clean value.

Clean Single-tube Heat Transfer Coefficient - U_{ref}

The thermal resistance to heat transfer of a *clean* tube consists of three major components:

i. Tube wall resistance
ii. Water side film resistance
iii. Shell-side film resistance

Note that both (i) and (ii) are referred to the outer tube surface.

i. Tube Wall Resistance

The thermal resistance of the tube wall (R_w) is calculated using the Kern[4] relationship:

$$R_w = \frac{d_o}{24k_m}\ln\left[\frac{d_o}{d_i}\right] \qquad (1)$$

ii. Water-side Film Resistance

The value of the water side film thermal resistance (R_t) is calculated using the Rabas-Cane correlation:[5]

$$R_t = 0.0450357\left[\frac{\mu^{0.373}}{\rho^{0.835}k^{0.538}C_p^{\ 0.462}}\right]\left[\frac{d_i^{\ 0.165}}{v^{0.835}}\right]\left[\frac{d_o}{d_i}\right] \qquad (2)$$

iii. Shell-side Film Resistance

The Nusselt factor (h_f) is the condensate film heat transfer coefficient calculated from the properties of water at the saturation temperature that corresponds to the compartmental backpressure, the Nusselt equation being:

$$h_f = 0.725\left[\frac{k_f^{\ 3}\rho^2 g\ \lambda}{\mu_f\ D_o\ (\Delta T)}\right]^{0.25} \qquad (3)$$

The shell side thermal resistance is1/h_f.

Overall Heat Transfer Coefficient - U_{ref}

The overall heat transfer coefficient for a clean tube R_{ref} can be calculated from the values of resistances R_w and R_t and Nusselt factor h_f as follows:

$$U_{ref} = \frac{1}{R_w + R_t + 1/h_f} \qquad (4)$$

Effective Condenser Heat Transfer Coefficient - U_{eff}

A steam surface condenser used in the Rankine Cycle is essentially a cross-flow heat exchanger. The effective heat transfer coefficient (U_{eff}) is computed from present steam and water temperatures and cooling water flow rate and, by rearranging the well-known Fourier equation for heat transfer, can be calculated from:

$$U_{eff} = \frac{Q}{A*LMTD} \qquad (5)$$

in which

$$LMTD = \frac{T_{out} - T_{in}}{\ln\left(\frac{T_v - T_{in}}{T_v - T_{out}}\right)}$$

To calculate an accurate value of U_{eff} requires knowledge of cooling water flow rate, representative values of the inlet and outlet water temperatures, together with the compartmental backpressure. For multi-compartment condensers, especially those operating at different back pressures, this set of information is required for each compartment.

Performance Factor - PF

The design data sheet for a condenser designed in accordance with the Standards published by the Heat Exchange Institute[(6)] contains information from which the HEI tube bundle heat transfer coefficient U_{HEI} may be calculated. It also contains information to calculate the value of the effective heat transfer coefficient U_{eff} as well as the design cleanliness factor CF_{des}, this being:

$$CF_{des} = \frac{100 * U_{eff}}{U_{HEI}} \qquad (6)$$

Thus the design cleanliness factor may be thought of as discounting the design HEI tube bundle U-coefficient so as to reflect the operating conditions that will be experienced in practice.

When the thermal resistance method is used to calculate the single-tube heat transfer coefficient, Tsou[(3)] recommends that the term *performance factor* be used in place of cleanliness factor. Thus, Performance Factor may be calculated from:

$$PF = 100 \frac{U_{eff}}{U_{ref}} \qquad (7)$$

A condenser designed with a cleanliness factor of 85% based on the Heat Exchange Institute method[(6)] will have an equivalent performance factor of about 74% when the thermal resistance method is used as the reference value. It should also be noted that the performance factor has been found by Putman and Karg[(7)] to vary linearly with load, or:

$$PF = a_0 + a_1 MW \qquad (8)$$

This must be taken into account when evaluating the condenser performance at partial load. A typical plot of Performance Factor vs. load is shown in Figure 1.0 and it is necessary to establish the relationship between Performance Factor (PF) and load *when the condenser is clean*. To develop equation (8), the unit is run for an hour at steady state at various loads over the normal operating load range and the Performance Factor calculated for each case. The essentially straight-line relationship of equation (8) can then be obtained using regression analysis. Note that, although displaced from one another, both cleanliness factor and performance factor follow similar linear relationships with respect to generated power and have similar slopes.

Condenser Fouling Factor

Fouling factor R_{fc} has been defined as the thermal resistance which can be attributed to fouling and, when testing a single tube in a heat transfer rig, is calculated from the overall tube heat transfer coefficient U_{tot} from an expanded version of equation (4) as follows:

$$U_{tot} = \frac{1}{R_w + R_t + 1/h_f + R_{fc}} \qquad (9)$$

To calculate condenser fouling resistance from the condenser effective heat transfer coefficient U_{eff}, it is necessary to perform the following transformation in order to convert the value of U_{eff} to the reference conditions:

$$U_{tot} = \frac{100 * U_{eff}}{PF} = \frac{1}{R_w + R_t + \frac{1}{h_f} + R_{fc}} \tag{10}$$

from which

$$R_{fc} = \frac{PF}{100 * U_{eff}} - \left[R_w + R_t + \frac{1}{h_f} \right] \tag{11}$$

or

$$R_{fc} = \frac{PF}{100 * U_{eff}} - \frac{1}{U_{ref}} \tag{12}$$

where U_{eff} is now a function of both fouling and excessive air ingress (if any) and following historical practice, may be calculated using equation (5).

Thus the condenser fouling resistance R_{fc} with reference to the clean single tube conditions can be calculated from the condenser effective heat transfer coefficient modified by the performance factor; minus the reciprocal of U_{ref}.

If air is present having an air resistance R_a and there is a tube fouling resistance R_f, then the total correctable resistance R_{fc} may be written as:

$$R_{fc} = R_a + R_f$$

Clearly, if R_{fc} and R_f are known, then: $R_a = R_{fc} - R_f$ (13)

Alternatively, if R_{fc} and R_a are known, then: $R_f = R_{fc} - R_a$ (14)

Methods of Measuring Fouling Resistance Alone (On a Tube Basis)

It is clear that the value of R_{fc} includes the effect on heat transfer not only of fouling but also of any air ingress. To distinguish between these two effects quantitatively, approach (a) above would measure the fouling resistance directly in some manner and subtract this from R_{fc} in order to calculate the resistance that can be directly attributed to air ingress. Two methods for estimating resistance due to fouling alone are available. The first of these methods was developed as a part of the new ASME Performance Test Code for Steam Surface Condensers[(2)] and the other as an instrumentation system developed under the auspices of EPRI in conjunction with Bridger Scientific[(8)]. However, these methods alone could lead to error if the condenser model described by Harpster [(13)], to be discussed in a later section, is not properly taken into consideration.

ASME Method for Estimating Tube Fouling Resistance

The Foreword to the new ASME Performance Test Code on Steam Surface Condensers[(2)] states that “to be certain that condenser performance results are not predestined, a mandatory cleanliness test is now required by this Code.” The method is illustrated in Figure 2.0, in which the inlet and outlet temperatures are measured on sets of two adjacent tubes. One of the tubes in each set remains in the as-found fouled condition while the neighboring tube has either been cleaned or replaced with a new tube. It is claimed that both tubes in the pair experience identical heat transfer dependency on steam pressures, cooling water flow rates and velocities. The latter two may, however, be questionable in view of one tube containing water side fouling.

The Code suggests that the number of pairs of tubes selected for the fouling test be one per 2000 tubes per tube bundle: but not fewer than four pairs or more than 16 pairs per bundle. The pairs are to be located at the approximate centroids of equal tube sectors within the tube bundle pattern. However, pairs should not be placed within three tube rows of the bundle periphery.

To perform the fouling resistance test, not only are the outlet water temperatures to be measured on each tube pair but also the common shell pressure, together with the cooling water inlet temperature and flow. The fouling resistance of the pair in a single-compartment, which may have one or two passes, is calculated as the difference between the heat transfer coefficients for each tube in a pair, thus:

$$R_f = \frac{1}{U_{foul}} - \frac{1}{U_{clean}} \qquad (15)$$

Expanding and rearranging the Fourier equation (5) reduces to:

$$R_f = \frac{A}{nwC_p}\left[\frac{1}{\ln\left(\frac{T_v - T_{in,\,f}}{T_v - T_{out,\,f}}\right)} - \frac{1}{\ln\left(\frac{T_v - T_{in,\,c}}{T_v - T_{out,\,c}}\right)}\right] \qquad (16)$$

For 'j' pairs of tubes, the mean fouling resistance R_{fmean} for the condenser tubes is given by:

$$R_{fmean} = \frac{1}{j}\sum_{i=1}^{i=j} R_{fi} \qquad (17)$$

As an on-line method for establishing fouling resistance for a short time after the one tube in each pair has been cleaned, the method would seem to have possibilities. Further, it has been assumed that the fouling resistance so calculated will not include the effects of any air ingress, since the same effect will apply to both tubes in the pair. However, any fouling of the reference (clean) tube will corrupt the values, as will errors in the estimation of cooling water mass flow rate. Further, by measuring the inlet water temperature at only one point, any water stratification will not be reflected in the results. However, inlet waterbox stratification is not anticipated to have error within the same magnitude expected for outlet temperature measurements. This is because temperature sensor measurements are strongly dependent on their location in the radial thermal gradients of exiting water from individual tubes. Clearly, some may believe that a more accurate result would also be obtained if the shell pressure adjacent to each pair were to be measured rather than, again, measuring it at only a common point.

EPRI/Bridger Scientific Method for Estimating Tube Fouling Resistance

The principle of the method developed by Bridger Scientific under EPRI sponsorship for estimating tube fouling resistance is illustrated in Figure 3.0. Tube pairs are again used but, in this case, one of the pair is a tube with blanked off ends through which no water flows: while the other, the fouled tube, not only has sensitive temperature measuring devices at both ends of the tube but is also provided with a turbine or ultrasonic type flow meter for accurate measurement of the water flow rate through the tube. The blanked off tube is used to measure the mean shell temperature in the vicinity of the fouled tube so that any vapor pressure loss through the tube bundles can be allowed for. Several pairs of tubes are placed strategically throughout the tube bundle(s) so that a mean fouling resistance can be estimated.

The interpretation of the data begins with substituting the water flow, water inlet and outlet temperatures and steam temperature in the Fourier equation stated above in equation (5) and calculating the effective heat transfer coefficient U_{eff}. After having calculated the values of R_w, R_t and h_f for the known tube operating conditions, the value of the fouling resistance R_f can be computed using equation (15). The mean fouling resistance R_{fmean} can then be computed from equation (17).

This apparatus avoids some of the criticisms that can be leveled at the method outlined in PTC.12.2-1998[2]. The water flow and temperature rise can be accurately measured: while the blank tube allows the vapor temperature in the locality of the fouled tube also to be measured with precision. Unfortunately, the cost of the apparatus and its computer and instrumentation system can be high but some economies might be possible if the calculations were executed within the data acquisition system for the unit being monitored.

Calculating Fouling Resistance due to Air Ingress Using a Fouling Monitor

Once the values of R_{fc} and R_{fmean} are known, the apparent fouling resistance due to air ingress may be calculated from:

$$R_a = R_{fc} - R_{fmean} \qquad (18)$$

The Measurement of Thermal Resistance Due to Air Ingress

It has been shown, using a model and theory by Harpster[10, 11], that air does not get trapped on tubes throughout the condenser but forms a steam rich region and an air rich region within the tube bundles. Referring to figure 4.0, the first of these regions is referred to as the “Steam Wind” (SW) region and the second is referred to as the “Stagnant” (S) region[10]. It was also shown, by Henderson, that if the mass ratio of steam vapor to air on a condensing tube is greater than 300 the heat transfer coefficient is greater than 90% of its U_{foul} condition. If this ratio falls below 3, then the heat transfer coefficient will fall below 10% of its U_{foul} value. In operating condensers, the mass ratio can vary, practically, between 50,000 to 0.2 depending on location in the tube bundle.[11]

In a normal operating condenser having an exhauster removing air in equilibrium with the in-leakage, sufficient to prevent condenser excess back pressure, the fraction of tubes essentially unaffected by this air is greater than 96%. Further, air in-leakage nearly 6 times the above pump capacity value, sufficient to cause an additional 0.9"HgA in the measured condenser pressure, will still have nearly 50% of its tubes, located in the outer regions of the tube bundle unaffected in their measured heat transfer coefficient because the mass ratio in this region is greater than 1,000. These results are provided in the referenced literature.[11]

Another perturbing feature about condensers explained by Harpster[12, 13] is air binding. This phenomena results from condenser design that promotes steam flow completely surrounding a tube bundle subsection and having no escape for scavenged air. Although these problems can be overcome by design (patent pending) unawareness of their affects can give rise to a lack of comfort regarding utilization of recorded data.

For these reasons, measurements using tube pairs should consider the above model result, regardless of the array pattern recommended by the ASME test code. Figure 4 shows what might be expected in a typical single shell, single pass condenser. The tube bundle consists of three subsections separated by a vertical crevice between Subsection I and II where the air removal section (ARS) vent line is placed. Another separation is caused by a horizontal condensate tray above Subsection III. The condensate tray provides drainage of the condensate from the above subsections to the sides of the tube bundle, from which condensate is allowed to fall into the hotwell. These trays prevent inundation of tubes in the lower part of Subsection III.

The anticipated air bound (AB) regions, which, out of necessity grow in size and then collapse, are shown in the tube bundle subsections at their most likely locations considering the bundle configuration. Also shown is the "S" region having a high concentration of air. This "S" region near the ARS is variable in size with air in-leakage and can be changed by admitting more or less air (or an inert noncondensable gas like N_2) at fixed flow rates into the condenser. Between each adjustment, about 20 minutes is needed to establish equilibrium. A typical relationship between the apparent increase in fouling resistance due to air ingress and the air removal rate is shown in Figure 5.0, the data being derived from the back pressure vs air in-leakage presented by Harpster[10,11].

Suggested tube pair areas to measure the effects of tube fouling without the impact of air are shown in Figure 4 as rectangular areas containing an "X" and labeled "SW" indicating these areas are in the "Steam Wind" region of the tube bundle.

Suggested tube pair areas for measuring the combined heat transfer coefficient of air and fouling or just fouling are shown as square areas with an "X". These areas for a tight condenser i.e., for air in-leakage below the excess back pressure threshold, will measure the effect of tube fouling only. With the addition of noncondensables, the stagnant zone will expand into these areas and the pairs will record the effects of increased air concentration on the heat transfer coefficient of these tubes.

Baselining and Controlling Air In-leakage

Following installation of the tube pairs the amount of background air in-leakage, water vapor to air mass ratio and exhauster capacity for noncondensables are to be measured. These are easily determined using a Multi-Sensor Probe (MSP) measurement system, shown in Figure 6.0. This system permits simultaneous measurement of air and water vapor flowing from the condenser in each vent line penetrating the shell. Typically, there are more than one shell and these may have the same or different pressures. Each vent line must be measured independently for air in-leakage, since the amount of noncondensables are generally different in each line.

The total amount of background air should be well below the exhauster capacity and the condenser pressure at its pressure saturation value. If not, a leak search should be made and leaks repaired before starting tests. Under this condition all tube pairs should be measured as a baseline. It is expected that all pairs should have the same determined U_{foul} values. If not these values will serve as a bases for determining changes in measured heat transfer coefficients $U_{foul,air}$ upon introduction of air.

Air may be introduced into the condenser at any convenient location. It should be recognized that this air will be scavenged by the steam to the closest ARS section. If uniform effects are desired the air should be introduced on the turbine floor near the LP turbine exhaust annulus.

A convenient means to introduce air, or other gas, is to pass it through a rotameter calibrated for atmospheric pressure at its inlet. If air is used, a control valve in the line between the top of the rotameter and the shell is all that is needed. The plant air being drawn in through the bottom opening of the rotameter should be free of steam and large amounts of dust or dirt to prevent error in readings.

Estimating Tube Fouling Resistance from Air Removal Rate

Once the condenser has been calibrated to provide an estimate of the apparent increase in combined fouling resistance due to air ingress R_a, the current value of R_a may be obtained from the plot shown in Figure 5, the thermal resistance due to tube fouling then being estimated from:

$$R_f = R_{fc} - R_a \qquad (19)$$

Distribution of Condenser Losses

Using a Newton-Raphson model of the condenser/turbogenerator subsystem, Putman and Saxon(9) showed how total condenser losses in MBTU/h (*Loss*) can be calculated from the present condenser duty minus the condenser duty calculated if the condenser were clean and operating under the same cooling water inlet temperature and flow conditions and the same generated power. If the fuel cost is *$Cost* ($/MBTU), then the distribution of these losses between fouling and air ingress can be accomplished as follows:

$ Cost of Fouling: $$Loss_f = \$Cost * Loss \frac{R_f}{R_{fc}} \tag{20}$$

$ Cost of Air Ingress: $$Loss_a = \$Cost * Loss \frac{R_a}{R_{fc}} \tag{21}$$

Conclusions

Tube fouling and air ingress have a similar effect on condenser performance degradation. However, methods for distinguishing between these two causes of performance degradation have been restricted by the absence of suitable instrumentation. Three methods for quantifying the contribution of these two sources of performance degradation are outlined as well as how they can be converted to the equivalent economic loss.

NOMENCLATURE

a_0, a_1	=	Constants in equation (8)	
A	=	Total tube surface area for compartment	ft^2
C_p	=	Specific heat of water	BTU/(lb.°F)
CF	=	Cleanliness factor	%
d_i	=	Inside diameter of condenser tubes	inches
d_o	=	Outside diameter of condenser tubes	inches
D_o	=	Outside diameter of condenser tubes	feet
g	=	Acceleration due to gravity	
		= 417*10E+06 (ft-lb mass) / (h.h.lb force)	
h_f	=	Nusselt condensing film conductance	BTU/(ft^2.h.°F)
k	=	Thermal conductivity of cooling water	BTU/(ft^2.h.°F)
k_f	=	Thermal conductivity of condensate film	BTU/(ft^2.h.°F)
k_m	=	Thermal conductivity of tube material	BTU/(ft^2.h.°F)
LMTD	=	Log mean temperature difference	°F
MW	=	Generated power	MW
n	=	Number of passes in compartment	
Q	=	Heat transfer rate to cooling water	BTU/h
PF	=	Performance Factor	%
R_a	=	Thermal resistance attributed to air ingress	°F/(BTU/(ft^2.h))
R_f	=	Thermal resistance due to fouling for tube pairs	°F/(BTU/(ft^2.h))
R_{fc}	=	Increase in thermal resistance due to both fouling and air ingress	°F/(BTU/(ft^2.h))
R_{fmean}	=	Mean fouled thermal resistance of tube pairs	°F/(BTU/(ft^2.h))
R_t	=	Thermal resistance of cooling water film	°F/(BTU/(ft^2.h))
R_w	=	Thermal resistance of tube wall	°F/(BTU/(ft^2.h))
T_{in}	=	Cooling water inlet temperature	°F
T_{out}	=	Cooling water outlet temperature	°F
T_v	=	Vapor saturation temperature	°F
U_{clean}	=	Heat transfer coefficient calculated from clean tube pairs	BTU/(ft^2.h.°F)
U_{eff}	=	Overall condenser effective heat transfer coefficient	BTU/(ft^2.h.°F)
U_{foul}	=	Heat transfer coefficient calculated from fouled tube pairs	BTU/(ft^2.h.°F)
U_{HEI}	=	Heat transfer coefficient based on HEI tube bundle value	BTU/(ft^2.h.°F)
U_{ref}	=	Reference heat transfer coefficient based on	
		sum of clean-tube thermal resistances	BTU/(ft^2.h.°F)
v	=	Cooling water velocity	ft/s
w	=	Mass cooling water flow through compartment water boxes	lb/h
ΔT	=	Temperature gradient across condensate film	°F
λ	=	Latent heat of condensate	BTU/lb
μ	=	Viscosity of cooling water	lb/(h.ft)
μ_f	=	Viscosity at condensate film temperature	lb/(h.ft)
ρ	=	Density of cooling water	lb/ft^3
ρ_f	=	Density of condensate film	lb/ft^3

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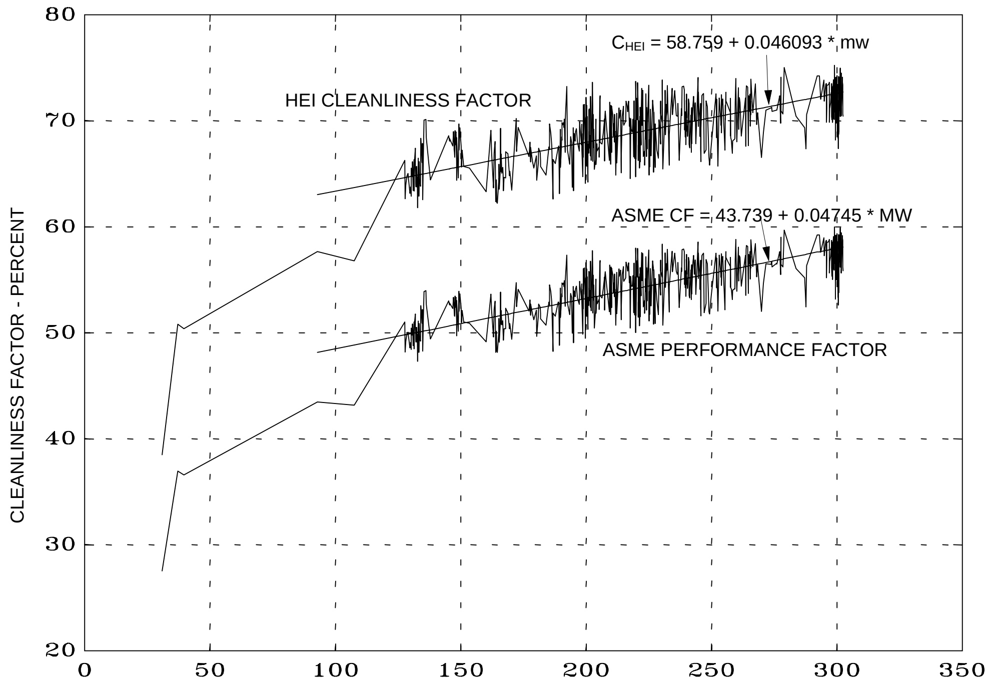
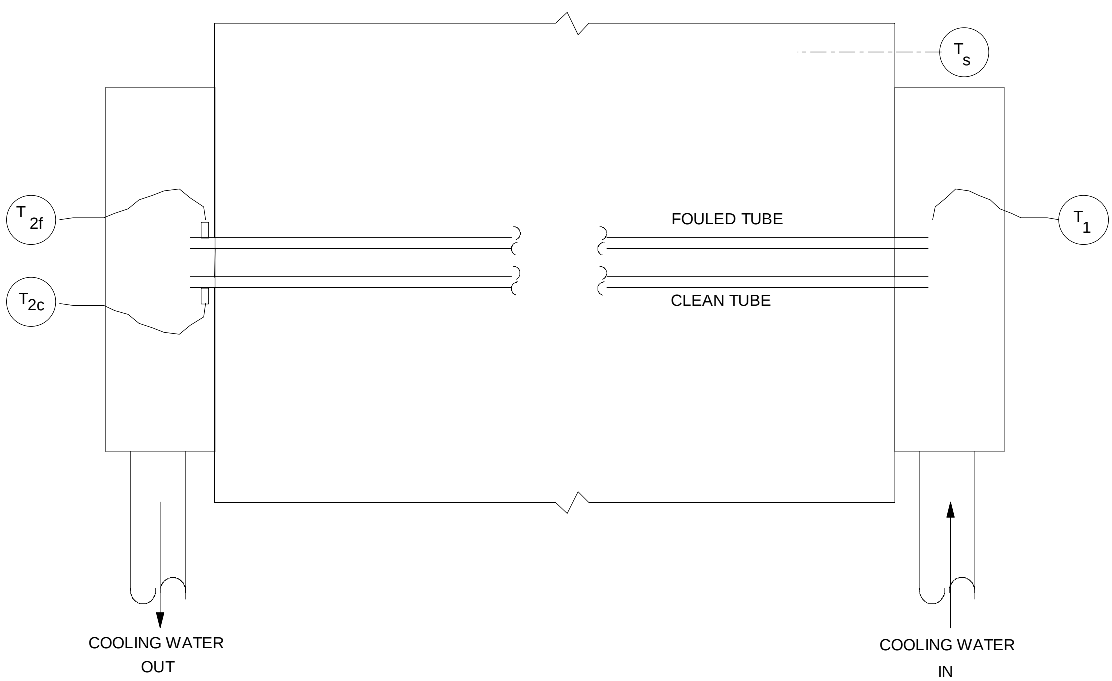
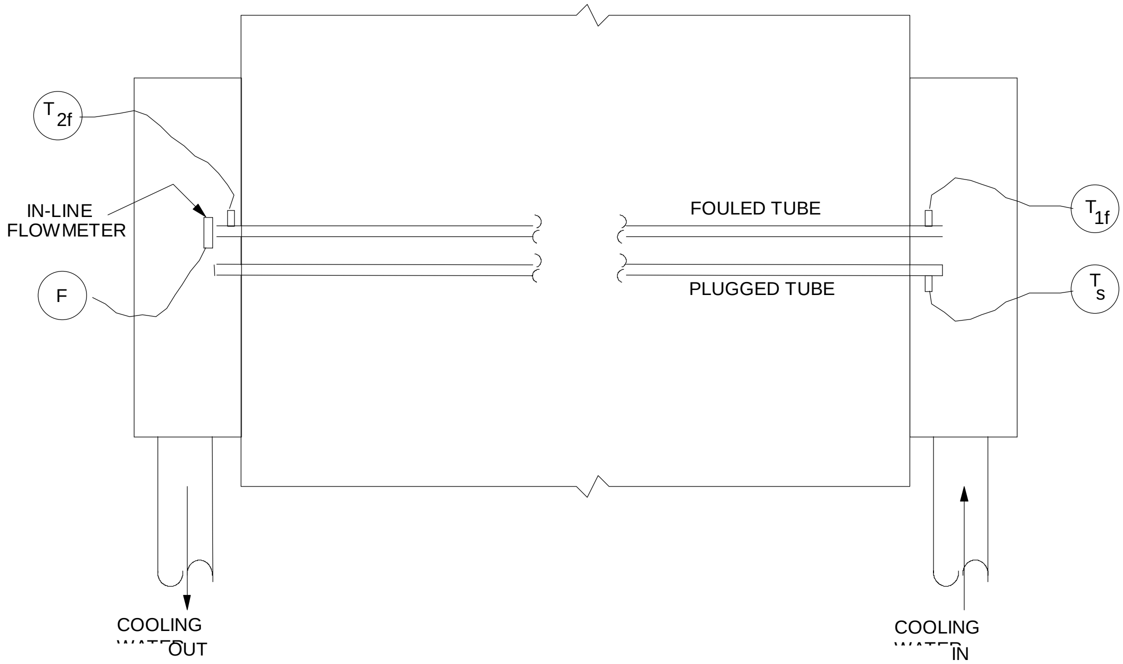


Generated Power – MW
Cleanliness Factors vs. Load
Winyah Unit #1
Figure 1.0



ASME METHOD OF ESTIMATING TUBE FOULING RESISTANCE

FIGURE 2.0



EPRI/BRIDGER METHOD OF ESTIMATING TUBE
FIGURE 3.0

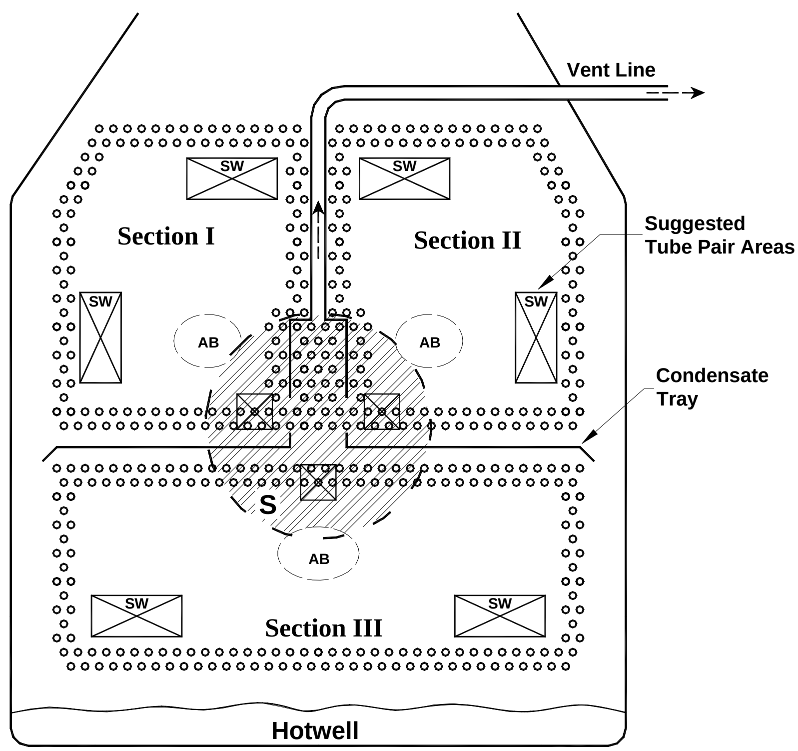


Figure 4. - Typical Single Shell Single Pass Condenser Configuration

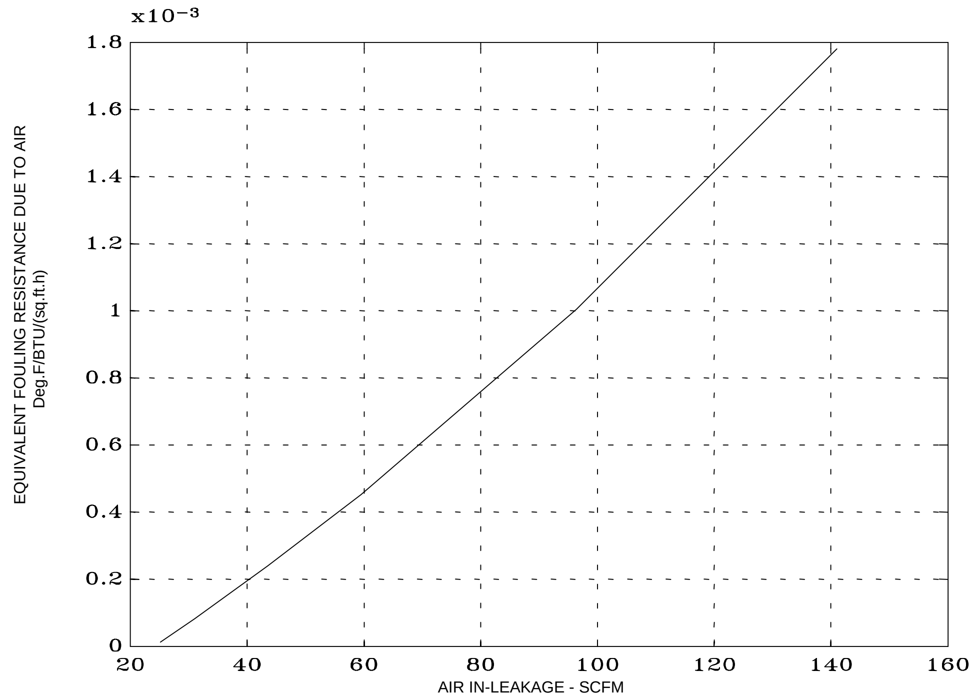


Figure 5. – Typical Relationship Between Air In-leakage and Apparent Increase in Tube Thermal Resistance

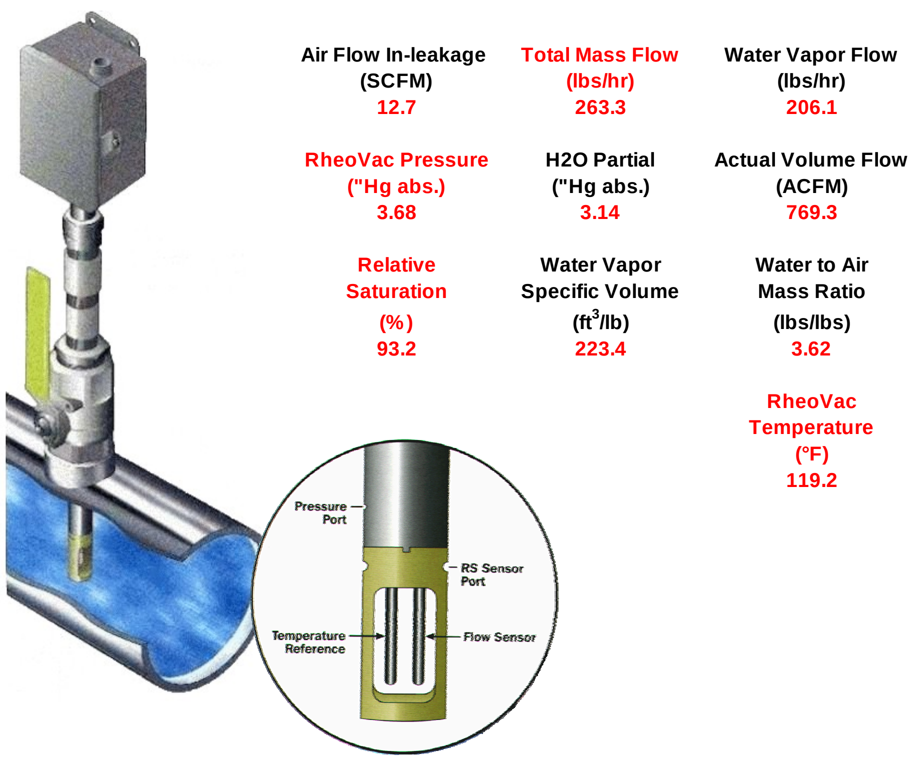


Figure 6 – Multi-Sensor Probe and Output Measurement Data (Courtesy of Intek, Inc.)