SECTION 5B

Replacement Feedwater Heater Rating Calculations Using PEPSE

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Abstract:

As feedwater heaters age and more tubes are plugged, their maintenance and operating expenses increase and justify replacement. When this occurs, the power producer often provides the original heaters name plate data to heater manufacturers and solicits bids on a new heater. However, other factors need to be considered to ensure that the replacement heater is sized and designed adequately to be capable of many years of useful service. A simple PEPSE feedwater heater sub-model can be a valuable tool to help accomplish this. The feedwater flow and inlet conditions are input to the model and PEPSE iterates to the steam flow required to match the input terminal temperature difference (TTD). The model can simulate unit operation changes since initial service data, overload conditions and even low load conditions. The model's flow calculations can then be utilized in the design process to calculate shell and tube side velocities, vibration potential, and bid evaluation.

Introduction

It is not uncommon to expect that feedwater heaters will require replacement at least once over a unit's service life. In some cases, mistakes in judgment and the absence of root cause analysis compounds the problems as heaters are replaced in-kind. This can result in the continuance of similar failures, and the ultimate need for replacement again. As they are replaced, the heater technical specification must not only address the obvious issues related to such areas as change in tube material, quality control, and references to the current HEI standards but also the mode of operation. An important factor in obtaining a replacement that will last for many years is to specify the correct heater that will be versatile enough to handle not only the normal base load operation, but to also handle higher loads, higher heat inputs, and other modes of operation reasonably expected. The replacement heater specification must define the full range of projected load impositions to allow the Manufacturer to consider and bias his internal layouts and physical geometries to accommodate them safely and conservatively. A poorly specified heater may only last 5 years or less, while a well specified heater may last as long as 50 years. Construction and quality play a large part in heater durability but if the specification is not correct and/or complete, the manufacturer is not given all the data necessary to compensate in his replacement design and a heater that meets all the limited specifications may still only last a short period of time before expensive tube leaks begin to occur.

The factors important to a heater design are:

- Feedwater flow – If the units rating has been increased since initial service date, the feedwater flow was most likely increased to correspond to the higher rating. As the unit ages the efficiency may reduce which will also tend to increase the feedwater flow for the same load point. Increase in feedwater flow obviously increases inlet tube velocities. The various tube materials utilized have specific maximum tube side velocities identified by the HEI guidelines. They should not be exceeded and should have enough conservatism to allow some margin for future plugging.

- Steam flow – If the feedwater flow increases, the required steam flow also increases. In addition, if the shell pressure or feedwater inlet temperature has changed, the steam flow will correspondingly change.
- Shell side velocities – Depending on the baffle design and spacing, shell side velocities within the respective DSH and DC zones of HP FWHs may be higher than the original and excessive tube vibration may result.
- Performance –Sometimes the terminal temperature difference(TTD) or drain cooler approach(DCA) temperatures are modified for the replacement heater. This will not only affect the steam flow required for the replacement heater but also for the heaters upstream and downstream in the cycle. Therefore a modification in the performance of one heater should be verified by checking the performance of the other heaters in the system.
- HP 3-zone FWHs, particularly those subjected to cyclic operation must be checked for a wet-wall condition. This can be a damaging situation as steam condenses before it exits the DSH zone.
- Changes in tube material for improved corrosion resistance usually relates to a decrease in thermal conductivity as per the replacement material and demands larger heaters with more tube surface, resulting in considerations for physical external limitations at the respective heater locations in the Plant.

The PEPSE model can be a valuable tool in specifying the new heater. A variety of cases can be executed in order to help determine the optimal design and its effects on the entire heater system. Constructing the performance mode model for a single heater is quick and relatively easy. The feedwater and drain inlet conditions (temperature, pressure and flow), the heater shell pressure, and the required TTD and DCA are inputs to the model which then calculates the required steam flow and the heater outlet conditions. Using this tool, the initial design, current full load, future full load, overloads and any other pertinent analysis may be accomplished.

A case study using PEPSE to analyze the requirements of a heater system is discussed next.

Analysis

The unit used in the analysis is an 850 MW turbine with dual boilers capable of independent operation. The unit has three stages of high pressure feedwater heaters for each boiler. The first two heaters in each string are suffering from stress corrosion cracking. The tube leakage incident frequency is increasing and eddy current results and tube samples have indicated the heaters are near end of life.

The first step is to construct a PEPSE performance mode model for each heater and then combine the individual heater models into the system model.

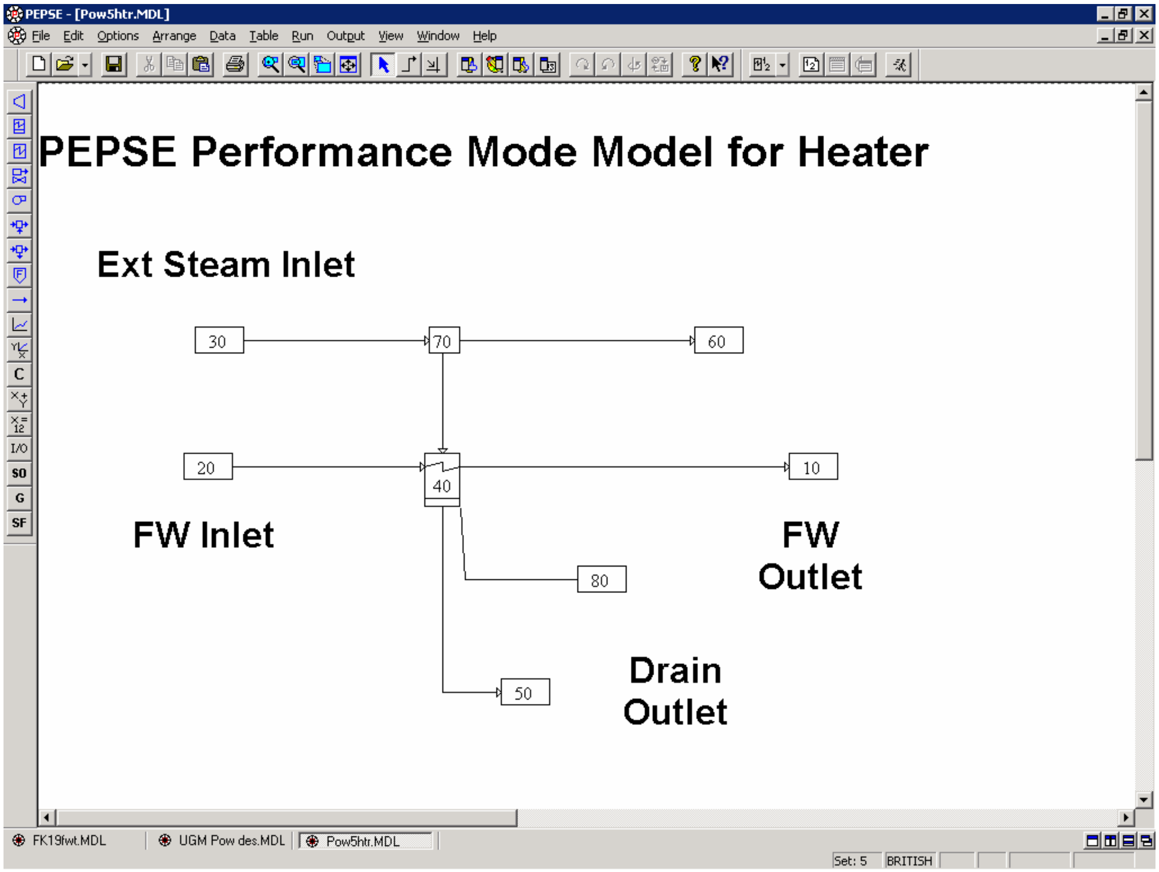
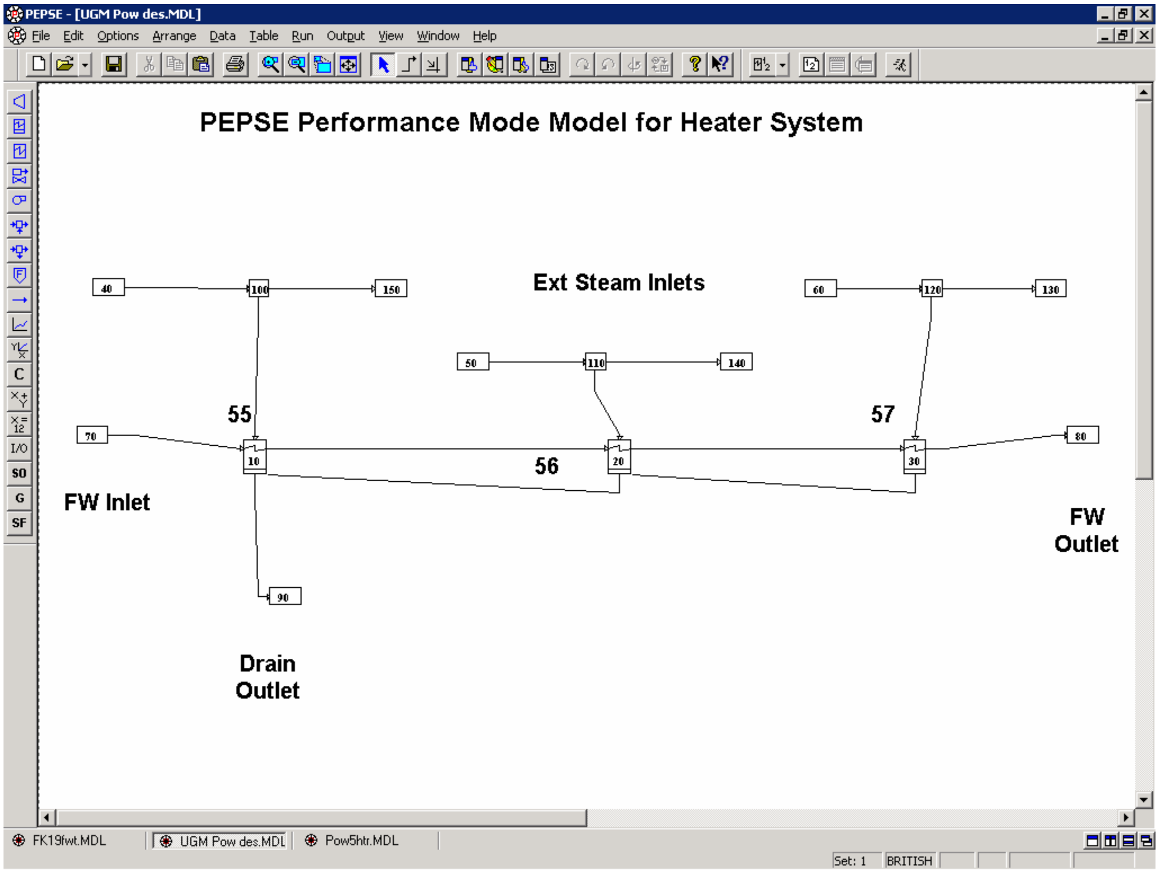


Figure 1- Base case model of single heater

The heater model like that shown in Figure 1 is constructed for each heater. The extraction steam originates from an infinite source(#30) and is regulated by a demand splitter(#70) and the steam sent to the heater(#40). The feedwater inlet originates from component #20 and exits the heater at component #10. The condensate leaving the heater exits through the drain outlet via component #50. The drain inlet originates from component #80.

The feedwater inlet temperature, pressure and flow is input into component #20. The drain inlet temp, pressure and flow is input to component #80. The heater performance data such as TTD and DCA is input into Component #40 and the model is executed. The important output is the steam flow required and the total flow to the lower heater (if applicable).

The individual heaters can be input into the system submodel where all the effects on modifying the individual performance system can be analyzed.



The next step in the modeling analysis is to utilize the system model to compare the design data to actual operating data. The unit is now rated at 850 MW and is not as efficient as it was when new. The revised feedwater flow to each boiler is now 3000 klbh which represents a 4% increase over the existing heaters nameplate rating. The heat duty on the #5 heater has increased because the steam/air preheating utilized previously is no longer used and the turbine is slightly less efficient. This has increased the #5 heaters extraction pressure and consequently increased the temperature rise and corresponding heat duty. This has caused the required steam flow to increase by 34%. The increase in the #5 heat duty has decreased the #6 feedwater temperature rise and heat duty by about 15%. The data is presented in table 1 and 2 as the new full load design point with 2 boilers in service..

Now that we have determined the base case design performance, it is very important to consider potential problems that may result due to abnormal operation. This could include operating with a depressed feedwater inlet temperature which will raise the required extraction steam flow and cause a corresponding increase in the shell side steam and condensate velocities in the desuperheating and drain cooling zones. Since the number 5 heater is the first heater above the dearator (DC heater), the overload condition chosen is a 20 deg F reduction in feedwater inlet temperature. The PEPSE model results for this case are also presented in table 1 as the thermal overload case with 2 boilers in service. The #6 heater is also analyzed with a 20 deg F reduction in inlet temperature in order to simulate a derated condition (such as an internal feedwater bypass) in the #5. This data was input to the PEPSE model and executed with the results presented in table 2.

Note that other potential cases for overload were discussed and considered. One method considered was to operate the number 6 with the #5 out of service. This was judged to incorporate more margin than realistically necessary because based on the system feedwater piping, the #5 cannot be removed from service by itself. The whole string must be removed. Therefore it was decided that a realistic overload condition was to consider the #5 operating in a derated condition instead of being taken out of service. These overload states for both heaters represent about 40% more steam flow than the re-rated design value for No. 5 and about 50% more for the No. 6. As a heater ages and more tubes are plugged, proactive measures to lengthen the life are taken. These could include drilling a hole in the pass partition plate to reduce the feedwater velocity and overall heat duty in the heater. Although this helps the aging heater, it can overburden the downstream heater if it is not designed for the increased heat duty and shell-side velocities due to decreased feedwater inlet temperature. Therefore, although it may not always be necessary to size a heater to operate safely without the upstream heater in service, it may be useful to build some conservatism into the design in case the heat duty does increase. Note that the heater is not expected to maintain the design TTD and DCA in the overload condition. The only requirement guaranteed is the full load with design flows. However, sizing for a realistic overload costs slightly more but can increase the service life significantly. It is important to note however that compensation for abnormal overload conditions <u>does not</u> require an increase in tube surface, with a proportional increase in heater purchase price. This is a commonly misunderstood point. This defining information provided in the performance schedules of the technical specification simply requires the manufacturer to check his mechanical design, physical internal geometries, and construction issues to insure that the replacement heater can safely and conservatively withstand the projected full range of load imposition and provide continuous, undamaging operation all the way up to the worst abnormal potentials. The model results are presented in Tables 1 and 2 where they are labeled Reference Case.

Another important realistic operation scenario that was modeled is the operation with only one of the two dual boilers in service. Since the unit is at half load, the DC heater operating pressure is lower which corresponds to a reduced feedwater inlet temperature. The shell operating pressure of the heaters is also lower, corresponding to less heating. Normally the feedwater flow and required extraction steam flow would be reduced at half load and an overload potential would not need to be considered. However, with the dual boiler configuration, the heating is not significantly less because the feedwater flow for the boiler in service is the same full load flow as with both boilers operating, 3000 klbh.. Even though the unit is at half load, the one boiler full load feedwater flow causes the PEPSE model to predict steam flow values close to the full load amounts with both boilers in service. Note that this is not a common relationship in most plants unless they have dual independent boilers. However this serves as an excellent example of analyzing the specific operating requirements of the heater system and specifying the replacement heater requirements accordingly.

Note that in the specification analysis, the objective is to consider all modes of operation reasonably expected and predict how long the heater will be operating in that mode. The PEPSE analysis results listed in tables 1 and 2 for the overloads is based on an estimate of the TTD and DCA that we predict based on operation experience in that mode. We then input that predicted TTD and DCA into the model and iterate to the required steam flow. The data that is calculated by PEPSE for the overload states is based on these assumptions but provides a reasonable flow estimate with which to calculate the shell and tube side velocities and other parameters required to ensure the heater life is not significantly shortened by operating away from the base case. The heater manufacturer chosen may be able to refine these calculations with his confidential heater rating programs.

Heater #5 Design Ratings and Overloads		Original Design Data	New Design Point 2 Boilers in service	Reference Case Max FW Flow 1 Boiler in service	Thermal Overload* 2 Boiler in service Fw inlet Temp 20 F Low
Gross Load MW	MW	850	850	390	850
Feedwater Flow	lb/hr	2,883,134	3,000,000	3,000,000	3,000,000
Feedwater pressure	psia	3215	3215	2800	3215
Feedwater Inlet Temp	deg F	322.8	325.7	271.6	305.7
Feedwater outlet temp	deg F	359.7	372.5	319**	371.5**
Steam Flow	lb/hr	87,693	117,639	112629**	161689**
Steam Press	psia	162.2	189.9	94.5	189.9
Steam Temp	deg F	724.5	728.0	727.8	728
Saturation temp	deg F	364.6	377.5	323.8	377.5
Drain Inlet Flow	lb/hr	361,060	349,316	303,758**	352,666**
Drain Inlet Temp	deg F	369.6	382.5	328.8**	382.5**
Drain Inlet Enthalpy	Btu/lb	342.6	356.3	299.6**	356.3**
Drain Outlet Flow	lb/hr	448,753	466,955	416307**	514355**
Drain Outlet Temp	deg F	332.8	335.7	281.6**	316.7**
TTD	deg F	4.9	5.0	5**	6**
DCA	deg F	10	10.0	10**	11**
FW Pressure Drop	psi	12.9	TBP	TBP	TBP
DSH Pressure drop	psi	1.8	TBP	TBP	TBP
DC pressure drop	psi	2.9	TBP	TBP	TBP

TBP = To be predicted by heater manufacturer
* Thermal overload data is approximated as 140% of design steam flow entering heater
** assumed data

Table 1 Heater Number 5 PEPSE Model Results

Heater #6 Design Ratings and Overloads		Original Design Data	New Design Point 2 Boilers in service Actual Plant Data	Reference Case Max FW Flow 1 Boiler in service	Thermal Overload* 2 Boiler in service Fw Inlet Temp 20 F Low
Gross Load MW	MW	850	850	390	850
Feedwater Flow	lb/hr	2,883,134	3,000,000	3,000,000	3,000,000
Feedwater pressure	psia	3215	3215	2800	3215
Feedwater Inlet Temp	deg F	359.7	372.5	318.8	352.5
Feedwater outlet temp	deg F	406.6	410.7	351.6**	409.0**
Steam Flow	lb/hr	116,874	99,750	80,073**	149,484**
Steam Press	psia	266.3	278.9	137.5	278.9
Steam Temp	deg F	844.4	869.4	878.7	869.4
Saturation temp	deg F	406.6	410.7	351.6	410.7
Drain Inlet Flow	lb/hr	244,186	249,565	223,685**	255,128**
Drain Inlet Temp	deg F	416.6	420.7	361.6**	419**
Drain Inlet Enthalpy	Btu/lb	393.1	397.9	334.3**	396**
Drain Outlet Flow	lb/hr	361,060	349,316	303,758**	404,613**
Drain Outlet Temp	deg F	369.7	382.5	328.8**	377.5**
TTD	deg F	0	0.0	0**	2.0**
DCA	deg F	10	10.0	10**	10**
FW Pressure Drop	psi	14.5	TBP	TBP	TBP
DSH Pressure drop	psi	2.3	TBP	TBP	TBP
DC pressure drop	psi	2.9	TBP	TBP	TBP

TBP = To be predicted by heater manufacturer
* Thermal overload data is approximated as 150% of design steam flow entering heater
** assumed data

Table 2 Heater Number 6 PEPSE Model results

Discussion

The heater vendors solicited are instructed to submit their bids based on the performance schedules presented in Tables 1 and 2. The bids are then technically evaluated for specification compliance at the new design point and the worst overload(s) conditions identified. Information requested in the proposals will allow checks and calculations of important parameters and compares them to the HEI and EPRI recommended guidelines referenced in the specification. Some of the major areas checked are as follows:

- Baffle details – The baffle types, configurations, percent cuts, and spacing selected for the desuperheating and drain cooling zones impact the heat transfer capability and pressure drop of the respective zone and the details as such are utilized to calculate shell-side mass flow rates and linear velocities. The baffle layouts define the available net flow areas as the steam or condensate flows into each zone, across the tube field array within the specific baffle spacing, and longitudinally through baffle window openings.

- Tube Pitch – The tube pitch is the center to center distance between the tubes and obviously impacts the net free flow area available for the cross mass flow rates calculated. Increasing the pitch is one way heater rating engineers can lower the shell-side velocities within the desuperheating and drain cooling zones. By limiting the tube side pressure drop, (approx. 10 psi max.), and limiting zone cross flow velocities at the overload parameters in the specification, usually forces the vendor to open the tube pitch as replacement heaters have a tendency to be proposed as shorter and fatter as opposed to longer and skinnier.
- Maximum unsupported tube span – The two main parameters affecting the vibration potential are the cross velocities and the maximum unsupported tube spans. If excessive, the tubes may actually collide or rub at the midspan and cause excessive wear and baffle fretting. Tube leaks can occur very quickly under these dynamic impositions.
- Vibration potential – Based on the above parameters, there are industry accepted empirical calculations that are utilized, authored by Sebald, Chen and Connors and others, to determine the potential for damaging vibration. Important parameters such as tube static deflection, fluid elastic whirling critical cross flow velocity ratios, and natural and vortex shedding frequency comparisons are some of the major calculation checks performed at design and checked at the worst overload conditions to assure non-damaging potentials across the full load range to be imposed.
- Shell inner diameter – Shell sizing must be based on the overload potentials to insure that conservative flow areas between the bundle outer tube limit and the shell ID exists that promote longitudinal distribution of the steam and lower penetrating velocities across the bundle as it condenses. Shell nozzle sizing may also increase based on the overload specification, and the shell sizing is also a function of those nozzle sizes. As the inner diameter increases, the shell-side velocities decrease. A shorter, larger diameter heater is generally more optimal than a longer smaller diameter heater.
- Desuperheating zone “wet wall” margin – The steam exiting the desuperheating zone must not be too close to its condensing temperature. Under all modes of operation, one must insure that the tube wall metal temperature at the exit of the DSH zone is at least 1 deg higher than the shell side saturation temperature corresponding to the pressure at that location. If not, localized condensation can develop, causing entrained water droplet impingement damage on tubes, supports and baffles.

Based on these parameters and the results of calculations, the heater bids may be compared on an equal basis for specification compliance and the requested heater design durability.

In addition to the heater mechanical integrity checks, the heat transfer of each zone should be checked and verified that it produces the required energy balance and that the respective heat transfer coefficients are reasonable. Comparing the heat transfer coefficients for all manufacturers together and using PEPSE to verify the energy balance through the simplified design mode is a useful verification. Utilizing the PEPSE JW method to calculate and compare all heater designs is also a useful option.

Conclusion

This paper has illustrated methodology to utilize PEPSE to model various heater operating scenarios representing the full range of projected operation. The key outputs are the steam and drain flows, which are utilized to calculate the internal shell side velocities in the desuperheating and drain cooling zones. If these velocities are higher than EPRI or HEI guidelines, the heater life may be reduced because tube vibration and other failure mechanisms may occur. The current conditions should be reflected as the new design flows and are provided to the heater manufacturer and utilized as the performance guarantee. However, the identification of the full range of operation and the resultant overload flows and velocities are critical because the manufacturer accounts for these predicted flows in checking the adequacy of his heater design to ensure a longer heater life. By limiting steam and fluid velocities within the various internal zones of the heater and from results of failure analysis of the root causes of the failures of the heaters been replaced, the specifying engineer has fulfilled his responsibility in providing the most important information necessary for the Vendor to provide a heater capable of longer reliable life.

Once the new heater is delivered and installed, a performance test should be conducted utilizing ASME PTC 12.1 as a guideline.

References

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