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Practical Application of Condenser System Performance Monitoring to Water Treatment Decision Making

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PRACTICAL APPLICATION OF CONDENSER SYSTEM PERFORMANCE MONITORING TO WATER TREATMENT DECISION MAKING

INTRODUCTION

Escalating fuel costs and more stringent government discharge regulations have resulted in an increased emphasis by the utility industry on maintaining clean condenser heat transfer surfaces and maximum cooling water flow rate. Reduced condenser performance is a major cause of increased fuel consumption and in many cases causes load limitations. Monitoring methods for condenser fouling are needed to assist the utility manager in decision making related to surface condenser maintenance and operation practices. These are also needed to perform the chlorination minimization studies required by the Environmental Protection Agency in conjunction with NPDES permit issuance and/or modification. Monitoring methods should be supplemented with a means to directly relate reduced condenser performance to operating costs, and data should be evaluated in a format conducive to the decision-making process.

Five basic monitoring methods are available to the utility engineer and management to provide a basis for decision making:

- Physical inspection of the condenser and transfer lines for microbiological, macrobiological and inorganic fouling.
- Water chemistry monitoring and saturation level calculation.
- Sidestream heat exchangers and deposition probe monitoring of fouling factors.
- Heat transfer performance of the condenser and measurement or calculation of cooling water flow rate.
- Pressure drop and friction factor calculations, a new experimental method.

This paper discusses the practical application of these methods to monitoring condenser waterside fouling and the effectiveness of control methods. The modifications to normal methods needed to obtain meaningful information with existing plant instrumentation are discussed and related to actual case histories. The use of a combination of these methods to cross-verify data is described for minimizing error. Standard results engineering methods are presented as a means to quantify the cost of fouling.

A payout/cost matrix approach is used to evaluate fouling control options and to predict the minimum cost or maximum payout.

All data presented are based upon actual field applications. The methods are applicable to most industrial cooling water systems with at most minor modification.

These methods can be used directly for:

- Chlorine minimization studies.
- Monitoring macrobiological fouling of condensers and determining the optimum time for mechanical cleaning based upon payout.
- Monitoring the effectiveness of cooling water treatment programs, optimizing treatment rates, and calculating their payout.
- Determining when a payout exists for chemically or mechanically cleaning heat transfer surfaces.
- Comparing the effectiveness and payout of different fouling control methods.

Physical inspection can be quantified by scraping one or more tubes and weighing the deposit after drying. The deposit can be related to the surface area of the tube:

Deposit Weight/Area in lbs/ft^2 or gms/m^2

This value can be converted to average deposit thickness if the density of the foulant is known.

Micrometers can also be used to measure the thickness of deposits on the benchmark tubes at each inspection, and foulant buildup rate quantified as deposit thickness increase per unit time.

$$\text{Buildup Rate} = \frac{(\text{Thickness T2} - \text{Thickness T1})}{\text{T2-T1}}$$

Where Thickness T2 is the deposit thickness measured at the current inspection

Thickness T1 is the deposit thickness measured at the previous inspection

and T2-T1 is the time elapsed between inspections.

Certain electrochemical phenomena should also be considered during inspections. Muntz metal tube sheets will scale heavily due to electrochemical effects, while Admiralty or other alloy tubes may remain scale free.

Unfortunately, a unit should be taken totally out of service for physical inspection. In utilities, this is a costly procedure which results in lost production or reliance on a less efficient unit. This shortcoming has been "overcome" in some utilities through the practice of bringing the unit to half load and taking one condenser section out of service. The exhaust steam flow is removed from the condenser, but in most cases auxiliary steam flow is continued. This can create a dangerous situation for two reasons. First of all, any solids present in water remaining in the tubes create a deposit when the water evaporates under a partial heat load. Secondly, the auxiliary steam heat load causes tube expansion. In some cases, this can lead to a break in the vacuum seal where the tubes are rolled into the tubesheet and, eventually, cooling water leaks into the condensate. Visual inspection is also difficult due to the hot, high-moisture atmosphere. Physical inspection is recommended only when a heat exchanger or condenser can be totally removed from service for these reasons.

Physical inspection is the ultimate verification of deposit presence or removal. Foulant buildup rates can be quantified during physical inspections to provide evidence for use in the decision-making process. Metallurgical samples can also be collected at this time to provide additional evidence in the form of cost estimates of corrosion such as localized attack.

PREDICTIVE MODELLING BY SATURATION LEVELS

Routine water chemistry monitoring is a second common method for monitoring fouling potential. Traditionally, indicators such as the Langelier Saturation and Ryznar Stability indices have been used to monitor calcium carbonate deposition potential. These methods are reasonably accurate for waters with a moderate to high scale-forming potential; and their use is limited to calcium carbonate.

The primary shortcoming of these methods is their treatment of corrections for ionic activity and the estimate of alkalinity present as carbonate.[1] The application of computerized calculations has led to more accurate calculation of scale potential by including these corrections. Computerization of these methods has also allowed for calculation of indices for foulants other than calcium carbonate.[1,2,3]

WATER TREATMENT DECISION MAKING

Making decisions involving water treatment alternatives can be a frustrating experience for the utility executive. For the most part, proposals are usually laden with promises of performance, but little tangible evidence is provided to make the choices clearly defined alternatives. Complicating the process is the uncertainty of future events such as weather variations, regulations imposed by agencies, demand of electricity and the availability of fuel.

One way of treating this problem is the payoff table, an example of which is shown in Exhibit 1. This is simply a two-dimensional array of numbers arranged in columns and rows. The rows labeled A_1, A_2, A_3 represent the different *alternatives* and the columns represent the possible *states of nature* labeled SN_1, SN_2, SN_3, etc. The entry in the space at the intersection of each row and column is designated the *payoff*. This can be discounted cash flow or cost or any other tangible number. This number is a measure of the *value* (utility) to the decision maker with the selection of a given alternative and the occurrence of a given state of nature.

Under conditions of risk, with some knowledge of the frequency of occurrence of several states of nature, the decision maker must weight the payoffs associated with each alternative to arrive at that one alternative with the highest probable combination of payoffs or the lowest probable combination of costs.

For example, our case in Exhibit 1 shows an expected value of $500,000 for this alternative. It is calculated as follows:

$$\text{Expected Value } (A_1) = (.2)(250)+(.6)(350)+(.4)(600) = 500$$

Had these numbers represented cost, alternative A_1 would be eliminated and alternative A_3 selected.

This paper will attempt to give a means of quantifying *value* of a treatment alternative, to aid the utility engineer in using this type of decision-making process. The net results are better decisions, better programs, and clearer explanations to boards of directors or Public Utility Commissions on the reasons for treatment expenditures.

PHYSICAL INSPECTION

Physical inspection is the most frequently used monitoring technique for fouling and corrosion of heat exchange equipment such as surface condensers. It provides the ultimate verification of foulant presence. The degree of fouling can be estimated visually. Sampling of deposits for chemical and microbiological analysis provides positive verification of foulant composition.

Visual inspection can be misleading unless precautions are taken. In the case of microbial slime, which is 90 to 98% water, the tubes should be inspected immediately after the cooling water is drained. If a unit is inspected after the slime film has been allowed to dehydrate and dry, only a dust-like residue will remain on the tubes.

Entrance and exit turbulence effects should also be considered during visual inspection. The high turbulence at these accessible areas of the tubes can hinder deposition. The tube inlet and outlet may be deposit free or clean relative to the areas further into the tubes. The use of fiber optic equipment and the removal of a tube for splitting is recommended for this reason.

Photographs and notes should be taken on tube status and tube sheets. Tube sheet maps are recommended if time allows. This helps eliminate "memory lag" between inspections. The use of "benchmarks" provides another aid. Deposits can be removed from small areas of the tube sheets and/or marked tubes. The benchmark area can be inspected each time the condenser or heat exchanger is open. The benchmark will remain clean if fouling is under control. If not, it can be used for a semi-quantitative indicator of deposit buildup between inspections.

The saturation level is defined for calcium carbonate as follows:

$$I_S = \frac{(Ca)(CO_3)}{K_{SP}}$$

where (Ca) is the activity of calcium at the temperature and ionic strength of the water

(CO_3) is the activity of carbonate at the temperature and ionic strength of the water

K_{SP} is the solubility product at the water temperature

This predictive method is applicable to most foulants. Similar calculations including the activity corrections can be used to calculate a more accurate $_pH_s$ for the Langelier and Ryznar indices. As a matter of fact, the Langelier index, corrected for activity, can be calculated as follows:[4]

$$I_{Langelier} = \log_{10} (I_S)$$

It should be noted that the I_S refers to an equilibrium condition and ignores kinetic effects.[5] Residence time of the water is a critical factor in determining whether or not a given foulant will present a problem in operating systems.

Saturation levels can be used to model foulant buildup in a system if fouling factor or deposit thickness increases versus time have been measured using the heat transfer coefficient method or physical inspection. The saturation level is assumed to be a rate function for this method of relating water chemistry to foulant buildup rate. It can be used to estimate the expected foulant buildup over a time period T_{ev} if the average saturation level of the foulant is known for this period, as follows:

$$\text{Buildup During } T_{ev} = \frac{(I_{SAV}-1)\, T_{ev}}{\sum_{i=1}^{n} (I_{S(i)}-1)\, T_{(i)}} \text{ Buildup } (X_n - X_L)$$

Where I_{SAV} is the average saturation level for a time period during which buildup is predicted

T_{ev} is the time elapsed during the period where buildup is being predicted

$I_{S(i)}$ is the average saturation level during a historic time period

$T_{(i)}$ is the time elapsed during the time period

Buildup $(X_n - X_i)$ is the thickness increase during the period

T_1 to T_n or fouling factor increase

In the case of monthly inspections over the period of a year this would simplify to:

$$\text{Mils Buildup in July} = \frac{(I_S \text{ in July-1})\ 30 \text{ days (Thickness Buildup from Jan-Dec)}}{(I_S \text{ in Jan-1})\ (31 \text{ Days}) + (I_S \text{ in Feb-1})\ (28 \text{ Days}) + \ldots + (I_S \text{ in Dec-1})\ (31 \text{ Days})}$$

Foulant buildup during any time period can be estimated in this manner using the average saturation level for the time under study. Fouling factor buildup rates can be used in place of thickness if the data is available.

A general index to include both equilibrium and kinetic effects is currently under development. This correlation is derived from computerized saturation level calculations and empirical data on fouling factor increases under both laboratory and field conditons.

Saturation level calculations provide an excellent tool for monitoring the inorganic fouling potential of a water. The equilibrium origin of the calculations should be considered and I_S is correlated to the actual fouling potential of a given system for most accurate predictions. Foulant buildup rates predicted by this model can be added to the evidence bank for use in the decision-making process. Saturation level projected changes can also be used as States of Nature (SN) for the payout matrix.

SIDESTREAM TEST HEAT EXCHANGERS

Test heat exchangers provide a simple option to physical inspection of a surface condenser or critical heat exchanger. They allow the engineer to collect current deposit samples, measure deposit thickness increases, and inspect a tube sample for corrosion type without taking a unit off line for physical inspection.

The test exchanger is a small surface area unit. Either plant low-pressure steam or an electrical cartridge heater is used for a heat source. Cooling water flow is normally on the tubeside for steam-heated units,[6,7] or in an annulus (shellside) for electrically-heated exchangers.

It is extremely important that the heat transfer and flow regime parameters be matched as closely to the surface condenser or critical exchangers as possible. These parameters can affect deposit type and degree as outlined in a later section of this paper. Tube metallurgy should also be matched when corrosion is a parameter under study. If tube metallurgy cannot be duplicated exactly for noble alloy deposition studies, another noble alloy may normally be substituted for monitoring fouling only.

The test heat exchanger can provide a source of deposits and metal specimens under heat transfer conditions. It should be noted that the composition of test heat exchanger deposits may differ from those encountered in an operational condenser or heat exchanger. Sidestream exchanger deposits contain only those foulants encountered during the time of exposure. The operational equipment has normally been exposed to water for a longer period of time. The deposit from the operational equipment reflects water chemistry changes over the history of the unit.

Sidestream exchangers provide a viable option to physical equipment inspection. As with all monitoring methods, the data should be verified by and correlated to physical condenser (or critical exchanger) inspection and subsequent metallurgical and deposit analysis. Deposit buildup rate information from the sidestream exchanger can be used as further evidence for decision making when frequent data from condenser inspection is not obtainable. Deposit rate buildup data can be used to project future condenser performance and the cost of fouling using the economics impact section of this paper.

DEPOSITION PROBE MONITORING

Sidestream test heat exchangers can be instrumented to measure the parameters required for overall heat transfer coefficients and foulant deposition rates measured. The resulting deposition probe has the following advantages:

- Deposition rate and foulant type can be monitored without taking an operational unit off line. Deposit samples can be removed for analysis at any time, as can metal specimens for metallurgical analysis.
- Treatment procedures and minimization studies can be evaluated on a small cooling water flow. A utility manager can implement a program with a positive indication of success and an accurate treatment cost at the minimum effective dosage.
- An ongoing treatment program can be monitored to assure success. Problems due to water chemistry changes can be identified on the sidestream deposition probe prior to a deposition problem occurring on the operating unit. Corrective actions such as treatment rate changes can be made before deposition occurs in the operating condenser or critical heat exchanger.

- Treatment rates can be optimized on line as water chemistry changes seasonally. In many once-through cooling systems, lower treatment rates will successfully control fouling in the winter versus the summer. The lower treatment rates can be evaluated on the sidestream deposition probe prior to making the change on the entire cooling system. This allows treatment at the minimum rate and reduces annual treatment costs.

- Sidestream studies can be conducted on a water source for a plant under construction to predict the waterside fouling problems that will be encountered and the optimum fouling prevention programs. Fouling factors from the deposition probes can be extrapolated to the condenser and the economic impact of fouling predicted before the condenser or other heat transfer devices are on line.

There are many configurations possible for deposition probes. They can vary from small exchangers with electrical or steam heat sources[8,9,10,11,12] to pilot surface condensers.[13] An example deposition probe (Pilot Heat Exchanger) is depicted in Exhibit 2.

All of the configurations are used in a similar manner. The cooling water flow regime and heat transfer conditions are matched as closely as possible to the operating unit they are to simulate, and the overall heat transfer coefficient (or waterside heat transfer coefficient) can be predicted accurately for the operating conditions. In the case of corrosion studies, the tube metallurgy should also be matched.

Operational parameters with a great impact upon deposit formation are:

- The mechanical shear force of the cooling water on the tube walls as it passes through the condenser or exchanger; a function of the flow regime present in the tubes (e.g., turbulent, transitional, or laminar).

- The bulk temperature of the cooling water and, more importantly, the tube wall temperature.

These parameters affect deposition by basic mechanisms:

- The *mechanical equilibrium* for deposition (or growth) wherein loose suspended solids deposits form and are removed by the mechanical action of the water. Initially, such deposits, or slime, form at a more rapid rate than the rate at which they are removed. As the loosely adherent deposit thickness increases, the shear force promotes a greater removal rate. This is the reason that slime or loose suspended solids deposits form rapidly and then reach a "plateau" or equilibrium point. Deposit thickness varies about this equilibrium value.[14,15]

- The *chemical equilibrium* for deposition of low solubility compounds in the water is affected by temperature. Foulants such as $CaCO_3$ may be soluble at the inlet water temperature but precipitate as the bulk water temperature increases in the condenser or exchanger. The tube wall is the highest temperature area of the equipment and the point where sparingly, inversely soluble compounds are most likely to precipitate and form deposits.

- The *growth rate* for microbiological slime is also controlled by the bulk water and tube wall temperatures. Microorganisms grow more quickly as temperature increases within the operational temperature range encountered in a surface condenser. The temperature of the cooling water also affects the flow regime and shear force for constant velocity.

Operational parameters are normally selected to either match the operating equipment simulated or to provide conditions more prone to fouling. Heat flux and tube wall temperature are run higher than the operating equipment when it is desired to have the deposition probe foul before the operating equipment such as a surface condenser. The deposition probe is also operated at a lower flow velocity (Reynolds Number) in this case. This allows a built-in "safety factor" so that corrective action can be taken on the condenser or other critical heat transfer device prior to fouling.

The important parameters must be monitored on a frequent basis (e.g., hourly). Data loggers are commonly used to collect the data for this reason. All of the operational parameters should be held as constant as possible to minimize correlation curve errors. A complete deposition monitor is depicted in Exhibit 3, including data acquisition and control equipment. This particular unit is equipped with two probes. This allows for comparison between treated and untreated conditions in a once-through cooling system.

The predicted overall heat transfer coefficient for the deposition probe must be calculated, as well as the observed overall heat transfer coefficient so that fouling factors can be calculated from the relationship:

$$r_{fouling} = \frac{1}{U_{observed}} - \frac{1}{U_{predicted}}$$

The actual overall heat transfer coefficient ($U_{observed}$) for the deposition probe is calculated from the familiar relationship:

$$U_{observed} = Q \,/\, (\text{Area} \cdot \text{LMTD})$$

Where Q is the heat flow in Btu/hour

Area is the outside heat transfer surface area

LMTD is the logarithmic mean temperature difference.

The predicted overall heat transfer coefficient is calculated from the following relationship:

$$U_{predicted} = \frac{1}{\frac{1}{h_{cw}} + \frac{x}{K} + \frac{1}{h_p}}$$

Where h_{cw} is the waterside heat transfer coefficient

x is the tube wall thickness

K is the thermal conductivity of the tube wall

h_p is the process side film coefficient derived empirically.

Fouling factor data from deposition probes can also be extrapolated to an operating surface condenser to predict the impact of fouling on its operation through the relationship:

$$\frac{1}{U_{fouled}} = \frac{1}{U_{SHEI}} + r_{fouling}$$

Where U_{fouled} is the operating value expected for a condenser fouled to the extent measured on the deposition probe

U_{SHEI} is the overall heat transfer coefficient for a condenser with perfectly clean tubes, as calculated by the Standards of the Heat Exchanger Institute.[16]

$r_{fouling}$ is the fouling factor measured on the deposition probe.

The derivation of the correlations and measurements required are covered in references 8 through 13. The application of statistics for data screening is covered by Bird.[17]

Deposition probes have primarily been used for the evaluation of inorganic deposition and biofouling. A strong correlation has been obtained between fouling factors measured on deposition probes and those observed on operating surface condensers for inorganic fouling and its control,[10] and for microbial slime fouling.[13,14]

These studies have demonstrated that the deposition probes foul with the same type of deposit as the operating equipment, and that control methods found effective on the deposition probes worked equally well on the operating equipment. Deposition probes can also be used effectively to predict and solve another problem of economic importance: underdeposit corrosion.

Deposition probe data can be used to collect decision-making evidence with regards to foulant type and buildup rate. They can also be used to determine the most cost-effective treatment program for use in the payout matrix.

HEAT TRANSFER COEFFICIENT MONITORING AND CONDENSER PERFORMANCE

Heat transfer coefficients can provide an excellent means for monitoring foulant buildup or removal on heat transfer surfaces. In certain cases, they can also be used as a basis for conservatively estimating the economic impact of cooling waterside fouling on operational costs. This section describes the practical application of heat transfer performance monitoring, and its use in providing tangible economic evidence for water treatment decision making.

Engineers use an overall heat transfer coefficient (U value) to describe the performance of a heat transfer device. As foulants build up on heat transfer surfaces, the U value decreases. As foulants are removed, the U value increases. The overall heat transfer coefficient describes the total thermal resistance between the process side and the cooling water, in the terms of thermal conductivity. It is defined as:

$U_{observed} = Q / (A \cdot TDF)$ and has the units Btu/(ft^2 hr F°)

where Q is the heat flow rate in Btu/hr

A is the surface area in ft^2 (normally the outside heat transfer surface area)

TDF is the temperature driving force or thermal potential difference between the process and the cooling water in F°. The logarithmic mean temperature difference is normally used for the calculation of this thermal driving force.

Factors other than fouling will affect the operating "U" value of a heat transfer device. For this discussion, these factors will be limited to an exchanger with a pure process stream—a surface condenser.

Cooling water flow rate and temperature are parameters with perhaps the greatest impact upon the overall heat transfer coefficient of a surface condenser. Both parameters directly affect the thickness of the water film boundary layer at the tube wall/cooling water interface. The thickness of this film is the primary controlling factor for the overall heat transfer coefficient of a clean condenser. It is the largest single variable resistance to heat transfer.

This boundary layer film thickness increases inversely to cooling water flow velocity. As a result, the overall heat transfer coefficient decreases when cooling water flow decreases. The higher the cooling water flow velocity, the higher the expected "U" value, all other parameters being constant.

Cooling water temperature also affects the boundary layer film thickness. The film thickness is a function of cooling water viscosity, as well as velocity. As temperature decreases and viscosity increases, the film thickness increases. So a surface condenser overall heat transfer coefficient will decrease in the winter and increase in the summer, as an effect of the changing cooling water temperature.

The cooling water flow velocity and temperature (viscosity) effects determine the waterside heat transfer coefficient. They can be modeled by establishing a relationship between waterside heat transfer coefficient and the dimensionless number groups: the Reynolds and Prandtl numbers.

The thermal conductivity of the tube wall in a condenser also affects the overall "U" value. This value does not vary significantly in the operational temperature range.

Condensate loading in a surface condenser also affects the operating "U" value. At low loads and condensate loading, the condensate flow down the tubes is "slow" in comparison to higher loads and condensate loading. The increase in condensate flow rate with increasing load results in a decreased condensate film thickness. As a result, the overall heat transfer coefficient for the condenser will increase with load.

One other factor can drastically affect the overall heat transfer coefficient of a surface condenser— noncondensible gas blanketing of the tubes on the steamside. Utility surface condensers are normally equipped with an air removal section to eliminate this problem. High air leakage can overwhelm the air removal equipment. The air (noncondensible gases) blankets the tubes and adds a resistance to heat transfer. Failure of the air ejectors can lead to extremely misleading performance monitoring. A condenser with noncondensible gas blanketing will behave as if it were heavily fouled on the waterside. Air ejectors should be checked prior to any performance monitoring of a condenser for this reason.

STANDARDS OF THE HEAT EXCHANGE INSTITUTE

The Standards of the Heat Exchange Institute for Surface Condensers outlines a procedure for calculating the performance of a condenser.[16] The measurement techniques required are covered in the "ASME Power Test Code for Steam Condensing Apparatus."[18]

The Standards of the Heat Exchange Institute allow the utility engineer to predict the overall "U" value for a condenser under the operating conditions at the time of measurement. This allows for the calculation of reduction in effective surface area due to fouling (cleanliness), and the calculation of fouling factors. The operating parameters impacting the expected "U" value (as previously described) are treated as corrections.

The operating or observed "U" value at the time of measurement is calculated from the relationship:

$$U = Q/(\text{Area} \cdot \text{LMTD})$$

as previously described.

The expected "U" value for a new condenser is calculated from the relationship:

$$U_{\text{expected}} = C \cdot C_{\text{MET}} \cdot C_T \cdot C_L \cdot V^{1/2}$$

where C is a geometry constant for the tubes (SHEI Fig. 3, p. 6)[16]

C_{MET} is a constant for the thermal conductivity of the tube metallurgy (SHEI Table 1, p. 4)[16]

C_T is a correction for cooling water temperature based upon the inlet temperature ($C_T = 1.0$ at 70F°) (SHEI Fig. 2, p. 6)[16]

C_L is a correction for condensate loading (Exhibit 4). $C_L = 1.0$ at a condensate loading of 8.0 lbs/(ft^2 hr)

V is the cooling water velocity through the tubes.

Data from performance measurements are normally presented in the form of percent cleanliness (% effective surface area) which is calculated from the relationship:

$$\% \text{ Cleanliness} = 100\% \, (U_{\text{observed}}/U_{\text{expected}})$$

The percent cleanliness (also called effective surface area) describes the reduction in surface area due to fouling. It is a conductivity ratio.

The same data can be used to more accurately describe fouling through the calculation of a fouling factor (r_f):[18]

$$r_f = (1/U_{observed}) - (1/U_{expected})$$

The fouling factor expresses the thermal resistance of the fouling. Cleanliness expresses the thermal resistance of the condenser as thermal conductivity. Thermal resistance is the more versatile form. Fouling factors can be converted to foulant thickness, for example, if the thermal conductivity of the foulant is known through the relationship:

$$x = r_f \cdot K$$

where x is the foulant thickness in inches:

r_f is the fouling factor in (ft^2 hr F°/Btu)

K is the thermal conductivity of the foulant at the average deposit temperature in (Btu in) / (ft^2 hr F°)

It is highly desirable that fouling factors be used to monitor deposit buildup or removal rather than percent cleanliness. Percent cleanliness is affected by both the overall thermal conductivity of the condenser and the thermal conductivity of the foulant; conductivity is not an additive function. Foulants such as calcium carbonate vary only slightly with changes of temperature. Therefore, the sole use of percent cleanliness monitoring can be misleading over an extended period of time.

Exhibit 6 depicts the theoretical variation in percent cleanliness over a year's period of time for a condenser fouled with a uniform 1/16" calcium carbonate deposit. The percent cleanliness (effective surface area) decreases as inlet temperature rises in the spring and again increases in the fall as temperature decreases. If percent cleanliness were the sole method used for monitoring, this curve might be interpreted as loss of microbial slime control (Exhibit 7).

Monitoring of this same system with both percent cleanliness and fouling factors would have prevented this misinterpretation of data. The fouling factor measured would be observed to increase slightly with the increase in inlet temperature. A conversion of this fouling factor to apparent foulant thickness would have revealed that the decrease in effective surface area was due to a constant average thickness of a deposit.

This is a theoretical example of a phenomena which has been observed on year-long plots of percent cleanliness for operating condensers. It is strongly recommended that fouling factor plots and their conversion to apparent foulant thickness be used for long-term monitoring of a condenser or other heat transfer device to assist in data interpretation.

Percent cleanliness monitoring alone is sufficient for short-term monitoring where only minor changes in inlet cooling water temperature (and cooling water flow) are expected. The additional time spent for the fouling factor calculation, however, does provide added insurance even during short-term evaluations, and is highly recommended.

DATA CROSSCHECKS

Many plants do not have the luxury of highly accurate flowmeters or temperature and pressure measurement devices with which to perform condenser tests. Ingenuity is required on the part of the engineer in many cases to obtain the required measurements with a degree of accuracy. Methods for calculating data which is unavailable and for checking measurements which are believed to be inaccurate are covered in Appendix I.

ECONOMIC EVALUATIONS USING CONDENSER PERFORMANCE

The effective condenser surface area can be used to estimate the cost of waterside fouling on operating costs using performance curves from the thermal kit. The observed back pressure and the expected back pressure for a clean condenser are calculated from the $U_{observed}$ and $U_{expected}$.[16,18] The unit heat rate at the load evaluated and the two back pressures are then derived from:

- Unit heat rate versus load curves at reference back pressure (Exhibit 8).
- Throttle steam flow versus load at reference back pressure (Exhibit 9).
- Vacuum (back pressure) correction to heat rate curves (Exhibit 10).

This method for calculating the heat rate penalty due to condenser fouling provides conservative estimates of increased fuel consumption for the following reasons. First of all, this method assumes that all components in the power generation unit are operating at design efficiency, and at the best value point setting. Most power generation units operate at a heat rate higher than design due to decreased performance of components other than the condenser. So the penalty estimated is less than would actually be encountered.

This method can also be used to project the economic impact of fouling based upon either observed percent cleanliness or a percent cleanliness calculated from fouling factors. It is recommended that these projections be calculated on a monthly average basis.

Further refinement can be used to predict deration due to high back pressure.

Condenser performance monitoring and the economic extrapolation of fouling-to-operating costs provides one of the more tangible "evidences" on which to base a decision. The same methods can be used to predict the cost of fouling with respect to increased fuel consumption and reduced capability, based upon fouling projection models.

FRICTION FACTOR (dP) MONITORING

Characklis[21] and his research team have demonstrated that the use of pressure drop across an exchanger (or tube) and friction factor is a valid monitoring technique for biofilm growth and accumulation on heat transfer surfaces. Johnson and Howells[22] have also used the method on a sidestream biofouling monitor successfully in the field to determine biocide effectiveness in removing microbial slime.

There are also strong theoretical indications that this technique may be useful in differentiating between smooth inorganic deposits and microbial slime films.[23]

The technique is based upon the observation that microbial slime films have an irregular surface which increases the frictional resistance to water flow. A higher pressure drop is required to overcome the resistance of a tube with a slime film than for a clean tube. The growth of a microbial slime film on the heat transfer surfaces is monitored by measuring the dP across the condenser (or exchanger). The pressure drop will increase as the slime film accumulates and decrease when the rough film is removed. Ideally, cooling water flow rates and water temperature will also be measured. This allows for correction of the data to a standard condition and for the calculation of friction factors.

Pressure drop changes alone can be used as a quantitative indicator of changes in systems where the cooling water flow is reasonably constant and where the cooling water temperature variation is minimal. The data should be corrected for flow variations by conversion to friction factors where significant flow rate changes occur during the period under study.

Friction factor (or dP) monitoring can be used to evaluate the efficacy of slime control methods and for determining the minimum treatment rate required for microbial slime removal.

PRACTICAL APPLICATION OF DEPOSITION PROBE MONITORING

A deposition probe study was used to link an extremely high incidence of tube failure to the presence of deposits in the following case history.

Brackish water from an estuary was used as a once-through cooling water source in a new condenser with 90-10 cupronickel tubes. Corrosion coupons and instantaneous corrosion rate measurement indicated corrosion rates of less than 1 mpy of the general etch form of attack. Deposits composed primarily of microbiological slime, silt, and traces of iron and calcium carbonate were observed on the tubes. Tube failures began occurring after nine months of operation. Metallographic analysis revealed an intergranular attack in the grain boundaries of the 90-10 cupronickel with indications of selective demetalification. The condenser was retubed within two and one-half years of operation. A second unit with the 90-10 cupronickel condenser tubes was already under construction

The discrepancy between the corrosion rates and type of attack on the coupons versus the condenser tubes was attributed to the deposits which formed on the heat transfer surfaces but not the coupons or instantaneous corrosion rate probes.

A management decision was needed to determine the best option for preventing a reoccurrence of the corrosion problem. A deposition probe was used to collect further data as a basis for decision making.

A sidestream study was initiated using test heat exchangers instrumented for heat transfer coefficient measurement. Heat transfer coefficients were used as a measure of fouling rate. Two exchangers were used for the study. One exchanger was maintained clean with a scale control agent and dispersant, as well as chlorination supplemented with a chlorination enhancer. No treatment other than plant chlorination was fed to the other exchanger.

The treated exchanger showed no signs of deposition or localized corrosion after 45 days of exposure under heat transfer conditions simulating the operational surface condenser. Intense pitting and intergranular attack were observed on the other exchanger beneath deposits. The absence of localized attack on the treated, deposit-free heat transfer surface is readily apparent in Exhibit 11, photomicrographs of the treated and untreated exchanger tubes. Localized corrosion and intergranular attack are clearly indicated in the photomicrograph of the untreated exchanger where deposits were present.

This sidestream study demonstrated that the intense localized attack observed could be prevented by maintaining deposit- and slime-free surfaces, and linked localized attack to the presence of deposits.

A treatment program, based upon the data derived from the sidestream deposition probe study, has been implemented to prevent a reoccurrence of the underdeposit corrosion.

The alternatives available to the plant in this case were as follows.

1. Retube with Cu/Ni alloy every 2½ years.

Cost =	Labor and Materials	$ 1,000,000
	Outage Penalty	210,000,000
	Total Cost	$211,000,000

2. Retube with AL-6X

Cost =	Labor and Materials	$ 2,000,000
	Outage Penalty	210,000,000
	Total Cost	$212,000,000

3. Treat for slime and deposit control

Cost =		
	Per Year	$ 650,000
	For Three Years	1,950,000

Here the choice is clear; it is much more economical to treat the system. The sidestream deposition probe satisfied the feasibility of the program, and the plant chose alternative three.

The sidestream deposition probe can also be used for chlorine minimization studies without jeopardizing an operating condenser or other heat transfer system. The data can then be applied to the operating system in conjunction with an NPDES chlorine minimization study.

PRACTICAL APPLICATION OF CONDENSER PERFORMANCE MONITORING: Acid Cleaning Effectiveness and Payout

A surface condenser, serviced by cooling water from an impounded lake, decreased to an effective surface area of 41% over a three-year period. Calcium carbonate was found to be the primary foulant. Condenser performance was checked before and after an on-line acid cleaning to determine its effectiveness.

It was observed that the effective condenser surface area increased to 51% after on-line pH reduction. This resulted in a decrease in projected fuel consumption of 458,000 million Btu's per year.

Effective surface area, fouling factor and apparent scale thickness are outlined before and after cleaning in Exhibit 12. Fuel consumption projections are outlined in Exhibits 13 and 14.

The alternatives facing the plant engineer are summarized in the payoff table in Exhibit 15. the pH is the most critical factor and fluctuates between 8.2 and 8.6, depending on rainfall. The pH usually is high and has been assigned a 0.6 probability. The cost figures are combinations of fuel penalties incurred due to higher back pressures, unit outage penalties, and chemical costs.

From analysis of the expected value data, alternative A_4 is the most economical one provided the condenser is thoroughly cleaned. Alternatives A_3 and A_4 do not have fuel penalty costs built in. The most logical decision would be to completely clean the condenser on line (estimated cost $50,000) and then use a scale inhibitor program such as alternative four.

The condenser performance program would verify how clean the condenser was to avert a shutdown for inspection, and the deposition probes would attest the feasibility of the inhibitor program, to make the choice much clearer for this particular plant.

PRACTICAL APPLICATION OF CONDENSER PERFORMANCE MONITORING: Macrobiological Fouling

Brackish water supplies a once-through condenser cooling system with a combined nameplate rating of 367,000 GPM for the three circulation pumps. Tube sheet fouling by shellfish and smaller debris increased the head pressure and decreased the cooling water flow rate. It was desired to monitor the macrobiological fouling to determine the best time for manual cleaning and to calculate the cost of fouling. Cooling water flow calculation by heat balance was used to monitor the fouling.

Inlet cooling water temperature was monitored with a mercury thermometer in the inlet pipe. Outlet cooling water temperature was measured by four mercury thermometers readable to ± 0.1°F. The outlet thermometers were installed in wells in the outlet pipe and the readings averaged. Back pressure was measured by a mercury manometer. A complete thermal kit was available for the unit for estimation of steam flows and heat rejection.

The calculated cooling water flow versus time is plotted in Exhibit 16. It was observed that cooling water flow decreased by approximately 35% within two weeks after manual cleaning. This reduction of cooling water flow was estimated to result in 355,000 million Btu's of additional fuel consumption per year. At a cost of $4.50 per million Btu's for oil, this would cost an additional $1.6 million per year at the median load of 700 megawatts. Exhibit 17 outlines the fuel penalty calculations.

In this situation, there is only one state of nature and four alternatives to choose from. These are listed in Exhibit 18. Alternative A_1, the do-nothing approach, looks attractive initially; however, the consequences of non-action on this problem can be disastrous. Condensate can be grossly contaminated requiring frequent shutdowns to plug leaking condenser tubes due to inlet area erosion. The $1.6MM penalty quickly escalates to $10MM or more. Cleaning manually either off line or on line at reduced loads is feasible, but alternative A_2 is by far the most attractive, even though it requires a $500,000 capital investment. Actually, over three years, the payback is better than $1.4MM per year, and the screens can easily be justified.

PRACTICAL APPLICATION OF dP (FRICTION FACTOR) MONITORING

The dP monitoring method was used successfully to monitor the effectiveness of an enzymic microbiological fouling control agent[24] in a 4,000-GPM open recirculating system servicing a chilled water system condenser.

This system performance and efficacy of the enzyme in removing microbial slime and preventing its reattachment and growth would normally have been monitored by heat transfer performance. Measurement error was excessively high, resulting in questionable results. The temperature rise across the condenser, for example, varied from 3 to 5°F. A 1°F measurement error would introduce a 20 to 33% error.

Inlet and outlet pressure gauges were available, however. It was observed that the dP across the condenser decreased to 10 psi after mechanical cleaning of the tubes. The pressure drop rose to 14 psi within three weeks of the cleaning. The enzyme was slug fed to the system. The pressure drop decreased to 10 psi within 12 hours of the enzyme application. Cooling water bulk temperature remained within 1°F during the time period where the pressure drop occurred.

This decrease in pressure drop to the levels observed immediately after cleaning indicated strongly that the microbial slime film was removed. The pressure drop observed was also of the magnitude expected for microbial slime film removal. The pressure drop measured versus time is plotted in Exhibit 19.

Heat transfer data, although deemed unreliable, also indicated a trend of decreased resistance to heat transfer. This further supported the pressure drop data.

A two-order of magnitude increase in total aerobic bacteria count in the recirculating water was also measured during the 48 hours after enzyme application. This would be expected based upon the mechanism observed in the laboratory. The enzyme *does not* kill bacteria and is *not* a biocide. It "catalyzes" the hydrolysis of the polysaccharide film, which holds the slime film together and allows it to adhere to the tube wall.

The combination of the pressure drop observations, the apparent increase in heat transfer performance, and the observed increase in total aerobic bacteria counts provided an extremely strong indication that the enzyme successfully removed the microbial slime film.

This evaluation demonstrates that pressure drop measurement can be a valuable tool in the field for monitoring biofilm development, as well as in the laboratory environment. Ideally cooling water flow and water temperatures should also be monitored in addition to pressure drop so that friction factors can be calculated and variations in dP related to parameter changes other than slime formation or removal can be factored out.

In this case, the dP method was used to provide evidence for a water treatment company's decision whether or not to pursue the application of enzyme technology to biofouling control. The approach is under further investigation.

CONCLUSIONS

Water treatment decisions in utility systems are business decisions involving millions of dollars of value to the utility executive. Quantification of alternative costs or payoffs by means of the monitoring approaches described above make the job easier and allows for better decisions. These same methods can be fitted to models for predicting future performance. Even in cases of uncertainty about the future states of nature, logical decisions can be made given some reasonable probabilities of the course of future events.

APPENDIX I

ALTERNATE METHODS FOR CALCULATING MEASUREMENTS NEEDED FOR H.E.I. CONDENSER PERFORMANCE

Q-Heat Flow

Heat flow is one of the more critical parameters required for condenser performance monitoring. Two methods are available for calculating this value: the waterside Q and the steamside Q. In theory both values should be identical (Conservation of Energy) if it is assumed that no heat energy is lost from the steamside through radiation from the condenser shell.

Cooling waterside Q is calculated from the relationship:

$$Q_{WS} = (T_{OUT} - T_{IN})\, W \cdot C_p$$

where Q_{WS} is the waterside Q (Btu/hr)

T_{OUT} is the outlet cooling water temperature (F°)

T_{IN} is the inlet cooling water temperature (F°)

W is the cooling water flow rate (lbs/hr)

C_p is the specific heat of water, which may be assumed to be 1.0.

The accuracy of a waterside Q is questionable in many plants. The following procedures are recommended prior to a performance test:

- Inlet cooling water temperature is normally the most accurate of the measurements. It is recommended that the thermocouples be calibrated prior to testing. In the case where temperatures are recorded by a computer, it is recommended that a test signal appropriate for the thermocouple type be injected at the measurement point. The signal should then be measured at the board in the computer and line loss calculated. The value calculated by the computer should also be verified.

- Outlet cooling water temperature accuracy is critical for any performance calculations. It is also highly subject to error. Most condensers are equipped with a thermocouple in the outlet water box where the water exiting from the tubes has not mixed thoroughly. Temperature stratification can result in large errors. It is recommended that several thermocouples be installed in the outlet area and averaged. Ideally, the thermocouples will be installed in the outlet pipe rather than in the water box. Acceptable accuracy has been obtained with four measurement probes in the outlet pipe. All thermocouples should be calibrated as outlined for the inlet temperature. In the event that only the water box temperature measurement device is to be used, the outlet temperature should be carefully cross-checked with the steamside Q as outlined later in this section.

- Cooling water flow rate can be an elusive value, especially in older pumps. The value can be obtained in several ways. The easiest method is to use the design cooling water pumps nameplate rating. This value, however, can be in error 20% or greater.

Reasonable accuracy can be obtained in many cases by measuring the head pressure and calculating the flow from pump performance curves. Flow rate can also be measured by injecting a readily measurable chemical tracer into the cooling water at the pump suction and measuring the increase in concentration across the condensers. The tracer ion must be environmentally acceptable (e.g., potassium). Cooling water flow can then be calculated as follows:

$$\text{Cooling Water Flow Rate (lbs/hr)} = \frac{\text{Pounds of Tracer Ion Injected per Hour}}{1{,}000{,}000\ (\text{ppm Tracer in Outlet} - \text{ppm Tracer in Water Source})}$$

Cooling Water flow may also be calculated from heat balance between the steamside and the waterside.

Cooling Water Flow (lbs/hr) = $Q_{steamside}/(T_{OUT} - T_{IN})$. Accurate inlet and outlet cooling water temperatures are required for this method. A reasonably accurate estimate of steamside Q is also required.

The ideal method is, of course, to install a flowmeter–if funds are available.

Steamside Q can be estimated with acceptable accuracy when a full thermal kit for the unit is available.

$$Q_{steamside} = WSX \cdot \text{Heat Rejection } 1 + WSAUX \cdot \text{Heat Rejection } 2$$

where WSX is the exhaust steam flow in lbs/hr

Heat Rejection 1 is the heat of condensation in Btu/lb of steam

WSAUX is auxiliary steam flow to the condenser (which can be considered negligible in most cases)

Heat Rejection 2 is the heat of condensation for the auxiliary steam

The exhaust steam flow can be estimated from heat balance diagram data in the thermal kit in cases where no measurement is available.

Heat rejection of the steam can be obtained in several ways. The simplest method is to assume a heat rejection of 970 Btu/lb.

Improved accuracy can be obtained by using the UEEP (Used Energy End Point) to calculate expected heat rejection versus back pressure and load (Exhibit 20). This particular table was developed by Archbold et al for a surface condenser design program.[19] Calculation of the UEEP is somewhat detailed.[20] Access to a computer is recommended for this method. The UEEP is the expected enthalpy of the exhaust steam. The expected heat rejection is then calculated as follows:

$$\text{Heat Rejection} = UEEP - H_L$$

where H_L is the enthalpy of the condensate at the back pressure observed.

The steamside Q and waterside Q for a given data point should be compared for each data point as a cross-check on data accuracy. In cases where a measurement such as cooling water flow rate or the outlet cooling water temperature is suspected of a high error, the questionable measurement can be calculated by heat balance.

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EXHIBIT 1

AN EXAMPLE PAYOFF TABLE

Alternatives	SN_1 P=0.2	SN_2 P=0.6	SN_3 P=0.4	EV
A_1	250,000	350,000	600,000	500,000
A_2	150,000	175,000	500,000	335,000
A_3	600,000	250,000	100,000	310,000

A_1, A_2, A_3 = Alternatives
SN_1, SN_2, SN_3 = States of Nature
P = probability of a particular state of nature occurring

EXHIBIT 2

Pilot Heat Exchanger

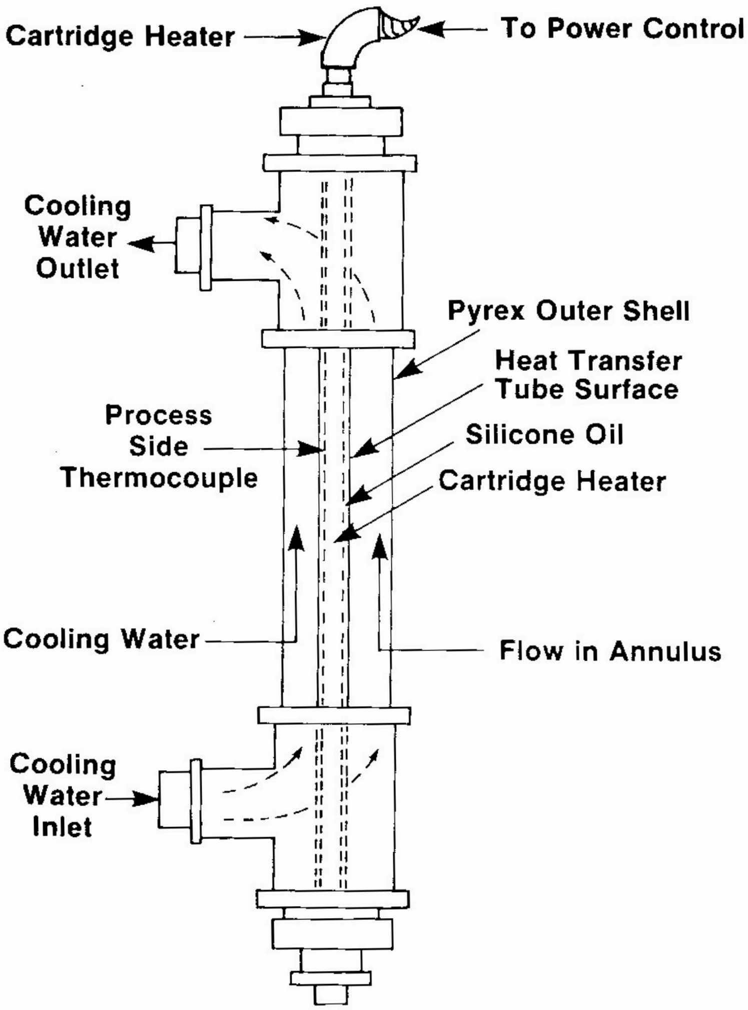
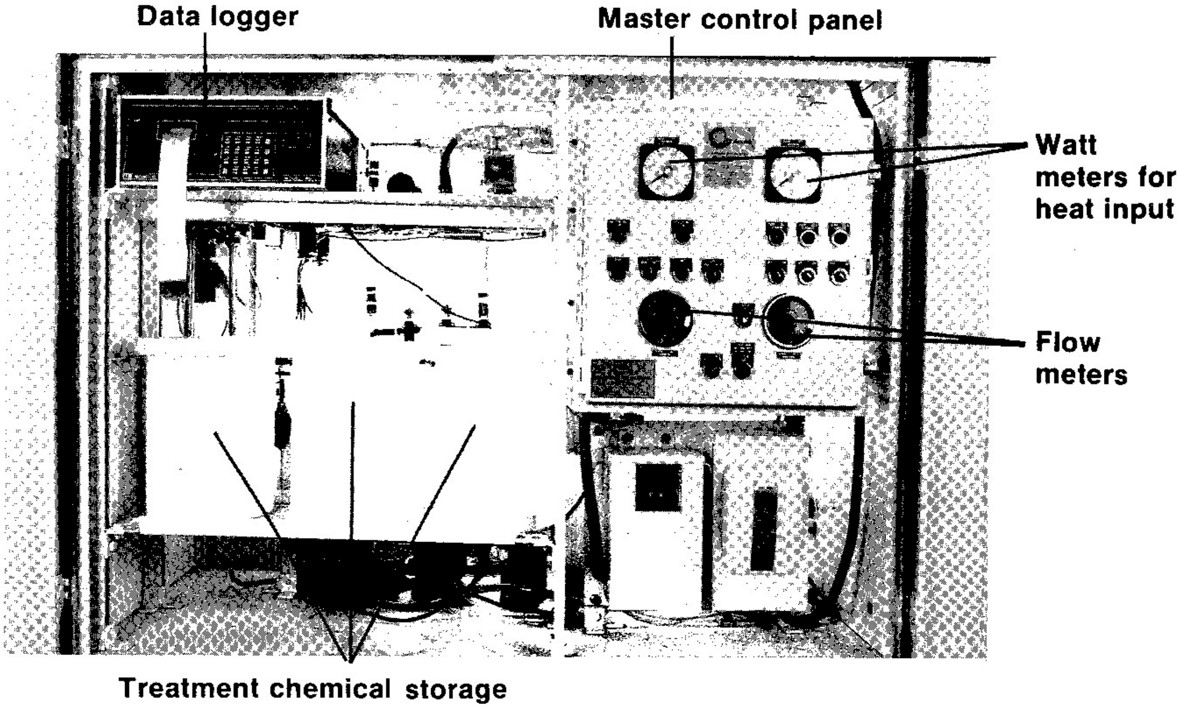


EXHIBIT 3

DEPOSITION MONITOR



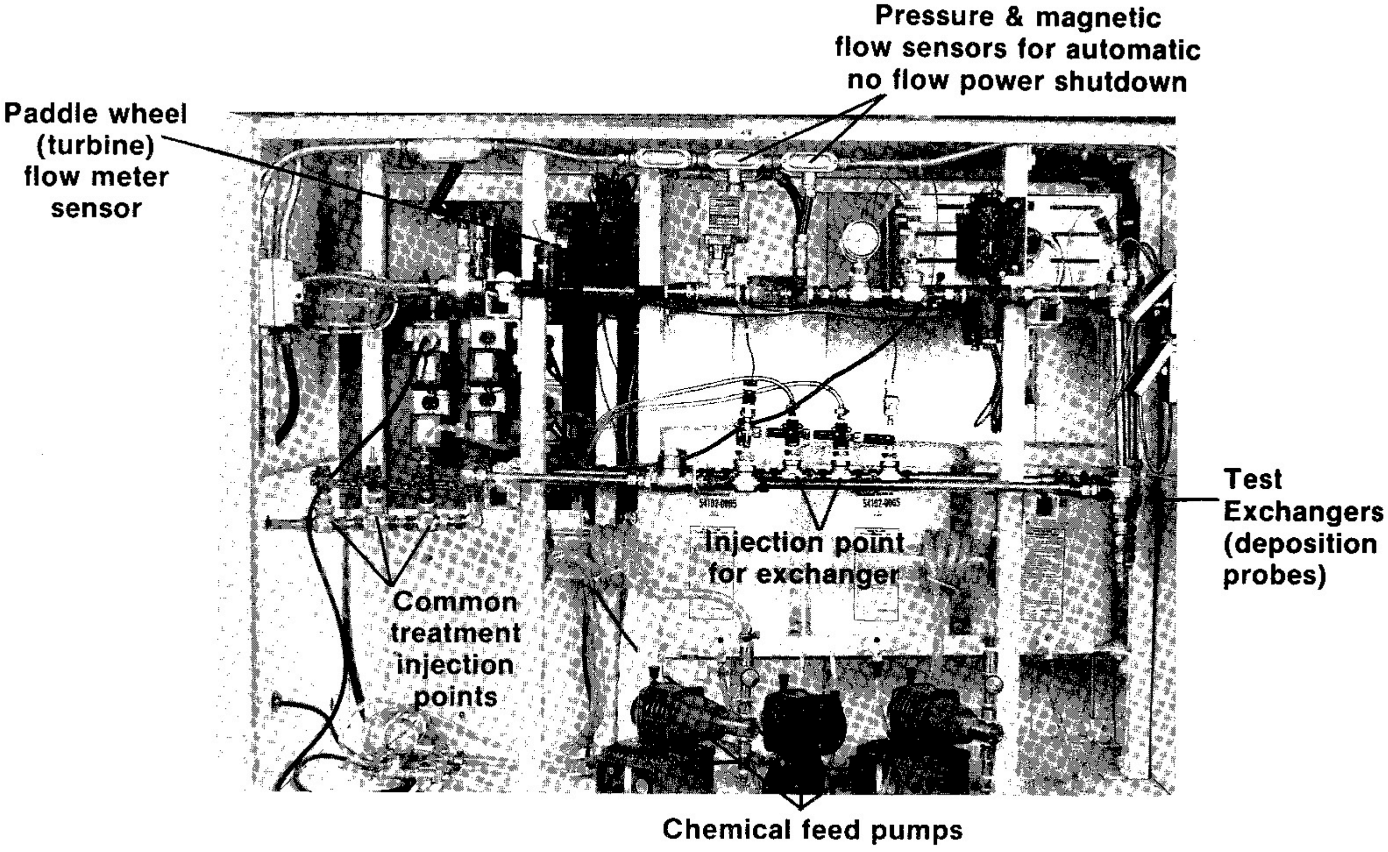


EXHIBIT 4

LOAD CORRECTION

Condensate Loading (Lbs/(Sq Ft · Hr)	Correction (C_L)	Condensate Loading (Lbs/(Sq Ft · Hr)	Correction (C_L)
4.0	0.841	10.0	1.057
5.0	0.889	11.0	1.083
6.0	0.931	12.0	1.107
7.0	0.967	13.0	1.130
8.0	1.000	14.0	1.150
9.0	1.030		

Condensate Loading is the Exhaust Steam Flow divided by the Outside Tube Surface Area. Exhaust steam flow can be estimated using heat balance diagrams. This load correction is not normally used for design purposes. It does, however, assist in narrowing data scatter for measurements taken at varying loads.

$C_L = (\text{Condensate Loading}/8.0)^{0.25}$

EXHIBIT 5

ANNUAL VARIATION IMPACT OF 1/16" $CaCO_3$ SCALE

CLEANLINESS IMPACT VERSUS LOAD AT 170,000 KWHR

COMPANY LOCATION

STATION UNIT

MONTH	% TOTAL SURFACE AREA	FOULING FACTOR X 1000	HISTORIC INLET WATER TEMP.	EXPECTED OUTLET WATER TEMP.	EXPECTED U VALUE FOULED SURFACE	EXPECTED U VALUE CLEAN SURFACE	EXPECTED BACK PRESSURE FOULED	RELATED HEAT RATE	EXPECTED BACK PRESSURE CLEAN	RELATED HEAT RATE	HEAT RATE PENALTY	DAILY FUEL PENALTY	MONTHLY FUEL PENALTY
JAN	55.4	2.030	35.0	50.3	237.3	428.5	1.34	9148.6	0.67	9080.8	67.7	276.3	8566
FEB	54.4	2.030	37.0	52.4	243.0	447.0	1.40	9159.4	0.71	9082.6	76.9	313.6	8780
MAR	52.3	2.040	41.0	56.5	253.0	484.1	1.52	9181.5	0.77	9085.7	95.8	390.9	12118
APR	49.4	2.060	47.0	62.6	265.8	538.1	1.72	9202.5	0.88	9092.9	109.6	447.0	13409
MAY	46.3	2.080	55.0	70.9	279.8	603.9	2.11	9254.6	1.08	9111.8	142.8	582.8	18066
JUN	43.8	2.100	64.0	80.3	291.4	665.1	2.72	9371.5	1.37	9153.8	217.7	888.1	26642
JUL	42.8	2.110	69.0	85.6	296.3	692.0	3.16	9476.7	1.59	9188.4	288.3	1176.1	36460
AUG	43.1	2.110	67.0	83.5	294.1	682.0	2.98	9432.8	1.50	9180.0	252.8	1031.5	31976
SEP	43.1	2.110	67.0	83.5	294.1	682.0	2.98	9432.8	1.50	9180.0	252.8	1031.5	30944
OCT	45.1	2.090	59.0	75.0	285.4	633.0	2.36	9297.2	1.19	9125.5	171.7	700.4	21713
NOV	47.4	2.070	52.0	67.8	275.2	580.3	1.95	9231.2	1.00	9103.3	127.9	521.7	15652
DEC	52.7	2.040	40.0	55.5	250.5	474.9	1.49	9177.8	0.74	9084.0	93.7	382.5	11857
AVE	48.0	2.072	52.7	68.6	272.2	575.9	2.14	9280.5	1.08	9122.4	158.1	645.2	19682
UNITS	PERCENT	I/U	DEGREES FAHRENHEIT		BTU/SOFT HR DEGF		IN–HGA	BTU/KWHR	IN–HGA	BTU/KWHR	BTU/KWHR	MILLION BTU	

THE TOTAL ADDITIONAL FUEL CONSUMPTION PER YEAR AT THIS LOAD IS 236182. MILLION BTU.

APOLLO TECHNICAL DATA PROCESSING CENTER
ONE APOLLO DRIVE, WHIPPANY, NEW JERSEY 07981

EXHIBIT 6

The Impact of 1/16" of $CaCO_3$ Scale on Condenser Effective Surface Area

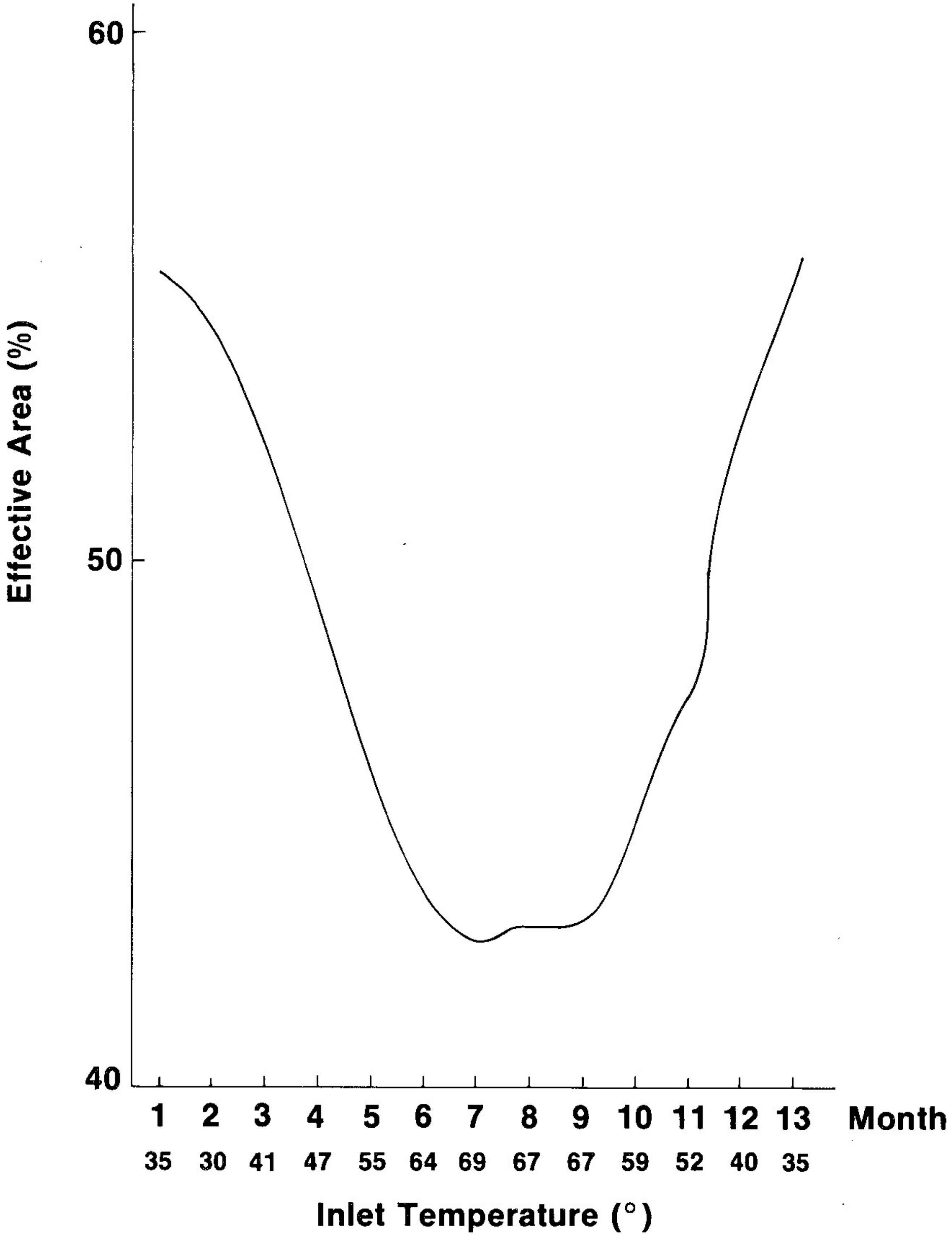


EXHIBIT 7

UNIT HEAT RATE
VS
Megawatt Load (Gross Generation)
at a Reference Back Pressure of 3.5 "HgA

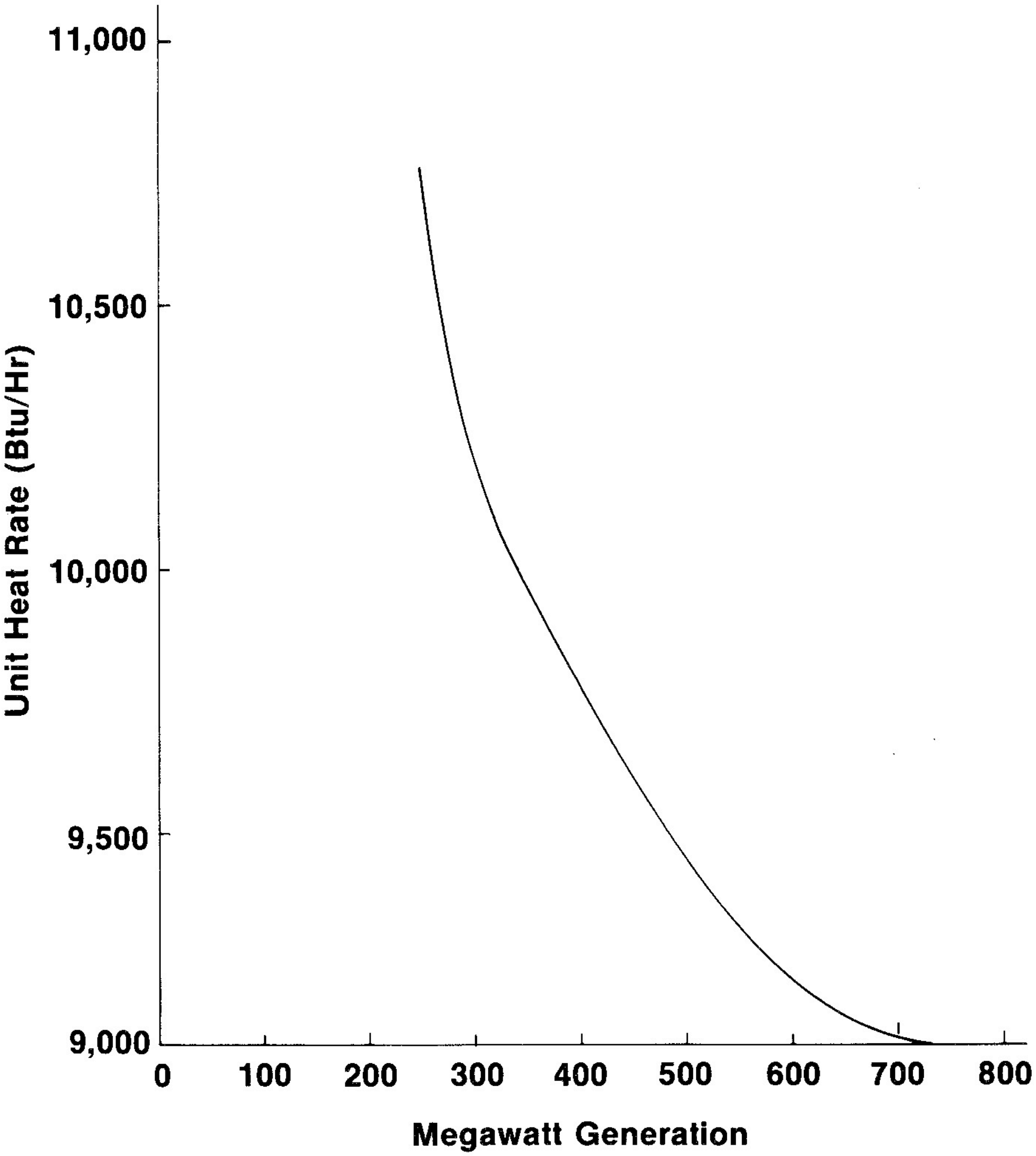


EXHIBIT 8

THROTTLE STEAM FLOW
vs
Megawatt Load (Generator Output)
at a Reference Back Pressure of 3.5 "HgA

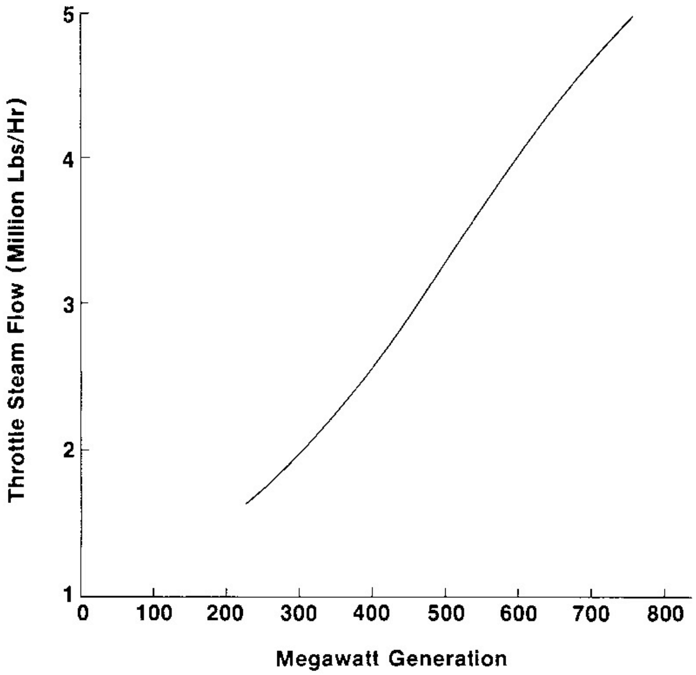


EXHIBIT 9

Vacuum (Back Pressure) Correction To Heat Rate

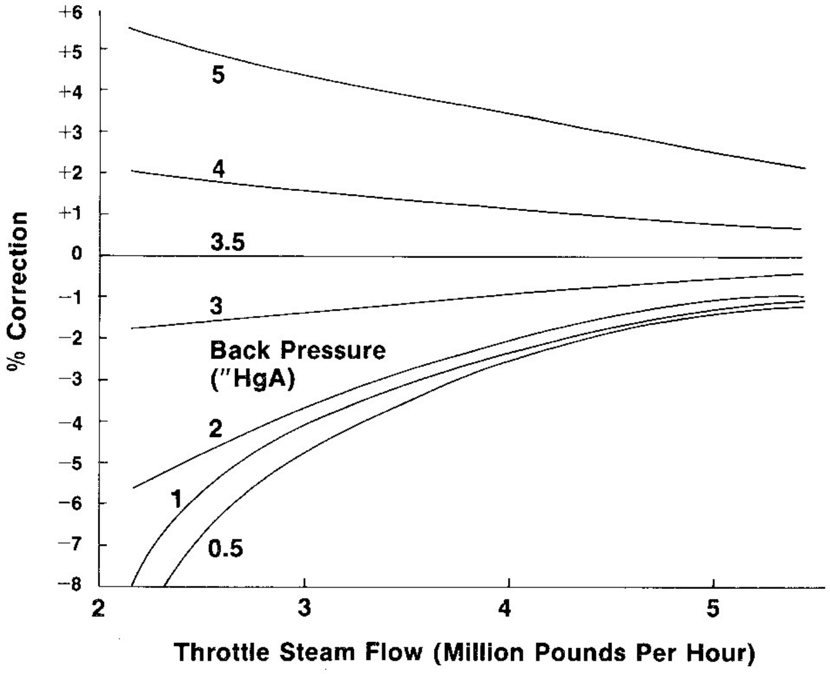
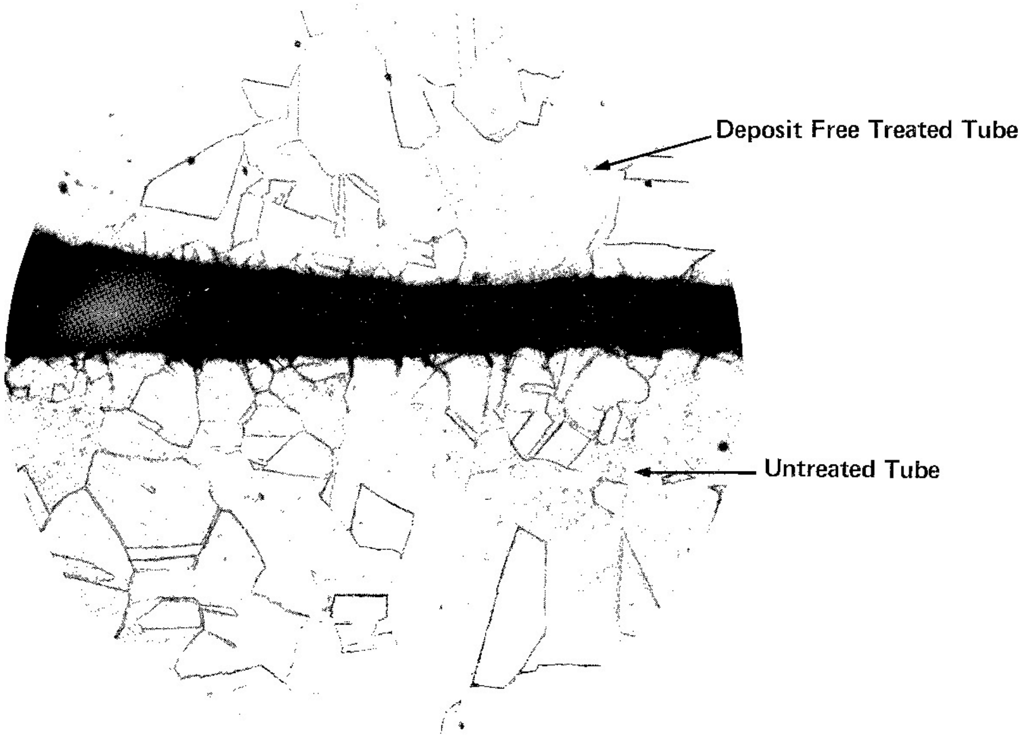


EXHIBIT 10



500 x magnification

EXHIBIT 11

ON-LINE CLEANING EFFECTIVENESS

	% Cleanliness (Effective Surface Area)	Fouling Factor X 1000	Apparent $CaCO_3$ Thickness (mils)
Prior to On-Line Cleaning	41	2.196	63.3
After On-Line Cleaning	51	1.446	41.8
Difference	+10	0.750	21.5

EXHIBIT 12

BEFORE ON-LINE CLEANING

CLEANLINESS IMPACT VERSUS LOAD AT 542,000 KWHR

COMPANY **LOCATION**

STATION **UNIT NUMBER**

MONTH	% TOTAL SURFACE AREA	FOULING FACTOR X 1000	HISTORIC INLET WATER TEMP.	EXPECTED OUTLET WATER TEMP.	EXPECTED U VALUE FOULED SURFACE	EXPECTED U VALUE CLEAN SURFACE	EXPECTED BACK PRESSURE FOULED	RELATED HEAT RATE	EXPECTED BACK PRESSURE CLEAN	RELATED HEAT RATE	HEAT RATE PENALTY	DAILY FUEL PENALTY	MONTHLY FUEL PENALTY
JAN	48.3	2.090	51.0	65.7	272.4	564.4	2.08	9978.0	1.00	9958.1	19.9	259.4	8041
FEB	46.8	2.100	55.0	69.8	278.7	595.6	2.28	9983.3	1.11	9961.1	22.2	289.3	8101
MAR	44.2	2.120	64.0	79.1	289.9	655.3	2.89	10027.8	1.43	9968.8	59.0	767.9	23804
APR	42.2	2.150	75.0	90.5	297.8	705.2	3.98	10166.5	1.95	9976.1	190.4	2477.1	74312
MAY	41.7	2.160	80.0	95.7	300.2	719.8	4.60	10250.6	2.24	9981.9	268.7	3494.8	108338
JUN	41.1	2.180	86.0	102.0	301.6	732.9	5.48	10366.5	2.67	10008.6	357.9	4656.2	139686
JUL	40.9	2.190	88.0	104.1	301.6	736.7	5.82	10410.0	2.83	10022.3	387.7	5042.8	156328
AUG	40.9	2.190	88.0	104.1	301.6	736.7	5.82	10410.0	2.83	10022.3	387.7	5042.8	156328
SEP	41.2	2.180	85.0	101.0	301.1	731.0	5.33	10346.7	2.61	10003.8	342.9	4460.2	133805
OCT	42.2	2.150	75.0	90.5	297.8	705.2	3.98	10166.5	1.95	9976.1	190.4	2477.1	76789
NOV	44.7	2.100	63.0	78.0	290.2	649.5	2.81	10020.5	1.39	9968.0	52.5	682.8	20485
DEC	45.5	2.110	59.0	73.9	284.2	624.1	2.53	9997.8	1.23	9964.3	33.5	436.3	13526
AVE	43.3	2.143	72.4	87.9	293.1	679.7	3.97	101770.0	1.94	9984.3	192.7	2507.2	76629
UNITS	PERCENT	I/U	DEGREES FAHRENHEIT		BTU/SQFT HR DEGF		IN–HGA	BTU/KWHR	IN–HGA	BTU/KWHR	BTU/KWHR	MILLION BTU	

THE TOTAL ADDITIONAL FUEL CONSUMPTION PER YEAR AT THIS LOAD IS 919,543 MILLION BTU

APOLLO TECHNICAL DATA PROCESSING CENTER
ONE APOLLO DRIVE, WHIPPANY, NEW JERSEY 07981

EXHIBIT 13

AFTER ON-LINE CLEANING

CLEANLINESS IMPACT VERSUS LOAD AT 542,000 KWHR

COMPANY LOCATION

STATION UNIT NUMBER

MONTH	% TOTAL SURFACE AREA	FOULING FACTOR X 1000	HISTORIC INLET WATER TEMP.	EXPECTED OUTLET WATER TEMP.	EXPECTED U VALUE FOULED SURFACE	EXPECTED U VALUE CLEAN SURFACE	EXPECTED BACK PRESSURE FOULED	RELATED HEAT RATE	EXPECTED BACK PRESSURE CLEAN	RELATED HEAT RATE	HEAT RATE PENALTY	DAILY FUEL PENALTY	MONTHLY FUEL PENALTY
JAN	58.5	1.380	51.0	65.6	330.4	564.4	1.63	9972.2	1.00	9958.1	14.1	183.5	5690
FEB	57.1	1.390	55.0	69.6	339.8	595.6	1.80	9974.3	1.11	9961.1	13.2	171.6	4805
MAR	54.5	1.400	64.0	78.8	357.3	655.3	2.26	9982.6	1.43	9968.8	13.8	179.2	5554
APR	52.4	1.420	75.0	90.2	369.7	705.2	3.09	10048.1	1.95	9976.1	72.1	937.6	28128
MAY	51.8	1.430	80.0	95.4	373.0	719.8	3.59	10112.4	2.24	9981.9	130.5	1697.8	52632
JUN	51.3	1.440	86.0	101.6	376.0	732.9	4.28	10207.5	2.67	10008.6	198.9	2587.0	77611
JUL	51.2	1.440	88.0	103.7	377.2	736.7	4.52	10239.9	2.83	10022.3	217.6	2830.3	87740
AUG	51.2	1.440	88.0	103.7	377.2	736.7	4.52	10239.9	2.83	10022.3	217.6	2830.3	87740
SEP	51.3	1.440	85.0	100.6	375.3	731.0	4.16	10191.1	2.61	10003.8	187.3	2436.4	73091
OCT	52.4	1.420	75.0	90.2	369.7	705.2	3.09	10048.1	1.95	9976.1	72.1	937.6	29065
NOV	54.7	1.400	63.0	77.8	355.6	649.5	2.20	9980.7	1.39	9968.0	12.7	165.5	4966
DEC	55.9	1.390	59.0	73.7	348.9	624.1	1.99	9976.6	1.23	9964.3	12.3	160.2	4965
AVE	53.5	1.416	72.4	87.6	362.5	679.7	3.09	10081.1	1.94	9984.3	96.8	1259.7	38499
UNITS	PERCENT	I/U	DEGREES FAHRENHEIT		BTU/SQFT HR DEGF		IN–HGA	BTU/KWHR	IN–HGA	BTU/KWHR	BTU/KWHR	MILLION BTU	

THE TOTAL ADDITIONAL FUEL CONSUMPTION PER YEAR AT THIS LOAD IS 461,985 MILLION BTU

APOLLO TECHNICAL DATA PROCESSING CENTER
ONE APOLLO DRIVE, WHIPPANY, NEW JERSEY 07981

EXHIBIT 14

PAYOFF – COST MATRIX FOR SCALE CONTROL DECISION

Alternative	pH = 8.2 p = 0.2	pH = 8.4 p = 0.2	pH = 8.6 p = 0.6	Expected Cost
A_1 Do Nothing Off line acid clean	600,000	950,000	1,500,000	1,210,000
A_2 Periodic On-line Cleaning	250,000	300,000	350,000	320,000
A_3 Feed H_2SO_4 for pH reduction	180,000	250,000	500,000	386,000
A_4 Feed Scale Control Agent	60,000	85,000	130,000	107,000

EXHIBIT 15

Calculated Cooling Water Flow

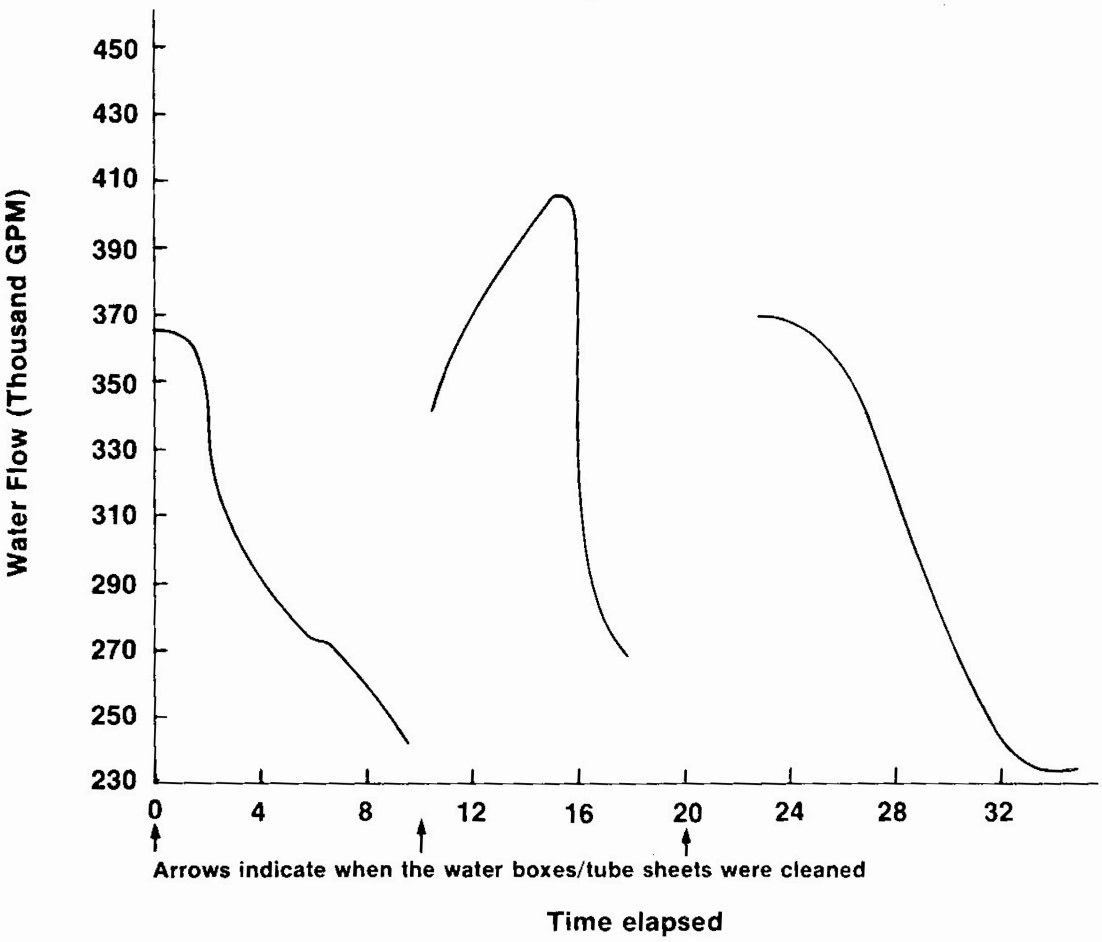


EXHIBIT 16

BACK PRESSURE & FUEL PENALTIES DUE TO COOLING WATER FLOW

CLEANLINESS IMPACT VERSUS LOAD AT 700,000 KWHR

COMPANY LOCATION

STATION UNIT 2

MONTH	% TOTAL SURFACE AREA	FOULING FACTOR X 1000	HISTORIC INLET WATER TEMP.	EXPECTED OUTLET TEMP. 240. MGPM	EXPECTED OUTLET TEMP. 367.MGPM	EXPECTED U VALUE 240.MGPM	EXPECTED U VALUE 367. MGPM	EXPECTED BACK PRESSURE 240. MGPM	RELATED HEAT RATE	EXPECTED BACK PRESSURE 367. MGPM	RELATED HEAT RATE	HEAT RATE PENALTY	DAILY FUEL PENALTY	MONTHLY FUEL PENALTY
JAN	100.0	0.000	47.8	72.7	63.7	312.5	591.0	1.37	9452.7	0.76	9450.4	2.3	38.9	1206
FEB	100.0	0.000	52.3	77.3	68.3	334.1	631.7	1.51	9454.8	0.88	9450.2	4.5	75.8	2122
MAR	100.0	0.000	65.3	90.7	81.5	385.2	728.0	2.10	9471.2	1.29	9451.8	19.4	325.3	10084
APR	100.0	0.000	69.6	95.2	85.9	397.7	751.4	2.34	9480.3	1.47	9454.1	26.2	439.8	13195
MAY	100.0	0.000	74.4	100.2	90.8	409.0	772.5	2.69	9499.4	1.71	9459.0	40.4	679.0	21050
JUN	100.0	0.000	81.4	107.6	98.0	421.2	794.7	3.27	9554.9	2.11	9471.5	83.4	1400.6	42019
JUL	100.0	0.000	84.6	111.0	101.3	425.5	802.2	3.58	9592.9	2.34	9480.3	112.7	1893.1	58686
AUG	100.0	0.000	85.2	111.7	101.9	426.3	803.5	3.66	9602.9	2.38	9481.8	121.1	2035.0	63084
SEP	100.0	0.000	81.3	107.5	97.9	421.0	794.4	3.26	9553.8	2.10	9471.2	82.6	1387.4	41623
OCT	100.0	0.000	83.5	109.8	100.2	424.1	799.8	3.47	9579.1	2.25	9476.8	102.3	1718.3	53268
NOV	100.0	0.000	81.8	108.0	98.4	421.8	795.7	3.32	9560.7	2.15	9473.0	87.7	1473.0	44191
DEC	100.0	0.000	58.4	83.6	74.5	360.4	681.3	1.76	9460.3	1.05	9450.4	9.9	165.8	5141
AVE	100.0	0.000	72.1	97.9	88.5	394.9	745.5	2.69	9521.9	1.71	9464.2	57.7	969.3	29639
UNITS	PERCENT	I/U	DEGREES FAHRENHEIT			BTU/SQFT HR DEGF		IN–HGA	BTU/KWHR	IN–HGA	BTU/KWHR	BTU/KWHR	MILLION BTU	

THE TOTAL ADDITIONAL FUEL CONSUMPTION PER YEAR AT THIS LOAD IS 355,669 MILLION BTU

APOLLO TECHNICAL DATA PROCESSING CENTER
ONE APOLLO DRIVE, WHIPPANY, NEW JERSEY 07981

EXHIBIT 17

DECISION ALTERNATIVES FOR MACROBIOLOGICAL FOULING

Alternatives		Cost
A_1	Do Nothing	$1,600,000 fuel penalties
A_2	Install Better Screens	$ 500,000 material + labor
A_3	Manual Clean On-line @ ½ Load Every 3 Weeks	$1,400,000 load penalty
A_4	Manual Clean Off-line Every 3 Weeks	$2,800,000

EXHIBIT 18

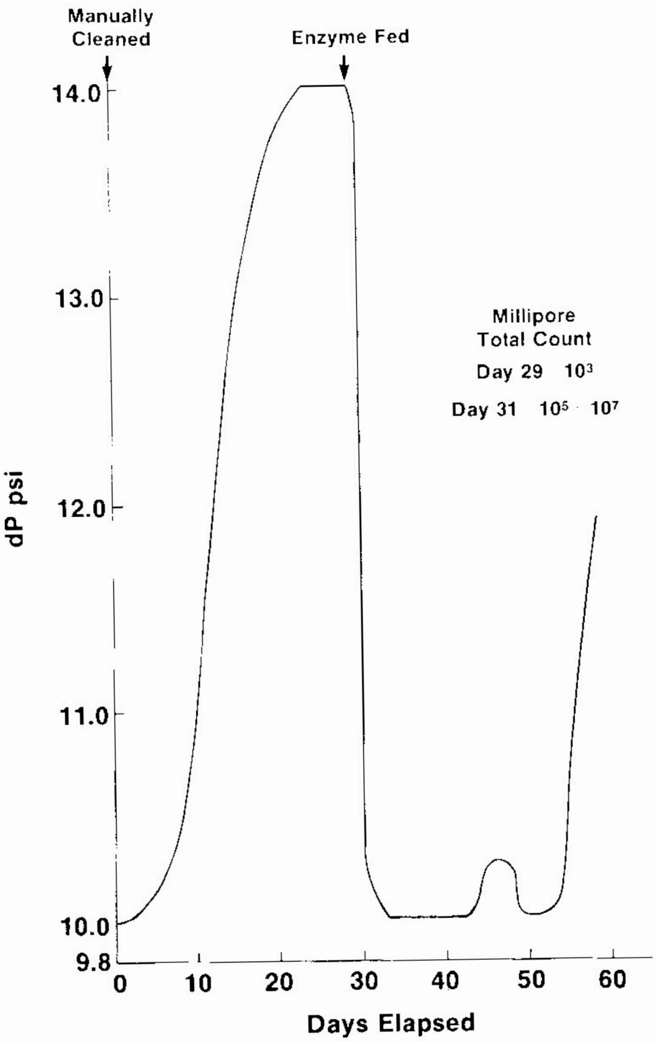


EXHIBIT 19

HEAT REJECTION VS LOAD AND BACK PRESSURE

Back Pressure	Megawatt Load			
HgA	325	284	203	144
0.3	982.0	984.3	988.2	991.5
0.5	972.4	974.0	981.5	992.6
1.0	956.6	961.7	979.0	997.5
1.5	954.4	962.8	984.7	1003.3
2.0	956.7	966.0	990.3	1010.0
*2.5	959.3	969.3	994.6	1015.3

Heat Rejection in Btu/Lb

***NOTE:** These curves are approximately linear for this 1.5" HgA design back pressure turbine generator above 2.5" HgA. Similar tables can be calculated for any unit using HEAT REJECTION = UEEP - H_L.

Reproduced from Archbold et al.[19]