Replacement of Steam Generators at Ulchin Nuclear Power Plant Units 1 and 2

Jeong Hoon Kim [a], Yoon Jae Choe [a], Kyong In Ju [a], Kyong Min Ko [a], Byung Jin Lee [a], Il Kon Kim [a]
Yook Geun Soo [b]
[a] *Korea Power Engineering Company, 150 Deokjin-dong,Yuseong-gu, Daejeon, 305-353*
[b] *Korea Hydro and Nuclear Power Company*
[*] *Corresponding author: jhkim11@kopec.co.kr*

1. Introduction

The steam generators operating at Ulchin nuclear power plant units 1 and 2 (UCN 1&2) are the Framatome-designed Model 51B. They have been in service since 1988, but the degradation due to primary water stress corrosion cracking is estimated to weaken their integrity and increase maintenance cost. In response to this concern, the utility, Korea Hydraulic and Nuclear Power Co., is planning to replace the original steam generator (OSG) with new one in September 2011 for unit 1 and in February 2012 for unit 2, respectively.

We have performed preliminary works which include preparing technical specification for bidders and evaluating proposals by a bidder to support the utility in selecting proposed replacement steam generator (RSG). This paper presents an overview of thermal sizing method, case studies, and technical evaluations on proposed RSG in view of thermal performance.

2. Thermal Sizing Method

The steam generator thermal sizing analysis is to find average heated tube length and primary side pressure drop for given conditions: thermal power, primary side mass flow rate, primary temperatures, steam pressure, number of tubes, tube outside diameter, tube wall thickness, tube material, and tube pitch.

The thermal sizing method is based on the equation of heat transfer across the tube wall:

$$Q = U_o \cdot A \cdot \Delta T_m \tag{1}$$

Rearranging Eq. (1), the heat transfer area becomes:

$$A = \frac{Q}{U_o \cdot \Delta T_m} \tag{2}$$

The log mean temperature difference is calculated directly from given primary and secondary temperatures.

$$\Delta T_m = \frac{T_{hot} - T_{cold}}{\ln\left(\dfrac{T_{hot} - T_{sat}}{T_{cold} - T_{sat}}\right)} \tag{3}$$

Meanwhile, the overall heat transfer coefficient is the inverse of a sum of various thermal resistances:

$$U_o = \frac{1}{R_i + R_w + R_o + R_f} \tag{4}$$

An inside film resistance is determined independently by primary fluid properties and operating conditions. An outside film resistance and a tube wall resistance vary according to heat flux, so their calculations are iterative. Because the tube surface area is not still fixed, initial heat flux is guessed and corrected at next step. The fouling factor has been taken into account. It depends on vendor's design practice.

If all resistances are known, the overall heat transfer coefficient can be obtained from Eq. (4). Then the heat flux through tube wall is determined by:

$$\phi_{cal} = U_o \cdot \Delta T_m \tag{5}$$

The difference between guessed and calculated heat fluxes is checked if it falls within a given tolerance or not. If not, the calculation is repeated again. If the difference converges within a user-specified allowance, the calculation is terminated. Now the heat transfer area can be obtained from Eq. (2). The average heated tube length and the tube bundle diameter are also calculated from tube surface area and geometric data.

The steam generator primary side pressure drop for average tube length is calculated with classical methods considering friction and form losses.

3. Evaluations on Technical Proposals

The utility required the RSG which performance at design conditions listed in Table 1 is identical to the OSG. The RSG also needs to assure no interference with upper lateral supports, inspection ports and access to cone-to-shell weld for in-service inspection. Constraints on overall dimensions will give us a limited choice of average tube length and number of tubes.

Table 1 RSG Full Load Operating Conditions [1]

Parameter	Value
NSSS thermal power, MWt	2785
Primary thermal design flow, lb/hr	34×10^6
Primary inlet temperature, °F	614.7
Primary outlet temperature, °F	545.5
Steam exit pressure, 0% plugging, psia	837
Primary side pressure drop, psi	36

We have performed preliminary case studies using method in Section 2. Case studies, as described in first paragraph of Section 2, are to search for a tube bundle geometry which meets the utility requirements in terms of steam pressure and primary side pressure drop. Two types of steam generators were chosen as design options: 0.875 inch tubes and 0.750 inch tubes. Both are arranged with a triangular array so as to house increased number of tubes, and their material is alloy 690 that have a strong resistance to corrosion. Two options are based on common ones in steam generator designs.

The ideal design will be such that its average heated tube length and tube bundle diameter are close to the OSG along with good performance. But it has been found out that the choice of design options would be very limited since no changes to primary flow and temperatures, secondary steam pressure, and even steam generator envelope are not allowed.

The results show two cases all need a larger tube surface area than the OSG. For thicker tubes, it may increase up to around 20% compared with the OSG. The principal contributors for this are as follows: The most important is the use of alloy 690 which thermal conductivity is lower than alloy 600. Next, the increased number of tubes results in decreased primary flow rate per tube. This increases inside film thermal resistance. Additionally, larger tube surface area leads to lower heat flux, and then a decrease in boiling heat transfer. These effects end up with a decreased overall heat transfer coefficient. Eq. (2) indicates that this needs a larger heat transfer area for the given heat load and temperature difference between the primary and secondary fluids.

As mentioned earlier, the use of a triangular pitch allows increased number of tubes without enlarging tube bundle diameter. But it is inevitable that the tube bundle is longer or shorter than the OSG depending on tube outside diameter. Specifically, for smaller tubes some dimension changes are necessary to adjust a new tube bundle to the existing envelope listed in Table 2. It has to be confirmed that new tube bundle geometry has no impacts on its interfaces with supports, connections and accesses.

Table 2 OSG Tube Bundle Geometry [1]

Parameter	Value
Tube surface area, ft^2	50,579
Number of tubes	34×10^6
Tube outside diameter, in.	0.750
Tube wall thickness, in.	0.050
Tube material	Alloy 690
Average heated tube length, ft	66.3
Type of tube array	Square
Pitch of tube array, in.	1.28
Tube bundle height, in.	416.8
Wrapper inside diameter, in.	123.5
Overall height, in.	812.9
Overall diameter, in	175.9

The steam generator with 0.875 tubes and triangular array has been proposed by a bidder first [2]. Its envelope is very similar to the OSG, which means no change to the existing envelope. The change to triangular pitch allows increase of number of tubes and heat transfer surface within the existing envelope, and also compensates for lower thermal conductivity of alloy 690. But it needs much larger tube surface area for the given performance. This means high cost.

The second proposal used thinner and smaller tubes compared with the previous proposal [3]. Thinner tubes have better heat transfer capacity than thicker tubes. This allows a smaller heat transfer area, but causes a higher pressure drop, which leads to a shorter tube bundle. Therefore, new steam generator envelope needs some adjustments to cone location to facilitate tube bundle assembly. But no interferences due to a lower elevation of cone were confirmed.

4. Conclusions

For replacement of UCN1&2 steam generators, we have performed technical reviews on two preliminary proposals: thicker and thinner tube models. Specifically, the evaluations were mainly focused on their thermal performance and external envelope. Both cases provide equal thermal performance at specified design conditions compared with the OSG. No interference with existing supports and connections was confirmed. These activities have been carried out as part of technical support works for the utility to make a decision.

NOMENCLATURES

Q	Heat transfer rate, BTU/hr
U_o	Overall heat transfer coefficient, hr-ft^2-°F/ BTU
ΔT_m	Log mean temperature difference, °F
T_{hot}	Primary inlet temperature, °F
T_{cold}	Primary outlet temperature, °F
T_{sat}	Secondary saturation temperature, °F
R_i	Inside film resistance, hr-ft^2-°F/ BTU
R_w	Tube wall resistance, hr-ft^2-°F/ BTU
R_o	Outside film resistance, hr-ft^2-°F/ BTU
R_f	Fouling factor, hr-ft^2-°F/ BTU
ϕ	Heat flux, BTU/ hr-ft^2

REFERENCES

[1] KEPCO, UCN 1&2 FSAR, p. 5.1-8, 5.4-58, May 1996.
[2] DOOSAN/ Westinghouse Electric Co., "1st Preliminary Technical Proposals for UCN 1&2 RSG," March 2008.
[3] DOOSAN/ Westinghouse Electric Co., "2nd Preliminary Technical Proposals for UCN 1&2 RSG," June 2008.