THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 E. 47 St., New York, N.Y. 10017

91-JPGC-NE-5

Heat Exchanger Analysis in Response to USNRC Generic Letter 89-13

Kim Iulianetti

ABSTRACT

Discovery of excessive fouling in equipment in the service water systems at several nuclear power plants in recent years led to the issuance of Generic Letter 89-13 by the USNRC. This letter requires all holders of Nuclear Operating Licenses to regularly evaluate the condition of the emergency service water systems to assure that they can remove the heat loads specified for a postulated event. The components in a service water systems include piping, valves, pumps, reservoirs, and heat exchangers. The focus of this paper is on some key considerations in the evaluation of the thermal performance of heat exchangers.

INTRODUCTION

Nuclear power plants are designed with a service water system which acts as the ultimate heat sink for the heat loads which are generated throughout the plant. The components in the service water system are designed to remove the heat loads which are expected under a variety of accident scenarios. In most plants, the source of service water is a nearby body of water such as a lake, river, bay, or ocean, collectively referred to as the heat sink. Water which is pumped in from the heat sink is strained and filtered before it enters the heat exchangers. However, many particulate forms still get through such as sand and biological fouling.

These particles become deposited on the walls of the piping and the system components. Over time, they build up in the heat exchanger tubes, and coagulate into a solid layer known as a "fouling layer". The fouling layer adds a resistance to the tube wall which degrades the thermal performance of the heat exchanger. Fouling can also exist on the shell side of a heat exchanger, although it is most common to have the degradation on the tube side (because the service water is typically assigned to the tube side).

There are other conditions which result in reduced thermal performance such as tube plugging, tube leakage, and flow bypass. Additionally, in the reactor building containment fan coils, it is not uncommon to have a build-up of boron on the tube fins which will also degrade the thermal performance.

The degradation of the thermal performance of heat exchangers in the service water system is an ongoing problem in any nuclear plant with consequences to plant safety. This inspired the Nuclear Regulatory Commission to issue Generic Letter 89-13. The letter requires all nuclear plants to assure the safe and adequate operation of all components in the service water system. This paper will address some key issues which should be considered when implementing the Generic Letter requirements for heat exchangers.

One objective of Generic Letter 89-13 is to ensure that each plant can provide adequate assurance that the heat exchangers can function under all postulated conditions. The ideal way to provide such assurance would be to test the heat exchanger under design (accident) conditions and demonstrate that the heat load removed by the exchanger meets the minimum design requirements. However, it is typically impossible or impractical to establish accident conditions for test purposes.

Presented at the International Power Generation Conference
October 6-10, 1991—San Diego, CA

The typical plant can only test at off-design conditions and then "extrapolate" the results to the design conditions. Regardless of instrumentation quality and evaluation method accuracy, the final results will always contain some degree of uncertainty. The objective of this paper is to highlight evaluation methods which will reduce the uncertainties. Subsequent sections of this paper will address some key considerations in acquiring test data with the primary focus on evaluation of data.

TEST MEASUREMENTS

In order to evaluate the thermal performance capacity of a heat exchanger in its day-to-day operation, the power plant engineer must measure the pertinent thermal performance parameters such as inlet and outlet temperatures, flow rates, pressures, and air relative humidity where applicable.

As a first step, the instrumentation should be evaluated to assure that the measurements will be sufficiently accurate for the expected test conditions. For example, the instrumentation used for measuring temperatures (such as a thermocouple, RTD, etc.) should be selected such that the measurement error is significantly less than the total temperature difference. This would ensure that the error introduced by the temperature measurements would be relatively small. Of course, one should always make certain that all instrumentation is properly calibrated in an attempt to minimize errors.

Other concerns which should be considered when gathering test data are summarized below:

(1) The difference between the inlet and outlet temperatures on each side should be sufficiently large such that small inaccuracies in temperature measurements will have a relatively insignificant impact.

(2) Flow rates on each side should be high enough to result in turbulent flow, since correlations for calculating heat transfer coefficients are most accurate in the turbulent flow regime.

Reference [1] suggests that in order to limit the error introduced by test measurements to 5%, the measurements should be accurate within the following ranges:

(1) A temperature measurement should be within ± 3% of the <u>smallest</u> measured temperature difference.

(2) A flow measurement should be within ± 3% of the measured value.

(3) Relative humidity should be within ± 2% of the measured value.

(4) Air pressure should be within ± 5% of the measured value.

Another consideration in collection of test data is the location of the instrumentation relative to the heat exchanger. For example, the exiting temperature should be measured at a location where the fluid is adequately mixed to ensure that the temperature reading is not a localized data point. For a more detailed discussion of instrumentation location, refer to Reference [1].

Evaluation of Data

The performance capacity of a heat exchanger under a certain set of conditions can be determined if the performance characteristics of that same exchanger are known for any other conditions. Using the measured inlet and outlet temperatures, flow rates, pressures, and relative humidity (for air coolers), in addition to the detailed geometric characteristics of the exchanger, the component parts of the overall heat transfer coefficient can be calculated. Additionally, one can calculate the LMTD, the LMTD correction factor, the surface area, and the total heat duty.

By equating these parameters, one can calculate the actual overall heat transfer coefficient, U, from the classical heat transfer formula:

$$U = \frac{Q}{A\,\Delta T_m}$$

where: U is the overall heat transfer coefficient, Btu/hr ft^2 °F

Q is the total heat duty Btu/hr

A is the effective heat transfer surface area, ft^2

ΔT_m is the corrected Log Mean Temperature Difference, °F

The inverse of the actual overall heat transfer coefficient is the total thermal resistance of the heat exchanger. By taking the total thermal resistance and subtracting each of the individually calculated thermal resistances (such as the tube metal resistance, the inverse of the shell side heat transfer coefficient, and the inverse of the tube side heat transfer coefficient), the remaining resistance can be attributed to fouling.

This resistance is what is most commonly referred to as the fouling resistance or fouling factor.

Since the overall heat transfer coefficient is typically referred to the outside surface, the fouling resistance which is calculated by this method will also be referred to the outside surface. However, typical power plant heat exchangers are designed for high tube side fouling resistances. To put the fouling resistance into the tube side perspective, it should be multiplied by the ratio of the inside surface area to the outside surface area.

Once this fouling factor is determined, it can be used in the same equations to calculate the performance parameters (such as outlet temperatures and heat duty) for a different set of inlet conditions.

Tube side Heat Transfer Coefficient. There are many available references for equations to calculate the tube side heat transfer coefficient. In order to most accurately calculate the actual tube side coefficient, the Reynold's number should first be calculated. Most sources would agree that if the Reynold's number is 4,000 or higher, the flow should be in the turbulent regime, and that below 2,000, the flow is probably in the laminar flow regime. For values of Reynold's number which falls between the two, it can be considered as a transitional regime, where the flow behavior is most unpredictable.

Of course, since the turbulent flow correlations are the most reliable, it is desirable to test under these conditions. Many times, however, it is impossible to produce the flow rates which would result in turbulent flow and still maintain sufficiently high temperature differences for a meaningful evaluation. Equations which are most accurate for the actual flow regime should be used when calculating the tube side heat transfer coefficient.

Shell Side Heat Transfer Coefficient. The calculation of the shell side heat transfer coefficient is probably the most critical parameter to be considered when evaluating the performance of a heat exchanger. There are many geometric parameters which should be considered when evaluating the shell side performance characteristics.

In the following sections, some key considerations in the evaluation of the shell side heat transfer coefficient will be discussed for shell-and-tube heat exchangers and air dehumidifying heat exchanger. It will be shown that for these two types of heat exchangers, the approach to this calculation is significantly different.

Shell-and-Tube Single-Phase Heat Exchanger. There are many approaches to calculating the shell side performance of a shell-and-tube exchanger. Some are very simplified methods which allow an engineer to quickly calculate an approximate value. Other methods, such as design based methods, account for the non-ideal flow paths which exist in every shell-and-tube exchanger.

In a TEMA type "E" shell (one shell pass) with one tube pass, the flow can be either cocurrent or countercurrent depending upon the arrangement of the inlet and outlet nozzles. Ideally, the cross baffles in a shell-and-tube exchanger would direct the shell side flow such that it is also in cross flow (see Figure 1).

However, during fabrication and assembly of shell-and-tube exchangers, the only reasonable and cost-effective way to design an exchanger, is to ensure that the baffle hole is sufficiently larger than the tube outer diameter such that the tubes can easily be laced through the baffle holes. Once the bundle is assembled, it can then be inserted into the shell. To make the insertion physically possible and practical, the shell diameter must be slightly larger than the baffles.

These clearances open up other areas within the shell where the shell side fluid can flow. The flow will now be split up into "flow streams" such that most of it follows the ideal path, but some of the fluid flows through the space between the tube and baffle holes or between the shell and baffle, thus bypassing the tube bundle completely (See Figure 2).

Referring to Figure 2, streams A through F are described below:

Stream	Description
A	Leakage between tube and baffle hole
B	Main effective cross flow stream
C	Main bypass stream (between bundle and shell)
E	Leakage between baffle and shell
F	Bypass stream in flow channels due to tubes omitted for pass partition plates

As the tube bundle is bypassed, the flow which would normally be available for heat transfer does not exchange heat with the tube side fluid at all. Thus, the heat exchanger can not perform as well as it could if the clearances did not exist. If the clearances are not properly considered in the thermal analysis, the decreased performance will be interpreted as excessive fouling. Additionally, the effects that the clearances will have on the performance will vary with the shell side flow rate.

Other considerations in the evaluation of shell side performance are the effects of the bundle layout, the baffle arrangements, and whether sealing strips are used to redirect the shell side flow.

Finned Tube Air Coolers. In air dehumidifying coils, there are several combined parameters which make up the heat transfer coefficient. As the air flows through the space between the fins, some heat is transferred by convection. The air is cooled and may eventually begin to condense on the fins, developing a condensate layer through which heat is transferred by conduction. Finally, the efficiency of the fins must be evaluated. The fin efficiency is used to correct the total shell side heat transfer coefficient.

In order to evaluate the convection coefficient, the shell side geometry must be considered. The fin geometry, tube diameter, and tube pitch must be known to establish the total cross sectional flow area. In addition, the fin shape will have an effect on the heat transfer coefficient. For example, a corrugated fin will increase the turbulence and, therefore, increase the heat transfer capacity when compared to a smooth (or flat) fin.

The shape and pitch of the fins combined with the air flow velocity will have an effect on the thickness of the condensate layer which will build up between the fins. If the air velocity is very slow, the condensate will tend to drip from the tubes by force of gravity. However, if the air velocity is sufficiently high, the shear forces will effectively blow much of the condensate off of the tube surface. The fin geometry will dictate the magnitude of the tendency to retain the fluid on the tube surface.

The fin efficiency will depend upon the style of the fins, the method of attachment of the fins to the tubes, and the operating conditions. The fin efficiency represents a quantitative assessment of how well the fin, as an extended surface of the tube, can convect heat.

Overall, the calculation of the shell side heat transfer coefficient for an air cooler would not be too difficult if either all of the heat transfer was sensible, or if the moisture began to condense from the air on the first row of tubes. In such a case, it would be easier to justify the assumption that the heat transfer parameters could be averaged over the entire coil.

However, it is most likely that the air may enter the coil with a rather high relative humidity (such as 80% or more), and the first few rows of tubes that it crosses would show only sensible heat transfer. Once the air stream approaches the dewpoint temperature, the cold tube and fin surfaces will cause the vapor to condense. Thus, it is considerably more accurate to model the coil by evaluating each row of tubes individually.

Sometimes, an individual row analysis is still insufficient. Due to the characteristics of the fan, the duct geometry, localized air side fouling, or possibly an obstruction, the air flow may not be evenly distributed along the length of the coil. The air flow distribution can significantly affect the evaluation of heat exchanger capacity since flow maldistribution results in decreased performance.

In cases where there is an uneven distribution of the air flow due to the geometric design of the unit, the performance will be affected differently as the flow rate and air moisture content vary.

CONCLUSIONS

Evaluation of thermal performance of heat exchangers in nuclear power plants requires a level of accuracy which may sometimes be difficult to achieve. When taking test measurements at less than ideal conditions, it is important to consider the following details:

(a) accuracy in the selected instrumentation,
(b) use flow rates which result in turbulent flow,
(c) location of instrumentation, and
(d) proper calibration of instrumentation.

If the test data is reasonably accurate, then theoretical methods can be implemented to evaluate the condition and thermal capacity of the heat exchanger. The theoretical method used to evaluate the data should consider non-idealistic situations such as laminar vs. turbulent flow regimes, flow bypass, and tube plugging. The baffle type configuration (type, cut and spacing) will have a significant effect on the heat transfer in shell-and-tube heat exchangers. All of these geometric and other constraints should be considered when evaluating the performance of heat exchangers.

REFERENCES

1. Inservice Performance Testing of Heat Exchangers in LWR Plants, ASME Operations and Maintenance Part 21, Draft Issue, November 28, 1990, (N. Stambaugh, Chairman).
2. Theory and Practice of Heat Exchanger Design, by K. P. Singh, Arcturus Publishers (in Print).

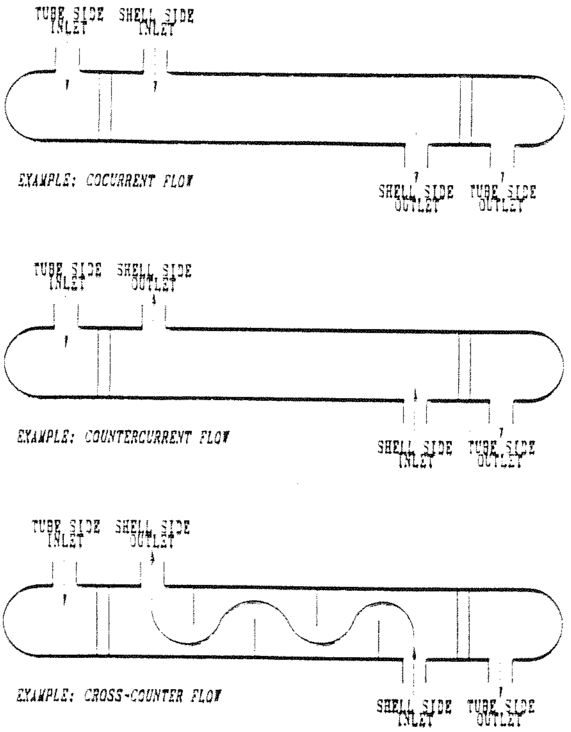

figure 1: FLOW ARRANGEMENTS

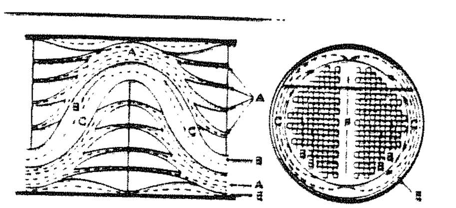

Figure 2: Shell Side Flow Patterns in Baffled Heat Exchangers