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Effects of Fouling and Corrosion on Heat Transfer

edited by

Y. MUSSALLI

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IMPROVEMENTS IN STEAM CONDENSER PERFORMANCE MONITORING

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ABSTRACT

Historically, monitoring of steam condenser performance at electric generating stations has suffered from an inability to detect either small changes in performance or in detecting significant changes in a timely manner. A program to improve condenser monitoring has been successfully applied at Southern California Edison's Huntington Beach Generating Station in Southern California. By linking a small personal computer to inexpensive sensors and applying a novel method of analysis, very small changes in condenser backpressure, on the order of 0.1 "Hg, can be identified almost immediately. In addition and of equal importance, is that these changes can be quickly linked to their underlying causes. The technique provides plant operators and engineers with real time condenser performance data upon which timely cost-effective decisions can be made.

NOMENCLATURE

A = Condenser tube surface area

C = Heat Transfer coefficient based on tube diameter from HEI Standards

C_p = Specific heat of cooling water

d = Subscript for design condition

E = Slope function [Exp (UA/WCp)], or $(T_s - T_1)/(T_s - T_2)$

F_1 = Heat transfer coefficient inlet temperature correction factor from HEI Standards

F_2 = Heat transfer coefficient tube material and gauge correction factor from HEI Standards

F_3 = Heat transfer coefficient cleanliness factor

LMTD = Log Mean Temperature Difference $(T_2 - T_1)/\ln[(T_s - T_1)/(T_s - T_2)]$

m = Subscript for measured condition

P_s = Condenser Pressure

T_1 = Inlet cooling water temperature

T_2 = Outlet cooling water temperature

T_s = Saturated steam temperature at condenser pressure

U = Overall heat transfer coefficient = $C\sqrt{V}\,F_1 F_2 F_3$

V = Average tube velocity

W = Weight flow of cooling water

INTRODUCTION

Power plant engineers at electric generating stations have traditionally monitored turbine steam condenser performance using information developed during the design and purchase phase of the plant equipment (1). The principal performance criteria is a graph of expected condenser pressure versus unit electric output at various inlet cooling water temperatures. Calculations for the graph are made assuming constant design cooling water flow, surface area and average tube velocity, a cleanliness factor of 0.85 and constant design heat balance condenser heat load versus unit electric output. Measured condenser pressure is compared to the expected pressure at the measured unit output and inlet cooling water temperature. The difference between actual and expected pressure is taken as a measure of condenser degradation(2). This difference, or Deviation from Standard, has generally not provided timely or accurate information of condenser performance deterioration for backpressure values much less than 0.5 "Hg.

Optimum condenser performance is recognized as an important factor in maintaining plant efficiency which in turn minimizes fuel costs. Recent rises in fuel costs have placed increased importance on maintaining plant efficiency at the highest practical level(3). Simultaneously, environmental concerns and resulting governmental regulations restrict condenser performance control measures such as chlorination, to extremely low discharge levels which places even greater emphasis on the need for accurate and timely determination of condenser performance degradation.

INSTRUMENTATION

The role of marine fouling in condenser performance has been under study at the Huntington Beach Generating Station of Southern California Edison Company since the early 1980s. Conventional condenser performance monitoring methods were initially used in conjunction with inexpensive pressure and temperature sensors and a microcomputer system. A Hewlett Packard (HP) 85 Microcomputer linked to a HP 3421A Data Acquisition and Control Unit recorded resistance measurements from Yellow Springs Instruments Company's Series 400 thermistors and voltage measurements from Data Instruments Company's Models EA and SA pressure transducers. All temperature probes were individually calibrated using an Orion Instruments platinum-resistance thermometer. The pressure transducers were calibrated using test pressure gauges available at the generation station. Raw data from each probe and transducer were then converted to engineering units by using the individual calibration functions developed prior to the test. Parameters monitored were unit electric load, condenser pressure, cooling water inlet and outlet temperature, hotwell condensate temperature, cooling water pump total head and condenser tube sheet differential. Data were recorded at 12-minute intervals by the micro-computer and then transferred to a larger computer for detailed analysis and program development.

INITIAL INVESTIGATIONS

The first significant discovery was made by examining condenser performance as measured by Deviation from Standard. Data for a 1 week period was plotted against net generator load and compared with a succeeding week's data set. Each data set plotted as a nearly straight line with the second week's data rotated counterclockwise from the first week's data (Figure 1). The rotation suggested that as condenser fouling occurs, the performance line rotates in proportion to the amount of fouling.

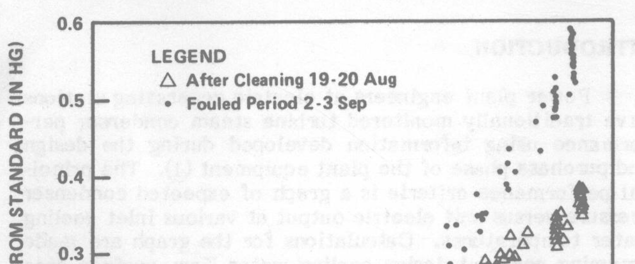

Figure 1. Backpressure deviation from standard vs. net generator load for a clean and fouled condenser.

To test this hypothesis, a base line was established based on the first week's data, where a special effort was made to have as clean a condenser as practical. The baseline value was then subtracted from the Deviation from Standard value and plotted with time. This methodology significantly decreased the data scatter (Figure 2).

A further refinement was made by multiplying the difference between Deviation from Standard and the base line, by the ratio of 200 MW to the load at the data point. Plotting the load corrected difference against time and eliminating periods when generator load was below 70 MW produced a much more consistent measure of condenser performance. The 70 MW cutoff was necessary due to possible air leakage problems and condenser cutoff pressure characteristic at low loads.

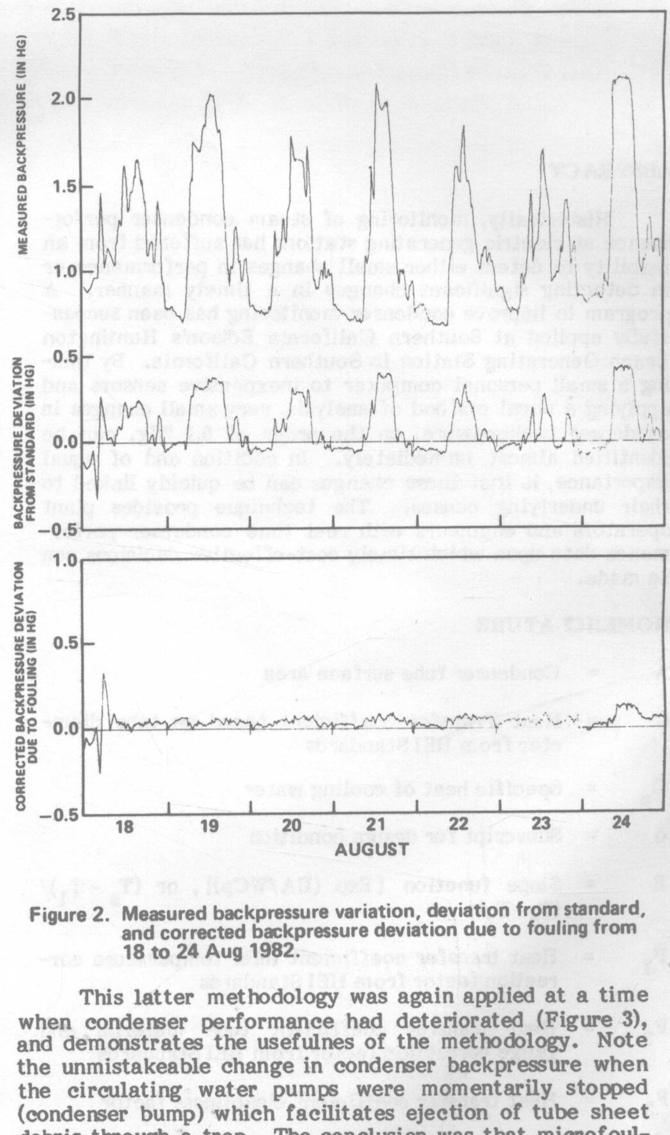

Figure 2. Measured backpressure variation, deviation from standard, and corrected backpressure deviation due to fouling from 18 to 24 Aug 1982.

This latter methodology was again applied at a time when condenser performance had deteriorated (Figure 3), and demonstrates the usefulnes of the methodology. Note the unmistakeable change in condenser backpressure when the circulating water pumps were momentarily stopped (condenser bump) which facilitates ejection of tube sheet debris through a trap. The conclusion was that microfouling was not significant but marine shells were the source of deteriorating condenser performance. This change would have been very difficult to detect and measure using conventional monitoring techniques.

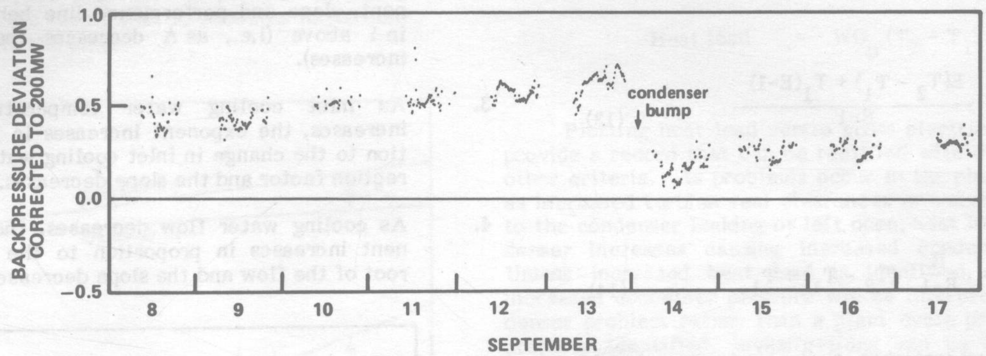

Figure 3. Backpressure deviation at loads above 70 MW and corrected to 200 MW from 8 to 17 Sep 1984.

Figure 3 demonstrates the utility of an accurate real-time display of condenser performance in making timely operational decisions. Had the operator been able to detect changes in condenser performance, the condenser bump could have been applied earlier with consequent significant savings in fuel costs.

THEORETICAL DEVELOPMENT

In order to better understand the basis for the success of the analytical method noted above and possibly make improvements, a theoretical development of condenser performance was made.

Setting heat load to the condenser equal to the total heat absorbed by the cooling water, an expression for condenser steam temperature can be derived.

$$UA(LMTD) = WC_p(T_2 - T_1) \tag{1}$$

Where:

$$LMTD = \frac{T_2 - T_1}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)} \tag{2}$$

Substituting for LMTD, Equation (1) becomes:

$$UA\,\frac{T_2 - T_1}{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)} = WC_p(T_2 - T_1) \tag{3}$$

Rearranging:

$$\ln\left(\frac{T_s - T_1}{T_s - T_2}\right) = \frac{UA\,(T_2 - T_1)}{WC_p\,(T_2 - T_1)} \tag{4}$$

or:

$$\ln\left(\frac{T_s - T_1}{T_s - T_2}\right) = \frac{UA}{WC_p} \tag{5}$$

and:

$$\frac{T_s - T_1}{T_s - T_2} = e^{\left(\frac{UA}{WC_p}\right)} \tag{6}$$

Setting:

$$e^{\left(\frac{UA}{WC_p}\right)} = E \tag{7}$$

Equation (6) becomes:

$$E = \frac{T_s - T_1}{T_s - T_2} \tag{8}$$

Rearranging:

$$ET_s - ET_2 = T_s - T_1 \tag{9}$$

Solving for T_s:

$$T_s = \frac{ET_2 - T_1}{E - 1} \tag{10}$$

adding $0 = ET_1 - ET_1$ to the numerator in Equation 10:

$$T_s = \frac{ET_2 - T_1 + ET_1 - ET_1}{E-1} \tag{11}$$

Rearranging:

$$T_s = \frac{ET_2 - ET_1 + ET_1 - T_1}{E-1} \tag{12}$$

or:

$$T_s = \frac{E(T_2 - T_1) + T_1(E-1)}{E-1} \qquad (13)$$

Which becomes:

$$T_s = \frac{E}{E-1}(T_2 - T_1) + T_1 \qquad (14)$$

Equation (14) demonstrates that condenser steam temperature, at a given cooling water inlet temperature varies linearly with cooling water temperature rise, at a slope of $E/(E-1)$ and y intercept at the inlet cooling water temperature, T_1. This is a very important observation. The equation, which applies above the cutoff pressure, states that every condenser and any condition of a particular condenser operates on a straight line that rotates about the inlet cooling water temperature, T_1. The single distinguishing feature between condensers of performance states of individual condensers is the slope or more precisely, the slope function E.

It is also important to note that the actual slope function E, for any point in time, can be determined from measurements of condenser pressure, cooling water inlet and outlet temperatures, the steam tables and Equation (8).

As performance degrades, the slope changes and establishes a new performance line. The slope function E therefore provides the key to determining the specific condition of a condenser, separating the causes of degradation from each other and developing a rationale for monitoring condenser performance. The first step toward this goal is an examination of the properties of the slope function E.

Recalling:

$$E = e^{\left(\frac{UA}{WC_p}\right)} \qquad (7)$$

and substituting $C\sqrt{V}\,F_1\,F_2\,F_3$ for U, Equation 7 becomes:

$$E = e^{\left(\frac{C\sqrt{V}\,F_1F_2F_3A}{WC_p}\right)} \qquad (15)$$

Examination of the exponent in Equation (15) reveals the following characteristics for the condenser performance line as the values of the parameters in the exponent change. These characteristics are depicted graphically in Figure 4.

1. As cleanliness decreases, cleanliness factor F_3 decreases which in turn causes the exponent to decrease and therefore the slope to increase (Note that for E greater than 1, the slope varies inversely with the slope function E).

2. As tube surface area A decreases, the exponent, slope and performance line behave as in 1 above (i.e., as A decreases the slope increases).

3. As inlet cooling water temperature T_1 increases, the exponent increases in proportion to the change in inlet cooling water correction factor and the slope decreases.

4. As cooling water flow decreases, the exponent increases in proportion to the square root of the flow and the slope decreases.

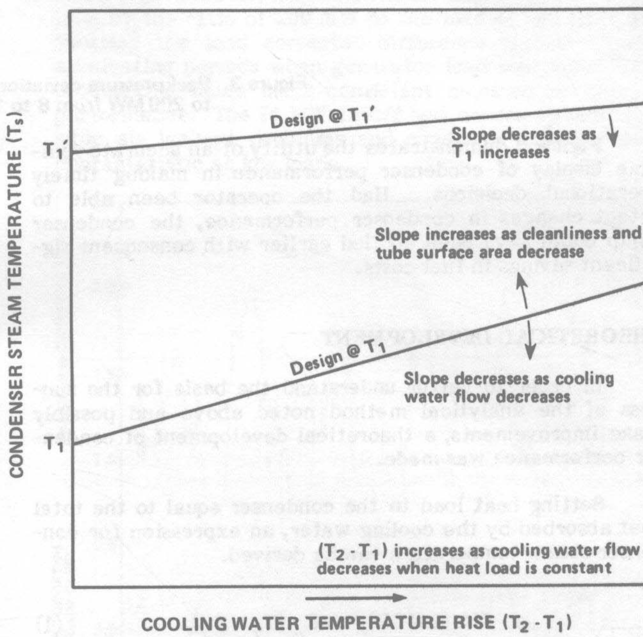

Figure 4. Slope variations in condenser steam temperature vs. cooling water temperature rise with changes in operating conditions.

Noting that 1) the performance line rotates about the inlet cooling water temperature, and 2) noting the causes for the rotation, it becomes obvious why conventional condenser performance monitoring methods produce so much scatter of the monitored parameter Deviation from Standard. Figure 5 was constructed to illustrate this problem which prevents timely and accurate determination of condenser degradation.

Two performance conditions of a condenser, operating at two different inlet cooling water temperatures, are plotted using the conventional parameters of condenser pressure and unit electric output. The absolute pressure limit cut off is not incorporated in the figure for reasons of clarity.

Note that the pressure difference between the two conditions varies considerably with both inlet cooling water temperature and electric output. Also note that using the coordinates of pressure and unit load assumes constant cooling water flow and a constant relationship between electric output and heat load to the condenser. If at any given electric output, cooling water flow or heat load to the condenser has changed from the assumptions made, still further variations in the Deviation from Standard will occur. Establishing a condenser performance standard based on these assumptions combine changes in performance due to degradation with normal changes unrelated to degradation, with the result that it becomes impossible to distinguish one from the other.

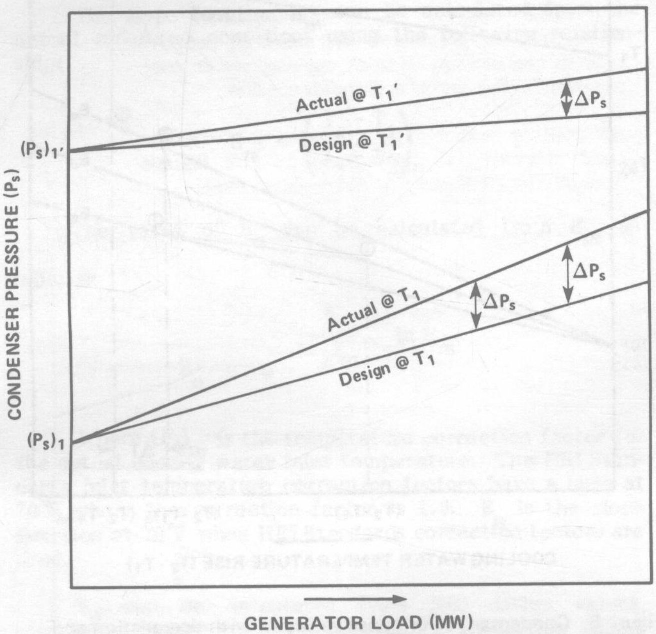

Figure 5. Condenser pressure variation with respect to load for two inlet water temperatures and two performance conditions.

DEVELOPMENT OF A MONITORING PROGRAM

Recognizing how changes in the parameters affecting performance occur and knowing the mathematical relationship between each parameter, a method has been devised which will separate the underlying cause for changes in condenser pressure and yield consistent monitoring results that can be evaluated in a timely and accurate manner.

Condenser pressure is the result of heat load to the condenser, inlet cooling water temperature, cooling water flow, tube surface area and the heat transfer coefficient at a point in time. Performance is degraded when: 1) Problems with cooling water flow occur, 2) The tube sheets become fouled, 3) Heat transfer across the tube surface is impaired by chemical or biological deposits, 4) Air blanketing on the steam side occurs (thus reducing the effective tube surface area), or 5) Heat load is increased for any given value of electrical output. These effects can be separated and quantified as described in the following analysis.

The first step is to obtain synoptic measurements of inlet cooling water temperature, discharge cooling water temperature, condenser pressure, cooling water flow and gross electric output. These parameters should be measured with an accuracy of better than 1 percent. If such precision is not available, differences from an established baseline value can generally be determined with reasonably small inaccuracies. Any change, once measured, can be investigated, the cause determined and corrections made.

Heat input to the condenser can be calculated from cooling water flow, temperatures and specific heat as follows:

$$\text{Heat load} = WC_p(T_2 - T_1) \tag{16}$$

Plotting heat load versus gross electrical output will provide a record that can be rectified with design or some other criteria. As problems occur in the plant cycle, such as increased turbine seal clearances or valves in pipelines to the condenser leaking or left open, heat load to the condenser increases causing increased condenser pressure. Unless increased heat load is identified, the resulting increased condenser pressure will be interpreted as a condenser problem rather than a plant cycle problem. When properly identified, investigations can be initiated, and where practical, the condition corrected.

If the tube sheets are relatively clear of debris and air leakage into the condenser is low, the cleanliness factor can be determined from the following relationship(4):

$$F_3 = \frac{U_m}{U_n} \tag{17}$$

Where:

U_m = Actual heat transfer coefficient at operating condition

U_n = Design heat transfer coefficient at 1.0 cleanliness factor, actual cooling water temperature and actual cooling water flow

Recalling Equation (5),

$$\frac{UA}{WC_p} = \ln\left(\frac{T_s - T_1}{T_s - T_2}\right) \tag{5}$$

The value of $\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)$ is determined from measurements of condenser pressure and cooling water inlet and outlet temperature. Therefore:

$$\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)_m = \left(\frac{UA}{WC_p}\right)_m \tag{18}$$

The value of $\left(\frac{UA}{WC_p}\right)_d$ is determined from design values, 1.0 cleanliness factor, 1.0 temperature correction factor and design cooling water flow. $\left(\frac{UA}{WC_p}\right)_n$ can be determined using the temperature correction factor at the actual cooling water inlet temperature and the actual cooling water flow in the following expression:

$$\left(\frac{UA}{WC_p}\right)_n = \left(\frac{UA}{WC_p}\right)_d (F_1)_m \sqrt{\frac{W_d}{W_m}} \tag{19}$$

Where $(F_1)_m$ is the temperature correction factor at the measured inlet cooling water temperature.

For the same condenser tube surface area, cooling water flow, inlet cooling water temperature and specific heat, the following expression is developed:

$$F_3 = \frac{U_m}{U_n} = \frac{\left(\frac{UA}{WC_p}\right)_m}{\left(\frac{UA}{WC_p}\right)_n} \qquad (20)$$

Using Equations 18, 19, and 20:

$$F_3 = \frac{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)_m}{\left(\frac{UA}{WC_p}\right)_d (F_1)_m \sqrt{\frac{W_d}{W_m}}} \qquad (21)$$

Equation 21 provides a means for calculating actual cleanliness factor from measurements of condenser pressure, cooling water inlet and outlet temperature and cooling water flow.

It should be noted that for divided water box condensers, Equation 21 can be applied to each condenser half separately by using the surface area and cooling water flow for a single side. This characteristic makes the method even more valuable as each condenser half can be monitored and analyzed independently.

The effect of reduced condenser tube surface area cannot be separated from cleanliness impact without additional information. If the tube sheets are relatively clean and excessive air leakage is not present, it is safe to assume that surface area has not changed and therefore a valid cleanliness factor can be determined. Empirical methods could be used to determine the affect of tube sheet blockage or air binding on condenser surface area. If determined, then a valid cleanliness factor could be calculated using the following expression:

$$F_3 = \frac{\ln\left(\frac{T_s - T_1}{T_s - T_2}\right)_m}{\left(\frac{UA}{WC_p}\right)_d (F_1)_m \sqrt{\frac{W_d}{W_m}} \; \frac{A_m}{A_d}} \qquad (22)$$

The performance parameters of heat load and cleanliness factor as determined from the above, and cooling water flow are not encumbered by the normally expected changes in condenser steam temperature and pressure, which result from changes in inlet cooling water temperature and condenser heat load. Monitoring of these parameters is recommended as a superior method over the conventional Deviation from Standard methodology.

CONDENSER PERFORMANCE COMPARISONS

The process of change and degradation of condenser performance can be better understood by examining Figure 6, which uses the information developed in the preceding sections. Assume a starting point at design conditions,

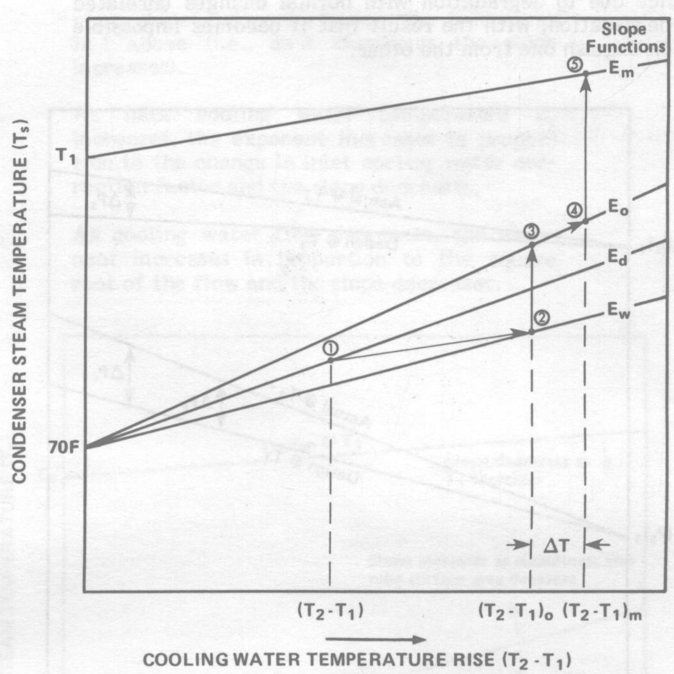

Figure 6. Condenser performance variation with degradation and changes in normal operating conditions.

100 percent cleanliness, 70°F inlet cooling water temperature, design cooling water flow, design surface area and a pre-chosen cooling water temperature rise. This point is designated as point 1 in Figure 6. The slope function at this point is E_d. If cooling water flow is allowed to degrade, the slope of the performance line decreases and the slope function becomes E_w. At the same time, cooling water temperature rise will increse because the heat load to the condenser has essentially not changed. If the temperature rise at point 1 was $(T_2 - T_1)$, then the temperature rise at point 2 will be inversely proportional to the decrease in cooling water flow or

$$(T_2 - T_1)_o = (T_2 - T_1)\frac{W_d}{W_m} \qquad (23)$$

If cleanliness factor and/or condenser surface area were to decrease, the slope would increase to a value corresponding to slope function E_o and condenser temperature would increase from point 2 to point 3. Cooling water temperature rise would remain essentially constant. However, if turbine load or some other heat source were to increase heat load to the condenser, condenser temperature would increase to point 4 with a corresponding increase in cooling water temperature rise.

Point 4 lies on a condenser performance line, degraded from a combination of decreased cooling water flow, cleanliness and surface area. If inlet cooling water temperature were to rise, the condenser performance line would rise to a higher level of condenser temperature and the new performance line would be at a decreased slope corresponding to slope function E_m.

Each of the slopes and condenser steam temperatures in Figure 6 can be calculated as shown in the following paragraphs. These calculations provide the basis for making valid performance comparisons, i.e., performance on the basis of duplicate inlet water temperature and heat load. It should be noted that points 1, 2 and 3 are on a comparable basis because the inlet water temperature and heat load for each point is the same.

The slope function E_m can be calculated from the actual measured conditions using the following relationship:

$$E_m = \left(\frac{T_s - T_1}{T_s - T_2}\right)_m \qquad (24)$$

The value of E_o can be calculated from E_m as follows:

$$E_o = e^{\frac{1}{(F_1)_m} \ln E_m} \qquad (25)$$

Where $(F_1)_m$ is the temperature correction factor for the actual cooling water inlet temperature. The HEI Standards inlet temperature correction factors have a base at 70°F where the correction factor is 1.0. E_o is the slope function at 70°F when HEI Standards correction factors are used.

E_d can be calculated from HEI design values, cleanliness factor of 1.0, temperature correction factor of 1.0, the design cooling water flow, and the following expression:

$$E_d = e^{\left(\frac{UA}{WC_p}\right)_d} \qquad (26)$$

Having calculated E_d, E_w can be calculated from:

$$E_w = e^{\sqrt{\frac{W_d}{W_m}} \ln E_d} \qquad (27)$$

Having all the slope functions from the above, condenser steam temperatures can be calculated as follows (continue to refer to Figure 6):

$$(T_s)_1 = \frac{E_d}{E_d - 1}(T_2 - T_1) + 70 \qquad (28)$$

$$(T_s)_2 = \frac{E_w}{E_w - 1}(T_2 - T_1)\frac{W_d}{W_m} + 70 \qquad (29)$$

$$(T_s)_3 = \frac{E_o}{E_o - 1}(T_2 - T_1)\frac{W_d}{W_m} + 70 \qquad (30)$$

$$(T_s)_4 = \frac{E_o}{E_o - 1}\left[(T_2 - T_1)\frac{W_d}{W_m} + \Delta T\right] + 70 \qquad (31)$$

$$(T_s)_5 = \frac{E_m}{E_m - 1}(T_2 - T_1)_m + T_1 \qquad (32)$$

The steam temperatures above can be converted to condenser pressures using the steam tables for saturated conditions.

FIELD MEASUREMENTS

Application of these theoretical calculations were initially unsuccessful. A sensitivity analysis for the test data revealed that a 0.1°F change in outlet cooling water temperature changes cleanliness factor by 0.01 or 1 percent. Because of this finding, the measurement of the outlet cooling water temperatures was investigated.

Cooling water outlet temperatures in each leg of a divided water box condenser were originally measured using a single accurate mercury-in-glass thermometer. Inconsistencies in cooling water temperature rise data from those sensors led to the installation of an additional temperature probe. Five thermistors were spaced evenly across the discharge conduit from one water box which was then left for a 4-day interval, recording water temperatures every 12 minutes. When the single point temperature was compared to the average, a significant deviation was found. The data were plotted to show the temperature difference between the plant sensor and the average of the 5 points over the load range (Figure 7). It is apparent that the single point does not track unit load and varies from zero to one degree Fahrenheit. This variation with load caused considerable scatter in the calculated data which could amount to as much as a 10 percent deviation in cleanliness factor. The condition was corrected by using the average of 5 points for the value of the discharge cooling water temperature from each condenser half.

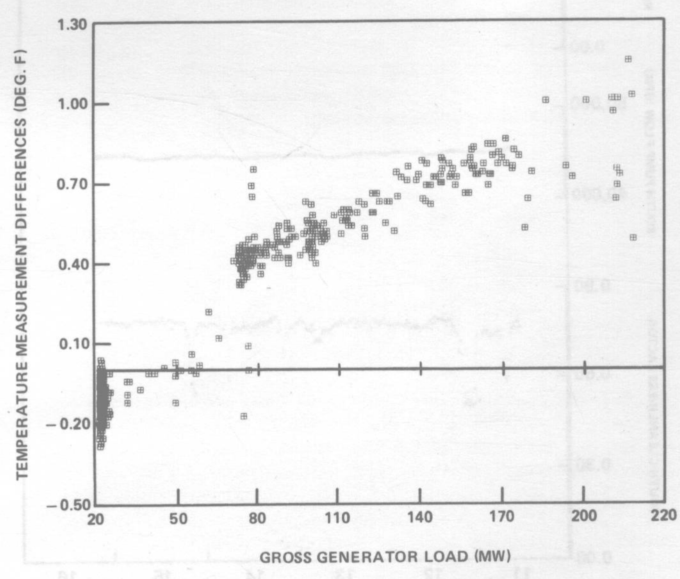

Figure 7. Temperature difference (Average conduit temperature minus single point temperature) vs. unit load.

In order to calculate a cleanliness factor it is necessary to measure cooling water flow or a flow index sensitive to changes in flow. Our studies at Huntington Beach utilized cooling water pump total head which had been previously calibrated to flow using a dye test. Total head was determined from measured discharge pressure and suction head. Suction head was calculated from tide elevations taken from tide tables. Cleanliness factor was then calculated using Equation 21.

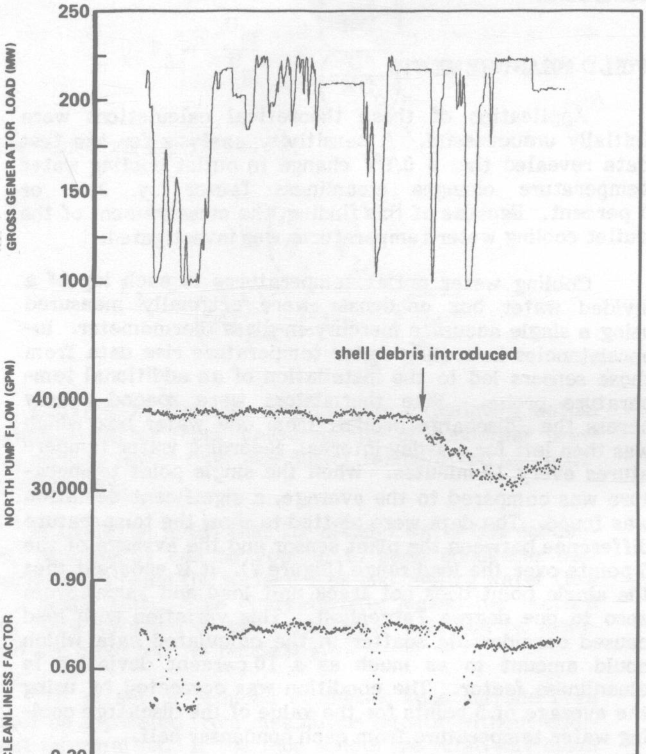

Figure 8. North and South condenser half calculated cleanliness factor, unit load, and pump flow from 11 to 16 Mar 1985.

Figure 8 demonstrates the calculation of actual cleanliness factors for each condenser half during simultaneous operation. These data cover a stable operating period during which a known quantity of marine shell debris was artifically introduced into one half of the condenser as shown.

Note that prior to the introduction of shell debris, performance above 175 MW is steady with cleanliness factor remaining in a band of about 2 to 3 percent. As the debris is introduced, the north cooling water flow decreased as the tube sheet becomes progressively plugged. Cleanliness factor during the initial fouling remains essentially unchanged demonstrating the validity of our calculations even though a decrease of about 10 percent in cooling water flow takes place. It is not until a significant number of tubes have become plugged and condenser tube surface area has been significantly impacted that the calculation of cleanliness factor becomes unreliable. Steady conditions in the south condenser half during this period support our belief that cleanliness had not changed and valid cleanliness factors were calculated. Calculated cleanliness factors at loads below 150-170 MW appear to be affected by inconsistencies in measuring T_2.

CONCLUSIONS

We have concluded that the above techniques can be incorporated into a monitoring package that will greatly enhance the plant engineers' ability to monitor plant performance and optimize maintenance activities. Such a package will also provide the researcher with a means for evaluating the need and impact of biofouling control measures. These activities can be accomplished with a level of precision not attainable with conventional means. Costly test apparatus is not required nor do the results contain the uncertainty inherent in a separate condenser simulation device. Inexpensive sensors can be used to directly measure actual conditions in an operating condenser.

As an example, because the normal changes in condenser backpressure due to changes in electric load and inlet water temperature cover a range of 2 to 3 inches of mercury, the operator is unaware of a performance degradation until a deviation on the order of 0.5 inches of mercury has occurred. When it does, it is not known if the problem requires chlorination, cleaning of the condenser tube sheets, or a search for a steam or air leak into the condenser. By monitoring cleanliness factor, cooling water flow, and condenser heat input on a real-time basis, the corrective action can be prescribed almost immediately.

In the same way, researching the application of chlorine may be conducted at precision levels not achievable with conventional monitoring. Changes on the order of 0.1 to 0.2 inches of mercury can be detected immediately. Since cleanliness factor is measured directly, uncertainly due to changes in load or cooling water flow are eliminated. It is possible that the actual results of variations in chloration intensity or duration can be judged immediately. In any case, the condenser need not be drained and physically examined to determine the impact of a modified chlorination program as is currently practiced. Indeed, because the purpose of a chlorination program is to preserve performance, precise measurement of performance makes inspections to a large extent superflous.

The method in its present state can be further refined. We expect that even better results can be gained through refinements in measuring cooling water outlet temperature and cooling water flow. We also suspect that the relationship of corrected heat transfer coefficient with tube velocity and inlet cooling water temperature can be improved over the information given in standard condenser heat transfer references.

In addition to the need to further refine our chlorination practices, we have noted unusual performance at low turbine generator loads. Condenser performance at low loads appears to be much less than the literature would predict. It is suspected that air blanketing of the condenser is a much more serious problem at low loads than was previously assumed. Significant gains in heat rate at low load may be obtainable through a better understanding of performance gained through these improved monitoring techniques.

The most vexing problem that remains is a means for detecting and quantifying the changes in effective condenser tube surface area, as may occur when air blanketing or severe tubesheet pluggage is present. If such information could be developed, the monitoring scheme would be complete. Further efforts are being directed toward developing answers in these areas.

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COMPUTERIZED THERMAL PERFORMANCE MODELING OF UTILITY POWER CONDENSER FOULING

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ABSTRACT

Condenser biofouling can have a large effect on the cost of electric generation from steam electric generating units. Determination of the optimum frequency of condenser cleanings can minimize this cost.

This paper presents a method which compares the cost of cleaning to the benefits of improved condenser performance to find the optimum condenser cleaning frequency. The method, based on "discrete" condenser cleaning, is simple and applicable to any steam electric generating unit with the requisite computerized thermal performance monitoring (TPM). A simple numerical example is presented to demonstrate the calculations and the applicability of the method.

METHODOLOGY

There are two factors--cost of cleaning and the total fuel cost penalty due to fouling--that determine the optimum number of cleanings in a given time period.

If the cost of a discrete cleaning is assumed to be constant, the total cleaning cost will be proportional to the total number of cleanings, n. The cost of a cleaning, C, will comprise the following components:

- Labor and materials costs necessary for the cleaning
- Unit restriction cost (replacement energy cost and replacement of spinning reserve, if applicable)
- Decrease in unit efficiency during the cleaning due to out-of-service water box or backwash thermal effects of circulating water

The thermal performance of the unit will improve significantly with more frequent cleaning. This will result in a decrease in heat rate which is caused by condenser biofouling (see Figure 1). The curve shown in Figure 1 is a function of the number of cleanings per unit time, n, and will be referred to as F (n). The shape of the curve is not as important as the fact that it decreases continuously.

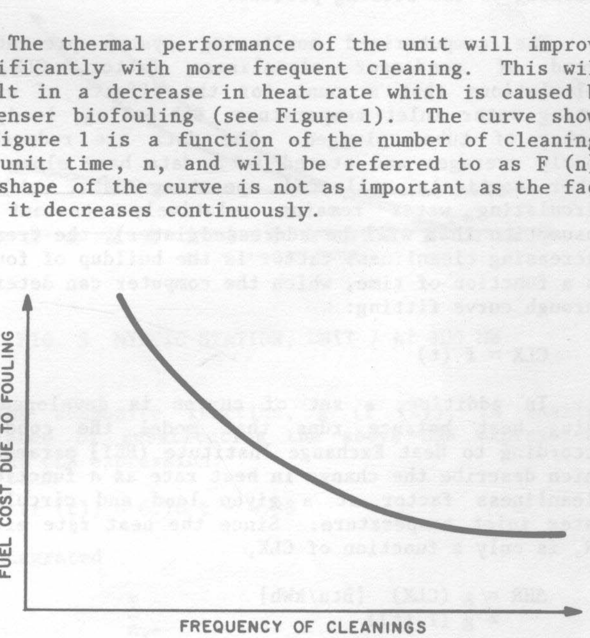

FIG. 1 FUEL COST VS CLEANING FREQUENCY

The optimum frequency of cleanings occurs when the sum of the cost of cleaning and the fuel cost penalty is minimized (see Figure 2). This can be found by setting the first derivative of the total combined cost, C (n) + F (n), to zero and solving for n.

$$\frac{d\,C\,(n)}{dn} + \frac{d\,F\,(n)}{dn} = 0$$

[1]The term "discrete" is used to describe noncontinuous activities that require some disruption in the operation of the unit, such as backwash or manual cleaning, as opposed to "on-line" methods, such as chlorination or continuous ball cleaning.

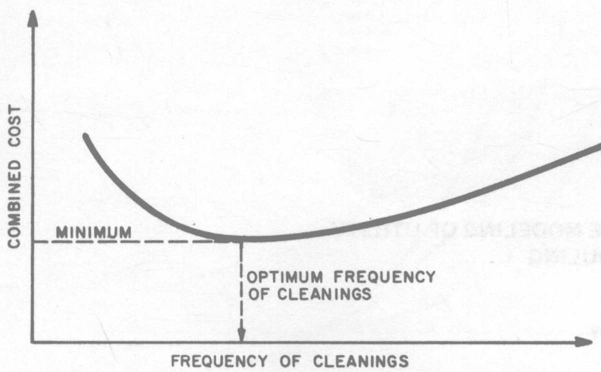

FIG. 2 TOTAL COST VS CLEANING FREQUENCY

FORMULATION OF FOULING COST F (n)

The focus of this investigation is the application of a computerized performance monitoring system to determine F (n), which, in turn, will determine the optimum cleaning schedule. The approach is similar to that used by Nery and Bell (1), where the authors used an implicit "trial and error" method to determine which of several proposed cleaning schedules minimized cost. In the method described herein, the optimum cleaning schedule is found explicitly through mathematical modeling of the fouling process.

The computerized monitoring system presents a trend of condenser cleanliness factor, CLX, by calculations that account for the effects of circulating water inlet temperature, subcooling, load, and number of tubes plugged. The data are reduced to hourly averages and stored in a data base along with other routinely collected operating data. If the circulating water remains relatively constant (an assumption that will be addressed later), the trend of decreasing cleanliness factor is the buildup of fouling as a function of time, which the computer can determine through curve fitting:

$$CLX = f\ (t)$$

In addition, a set of curves is developed (by using heat balance runs that model the condenser according to Heat Exchange Institute (HEI) parameters) which describe the change in heat rate as a function of cleanliness factor at a given load and circulating water inlet temperature. Since the heat rate effect, HR, is only a function of CLX,

$$\begin{aligned}\Delta HR &= g\ (CLX) \quad [\text{Btu/kWh}] \\ &= g\ (f\ (t))\end{aligned}$$

At this point, it may be necessary to modify function g (CLX) to take into account any cycling of the unit in question; otherwise, CLX is used directly from the full-load curve for a base-loaded unit, as in the sample case in reference (1).

By multiplying by daily MWh production and fuel cost, the cost of fouling is determined on a daily basis as:

$$G \cdot g\ (f\ (t))$$

where G = average daily MWh production x fuel cost

The total cost can be determined over time, T, by integrating the daily cost over time.

$$F\ (t) = G \int_{o'}^{t} g\ (f\ (t)\ dt$$

In this integral, the upper limit, t, is the time between cleanings. Since the analysis is based in part on a constant circulating water inlet temperature, an overall time period, T, must be selected from which the circulating water temperature can be taken as constant. In this manner the assumption that heat rate penalty and rate of fouling are independent of circulating water temperature is validated. Thus, the period between cleanings is the overall time divided by the number of cleanings in that time.

$$t = \frac{T}{n}$$

This now becomes the upper limit of the integral and F (t) becomes F (n).

$$F\ (n) = G \int_{o}^{\frac{T}{n}} g\ (f\ (t))\ dt$$

The cost penalty due to fouling is now in the form to be differentiated by the number of cleanings, n, to find the optimum number of cleanings in time period, T.

At the end of the total time period, T, the analysis should be repeated using a new average circulating water temperature, and, if applicable, a new trend for fouling and a new loading scheme.

THERMAL PERFORMANCE MONITORING PROGRAM

The tool used for trenching cleanliness factor is Stone & Webster Engineering Corporation's TPM computer program. This program is a microcomputer-based software package that uses operating data taken from plant instrumentation and plant design data to determine the actual and expected heat rate, and the actual and expected performance of major pieces of equipment. For an analysis of condenser performance, the following operating data is entered:

- Generator Load
- Circulating Water Inlet Temperature
- Circulating Water Outlet Temperature
- Condenser Pressure

The program uses design data to approximate the condenser duty. Cleanliness factor and expected backpressure are calculated using the HEI method. The heat rate effect of fouling is determined from heat rate correction curves that are calculated, using heat balance techniques, and are stored permanently in the program.

EXAMPLE

Boston Edison Company's (BECO) Mystic Station Unit 7 is a 565 MW oil/gas-fired steam electric unit located on the Mystic River in Everett, Massachusetts. The station has a tandem compound, four-flow General Electric turbine, with 30-inch last stage buckets, that exhausts to a 23,166-tube, 242,000 sq ft condenser. Cooling water is brackish and is pumped at 290,000 gpm.

Data collected and reduced by the station's Foxboro process computer and interpreted with the aid of Stone & Webster Engineering Corporation's TPM program will be used to demonstrate the optimization method.

The equations used are simplified to linear equations so that the example is more easily followed. In practice, at least second degree polynomials should be used to ensure accuracy, especially in the case of heavily last stage end-loaded machines.

A cleanliness factor trend developed by the TPM program (Figure 3) was approximated by the following linear equation:

$$CLX = 0.935 - 0.13\ t$$

where t = number of days since last cleaning.

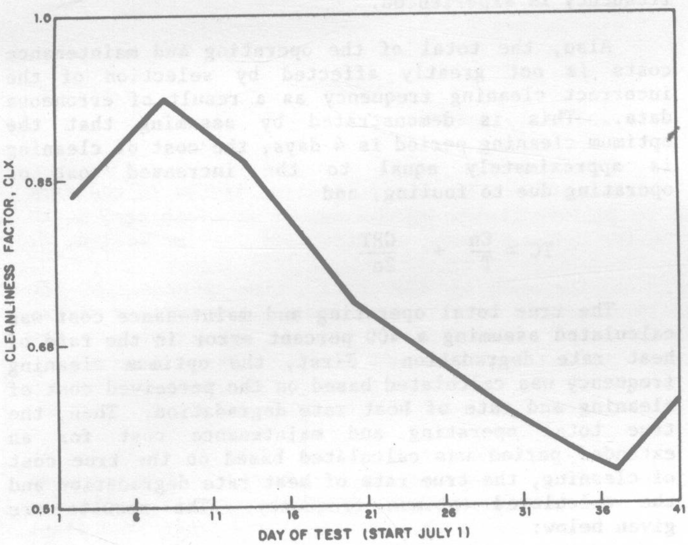

FIG. 3 CLEANLINESS FACTOR VS TIME

Likewise, the curves of heat rate effect as a function of cleanliness factor (Figures 4 and 5) at 70°F circulating water inlet temperature were developed from TPM output. They can be approximated by:

$$\Delta HR\Big|_{565\ MW} = 53.99 - 63.52\ (CLX)$$

$$\Delta HR\Big|_{100\ MW} = 23.42 - 27.55\ (CLX)$$

Mystic Unit 7 typically cycles between 565 MW (day) and 100 MW (night). A prorated curve to represent this loading is as follows:

$$\Delta HR\Big|_{cycle} = 43.80 - 51.53\ (CLX)$$

This loading would generate 9,840,000 kWh daily. During the heavy fouling season, Mystic 7 burns natural gas, which costs approximately \$3.75 per million Btu. This makes the multiplying factor, G, equal to 36.90 (kWh \$/Btu day).

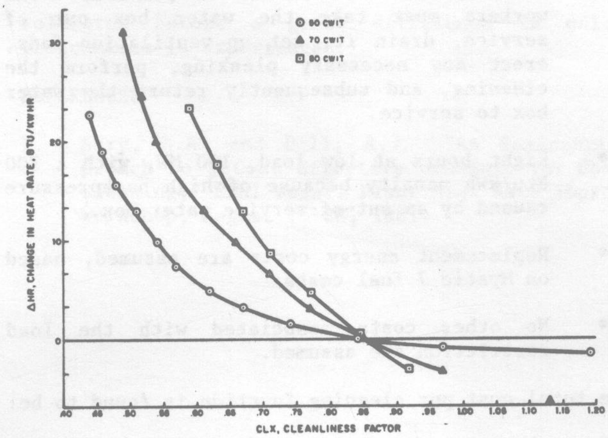

FIG. 4 MYSTIC STATION, UNIT 7 AT 565 MW

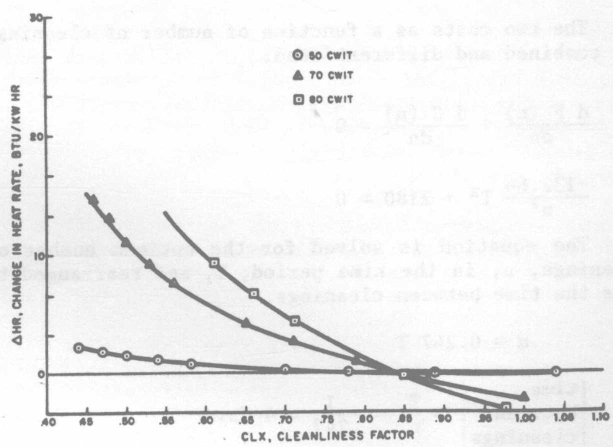

FIG. 5 MYSTIC STATION, UNIT 7 AT 100 MW

The composite function, g (f (t)), is easily calculated by substituting the above ΔHR expression with the CLX expression.

$$g\ (f\ (t)) = 6.70\ t - 4.38$$

and integrated

$$F\ (n) = nG \int_{0}^{\frac{T}{n}} (6.70\ t - 4.38)\ dt$$

$$= \frac{3.35\ GT^2}{n} - 4.38\ GT$$

$$F\ (n) = \frac{132.64\ T^2}{n} - 173.45\ T$$

where G = 36.9 as shown above.

The following assumptions can be used to determine a cleaning cost: