HEAT-RATE IMPROVEMENTS OBTAINED BY RETUBING CONDENSERS WITH NEW, ENHANCED TUBE TYPES*

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ABSTRACT

Significant fuel savings can be achieved at power plants by retubing the condensers with enhanced tubes. Because of the higher overall heat-transfer coefficient, the exhaust steam is condensed at a lower pressure and the plant efficiency is therefore increased or plant heat rate is reduced. Only the spirally indented type of enhanced tube is currently being used in the U.S. and most other countries; however, different types of enhanced tubes have been proposed for power-plant condensers, each with their own set of attributes. This paper determines what attributes and their magnitudes of enhanced tubes lead to the most energy savings as measured by reduction of the plant heat rate. The particular attributes considered are the inside and outside enhancement levels, the inside efficiency index (inside enhancement level divided by pressure-drop increase), and the enhanced-tube fouling-rate multiplier. Two particular condensers were selected because all necessary information were known from previous heat-rate studies such as the condenser geometry, the circulating-water pump and system information, and the low-pressure turbine characteristics. These are "real-world" condensers and therefore the finding will be representative for many other condenser-retubing applications. However, the authors strongly recommend that an economic evaluation be performed at each site to determine the energy savings and payback time.

This generic investigation showed that the outside enhancement level is the most important attribute, and a value of about 1.5 can lead to heat-rate savings of about 20 to 40 Btu/kW-hr. Increasing the inside enhancement is less effective because of the increased pressure drop that leads to a reduction of the coolant flow rate and velocity. The efficiency index, which is a measure of the pressure drop increase, is only important if the outside enhancement level is close to unity. With an outside enhancement level of 1.5, two other interesting findings were obtained: enhanced tubes lead to greater energy savings if (1) the cooling water has more rather than less fouling (even with twice the fouling rate for the enhanced tubes), and if (2) the condenser is a poor performer (a poor design) rather than a good design. The major reason for these two effects is the shape of the low-pressure turbine heat-rate correction curve.

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INTRODUCTION

One method to improve the heat rate and increase the output of power plants is to reduce the steam-turbine back pressure by improving the thermal performance of the surface condenser. A method often discussed but seldom used is replacing the plain tubes with enhanced tubes, which was first proposed by Withers and Young (1971).

Only one particular enhanced tube type is currently used and by only one U.S. utility. This enhanced tube is the spirally indented type – also called rope or corrugated – and is shown in Figure 1. It can be obtained from three tubing suppliers and empirical prediction methods are available from supplier catalogs such as Wolverine Tube, Inc. (1984) and Yorkshire Alloys Limited (1983). These tubes can be fabricated using all commonly used tube materials including titanium and stainless steel.

Table 1 (Rabas and Schaefer, 1993), shows the 14 power-plant condensers that were retubed (some partially) with a particular spirally indented tube called Korodense LPD by Tennessee Valley Authority (TVA) and the retubing dates. Note the dramatic long-term demonstration at the Gallatin Unit 1 condenser, about fourteen years of successful operation.

Other countries have successfully retubed power-plant condensers with enhanced tubes, especially the former Soviet Union. Katsman and Kishinevsky (1983) reported that 15 condensers with capacities ranging from 25 MW to 200 MW have been retubed with spirally indented tubes and have operated for more than 10 years without major problems. Nikolaev et al. (1983) reported on the successful partial retubing of a condenser with spirally indented tubes. In the United Kingdom, Rowe (1983) described three prototypical surface condenser experiments that were performed on condensers that were partially retubed with only spirally indented tubes. More recently, Pearce (1990) described some additional experiments with condensers that were also retubed with different types of spirally indented tubes. It is very evident that the spirally indented tube type has received almost all the attention for power-plant retubing applications throughout the world.

Table 2 lists the authors best estimates of the attributes of Korodense LPD, the most commonly used spirally indented enhanced tube for power-plant condensers in the U.S. The important point is that most of the enhancement occurs at the inside of the tube and there is very little outside enhancement. It has been suggested that spirally indented tubes would show some steamside enhancement due to the grooves that will aid in condensate drainage and/or to surface tension effects. However, a review of the existing data by Rabas (1987) showed that the groove spacing was too large to obtain any significant enhancement.

A number of investigations such as Jaber and Webb (1993), Jaber et al., (1991) and Webb et al. (1991b) stressed that the tubes used for retubing of power-plant condensers should have both inside and outside enhancements. The increase of the overall heat-transfer coefficient is limited if only one side is enhanced because both the inside forced-convection coefficient for water and the outside condensing coefficient of steam are about the same order of magnitude. The basis for their recommendation is that similar enhancement levels are required for both in order to obtain the highest overall heat-transfer coefficient. These studies focused on enhanced-tube types that supplied increased outside enhancement because they were very aware of the limited outside enhancement obtained with the spirally indented tubes. Typical values of the outside enhancements measured in these experiments were 2.8, 1.75, 1.64, and 1.70 for four different copper-base tube materials. The first three were obtained with enhancements of the low-fin type and the fourth consisted of copper particles bonded to a

tube with a plain outside diameter. The outside enhancement mechanisms are a combination of both additional surface area and condensate film thinning.

Previous investigations (Webb et al., 1984 and Rabas et al., 1990) of the "optimum" geometry or attributes of enhanced tubes started from performance data of tubes obtained from tubing suppliers. These tubes have different levels of both the inside and outside enhancements, different efficiency-index values, and fouling characteristics. However, no attempt was ever conducted to identify and quantify which attributes of various enhanced tubes lead to the greatest reductions of the plant heat rate. This investigation does identify and quantifies the energy savings as measured by the heat-rate reductions of two actual condensers. Heat-rate savings are directly related to dollars and this conversion factor is known by every utility.

The different attributes of the enhanced tubes that are addressed are the following: the inside enhancement, e_i ; the outside enhancement, e_o ; the efficiency index, η , and the fouling-rate multiplier, f_m . The inside and outside enhancements are obvious attributes. The increase of the inside heat-transfer coefficient is always accompanied with an increase in the cooling-water pressure drop. The pressure-drop increase leads to a flow reduction, the magnitude of which depends on the cooling-water system. As a result, enhanced tubes lead to a reduced condenser saturation temperature or back pressure only if the reduction of the flow does not offset the improved thermal performance. The efficiency index is used to account for this negative attribute of enhanced tubes. The efficiency index is the ratio of the inside enhancement level divided by the pressure drop increase and is defined as $e_i/(\Delta P_o/\Delta P_i)$.

The fourth attribute is the fouling multiplier. The plain-tube fouling rate is dependent on the particular site. Different plain-tube fouling rates will be assumed, and the fouling rates of the enhanced tube will be obtained using fouling-rate multipliers based on measured data from some typical sites.

The impact of each attribute will be determined for different levels of the outside bundle performance characterized by a bundle factor, different plain-tube fouling rates, and different pressure-drop-increase values due to fouling. Before the results are discussed, a brief overview of the heat-rate calculation method for both the plain and enhanced tubes is presented. Appendix 1 presents the details of the two condensers that were considered for these retubing investigations.

HEAT-RATE CALCULATION METHOD

The input information needed to evaluate the heat-rate change by retubing with enhanced tubes are discussed first and then an overview of the calculation method is given with emphasis on the bundle-factor rating method.

Input Information

Table 3 shows the factors that must be considered to determine if retubing with enhanced tubes will be cost effective. The enhanced tubes attributes have already been discussed. The effects of the fuel cost and the incremental increase of the tubing cost are obvious. The cost increase for enhanced tubes of the spirally indented type are not large (Rabas et al. 1990); are somewhere between a 1.15 to 1.35 increase over the plain-tube cost. The magnitude depends on the tube material type, less for the more expensive tube materials such as stainless and titanium. The cost increase for other enhanced tube types such as low finned tubes will be larger because additional metal is required.

The condenser design conditions – the tube length and diameter, tubeside velocity, and size (number of tubes) – are necessary input. It has been shown (Rabas and Schaefer, 1993) that designs with tube lengths less than about 10 m (35 ft) and cooling-water velocities greater than about 1.5 m/s (5 ft/s) are good candidates for retubing with enhanced tubes. For condensers with long tubes and/or low design velocities, the reduction in the flow due to the increased pressure drop can negate the advantage of the improved thermal performance.

The magnitude of the flow reduction due to the increased pressure drop of the condenser depends on the existing circulating water system. This system includes the pump characteristics (head-capacity and horsepower-capacity curves) and the magnitude of all the external pressure losses in the system. Enhanced tubes are usually cost effective for units with steep pump curves (large negative slope of head with capacity) and/or with large external piping losses because of the smaller flow reductions. Figure 2 shows the pump curves for the two condensers considered in the study. Note that pump curve for condenser #1 is much steeper than for condenser #2. The range of typical flow reductions is about 0.80 to 0.99. There usually is a small change of the pumping power because of the increased pumping head and reduced flow rate that must be considered in the overall evaluation.

The heat-rate correction curve, which is a plot of the percent change in the heat rate with back pressure for a fixed steam flow rate, is crucial for determining the cost effectiveness of enhanced tubes. Enhanced tubes are a good candidate for retubing if the back pressure falls in the ramp portion of the curve, that is away from the flat portion or the choked condition. This characteristic of the low-pressure turbine is critical in the evaluation of the cost effectiveness of enhanced tubes. The heat-rate correction curves for the two condensers are also shown in Figure 3.

Condensers that experience a large variation in the inlet cooling-water temperature are good candidates for retubing with enhanced tubes. The reason is that for a large portion of time the plant will be operating with higher back pressures and on the steep portion of the heat-rate correction curve. Another important aspect is that the maximum benefits are obtained in the summer months when the need for the power is the greatest.

Calculation Method

A computer program was previously developed (Rabas et al., 1990) and made available to utilities that performs the calculations to determine the heat-rate change; however, it was restricted to just one enhanced tube type, Korodense LPD, and the Heat Exchange Institute (HEI) prediction method (1984) for the overall heat-transfer-coefficient calculation. All the necessary calculations are discussed in detail in this publication. For this investigation, the bundle-factor method was used to replace the calculation from the Standards for Steam Surface Condensers (the HEI method) for the overall heat-transfer coefficient and the general set of enhanced-tubes attributes replaced those of Korodense LPD.

Bundle Factor Method. The performance of large power-plant condensers can not be accurately predicted with an overall heat-transfer coefficient method – the standard text-book method – unless some correction is made for the complexities existing on the shellside of the condenser. It is well known that there are substantial variations of the condenser performance throughout the tube bundle resulting from local variations of the outside condensing coefficients. The reasons for these variations are vapor shear, condensation inundation, the saturation-temperature depression due to the vapor pressure drop, and the presence of noncondensable gases. It is obvious that very complex three-dimensional computer codes are needed if these important effects are to be considered. However, there are some

empirical rating methods that have been developed that permit the use of much simpler overall heat-transfer-coefficient method. One is the often used but much maligned HEI method; however, there is a general movement away for this method.

A method called the bundle-factor method, which was initially proposed by Sklover and Grigor'ev (1970), is now becoming accepted. The bundle factor is a multiplier on the single-tube Nusselt prediction, which is experimentally determined, to account for complex shellside effects listed above. The basis of this method is as follows. Calculate the outside heat-transfer coefficient based on the single-tube Nusselt condensing equation and an empirical multiplier, the bundle factor. Then calculate the inside heat-transfer coefficient using an accepted correlation based on the assumption of uniform flow rate of the coolant to all the tubes. Finally, calculate the tube metal resistance. The overall heat transfer coefficient is then calculated in the normal manner by adding the four resistances: shellside, inside, metal resistance, and inside fouling factor. This bundle factor method has received considerable attention lately and has been proposed for computing the fouling resistances in the revised ASME Surface Condenser Code (Burns et al., 1991) and used by Rabas et al. (1993) and Rabas and Elliott (1993) to obtain fouling rates of power-plant condensers with plain and enhanced tubes. Also, this method was proposed for sizing power-plant condensers by Taborek and Tsou (1991).

The magnitude of the bundle factor depends on such factors as the condenser size, tube material, internal design, air infiltration, and the performance of the gas-removal equipment. For more discussion on the magnitude of the bundle factor, see Taborek and Tsou (1991), Rabas (1992), Rabas et al. (1993), and Rabas and Elliott (1993). Typical bundle factors range from about 0.35 to 0.8 for power-plant condensers. An important observation for large power-plant condensers is that the condensing heat-transfer coefficient of steam is lower than the inside forced-convection heat-transfer coefficient of water because of the magnitude of bundle-factor correction. The outside heat-transfer performance is therefore controlling. This point is often overlooked by some previous investigators because the experimental facility consisted of a single and/or a small number of tubes where bundle inundation, saturation-temperature depression due to the vapor pressure drop, and noncondensable gas pockets do not exist.

Enhanced Tube Heat Transfer Coefficients. For this study, the plain-tube inside heat transfer coefficient was evaluated with the commonly used Petukhov and Popov correlation (Petukhov, 1970). For the enhanced tubes, the same correlation was used; however, the inside heat-transfer coefficient was increased with the inside enhancement-factor multiplier, e_i . The plain-tube pressure drop was calculated with the commonly used Blasius friction-factor correlation. For the enhanced tubes, this friction-factor correlation was used but corrected by the efficiency index (actually a multiplier of e_i/η). The plain-tube outside condensing heat-transfer coefficient was calculated with the single-tube Nusselt equation and different assumed values of the bundle factor. For the enhanced tubes, again the same prediction method was used but now the bundle factor was increased with the outside enhancement factor multiplier, e_o . This calculation procedure therefore permits the evaluation of the overall heat-transfer coefficients and tubeside pressure drop in a generic manner for the enhanced tubes.

Appendix 2 contains sample output values for two months that were obtained with this modified computer program. These results clearly show some interesting characteristics such

as the reduced flow with enhanced tubes, the substantially superior performance in the summer, and the general magnitude of the flow parameters.

RESULTS

The effect on the four attributes of the enhanced tube - inside enhancement, e_i ; the outside enhancement, e_o ; efficiency index, η ; and the fouling-rate multiplier, f_m - on the heat rate for the two condensers are now presented. The results are divided into three categories: the condenser design, the plain-tube inside fouling rate, and the fouling impact on the pressure drop.

Condenser Design

Tables 4 and 5 list the heat-rate changes (negative sign means a reduction) for condensers #1 and #2, respectively, for the case of no tubeside fouling and with bundle factors of 0.45 and 0.6. First note that the outside and/or inside enhancements are more effective in reducing the heat rate if the condenser is a poorer performer - a lower bundle factor. Also note that enhancing the outside is more effective at reducing the heat rate than enhancing the inside; however, an asymptotic behavior of the heat-rate reduction is obtained with an increasing outside enhancement. An outside enhancement value of about 1.5 is sufficient to obtain almost the maximum possible gain. This level is possible with low-fin tubes that have already been used in Japan (Nosetani et al., 1989) or with some coating as proposed by Webb et al. (1991b).

Table 4 shows that increasing the inside enhancement leads to a heat-rate increase when there is no outside enhancement or $e_o = 1$. There appears to be no advantage to have inside enhancement levels greater than about 1.5. Also, the efficiency-index attribute appears to be important only if there is little outside enhancement or e_o is close to unity as is the case with the currently used spirally indented tube type.

Plain-Tube Fouling Rates

Consider next the fourth attribute of an enhanced tube, the fouling-rate multiplier. For this investigation, the bundle factor will be set to 0.5 and efficiency index to unity. These are reasonable values and we have already established how changes in their magnitudes impact on the heat rate.

Two straight-line, plain-tube fouling rates that are typical for river waters (Rabas et al. 1993) are used for the following investigation: 5.3×10^{-5} and 10.6×10^{-5} ($\text{m}^2 \text{ } ^\circ\text{C/W}$)/year (0.0003, and 0.0006, ($\text{hr } ^\circ\text{F ft}^2/\text{Btu}$)/year). A straight-line fouling rate is a reasonable assumption based on these limited data from three different rivers although there was some seasonal variations due to the river-water temperature variations. Normal cleaning cycles for plants using river water as the coolant usually are about one year or more. The heat-rate analysis will be performed assuming these plain-tube fouling rates for one full year of operation without cleaning.

A straight-line fouling rate of 63.5×10^{-5} ($\text{m}^2 \text{ } ^\circ\text{C/W}$)/year (0.0036 ($\text{hr } ^\circ\text{F ft}^2/\text{Btu}$)/year) will also be considered but with different cleaning periods. This fouling rate is representative for tubes with high-fouling estuary sea waters (Rabas and Elliott, 1993). Common practice is to use monthly and, in some special cases, even daily cleanings for these difficult cooling-water sources. The actual fouling rates and cleaning cycle lengths vary from site to site; however,

these fouling rates were obtained by analyzing data taken to monitor the performance of different condensers and therefore are representative of "real-world" cooling-water sources.

Two different enhancement fouling-rate multipliers were selected: the same fouling rate, $f_m = 1.0$ and a fouling rate that is twice that of a plain tube, $f_m = 2.0$. A comparison of the fouling-rate data for plain and spirally indented enhanced tubes presented by Rabas and Schaefer (1993) clearly showed the following: in many cases, they were essentially the same, and at worst, they are at about twice as great as smooth tubes. These comparisons were obtained with tubes of the spirally indented type, which has been identified in laboratory experiments by Somerscales et al. (1991) and Webb and Chamra (1991a) as having a low fouling propensity. Because there is no field data to confirm or contradict the use of these values for other enhanced tube types, the above f_m values will be used although they were obtained for the spirally indented tube types.

Consider first the river-water applications. Tables 6 and 7 show that the heat rates are reduced if the plain-tube fouling rate is increased and if the fouling rates of the plain and enhanced tubes are similar. If no outside enhancement exists or $e_o = 1$, note that the enhanced tubes are not cost effective because of the cooling-water fouling, especially if the enhanced-tube fouling multiplier exceeds that of a plain tube. If the level of the outside enhancement is 1.5 and above, enhanced tubes are indeed cost effective (heat-rate reductions of about 20 Btu/KW-hr) even with minimal inside enhancement. The magnitude of the inside enhancement level, 1.5 or 2.0, has almost no impact on the heat-rate improvement, which is somewhat of a surprise.

Consider next the impact of different cleaning periods for the higher fouling-rate coolant. Tables 8 and 9 show that the heat-rate reduction increases (more negative) with increasing cleaning periods if the fouling rates are assumed to be identical. With twice the fouling rate, enhanced tubes still produce a significant heat-rate reduction if a one or two month cleaning period is selected; however, this advantage disappears for a six month period. The heat-rate reduction obtained with enhanced tubes for high-fouling-rate cooling waters was at first a surprise but it is consistent with the finding that a poorer performing condenser due to design and/or fouling will always benefit more (larger heat-rate reduction) by retubing with enhanced tubes. Again note that there is no need to obtain an inside enhancement level greater than about 1.5 and the outside enhancement level of about 1.5 also appears to be near the optimum.

Fouling Impact on Pressure Drop

The only data that the authors are aware of dealing with the fouling of plain and enhanced tubes was presented by Rabas et al. (1990) and is reproduced in Figure 4. The normal cleaning frequency at the site where these data were obtained was about one year. It will be assumed that the friction factor values labeled TVA correlation - 51% clean was obtained in either one-half or a full year, the initial values being obtained from the Blasius correlation. For the enhanced tubes, it will be assumed that the friction factor increases from the Wolverine correlation (100% clean) to the TVA correlation labeled 73% clean in either one-half and a full year.

Tables 10 and 11 show heat-rate changes obtained with enhanced tubes with the bundle factor fixed at 0.5 and the efficiency index equal to unity. Because the friction-factor data were obtained from a condenser with river-water as the coolant, only the $5.3 \times 10^{-5} \text{ (m}^2 \text{ }^\circ\text{C/W)}/\text{year}$ ($0.0003 \text{ (hr } ^\circ\text{F ft}^2/\text{Btu)}/\text{year}$) fouling rate is considered in these tables. Again note the pattern previously obtained – the heat-rate reduction increases due to any factor that

reduces the condenser performance. The increased pressure drop due to fouling impacts negatively on the thermal performance and leads to greater heat-rate reductions with enhanced tubes. Note that this impact is not substantial except for low outside enhancement values. Tables 10 and 11 show that the friction-factor increase due to fouling leads to a small but favorable reduction of the heat rate if the condenser is retubed with enhanced tubes.

OUTSIDE HEAT-TRANSFER ENHANCEMENT

The consistent finding from these heat-rate calculations is that the most important attribute of enhanced tubes is the outside heat-transfer enhancement level. The apparent reason is that the outside condensing heat-transfer resistance is controlling when the bundle-factor correction is applied. However, Table 2 shows that the currently used spirally indented type produces very little or no outside enhancement. This concept – the outside heat-transfer resistance is controlling – is not apparent especially when experiments are performed on single-tubes or small experimental facilities because the adverse effects of condensate inundation, saturation-temperature depression due to the pressure drop, and noncondensable gas pockets are not present. For "real world" condensers, the outside performance can be reduced by about 50% because of these effects.

Is it possible to actually obtain outside enhancement levels of about 1.5 or greater if these adverse factors are present? The best guide is the local heat-transfer-coefficient data presented by Nosetani et al. (1989) obtained with 35 low-fin tubes selectively placed in a large power-plant condenser. The fin density and height were 354 fins/m (9 fins/in) and 0.7 mm, respectively, and there was no inside enhancement although the inside diameter was slightly smaller than the plain tubes. Adjacent to each instrumented enhanced tube was an instrumented plain tube. As a result, the local overall heat-transfer-coefficient enhancements can be measured directly and values ranging from 1.18 to 1.51 were obtained for the 35 tubes. Obviously, the outside heat-transfer coefficient must be larger suggesting that e_o equal to 1.5 are indeed possible.

Most of the outside enhancement was obtained in the inner portion of the bundle and, in particular, in the bottom half. Around the perimeter of the bundle, the enhancement levels were all close to unity. Nosetani et al. (1989) argued that the large vapor velocities existing as the steam enters the bundle shear the condensate from the tubes regardless of the type. However, within the bundle, the falloff of the performance with bundle penetration is significantly less with the finned tube. This finding existed in the lanced portion of the bundle as well as in the close-packed center portion. Low-fin tubes appear to be one type of enhanced tube that can be used to achieve large heat-rate reductions.

CONCLUSIONS

This investigation showed that the outside enhancement is the most important attribute for typical power-plant condensers (surface areas greater than about 2300 m² (25,000 ft²)). For the two condensers studied, an e_o value of about 1.5 leads to heat-rate savings of about 20 Btu/kW-hr. Experiments conducted with low-fin tubes by Nosetani et al. (1989) in a power-plant condenser suggest that outside enhancement levels of this magnitude are possible. Increasing the inside enhancement is less effective because of the increased pressure drop that leads to a reduction of the coolant flow rate and velocity. The efficiency index, which is a measure of the pressure drop increase, is only important if the outside enhancement level is

very low. With an outside enhancement level of 1.5, two other interesting findings were obtained: enhanced tubes lead to greater energy savings if (1) the cooling water has more rather than less fouling (even with twice the fouling rate for the enhanced tubes), and if (2) the condenser is a poor performer (a poor design) rather than a good design. A general pattern was observed – the heat-rate reduction increases due to any factor that reduces the condenser performance such as a poor design, more rather than less fouling, and increased tubeside pressure drop due to the fouling. The major reason for this pattern is the shape of the low-pressure turbine heat-rate correction curve.

NOMENCLATURE

e_i	inside enhancement level
e_o	outside enhancement level
f_m	enhanced-tube fouling-rate multiplier
f_r	plain-tube fouling rate
Pr	Prandtl number
U	Average overall heat-transfer coefficient
η	efficiency index, $\eta = e_i / (\Delta P_e / \Delta P_i)$
ΔP	pressure drop increase multiplier due to fouling
ΔP_p	tubeside pressure drop of a plain tube
ΔP_e	tubeside pressure drop of the enhanced tube

APPENDIX 1

This appendix presents the detailed information of the two condensers used for these incremental heat-rate evaluations. These comparisons were for straight tube-for-tube replacements with the same tube diameter and length values. All the information for both condensers is contained in Table 12. Curves of the cooling-water pump-head variation and the % heat-rate with backpressure are shown in Figures 2 and 3, respectively.

APPENDIX 2

Table 13 presents monthly outputs – January and July – for condenser #1. These results are presented to help the reader understand the enhanced tubes alter the characteristics of the condenser and plant operation. First note that the heat rate is increased in January but is reduced by 69.25 Btu/KW-hr in July. The major reason is, of course, is the coolant inlet temperature; the low-pressure turbine is choked (flat portion of heat-rate correction curve shown in Figure 3) in January but not in July. Also note the larger tubeside fouling resistance and the % pressure-drop increase due to fouling in July. These factors also are responsible for the dramatic heat-rate reduction in July. Table 13 also shows that the plain-tube overall heat-transfer coefficient, U , is not significantly changed because the increased fouling is offsetting the higher heat-transfer coefficients that exist at higher temperature levels. However, note the significant reduction of U for the enhanced tube because fouling can not offset the higher temperature level with twice the fouling rate and higher initial U values. Also note the flow reduction caused by the increased pressure drop; however, it is not large because of the steep pump curve and low level of pressure-drop increase with $e_i = 1.5$ and $\eta = 1$.

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Figure 1 Spirally indented tube illustration. Dimensions of Korodense LPD used by TVA (mm): $d_o = 22.2$, $e = 0.51$, $p = 9.3$.

Figure 4 Friction-factor correlations for Korodense LPD and plain tubes and the effect of the fouling rates.

Figure 3 Cooling-Water Head-Capacity Curves (Pump Curves) for Condenser #1 and Condenser #2

Figure 4 % Heat-Rate Correction Curves for Condenser #1 and Condenser #2

**TABLE 1 TENNESSEE VALLEY AUTHORITY CONDENSERS
RETUBED WITH KORODENSE LPD TUBES**

<u>Unit</u>	<u>Bundle 1</u>	<u>Bundle 2</u>
Gallatin 1	7/80	7/80
Gallatin 2	5/82	
Shawnee 5		5/87
Shawnee 6		4/85
Shawnee 7		11/83
Shawnee 8	5/88	5/86
Shawnee 9	9/87	9/87
Shawnee 10	10/87	10/87
Kingston 1	4/90	4/90
Kingston 2	10/91	10/91
Kingston 4	5/91	5/91
Kingston 7	9/90	9/90
Kingston 8	7/91	7/91
Kingston 9	4/91	4/91

**TABLE 2 ATTRIBUTES OF KORODENSE LPD
INSIDE ENHANCEMENT AND EFFICIENCY INDEX**

Pr No.	Reynolds Number					
	20,000		40,000		80,000	
	e_i	η	e_i	η	e_i	η
5	1.71	0.78	1.66	0.74	1.61	0.69
8	1.80	0.82	1.73	0.77	1.67	0.71

Based on correlations in the Wolverine Catalog (Wolverine Engineering Data Book II, 1984) and the Petukhov and Popov correlation (Petukhov, 1970)

Outside Enhancement, $e_o \equiv 1.0$ to 1.10 (Rabas, 1989)

Fouling Rate Multiplier, f_m in the 1 to 2 range (Rabas et al., 1993)

Table 3 Factors Influencing The Cost Effectiveness of Retubing with Enhanced Tubes

- Enhanced tube attributes
 - inside heat-transfer enhancement
 - outside heat-transfer enhancement
 - efficiency index
 - fouling multiplier
- Cost factors
 - fuel cost
 - incremental tube-cost increase
- Condenser design conditions
 - tube diameter
 - tube length
 - tubeside velocity
 - tube number
 - tube passes
 - tube material
- Circulating water pump information
 - head-capacity curve
 - horsepower-capacity curve
 - external piping head losses
- Heat-rate correction with back pressure
- Cooling-water characteristics
 - monthly average temperatures
 - cleaning frequency

**TABLE 4 HEAT-RATE CHANGES FOR CONDENSER #1
WITH DIFFERENT BUNDLE FACTORS AND NO FOULING**

No Fouling		Bundle Factor is 0.6		
e_i	η	$e_o=1.0$	$e_o=1.5$	$e_o=2.0$
1.0	1.00	0.0	-12.24	-15.22
1.5	0.75	-0.52	-11.31	-14.12
1.5	1.00	-4.62	-14.21	-16.38
1.5	1.25	-6.50	-15.21	-17.54
2.0	0.75	0.52	-9.68	-13.84
2.0	1.00	-4.80	-13.36	-15.53
2.0	1.25	-7.35	-15.30	-17.12
No Fouling		Bundle Factor is 0.45		
1.0	1.0	0.0	-30.73	-37.73
1.5	0.75	-3.13	-30.71	-36.80
1.5	1.00	-9.17	-34.16	-39.45
1.5	1.25	-12.49	-35.93	-40.70
2.0	0.75	-2.50	-29.45	-35.15
2.0	1.00	-10.07	-34.03	-38.85
2.0	1.25	-14.27	-36.38	-40.79

**TABLE 5 HEAT-RATE CHANGES FOR CONDENSER #2
WITH DIFFERENT BUNDLE FACTORS AND NO FOULING**

No Fouling		Bundle Factor is 0.6		
e_i	η	$e_o=1.0$	$e_o=1.5$	$e_o=2.0$
1.0	1.0	-0.15	-30.54	-40.17
1.5	0.75	-9.80	-36.12	-44.39
1.5	1.00	-12.35	-37.53	-45.96
1.5	1.25	-14.66	-39.00	-47.37
2.0	0.75	-12.54	-37.98	-46.18
2.0	1.00	-17.18	-41.14	-48.98
2.0	1.25	-18.84	-42.41	-50.06
3.0	0.75	-10.40	-35.48	-43.47
3.0	1.00	-17.92	-41.37	-48.65
3.0	1.25	-22.01	-44.63	-51.20
No Fouling		Bundle Factor is 0.45		
e_i	η	$e_o=1.0$	$e_o=1.5$	$e_o=2.0$
1.0	1.0	0.70	-59.19	-77.08
1.5	0.75	-10.87	-65.88	-88.66
1.5	1.00	-13.79	-60.03	-84.07
1.5	1.25	-17.63	-70.65	-85.55
2.0	0.75	-15.77	-69.35	-84.53
2.0	1.00	-21.75	-73.35	-87.69
2.0	1.25	-24.20	-74.80	-88.96
3.0	0.75	-14.23	-67.23	-82.02
3.0	1.00	-24.66	-74.26	-87.92
3.0	1.25	-30.28	-77.93	-91.18

**TABLE 6 EFFECT OF RIVER-WATER FOULING RATES
ON THE HEAT-RATE FOR CONDENSER #1
BUNDLE FACTOR EQUAL TO 0.5 AND $\eta = 1.0$**

e_i	f_r	f_m	$e_o=1.0$	$e_o=1.5$	$e_o=2.0$
1.5	0.0	1.0	-6.81	-24.50	-28.46
1.5	0.0003	1.0	-8.41	-31.52	-37.04
1.5	0.0003	2.0	3.81	-25.11	-32.58
1.5	0.0006	1.0	-10.67	-39.59	-47.06
1.5	0.0006	2.0	23.27	-20.41	-32.70
2.0	0.0	1.0	-7.78	-24.18	-27.68
2.0	0.0003	1.0	-10.32	-31.53	-36.62
2.0	0.0003	2.0	1.19	-25.44	-32.50
2.0	0.0006	1.0	-13.29	-39.92	-46.98
2.0	0.0006	2.0	20.21	-22.02	-38.30

**TABLE 7 EFFECT OF RIVER-WATER FOULING RATES
ON THE HEAT-RATE FOR CONDENSER #2
BUNDLE FACTOR EQUAL TO 0.5 AND $\eta = 1.0$**

e_i	f_r	f_m	$e_o=1.0$	$e_o=1.5$	$e_o=2.0$
1.5	0.0	1.0	-14.97	-55.49	-67.72
1.5	0.0003	1.0	-13.95	-62.70	-77.38
1.5	0.0003	2.0	7.56	-46.90	-64.99
1.5	0.0006	1.0	-17.16	-71.62	-89.71
1.5	0.0006	2.0	41.44	-31.24	-55.89
2.0	0.0	1.0	-20.23	-59.02	-70.95
2.0	0.0003	1.0	-22.69	-67.53	-81.66
2.0	0.0003	2.0	-0.61	-53.17	-69.50
2.0	0.0006	1.0	-25.33	-77.89	-94.22
2.0	0.0006	2.0	29.39	-38.95	-61.72

**TABLE 8 EFFECT OF CLEANING PERIOD (MONTHS) ON
THE HEAT RATE FOR CONDENSER #1**

**BUNDLE FACTOR EQUAL TO 0.5 AND $\eta = 1.0$
FOULING RATE EQUAL TO 0.0036 (HR F FT²/BTU)/YEAR**

e_i	months	f_m	$e_o=1.0$	$e_o=1.5$	$e_o=2.0$
1.5	∞	1.0	-6.81	-24.50	-28.46
1.5	1	1.0	-8.29	-30.74	-36.36
1.5	1	2.0	1.78	-25.39	-32.36
1.5	2	1.0	-9.96	-38.04	-45.05
1.5	2	2.0	19.99	-20.65	-31.92
1.5	6	1.0	-16.42	-68.53	-85.46
1.5	6	2.0	∞	∞	∞
2.0	∞	1.0	-7.78	-24.18	-27.68
2.0	1	1.0	-10.04	-30.66	-35.55
2.0	1	2.0	-0.73	-25.67	-32.21
2.0	2	1.0	-12.86	-38.45	-45.08
2.0	2	2.0	15.89	-21.71	-32.56

**TABLE 9 EFFECT OF CLEANING PERIOD (MONTHS) ON
THE HEAT RATE FOR CONDENSER #2**

**BUNDLE FACTOR EQUAL TO 0.5 AND $\eta = 1.0$
FOULING RATE EQUAL TO 0.0036 (HR F FT²/BTU)/YEAR**

e_i	months	f_m	$e_o=1.0$	$e_o=1.5$	$e_o=2.0$
1.5	∞	1.0	-14.97	-55.49	-67.72
1.5	1	1.0	-13.78	-61.22	-76.79
1.5	1	2.0	3.14	-49.79	-66.75
1.5	2	1.0	-25.46	-76.41	-92.22
1.5	2	2.0	24.47	-43.29	-65.46
1.5	6	1.0	-26.63	-114.9	-145.8
1.5	6	2.0	∞	∞	∞
2.0	∞	1.0	-20.23	-59.02	-70.95
2.0	1	1.0	-22.46	-66.77	-80.27
2.0	1	2.0	-3.56	-54.90	-70.68
2.0	2	1.0	-25.93	-76.41	-92.22
2.0	2	2.0	24.47	-43.29	-65.45

TABLE 10 EFFECT OF FOULING OF THE PRESSURE-DROP INCREASE DUE TO TUBESIDE FOULING ON THE HEAT RATE OF CONDENSER #1

**BUNDLE FACTOR EQUAL TO 0.5 AND $\eta = 1.0$
FOULING RATE EQUAL TO 0.0003 (HR F FT²/BTU)/YEAR**

f_m	ΔP	$e_0=1.0$	$e_0=1.5$	$e_0=2.0$
1	NA	-8.41	-31.52	-37.04
1	1	-9.76	-35.26	-36.45
1	2	-9.94	-37.47	-44.80
2	NA	3.81	-25.11	-32.58
2	1	4.09	-27.65	-36.45
2	2	5.10	-29.19	-38.75

TABLE 11 EFFECT OF FOULING OF THE PRESSURE-DROP INCREASE DUE TO TUBESIDE FOULING ON THE HEAT RATE OF CONDENSER #2

**BUNDLE FACTOR EQUAL TO 0.5 AND $\eta = 1.0$
FOULING RATE EQUAL TO 0.0036 (HR F FT²/BTU)/YEAR**

f_m	ΔP	$e_0=1.0$	$e_0=1.5$	$e_0=2.0$
1	NA	-13.95	-62.70	-77.38
1	1	-15.02	-57.64	-70.31
1	2	-15.67	-58.50	-71.21
2	NA	7.56	-46.90	-64.99
2	1	-12.04	-55.68	-68.72
2	2	-12.68	-56.53	-69.61

**TABLE 12 PLANT AND CONDENSER DESIGN
PARAMETERS FOR CONDENSERS #1 AND #2**

<u>Design Parameters</u>	<u>#1</u>	<u>#2</u>
Condenser Geometry		
tube diameter, mm	25.4	25.4
tube wall thickness, mm	1.25	0.71
tube length, m	9.144	10.97
tube number	7720	10092
tube passes	2	1
tube material	admiralty	stainless
Plant Information		
power generation, MW	157.4	250.0
plant heat rate, Btu/KW-h	8612.0	7853.
Cooling Water Information		
design flow rate, m ³ /s	3.39	11.10
pump number	2	2
water-box nozzle diameter, m	10.97	18.29
number of nozzles	2	2
pump efficiency, %	85.	85.
monthly temperatures, °C		
January, February,	1.7, 2.2,	17.2, 18.3,
March, April,	5.0, 10.5,	20.6, 23.3,
May, June	16.7, 22.2,	26.7, 28.9
July, August,	24.4, 24.4,	30.0, 30.6,
September, October,	19.4, 13.3,	30.6, 28.9,
November, December	7.2, 2.8	23.9, 19.4

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**TABLE 13 MONTHLY AVERAGE OUTPUT COMPARISONS
OF OPERATION DATA FOR CONDENSER #1**

Condenser and Plant <u>Characteristics</u>	Plain <u>Tube</u>	Enhanced <u>Tube</u>	<u>Diff.</u>
January			
Heat Rate, Btu/KW-hr	8599.	8606.	7.31
% Heat-Rate Correction	-0.147	-0.0658	0.0815
Power Output, MW	157.6	157.5	-0.1281
Total Pump Power, KW	730.15	722.60	-7.54
Coolant Flow Rate, m ³ /s	3.410	3.246	-0.164
Condenser Total Drop, m	5.568	7.077	1.509
System Pressure Drop, m	18.58	19.31	0.729
% Pressure Drop Increase	5.76	4.40	-1.36
Tubeside Velocity, m/s	2.142	2.039	-0.103
Overall U, W/m ² °C	2452.	3787.	1335.
Fouling Resistance, m ² °C/KW	0.0044	0.0088	0.0440
Cleanliness Factor	1.01	1.60	0.59
Back Pressure, KPa	3.897	2.943	-0.954
Saturation Temperature, °C	28.67	23.91	-4.76
Temperature Rise, °C	16.76	17.59	0.83
Log-Mean Temp. Difference, °C	17.29	11.24	-6.05
Coolant Inlet Temperature, °C	1.67	1.67	
July			
Heat Rate, Btu/KW-hr	8884.0	8814.8	-69.25
% Heat-Rate Correction	3.159	2.353	-0.804
Power Output, MW	152.58	153.78	1.199
Total Pump Power, KW	712.7	697.7	14.95
Coolant Flow Rate, m ³ /s	3.200	3.039	-0.161
Condenser Total Drop, m	7.452	8.775	1.323
System Pressure Drop, m	19.48	20.07	0.594
% Pressure Drop Increase	49.19	42.91	-6.28
Tubeside Velocity, m/s	2.011	1.911	-0.100
Overall U, W/m ² °C	2565.	3067.	-502.5
Fouling Resistance, m ² °C/W	0.0572	0.1145	0.0572
Cleanliness Factor	0.652	0.800	-0.148
Back Pressure, KPa	13.67	12.41	1.257
Saturation Temperature, °C	52.23	50.28	1.95
Temperature Rise, °C	18.45	19.33	-0.89
Log-Mean Temp. Difference, °C	16.92	14.02	2.90
Coolant Inlet Temperature, °C	24.44	24.44	