USING TURBINE THERMAL KIT DATA TO BENCHMARK CONDENSER PERFORMANCE CALCULATIONS

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ABSTRACT

In the past, the performance of a condenser has been judged by empirical criteria such as cleanliness factor, terminal temperature difference and/or cooling water pressure drop. However, these criteria vary with load, cooling water inlet temperature and cooling water flow rate. Thus, to base maintenance or operating decisions on deviations in the empirical criteria requires that these other factors be taken into account.

One way to avoid this is to develop a benchmark or reference condenser duty. A source for this reference can be found in the analysis of turbine thermal kit data to create a design model of the turbine LP stage. Provided that the turbogenerator is operating with the equipment configuration on which the thermal kit data was based; and that the boiler operating conditions are close to design for a given load; then the exhaust enthalpy and flow entering the condenser can be calculated as a function of load and back pressure, and these used to estimate present condenser duty. Thus, the performance of a condenser can be compared against a calibrated and stable frame of reference, which changes only very slowly over time.

The method allows cooling water flow rates to be estimated and compared; while tube fouling factors and condenser ambient heat discharges can also be quantified. Further, the excess heat discharges due to the fouling of the condenser and its effect on the turbogenerator are initially calculated in MBTU/h. However, they can also be converted to the equivalent economic loss in $/h; as well as the equivalent lbs. of excess CO_2 emissions per hour.

THE RANKINE CYCLE

Almost without exception, fossil-fired boiler/turbogenerator units are based on the Rankine Cycle. Here, condensate from a condenser is heated and compressed before it enters the boiler. The water is evaporated by the heat transferred from the combustion chamber through the tube walls, the vapor then being passed through other tubes located in the gas path, in which the temperature is raised to produce superheated steam. In many plants, the exhaust from the high pressure stage of the turbine is passed back through reheater tubes before it enters the IP stage of the turbine, so improving the overall efficiency of the unit.

As the steam expands through the turbine, the associated drop in enthalpy is converted to electric power. While some of the steam is extracted to preheat the boiler feedwater, the balance of the steam leaves the exhaust of the low pressure stage of the turbine as a wet vapor.

In order for the vapor to be recovered and used again, it must be passed to a condenser in which the residual heat is removed, so creating a pool of condensate which can be pumped back through the system. Such thermodynamic cycles are usually plotted on Steam Heat/Enthalpy (H/S) Charts, similar to those published by ASME (1993), in which the ordinate is enthalpy (BTU/lb) and the abscissa is entropy in BTU/(lb.Deg.R), a typical example of a Rankine Cycle plot being shown in Figure 1.0.

TURBINE THERMAL KIT DATA

Figure 1.0 shows a Rankine Cycle plot for one particular load, but Figure 2.0 shows how the line plotted between the reheater outlet conditions and expansion line end point (ELEP) shifts with load. All of this data can be abstracted from the thermal kits supplied by the steam turbine manufacturer, the data usually having been verified during the original equipment acceptance tests. Clearly, it is possible to regress both the reheater outlet conditions and expansion line end point conditions with respect to load. The locus of the expansion of the steam between these two points is usually shown as straight line. Although Silvestri (1997) has termed this practice a "useful fiction", the linearity does help to simplify the calculations outlined below, without introducing any significant departure from reality.

The exhaust from the low pressure stage of a turbine is usually provided with an annulus through which the turbine exhaust vapor passes before it enters the condenser itself. Robinson(1933) and Spencer, et al(1962) have examined the effect of the annulus on exhaust steam conditions.

The pressure and enthalpy conditions before the annulus are often termed the turbine end point (TEP) and, from these, the other properties at the TEP can be calculated. Figure 2.0 also includes a typical plot of TEP, the value of which varies with load.

As would be expected, flow through the annulus causes an enthalpy drop across the annulus, usually referred to as the annulus loss, a typical plot of which is shown in Figure 3.0. Note that some vendors plot annulus loss with respect to annulus velocity while others plot the loss with respect to volumetric flow. The regression of annulus loss curves is performed by considering them as two curves separated by the lowest point in the velocity range. Finally, the conditions at the TEP, together with design exhaust flow, can also be abstracted from the thermal kit data and both regressed with respect to load.

Normally the expansion of vapor through the LP stage follows the path shown in Figure 4.0. However, if the condenser back pressure should fall to a very low value due, for example, to a low cooling water temperature during winter months, the annulus can become *choked*. When this occurs, the TEP can not fall below a minimum value, at which point the exhaust flow is also maximized. Figure 5.0 shows the path followed by the vapor when the annulus is under choked conditions and it will be seen that the enthalpy at the *throat* of the annulus is the same as that of the vapor at the current condenser back pressure. Clearly, in terms of a potential increase in generated power and reduced exhaust flow rate, full advantage can not be taken of the effects of a low back pressure when the annulus is operating under choked conditions.

The importance of these observations is that the possibility of annulus choking must be examined, before the latent heat to be removed from the vapor can be correctly estimated. This amount of latent heat is, of course, equivalent to condenser duty.

EXPANSION LINE ANALYSIS

Putman et al (1996,1997) have described methods of modelling Turbine LP stage/Condenser subsystems for one, two or three-compartment condensers to determine the fouling factor and cost of losses due to fouling. This method requires the condenser duty to be calculated as a function of condenser back pressure (or saturation temperature) as well as LP stage exhaust flow.

If the annulus is not choked, then the TEP can be determined from back pressure and annulus loss, based on velocity or volumetric flow, themselves functions of exhaust flow. However, in a fossil-fired turbogenerator unit, the governor responds automatically to changes in back pressure as will the exhaust flow and, to a first approximation, an adjustment can be made to the design exhaust flow as a function of design and current TEP's thus:

$$EXFL_{adj} = EXFL_{des} * (HTHROTTL - HTEP_{des}) / (HTHROTTL - HTEP_{calc}) \quad (1)$$

But, while the flow and turbine end point enthalpy ($HTEP_{calc}$) depend on back pressure, they can not be directly calculated from that parameter because of an uncertain value of annulus loss. This problem can be avoided by, first, taking the expansion line corresponding to a given load, and calculating the properties of the vapor at various values of TEP pressure, e.g. from 10.0 ins.Hg to 1 ins.Hg in 1 in.Hg steps. These properties include the associated exhaust flow, exhaust losses, and the enthalpy, pressure and saturation temperature at the expansion line end point. Once a table has been constructed containing this data, the exhaust flow and $HTEP_{calc}$ can now be regressed with respect to expansion line end point (ELEP) pressure or condenser shell temperature. Thus the analysis is first conducted with respect to TEP ("from the top, down"); while the regression analysis and the calculations within the model can be conducted with respect to ELEP ("from the bottom, up").

While conducting the above analysis, it is also possible to determine the TEP at which the exhaust annulus velocity, calculated as a function of TEP, equals the exhaust velocity calculated as a function of the annulus losses, using the well-known thermodynamic relationship (Lewitt,1953):

$$(EXFL_{throat} * SPVOL_{throat}) / (\text{Annulus Area}) = 3600 * 224 * SQRT(\text{Annulus Loss}) \quad (2)$$

The choking point occurs when these two quantities are the same and the associated values of the maximum condenser duty and minimum annulus throat pressure (or saturation temperature) are stored and then used as constraints during the convergence of the turbogenerator LP stage/condenser subsystem model described below. Figure 5.0 shows that these constraints come into play when the condenser back pressure is lower than the throat pressure of the *choked* annulus.

ESTIMATING COOLING WATER FLOW RATE

Cooling water flow rate is perhaps the most important parameter used in the calculation of both the present and reference heat transfer coefficients, whether by the HEI or ASME method. Unfortunately, it can seldom be *measured* directly, due to the large diameter piping normally involved and the costliness of an appropriate flow meter. Nor is it able to be *estimated* with any accuracy from the pump characteristic curves, due to impeller wear, change in number of pumps operating, variations in density due to water temperature or changes in salinity, all of which can affect the flow rate. Clearly, when estimating the performance of the condenser, to have a reliable method for estimating cooling water flow rate would also greatly enhance the accuracy of those calculations in which flow rate is included.

It has been shown above that condenser duty can be calculated for a given load and back pressure from an analysis of the expansion line for that load. A high confidence can be placed in this value, provided that the unit is not operating under sliding pressure control nor has a feed heater out of service. In the latter case, a conventional heat balance program (e.g. PEPSE) can be used to generate the data for the changed configuration, the regressed curves being adjusted from this data. The condenser duty must also take account of the heat in the exhaust from any boiler feed pump turbines, as well as gland and drain heat, all of which are included in the thermal kit data and vary with load.

Once condenser duty is known, cooling water flow rate can be estimated by dividing condenser duty by the water temperature rise, making due allowance for the density and specific heat corresponding to the bulk water temperature, thus:

$$CWFLOW = (DUTY * 62.3 * 1.0E{+}06) / (8.34 * 60 * (T_{out} - T_{in}) * SPHT * DENS) \qquad (3)$$

Provided the water temperatures are accurate and representative of the actual conditions, plots of the calculated flows clearly show the effect of fouling on flow and back pressure, can be used to distinguish the effect of tubesheet fouling from tube deposit fouling; and have also been able to confirm the number of circulating water pumps in operation. It is also possible to evaluate the effect on flow of several pumps operating in parallel.

In addition, the method can be used to calculate cooling water flow rate to the condenser of a nuclear plant, which also operates in accordance with the Rankine Cycle.

CONDENSER PERFORMANCE MONITORING

Clean Condenser - Operating Conditions

Putman and Saxon(1996) have shown how a model of the turbine LP stage and condenser subsystem can be used to calculate the conditions throughout the subsystem assuming that the condenser tubes were clean. The model, consists of a set of non-linear simultaneous equations which reflect the configuration of the condenser and its heat transfer relationships; and these can be solved using the Newton-Raphson iterative procedure.

The boundary conditions for this model of a clean condenser are not only the condenser design details but also the generated power, the inlet water temperature and cooling water flow, the latter usually being the design flow or, alternatively, the flow calculated when the condenser has just been cleaned. The relationships which were developed from thermal kit data and the analysis of the expansion line for the load of interest, are an important part of this model. On convergence, the results from the model include the estimated duty of a clean condenser, compartmental back pressures, water outlet temperatures and the associated compartmental ASME tube heat transfer coefficients, calculated in accordance with ASME PTC.12.2-1996 (ASME,1996). As discussed earlier, the calculations are checked at each iteration to test whether choking has occurred and, if so, the appropriate constraints are applied.

Clearly, a comparison of the clean condenser duty with the duty when fouled, is a measure of the excess amount of heat discharged into the water; and not only of the economic loss, but also the quantity of CO_2 emissions which can be ascribed to fouling. Further, the ASME Heat Transfer coefficients for the clean condenser can form the basis for calculating tube fouling factors and these then used to develop a fouling model for the unit.

Fouled Condenser - Compartmental Fouling Factors

In the course of estimating the present cooling water flow rate, the condenser duty as a function of load and back pressure(s) has been calculated, as well as the exhaust flow rate. Another model of the condenser/LP stage subsystem can now be written which accepts the estimated duty, cooling water and exhaust steam flows, and water and shell temperatures as boundary conditions. Among the data presented to the model are also included the tube diameter, gage, length and material.

This model is then used to estimate the inter-compartmental water temperature(s), the value of the ASME tube heat transfer coefficient (in accordance with ASME PTC.12.2-1996) and the compartmental fouling factors (FF_{tot}), all of which allow the turbine exhaust heat to be transferred through the condenser tubes to the cooling water, under the specified set of boundary conditions.

Note that fouling factors FF_{tot} include the effects of both tubesheet and tube deposit fouling; and may even include the effect of air ingress if that should be the case.

Fouled Condenser - Tubesheet Fouling Loss

As outlined in the previous section, it is the present value of cooling water flow rate which was used to calculate the compartmental fouling factors (FF_{tot}). However, if the fouling should occur to the tubesheet and it is removed, the water flow will rise, probably to its design value. If the design flow rate is substituted for the present cooling water flow rate in the previous model, the ASME heat transfer coefficients and fouling factors subsequently calculated will be those corresponding to tube deposit fouling alone (FF_{tube}). The percentage of contribution of tubesheet fouling (TSFpercent) to total losses can now be calculated from:

$$TSF_{percent} = 100.0 * (FF_{tot} - FF_{tube}) / FF_{tot} \quad (4)$$

BIG BEND CASE STUDY

Plant Design Details

Unit #4 at the Big Bend Power Plant of TECO Energy is a 444 MW unit equipped with a two-compartment once-through condenser. The cooling water is drawn directly from Tampa Bay and the condenser has shown a tendency for both tubesheet fouling and deposit fouling of the tubes. The frequency of cleaning varies, but is on the order of days rather than weeks or months.

Cooling Water Flow Rate

Figure 6.0 shows a plot of generated power, cooling water flow and back pressure vs. time based on data taken during June 1997 and, for a given load, demonstrates a strong correlation between the calculated flow and measured back pressure. For the two pumps which were running during this period, the plot shows that, while the back pressure tends to fluctuate with the load, the water flow rate is consistently declining due to what was known to be progressive tubesheet fouling. When the back pressure rises to 5 ins.Hg. the unit is shut down to remove fouling from the tubesheet. Immediately afterwards, the back pressure falls to about 3.5 ins Hg. while the flow is restored to its design value of about 225,000 GPM. Thus this method of estimating cooling water flow rate provides a valuable verification of other changes which are known to be occurring.

Using data taken from another plant, Figures 7.0 and 8.0 show the consistency of the results when plotted against *load*. Figure 7.0 shows the flows calculated when only one pump was running while Figure 8.0 is for the two-pump case. The estimated flows are remarkably consistent regardless of the load, and even though the inlet water temperature varies substantially for the various data sets, which were taken at different points in time.

Since it is rare for the cooling water flow rate to a condenser to be directly measured, and yet the flow is a vital part of condenser cleanliness factor and related calculations, the cooling water flow rate estimated in this way can provide more consistent values of cleanliness factors, heat transfer coefficients and fouling factors, so allowing the results to respond more closely to the changing conditions.

Condenser Performance Monitoring

An existing condenser performance monitoring program was adapted to match the configuration of the condenser on Big Bend Unit #4. The main menu included the functions shown in Table I and allowed condenser operating data sets to be entered, the performance calculated in the ways discussed below and, if required, the results stored in a file for later plotting and post-processing. Even the properties of steam and water can be calculated using the Steam Table function.

Table II shows the set of fixed and variable input data required to model this condenser. In order to calculate condenser performance, the only data which needs to be entered is the variable set shown at the bottom of Table II.

Table III shows the set of input data and calculated results associated with the estimation of cooling water flow rate.

Table IV displays the set of input and calculated data associated with the calculation of basic condenser performance. It will be seen that the load and cooling water inlet temperature are the only two variables which have the same value in both columns. The left hand column contains the present estimated cooling water flow rate while the cooling water flow rate in the right hand column is either the design value or the value calculated immediately after the condenser had been cleaned. The other data in the right hand column is that contained in the solution when the model converges under the Newton-Raphson procedure. Fouling losses are the difference between the condenser duty when fouled and that estimated if the condenser were to be cleaned.

Table V lists the present actual heat transfer coefficient as well as the heat transfer coefficients calculated according to both the HEI and ASME procedures. The fouling resistances shown towards the bottom of the left hand column are those calculated to bring the duty, water temperature rise, ASME coefficients, etc. into equilibrium when the fouled model has converged.

Table VI shows the distribution of losses between tubesheet

(Macro-fouling) and tube deposits (micro-fouling) and was calculated using the value the cooling water flow assumes after the condenser has been cleaned. Figure 9.0 is a plot of data calculated using this program and shows the progress of fouling resistance. This is consistent with the measured back pressure as well as the calculated flow rate.

Figure 10.0 includes the calculated hourly fouling losses, which are seen to rise as the fouling resistance increases. By multiplying the fouling losses (MBTU/h) by the cost of fuel ($/MBTU), the equivalent hourly cost in dollars can be calculated. To convert the losses due to fouling (MBTU/h) to equivalent carbon dioxide emission due to fouling (lb.CO_2/h), the data contained in Table VII may be used with equation (5). The two major fuel properties associated with the carbon dioxide emission calculation are carbon content of the fuel (weight) and fuel heating value. For the three major fuels they are typically as shown in Table VII.

Now 1 lb. carbon produces 3.6644 lbs CO_2 and, assuming a boiler combustion efficiency of 95%, the lbs. of carbon dioxide emissions (CE) per one MBTU change in condenser loss may be calculated from:

$$\begin{aligned} CE &= 3.6644 * C * 1.0E{+}06 / (0.95 * HV) \\ &= 3.8573E{+}06 * C / HV \qquad (5) \end{aligned}$$

The last column in Table VII indicates the equivalent carbon emissions per MBTU fouling loss, stated in accordance with accepted IPCC (IPCC,1995) practice.

CONCLUSIONS

Turbine thermal kit data can be analyzed and used to construct a model of turbine behavior with respect to generated power and condenser back pressure. The model will also detect whether the exhaust annulus is becoming choked and allow for that in the calculations. This turbine model is independent of both condenser design and waterside temperature and flow, so providing a condenser duty and exhaust flow benchmark with which to compare the performance of the condenser as it operates under a wide variety of conditions.

The condenser duty can be used to estimate cooling water flow rate with consistency, certainly allowing it to respond to changes in turbine operating conditions in a predictable manner, and regardless of whether the condenser is clean or fouled. The use of this estimated cooling water flow rate also enhances the quality of cleanliness factor and heat transfer coefficient calculations.

The turbine L.P. stage/condenser model can also take advantage of this predictability and the calculated values of fouling resistances and fouling losses can now be confidently compared against the benchmarked condenser duty.

An estimate of the equivalent CO_2 or carbon emissions can also be obtained.

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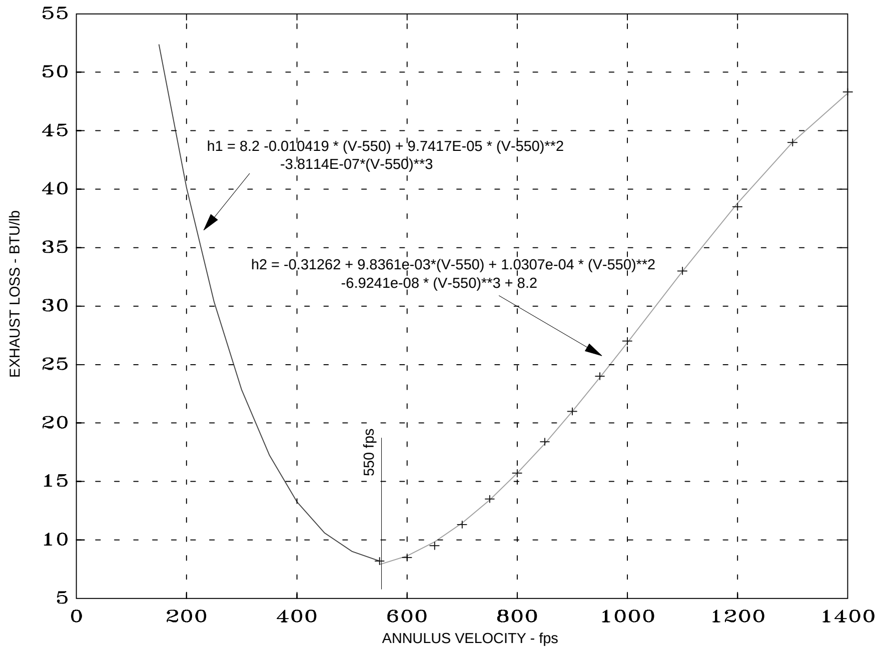
NOMENCLATURE

H	= Lower Heating value of fuel	BTU/lb
EXFL	= Exhaust Flow	lb/h
HTHROTTL	= Enthalpy of Throttle Steam	BTU/lb
HTEP	= Enthalpy of vapor at turbine endpoint	BTU/lb
SPVOL	= Specific Volume of steam at throat	cu.ft/lb
CWFLOW	= Cooling Water Flow rate GPM	
DUTY	= Condenser Duty	MBTU/h
T_{out}	= Mean Cooling Water Outlet Temperature	Deg.F
T_{in}	= Mean Cooling Water Inlet Temperature	Deg.F
SPHT	= Specific Heat of Water at Bulk Temperature	BTU/Deg.F
DENS	= Density of Water at Bulk Temperature	lb/cu.ft
FF	= Fouling Factor	Deg.F/(BTU/sq.ft.h)
TSF	= Tubesheet losses	percent
C	= Carbon content of fuel	lb C/lb.fuel

Fuel	C lb/lb fuel	H BTU/lb	lbs. CO_2 / MBTU loss	lbs. Carbon/ MBTU loss
Bituminous Coal	0.86	13930	238.1	64.987
Fuel Oil	0.863	18558	179.4	48.950
Natural Gas	0.749	25128	115.0	31.376

TABLE VII - CARBON DIOXIDE EMISSIONS - lbs. CO_2 per MBTU Losses



BIG BEND UNIT #4 - EXHAUST LOSS CURVE

FIGURE 3.0