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ADVANCEMENTS ON A COMPREHENSIVE THEORY OF CONDENSERS

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ABSTRACT

The importance of having complete knowledge of the performance of all subsystems in any process is undisputed. The optimization of a process is dependent on the response of each component to external perturbations which can be numerous and are unique to each component. Such is the case for the turbine exhaust steam surface condenser found in most power generating plants around the world. The lack of a detailed understanding of internal behavior of gas mixtures on the shell side of condensers is now recognized uniquely as being the primary reason for responsible engineers to think of condensers as their "biggest job-related headache."

The results of a new condenser model and theory that explains condenser behavior response to various amounts of noncondensables and the cause of dissolved oxygen uptake in condensate is reviewed. New advancements coupling the model with direct measurement of air in-leakage and its impact on dissolved oxygen, contribution to excess condenser back pressure, effects on a condenser with variable exhauster capacity and identification and quantification of tube fouling will be presented. Impact of a multi-sensor probe condenser diagnostic instrument on the model development and condenser measurement is also presented.

NOMENCLATURE

d_t	tube bundle density, tubes/ft²
$\dot{m}_r$	steam mass flow rate at r, lb/hr
$\dot{m}_{r,a}$	steam mass flow rate, air present, lb/hr
$\dot{m}_t$	steam mass flow rate per tube, lb/hr
$\dot{m}_s$	total steam mass flow rate, lb/hr
n	number of tubes in bundle
n_{act}	number of active tubes in bundle
p_a	partial pressure of air, "HgA
p_s	partial pressure of steam, "HgA
p_v	partial pressure of water vapor, "HgA
r	radial position in tube bundle, ft
r_s	radius of the stagnant zone, ft
t	tube thickness
ν_r	velocity at radius r, ft/sec
$\nu_{r,a}$	velocity at radius r with air present, ft/sec
ρ_a	density of air, lb/ft³
ρ_s	density of steam, lb/ft³
ρ_v	density of water vapor, lb/ft³
η_1	calibration parameter for determining percent of available condensing surface
η	percent of available condensing surface
ΔT_{lm}	logarithmic mean temperature difference, °F
ΔT_{cw}	circulating water temperature rise, °F
A	geometric condensing surface area, ft²
ARS	air removal section
P_{EX}	excess condenser pressure
P_T	total condenser pressure, "HgA
SCF	standard cubic feet
SCFM	standard cubic feet per minute, ft³/min
T_{HW}	hotwell temperature
T_{cw1}	inlet circ water temperature, °F
T_{cw2}	outlet circ water temperature, °F
TTD	terminal temperature difference, °F
T_s	steam temperature, °F
U	heat transfer coefficient, BTU/Hr×ft²×°F

INTRODUCTION

The basis for a comprehensive understanding of shell side steam and noncondensable gas dynamics was introduced in 2001[1,2] and it is summarized here as background. In this work sufficient theoretical effort was made to develop and describe, using the simplest form of algebraic equations, a very basic dynamic model of condenser behavior. A major contributor to this development effort was data from condenser measurements using a *RheoVac*® (hereinafter referred to as a Multi-Sensor Probe (MSP)[6]) air in-leakage instrument. The measured and derived data from this instrument was then used along with other supportive data found in the open literature to calculate condenser performance characteristics using this model. These theoretical results were then compared with condenser response measurements caused by various amounts of air admitted to a condenser; the measured data showed good agreement with the model.

This writing advances this model to include how vacuum pump performance affects condenser response and expands on the theoretical description of steam and steam/air dynamics in operating condensers. Particular attention is placed on condenser behavior that will minimize or eliminate hotwell subcooling with consequential reduction in the amount of dissolved gases in condensate.

BACKGROUND

Throughout the literature is noted apparent and conflicting arguments as to how air in-leakage affects condenser excess back pressure. Some believe that noncondensables add an overall partial pressure of these gases throughout the condenser above that of a steam caused pressure, giving rise to an overall higher back pressure than by that of the steam alone. Others believe that noncondensables converge uniformly on all tubes in the condenser, being pushed there by the flow of steam on its way to becoming condensate, thus causing air blanketing. This "tube blanket" is believed to provide a "resistance," much the same as water side fouling to heat flow, causing an increase in ΔT_{lm}, thus a higher steam temperature. This then results in higher condenser back pressure. Still others believe that thermal masking occurs over large areas, sometimes called "air binding," which may or may not be associated with an air partial pressure. None of the above beliefs have been incorporated into a dynamic description of steam and air within the condenser where the effects on observed behavior, other than increased back pressure, could be explained. This pseudo-engineering and lack of integration into an understandable condenser description has added to the confusion and perceived condenser shell side complexity that has spanned the whole of the twentieth century. Not having a comprehensive understanding of condenser shell side behavior has inhibited the ability of plant staff to confidently trouble shoot condenser problems, giving rise among engineers responsible for performance to state that condensers are their "biggest job related headache."

In earlier papers [1,2] a summary was provided of historical work describing the then known understanding of condenser behavior. It has been noted from several observations by the author, while evaluating the performance of many condensers using the MSP instrument, shown in Figure 1, that there were deficiencies in this understanding and discrepancies between observations and available theory[1]. During further work as an EPRI task force member engaged in the preparation of the "Condenser In-Leakage Guideline"[3], the author came to realize that little was truly known about the condenser shell side dynamics. A serious effort was begun in January of 2000 to understand this side of the condenser and develop a mathematical description of its dynamics. Because of the newness of this discovery, a brief overview is provided as background.

The starting point for the dynamic model is a simple hypothetical condenser as shown in Figure 2. It is assumed that it has no continuous source of air in-leakage or other noncondensables, eliminating the need for an air removal section (ARS). Since baffles, tube bundle voids, or condensation collection trays can interfere with the natural flow of condensing steam, they are not included in this basic model. Another feature of this structure is that turbine exhaust steam is permitted to flow around the sides of the tube bundle, allowing this steam to enter the bundle from the sides and up from the bottom, much in keeping with modern condenser design attempts. The arrows show the obvious flow of steam to be radial, converging to a point in the middle of the condenser. For performance computations, Table I provides a list of parameters for this condenser. Absent of noncondensables, this hypothetical condenser would be a practical condenser for condensing exhaust steam from a 535 MW unit.

Table I - Hypothetical Condenser Parameters

Steam load, $\dot{m}_s$	2.444×10^6 lb/hr	Circ water inlet temperature, T_{cw1}	85°F
Tube diameter, d	1.0 inch	Tube circ water velocity, v_{cw}	6.332 ft/sec
Wall thickness, t	22 ga	Area per tube, A_t	17.8 ft^2
Tube pattern	Hexagonal	Total tube area, A	360,889 ft^2
Tube bundle diameter, R	12.37 ft	Heater transfer coefficient, U	593.8 BTU ÷ (ft^2 × Hr × °F)
Number of tubes, n_t	20,272	Circ water flow rate, $\dot{m}_{cw}$	279,889 GPM
Tube length, L	68 ft	Hotwell temperature, T_{HW}	108°F
Tube bundle density, d_t	42.16 tubes/ft^2	Tube material	Stainless steel

It was shown that if a fixed amount of air (not a continuous flow) was injected, amounting to 4.3 SCF, this air would be immediately scavenged by the steam, forcing it to concentrate at the center of the tube bundle. Only after reaching the center of the tube bundle, this concentrated air shown in Figure 3 blankets the affected tubes in this region, reducing the amount of steam being condensed on them in agreement with Henderson and Marchello[4]. As a result there is little or no significant temperature rise of the circulating water within these tubes and in the limit, they will remain at the inlet circulating water temperature throughout their length. Therefore, the temperature of the vapor around these tubes is lowered, which lowers the water vapor pressure p_v, below that of the surrounding steam vapor pressure p_s, which is equal, very approximately, to the condenser pressure P_T. The differences between P_s and the lower temperature dependent p_v is the air partial pressure p_a, giving essentially no net pressure difference across this air/steam boundary. The air bound region containing inactive tubes that are effectively shut off from the condensation process is designated the "stagnant" zone. This is because of the substantially reduced condensation rate which in the limit stops all drift velocity. The region outside the stagnant zone, containing radially directed moving turbine exhaust steam, and containing active tubes, is called the "steam wind" region. Should the air bound region be subcooled by only 6°F, the condensation rate per tube is reduced by a factor of 10 from the condensation rate of tubes in the steam wind region. At about 6°F subcooling of gas entering the ARS vent line its water vapor to air mass ratio is 3. Computations show that at the inlet of the ARS, this ratio is more than 100, having subcooling of about 0.2°F, indicating that there is very little decrease in the heat transfer of these tubes. For this reason an MSP measurement of about 3 has been shown and validated to be the value associated with condenser back pressure threshold. From this point, increasing air in-leakage causes a decrease in the water vapor to air mass ratio and an increase in measured back pressure.

Equations describing the mass flow rate of steam and its velocity as a function of tube bundle radius for no air injection are:

$$\dot{m}_r = \pi\, \dot{m}_t\, d_t\, r^2 \tag{1}$$

and

$$v_r = (\dot{m}_t d_t r) / (2\rho_s L) \tag{2}$$

with

$$\dot{m}_t = \dot{m}_s / n \tag{3}$$

where air is injected into the hypothetical condenser in different amounts to change the stagnant zone boundary radius r_s, the following equations apply:

$$\dot{m}_{r,a} = \dot{m}_s [((r/r_s)^2 - 1)/((R/r_s)^2 - 1)]_{r \geq r_s} \tag{4}$$

and

$$v_{r,a} = \dot{m}_{r,a} / 2\pi \rho_s r L \tag{5}$$

with

$$\dot{m}_t = \dot{m}_s / n_{act} \tag{6}$$

The results of employing these equations to the hypothetical model is shown in Figures 4 and 5 for different percent of lost tubes located in the stagnant zone.

Because tubes are essentially removed from the condensation process, the amount of steam being condensed on each tube in the steam wind region must increase as noted in Equation (6). For this to occur ΔT_{lm} and *TTD* must increase. This then becomes the cause for increased condenser back pressure, i.e., the condenser back pressure increase is caused directly by a loss of condenser tube surface area. Since the stagnant zone tubes are not responsible for condensing any significant amount of steam, then the condenser tube surface area A is reduced to ηA where (1-η) is the fractional tube loss. New condenser operating conditions must now be calculated based on a smaller number of tubes. Therefore, not knowing how many tubes are partially or fully removed from the total condensing surface area A due to air binding or a stagnant zone is the major source of error in measuring the condenser heat transfer coefficient in operating condensers.

Computations show that condenser pressure increases as the size of the stagnant zone increases; i.e, more central tubes are included in the stagnant zone and effectively removed from participating in condensing steam. Table II provides a summary of computations using methods described elsewhere[2]. The established condition promoted hotwell subcooling by letting condensate, produced on tubes above the stagnant zone, to be cooled as it fell past tubes in the stagnant zone and then allowed to enter the hotwell without being regenerated by steam. This subcooling is noted in Table II, by the steam temperature increasing with condenser pressure and the hotwell temperature remaining constant at 108°F. Most condensers, however, constructed since about 1960, are designed to regenerate cooled condensate, essentially preventing hotwell subcooling. These results are presented later in this paper.

Further, in computing the data of Table II, the average circulating water outlet temperature was used to determine ΔT_{Lm} and *TTD*. The earlier computations used this average temperature because its approximate value is commonly recorded in operating plants. The result, however, is an η coefficient that is not in direct proportion to the percent of active tube surface area in the condenser. For this reason the values of η_{1} in Table II should be considered as corrupted by measurement limitation and not in proportion to active tubes. In the work to be presented, the circulating water outlet temperature from the steam wind (active tube) region of the condenser is used to determine these two important temperature parameters.

Although the stagnant zone size in operating condensers was discussed assuming a fixed pump capacity so as to allow air in-leakage to be easily computed, the effects of degraded pump capacity was not shown. This also is presented in this paper.

MORE ON THE HYPOTHETICAL CONDENSER AS IT RELATES TO REAL CONDENSERS

When the hotwell temperature falls below the steam temperature defined by the saturation pressure of the condenser, the hotwell is said to be subcooled. This condition decreases plant efficiency since extra heat must be added to this condensate to bring it back to the saturation temperature of the condenser. The condition results from stagnant and air bound zones in the condenser as well as the inability to regenerate the cold condensate, by the steam from the surrounding regions, before entering the hotwell.

Since these stagnant and air bound regions are themselves subcooled, dissolution of noncondensable gases in these regions into localized low temperature condensate is high. Since these noncondensables contain oxygen, carbon dioxide and ammonia, these subcooled regions in operating condensers become a major source of condenser tube corrosion. Incomplete regeneration and release of these gases from condensate before reaching the hotwell becomes an additional source for corrosion throughout the condensate and steam path of the plant.[5]

Table III is the results of recent computations made on the hypothetical condenser with various fixed amounts of injected air and under the conditions of complete regeneration. The outlet circulating water temperature emitted from the active condensing tube section of the condenser is used in determining ΔT_{lm} and *TTD* defined in Figure A1.

The factor η now can be seen to represent the fraction of active tubes located in the steam wind region of the condenser. It is the exact complement of the percentage of tubes lost in the stagnant zone. The active region heat transfer coefficient U_A is the same as for individual tubes within the active region. Because A in Equation A8 is no longer a constant, Equation A9 should be written as $TTD = f(\eta U)$. The results of Tables II and III plotting ηU vs TTD are shown together in Figure 6 for comparison. The upper curve obtained using the conveniently available average circulating water outlet temperature is labeled as η_1 to separate it from the more meaningful η which bears the noted direct relationship to percentage of available tubes.

The observed heat transfer coefficient U that is measured by a plant using a value for A representing the total physical tube surface area will be in error as shown, provided T_{cw2} in Equation A1 is measured as the active region outlet temperature. From this condenser description it is believed that condenser surface area lost due to stagnant zones or otherwise air bound zones are responsible for the known wide discrepancies between formally employed theory and condenser measurements of U.

OPERATING CONDENSER

The information provided here on a hypothetical condenser having no air removal section and with a stagnant zone created by injection of a fixed amount of air can be identically reproduced, within limitations, in operating condensers. In this case, air entering the shell space at some specific rate is removed by an exhauster having a capacity for air matched to the in-leakage rate through its suction pressure and performance rating curve. It is the suction pressure of the pump that varies the air density and water vapor density at the pump suction such that the pump's pressure dependent capacity for air component is compatible with the air in-leakage rate.

It has been noted, using the MSP instrument, that for low air in-leakage values below the back pressure saturation level of the condenser, the removed water vapor to air mass ratio at the suction of the vacuum pump is greater than three. In this condenser pressure saturation region, as air in-leakage increases, the air partial pressure increases with a corresponding decrease in the water vapor partial pressure. This condition does not result in a noticeable change in pump suction pressure or in condenser pressure. When air in-leakage is further increased, the condenser back pressure threshold is reached and with additional increases in air in-leakage the condenser back pressure begins to rise, and continues to rise monotonically above the pressure saturation value. At the air in-leakage point where excess back pressure begins, the water to air mass ratio becomes three (3), a condition that is almost always observed when using the MSP instrument. This mass ratio value becomes a better indicator of the onset of air in-leakage problems than using increases in condenser back pressure, as noise and periodic variations in the pressure measurement make the determination of a back pressure increase very difficult. As indicated in MSP description literature, a decreasing water vapor to air mass ratio approaching a value of three (3) is an indication that excess back pressure is about to be present, an early indication that corrective action should be initiated.

Another measured quantity from the MSP instrument is the volumetric flow rate or the total water vapor and air mass flow rate into the vacuum pump. Depending on the type of exhauster in use, one or the other can be used to measure the pump's operating capacity and compare it to the pump's rated capacity.

The model can be used to determine how the condenser responds to a failing pump that is performing below its rated capacity. Table IV presents data from an operating condenser having the performance and design parameters of Table I. At a pressure saturation value of 2.450 "HgA, the vapor saturation temperature of the steam T_s is 108°F. With subcooling of 6°F after passing through the ARS, the gases have a water vapor to air mass ratio of 3.2 which leads to a calculated air in-leakage of 25.1 SCFM using an exhauster having a capacity of 2000 ACFM. Note that even with this much air in-leakage, there is no contribution to excess condenser back pressure due to the 2000 ACFM operating capacity of the exhauster. Another, lower air in-leakage condition of 12.5 SCFM has been included in the table, also having zero excess back pressure, to show how water vapor pressure and air partial pressure exchange with no change in total pressure. In this latter case the water vapor to air mass ratio has increased to 6.9. Also shown in this table are the percentage of lost tubes and behavior of vacuum line gases yielding higher air in-leakage along with increasing excess back pressure, P_{EX}.

Table V shows operating and performance changes resulting from a degraded vacuum pump. The capacity of the exhauster has been reduced to 1000 SCFM. All of the data on the left side of the table remain the same as shown in Table IV because they are based on the loss of the same number of tubes in the stagnant zone. When computing the mass flow rates and levels of air in-leakage that produce the number of lost tubes these mass flows are halved, as well as the SCFM of air in-leakage, as might be expected by the lower pump capacity. It should be noted that the excess back pressure and water vapor to air mass ratio at the pump suction are directly related to the percentage of lost tubes, or size of the stagnant zone.

Having the ability with the MSP to measure the amount of noncondensable gases being removed from the condenser and independently the exhauster capacity, it now becomes easy to determine if either is contributing to excess back pressure on the turbine. Excess back pressure from one or the other will result in the measured water vapor to air mass ratio falling below the value of about 3 in the MSP monitored exhauster line. This determinable excess back pressure can be as small as zero and is independent of indicated condenser pressure or measured *TTD*. The MSP determined excess back pressure is free of measurement "noise" that plagues pressure gauge readings, and overcomes the error in outlet circulating water temperature measurements that is needed to obtain the value of *TTD* due to temperature stratification common in the outlet water box.

Any otherwise noted excess back pressure must be the result of tube fouling. If the water vapor to air mass ratio is above approximately 3 and condenser back pressure is greater than that which can be accounted for considering load, inlet circulating water temperature and flow rate and condenser design factors, then this difference in excess back pressure must be caused by fouling. Quantification of this pressure will be in relation to the type of fouling and its thickness following conventional heat transfer practices.

Review of this subject, by examining the often quoted literature, will distinguish the model presented here from others. Papers by Moore(7), Marto(8) and Silver(9) discuss the concept of tube blanketing on the overall heat transfer coefficient of the condenser. One of the models they present considers small amounts of air trapped about tubes in the condenser, establishing an air partial pressure at the tube surface. It is said that this air bound layer interferes with the steam approaching the cold tube surface, decreasing heat transfer much the same as internal tube fouling which has been summarized by Harpster(1). This concept, although suitable to mathematical description, ignores the scavenging phenomenon and the extremely dilute state of air in steam entering the periphery of the tube bundle. The model discussed in this paper is based on measured data which shows that air ingress affects condenser heat transfer in very specific and different ways. The heat transfer coefficient of tubes in the "steam wind" region of the condenser, where the water vapor to air mass ratio is greater than 100, is not greatly affected, therefore their heat transfer coefficient is not significantly decreased as demonstrated by the work of Henderson and Marchello(4). At the outer rows of tubes, the water to air mass ratio are of the order 10^4 and as also found by Henderson and Marchello, these tubes are essentially unaffected. It is only the "stagnant zone" tubes whose heat transfer coefficient is drastically reduced. This stagnant zone is accounted for in the model presented here and is mathematically described.(1) The zone is formed by the scavenging process, and only for high air in-leakage, serving as a repository for the air or noncondensable gas within the condenser, driven there by the steam.

Other concepts for condenser behavior and excess back pressure with air ingress exist, as mentioned in the background section. While these use descriptive words such as partial pressures, stagnation, and air-bound zones, they are not sufficiently similar to the model discussed here to require comparative comment.

CONCLUSIONS

The task of developing a comprehensive theory of condenser shell side dynamics has been supported with new analytical results. Prior work showing how the quantity of tubes effectively removed from active participation in the condensation process due to air in-leakage was dependent on experimental measurements giving rise to values of the coefficient η_1 not directly related to remaining percentage of active tubes. It was found that this lack of correlation resulted from using the conveniently available measurement of average condenser outlet circulating water temperature. As shown here, use of the circulating water outlet temperature from active tubes only to determine *TTD* and ΔT_{lm} provided an η coefficient in direct proportion to the percent of active tubes remaining in the condenser, and results in a direct determination of active surface area participating effectively in condensing steam.

The model and its description was also used to show how subcooling in the stagnant zone or other air bound zones are responsible for high dissolved corrosive noncondensable gases. Lacking sufficient regeneration, these subcooled gases would enter the condensate stream. This source of dissolved gases becomes a major contributor to plant corrosion.

The theory was used to describe condenser responses to vacuum pump performance degradation brought on by wear or internal failure. The description method has value in understanding the behavior and performance of the vacuum pump/condenser combination. Finally use of the MSP to show how tube fouling can be quickly and accurately determined as a cause for excess back pressure was presented.

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Table II - Summary of Hypothetical Condenser Performance

Constants: T_{HW} = 108°F; $U_{act\ area}$ = 594 BTU/ft^2 × Hr × °F; T_{cw2} (average) = 103.2°F

% tubes lost	Steam Load per Active Tube (lb/Hr)	T_s (°F)	Pressure ("HgA)	h_{fg}	ΔT_{cw} Active Tubes (°F)	T_{cw2} (Active Area) (°F)	TTD* (°F)	ΔT_{lm}* (°F)	U (Observed)	Apparent Area Coefficient η_1
0	120.56	108.00	2.450	1032.5	18.016	103.02	4.980	11.78	593.80	1.000
2	123.03	108.46	2.483	1032.2	18.386	103.35	5.442	12.33	567.01	0.955
6	128.26	109.45	2.556	1031.7	19.159	104.15	6.432	13.49	517.95	0.873
11.1	135.62	110.84	2.660	1030.9	20.242	105.24	7.822	15.08	462.98	0.780
22.2	154.97	114.45	2.950	1028.8	23.083	108.08	10.980	18.56	375.40	0.632
33.3	180.76	119.25	3.376	1026.1	26.854	111.84	16.232	24.13	287.96	0.485

*From T_{cw2} (average)

Table III - Heat Transfer Coefficient Derived from Active Tube and Region Derived Parameters

Conditions: No hotwell subcooling; T_{cw} = 85°F; Steam Load $\dot{m} = 2.444 \times 10^6$ lb/hr; $U_{act\ area}$ =594 BTU/ft^2 × Hr × °F

% tubes lost	Steam Load per Active Tube (lb/Hr)	T_s (°F)	Pressure ("HgA)	h_{fg}	ΔT_{cw} Active Tubes (°F)	T_{cw2} (Active Area) (°F)	TTD*	ΔT_{lm}*	U (Observed) (ηU_A)	η Real Area Coefficient
0	120.56	108.00	2.450	1032.5	18.016	103.023	4.977	11.77	594	1.00
2	123.03	108.46	2.483	1032.2	18.386	103.386	5.074	12.08	582	0.98
6	128.26	109.45	2.556	1031.7	19.159	104.159	5.291	12.52	558	0.94
11.1	135.62	110.84	2.660	1030.9	20.242	105.212	5.598	13.23	528	0.89
22.2	154.97	114.45	2.950	1028.8	23.083	108.083	6.367	15.07	462	0.78
33.3	180.76	119.25	3.376	1026.1	26.854	111.854	7.396	17.52	396	0.67

*From T_{cw2} in Active Region

Table IV - Condenser ARS and Stagnant Zone Response to Air In-Leakage

Conditions: Vacuum Pump Capacity 2000 ACFM

			ARS Exit Gas Properties					**ARS Exit Flow Rate**					
% Tubes Lost	T_s (°F)	P_T ("HgA)	T_v (°F)	p_v ("HgA)	p_a ("HgA)	ρ_v (lb/ft^3)	ρ_a (lb/ft^3)	$\dot{m}_T$ (lb/Hr)	$\dot{m}_v$ (lb/Hr)	$\dot{m}_a$ (lb/Hr)	AIL (SCFM)	P_{EX} ("HgA)	$\frac{\dot{m}_{wv}}{\dot{m}_a}$
0	108	2.450	105	2.245	.205	.00328	.00047	450.0	393.6	56.4	12.55	0	6.98
0	108	2.450	102	2.053	.397	.00302	.00094	475.2	362.4	112.8	25.10	0	3.21
2	108.46	2.483	101	1.992	.491	.00294	.00116	491.5	352.3	139.2	30.97	.033	2.53
6	109.45	2.556	98.9	1.870	.686	.00277	.00163	527.5	331.9	195.6	43.52	.106	1.70
11.1	110.83	2.650	96.3	1.728	.932	.00258	.00223	575.6	308.0	267.6	59.50	.210	1.15
22.2	114.45	2.950	90.7	1.453	1.497	.00218	.00361	694.9	261.7	433.2	96.40	.500	.605
33.3	119.25	3.376	85.0	1.213	2.163	.00184	.00528	854.4	220.8	633.6	141.0	.926	.348

Table V - Condenser ARS and Stagnant Zone Response to Air In-Leakage

Conditions: Vacuum Pump Capacity 1000 ACFM

			ARS Exit Gas Properties					**ARS Exit Flow Rate**					
% Tubes Lost	T_s (°F)	P_T ("HgA)	T_v (°F)	p_v ("HgA)	p_a ("HgA)	ρ_v (lb/ft^3)	ρ_a (lb/ft^3)	$\dot{m}_T$ (lb/Hr)	$\dot{m}_v$ (lb/Hr)	$\dot{m}_a$ (lb/Hr)	AIL (SCFM)	P_{EX} ("HgA)	$\frac{\dot{m}_{wv}}{\dot{m}_a}$
0	108	2.450	105	2.245	.205	.00328	.00047	225.0	196.8	28.2	6.27	0	6.98
0	108	2.450	102	2.053	.397	.00302	.00094	237.6	181.2	56.4	12.55	0	3.21
2	108.46	2.483	101	1.992	.491	.00294	.00116	245.7	176.2	69.6	15.49	.033	2.53
6	109.45	2.556	98.9	1.870	.686	.00277	.00163	263.7	165.9	97.8	21.76	.106	1.70
11.1	110.83	2.650	96.3	1.728	.932	.00258	.00223	287.8	154.0	133.8	29.77	.210	1.15
22.2	114.45	2.950	90.7	1.453	1.497	.00218	.00361	347.5	130.8	216.6	48.20	.500	.604
33.3	119.25	3.376	85.0	1.213	2.163	.00184	.00528	427.2	110.4	316.8	70.49	.926	.34

APPENDIX A

USEFUL EQUATIONS FOR UNDERSTANDING CONDENSER BEHAVIOR

The currently accepted description of a condenser and the formulas for determining its performance are as follows. The illustration in Figure A1 represents the temperature profile of cooling water passing through tubes in a condenser. Here, T_v is the vapor temperature which can be set equal to the hotwell temperature, T_{HW}. T_{cw1} and T_{cw2} are inlet and outlet circulating water temperatures. TTD is the terminal temperature difference, ΔT_{cw} is the rise in circulating water temperature, and ΔT_{lm} is defined as the Graph of logarithmic mean temperature difference which is the mean temperature driving force for heat flow between the exhaust steam vapor and cooling water in the condenser tubes. The relationship between ΔT_{lm} and others in the diagram is

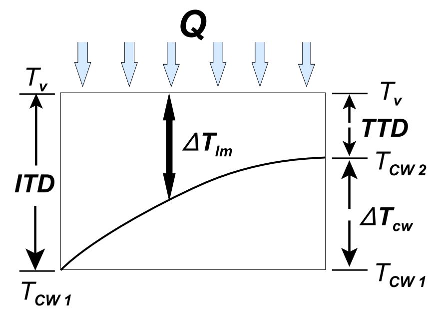

Figure A1. Condenser Temperature Profile and Thermal Parameter Definition

(All temperatures in °F)

$$\Delta T_{lm} = T_{cw2} - T_{cw1} \div \ln[(T_v - T_{cw1})/(T_v - T_{cw2})] \qquad \text{Eq. A1}$$

which can be written as:

$$\Delta T_{lm} = \Delta T_{cw} \div \ln[1 + (\Delta T_{cw} / TTD)] \qquad \text{Eq. A2}$$

Since ΔT_{cw} is due to a steam load Q (BTU/hr) from the turbine requiring energy removal sufficient to convert it to condensate, we can write

$$Q = \dot{m}_{cw} c_p \Delta T_{cw} \qquad \text{(Heat load to the circulating water)} \qquad \text{Eq. A3}$$

also,

$$Q = \dot{m}_s h_{fg} \qquad \text{(Heat load from steam condensation)} \qquad \text{Eq. A4}$$

where $\dot{m}_{cw}$ (lb/hr) is the mass flow rate of circulating water, c_p (BTU/lb•°F) the specific heat of water, $\dot{m}_s$ (lb/hr) the mass flow rate of steam, and h_{fg} (BTU/lb) the heat of condensation of incoming steam. Combining Equations A3 and A4, we have:

$$\Delta T_{cw} = \dot{m}_s \, h_{fg} / \dot{m}_{cw} c_p \qquad \text{Eq. A5}$$

which defines the rise in circulating water temperature in terms of mass ratio of steam flow to circulating water flow and two identifiable properties. Consistent with good engineering heat transfer practice in describing heat exchangers, Q is related to the exposed heat transfer surface area A, and ΔT_{lm}, with a proportionality factor characteristically called the heat transfer coefficient, U. This relationship is given by:

$$Q = U\,A\,\Delta\,T_{lm} \qquad \text{Eq. A6}$$

Combining Equation A6 with Equations A2 and A3, we have:

$$\dot{m}_{cw} = UA / c_p \ln\left(1 + \Delta T_{cw} / TTD\right) \qquad \text{Eq. A7}$$

which, following rearrangement, becomes:

$$TTD = \Delta T_{cw} \div [\exp(UA/\dot{m}_{cw}\, c_p) - 1] \qquad \text{Eq. A8}$$

Since c_p is nearly constant, and $\dot{m}_{cw}$ and ΔT_{cw} held constant through a fixed load Q and with A assumed constant, the terminal temperature becomes only a function of U, or: $TTD = f(U)$ Eq. A9

The theory goes on to say that the thermal resistance R, the inverse of U, can be described as the sum of all resistances in the path of heat flow from the steam to the circulating water, given by:

$$R = 1/U = R_a + R_c + R_t + R_f + R_w \qquad \text{Eq. A10}$$

where, a is air; c, condensate on tubes; t, tube; f, fouling and w, circulating water. Historically, much effort has gone into describing analytically each of these series resistances. The best characterized are R_w, R_f and R_t. Values of R_c, dealing with condensate on the tubes, have gained a lot of attention with some success; and R_a has essentially been ignored with the exception of near equilibrium diffusion limited experiment measurements and its associated theory.[3] The latter is generally believed to be very complex[1,2] and limited data is available. The general belief is that small amounts of air will dramatically affect the heat transfer coefficient, resulting in an increase in the values of ΔT_{lm}, TTD, and T_{HW}. The importance to the work presented is that R_a is assumed to be treatable in a manner similar to tube fouling, as shown in Equation A10. It is not.

A useful relationship for describing water and air mixtures derived from the Ideal Gas Law is:

$$\rho_v / \rho_a = .622\, p_v / p_a \qquad \text{Eq. A11}$$

where v and a are for water vapor and air respectively and ρ is density and p is partial pressures.
If P_T is the total pressure, then, $P_T = \rho_v + \rho_a$.