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Preliminary Analysis of Passive Residual Heat Removal System for a Nuclear-powered Ship

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1. Introduction

Recently, a major focus in the nuclear field lies on developing small and medium sized reactors (SMRs). Their purpose is not only restricted to electricity generation but also non-electric applications such as propulsion or desalination. In particular, nuclear propulsion has received considerable attention to date. One of the main reason is that high power source has become important for ships. For examples, in cases of icebreaker or commercial large cargo ship, around 100 MWth ~ 200 MWth of output is required, which is a typical value of that of SMRs. In addition, International Maritime Organization (IMO) requests the reduction of the greenhouse gas emission recently. In line with those trends, the nuclear marine propulsion has gained interest.

In order for the nuclear marine propulsion to be widely applied, safety of the system must be assured. Especially, the ocean condition restricts the outside intervention for mitigating accident cases, which implies that passive safety is essential. Among many passive safety systems, a passive residual heat removal system (PRHRS) is of interest in this study. In the land-based PRHRS, one of the most practical and fully-passive configuration is the system where the initial large decay heat is removed by outside pre-installed water tank, and after all the water evaporates air-cooling tower cools down the heat. This system can effectively remove the decay heat for a long-term without relying on any intervention, but it usually requires a large volume of the air-cooling tower. In the ocean-based PRHRS, the available space for the reactor system is very limited, meaning that the conventional large one is not suitable for this case.

Therefore, a new concept of PRHRS, which combines the air-cooled tower and the seawater-cooling is suggested. The basic idea is as follows. Some amount of decay heat is removed by the air-cooled tower, which is installed at the upper part, inducing the natural circulation flow. Then the seawater-cooling part effectively removes the rest of decay heat as temperature of the seawater is very low, although the flow rate is small. That is, the required size of the air-cooled tower is expected to be decreased because the seawater can reduce the heat removal load of the tower. However, the system characteristic will be different from that of conventional one, since two heat sinks are installed in a single loop. Therefore, the purpose of this study is preliminary designing the system and evaluating the overall size of it. The results can provide a first insight into the new concept of system. This paper describes the characteristic of buoyancy head of the system, air-cooled condenser design, and feasibility of the system.

2. Methods

In order to design the PRHRS, the following methods are used. The schematic diagram of the system is shown in figure 1. H represents the height of the components, and ρ represents the density at the inlet and outlet of the components. It is assumed that the height and location of seawater-cooling part are same as those of steam generator. Additionally, it is assumed that the seawater can remove all the rest of decay heat, and corresponding heat transfer area is calculated. The system design depends on environmental conditions such as ambient air and seawater temperature. Considering the purpose of the system, icebreaker, average temperature of the artic environment is used.

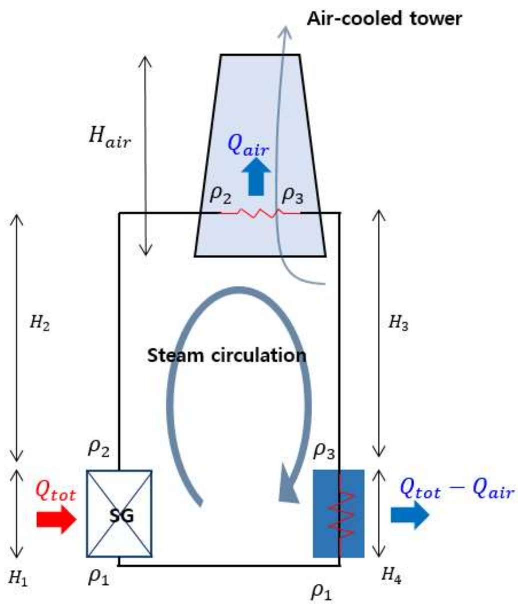

Fig. 1. Schematic diagram of the system

2.1 Buoyancy head

The buoyancy head of the steam in the system is calculated as follows.

$$\Delta P_{buoy} = \oint g\rho(s)\cos\theta\, ds$$

$$= \left[-\frac{\rho_1+\rho_2}{2}H_1 - \rho_2 H_2 + \rho_3 H_3 + \frac{\rho_1+\rho_3}{2}H_4\right]g \quad (1)$$

By utilizing the relationship (2), the buoyancy head is arranged as (3).

$$H_{tot} = H_1 + H_2 = H_3 + H_4 \tag{2}$$

$$\Delta P_{buoy} = \left[\frac{\rho_1-\rho_3}{2}(H_4 - H_1) + (\rho_3 - \rho_2)\left(H_2 + \frac{H_1}{2}\right)\right] g \tag{3}$$

2.2 Steam generator

A significant amount of pressure drop comes from the steam generator part. The information shown in table I is used for the calculation, which is based on SMART [1]. It is assumed that power of 2 MWth (1% of 200 MWth) is removed from the steam generator.

Table I: Design parameters of the steam generator

Parameter	**Value**
Tube outside diameter	17 mm
Tube inside diameter	12 mm
Number of tube	375
Tube length	25 m
Fluid at tube outlet	Saturated steam
System height	3.8 m

2.3 Air-cooled condenser

The air-cooled condenser is configured as horizontal type. For the tubes in the condenser, the L type finned tube was chosen. The configuration is cross-flow, staggered-tube heat exchanger. In order to design the heat exchanger, Log Mean Temperature Difference (LMTD) method is used. The overall heat transfer coefficient is obtained from equation (4).

$$\frac{1}{U_o} = \frac{1}{h_o\eta_W} + \frac{A_o}{A_i}\left(\frac{s}{k} + \frac{1}{h_i}\right) \tag{4}$$

where U, A, h, s, k, η_W is overall heat transfer coefficient, surface area, heat transfer coefficient, tube thickness, tube thermal conductivity, and weighted fin efficiency, respectively. The subscript i, o represent inner (steam) and outer (air) parts. The heat transfer coefficient and pressure drop of the air can be calculated using Briggs and Yong [2] and Robinson model [3], respectively. For the condensation heat transfer coefficient of the steam, Shah correlation [4] is used.

2.4 Calculation algorithm

The flow diagram of calculation is shown in the figure 2. Firstly, set the foot print area, relative heat removal capacity of air-cooling tower, and total loop height as input values, and assume the mass flow of steam. Next, assuming overall heat transfer coefficient, steam-side and air-side calculation in the heat exchanger such as pressure drop or heat transfer coefficient are conducted. Then, from the calculation, updated overall heat transfer coefficient is obtained. This procedure is iterated until the convergence is achieved. After the buoyancy head and pressure loss in the loop are calculated, the mass flow of steam is re-assumed based on the two values. Lastly, after all the iterations are converged, the required height of the air-cooling tower to induce the corresponding mass flow of air is obtained by the following relationship.

$$H_{air} = \frac{\Delta P_{fric}^{air}}{g(\rho_{cold}^{air} - \rho_{hot}^{air})} \tag{5}$$

where ΔP_{fric}^{air} is frictional pressure drop in the heat exchanger, ρ_{cold}^{air} is density at the outlet of the tower, and ρ_{hot}^{air} is density at the ambient temperature. When calculating the pressure loss of the steam and air, only the frictional pressure drop in the steam generator and heat exchanger are considered. In other words, no acceleration and form loss are considered. For the required heat transfer area of seawater-cooling part, 1-D steady state conduction equation is solved.

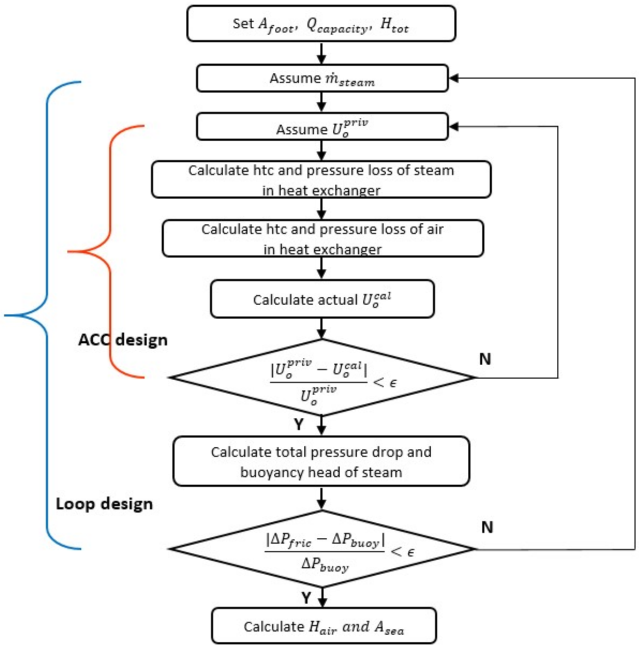

Fig. 2. The flow diagram of the calculation

3. Result

By applying the above method, design of the PRHRS was performed. Firstly, the air-cooled condenser design results are given in table II. The design results depend on several initial parameters, so one case is shown. For the design, the basic tube and fin geometry such as diameter or pitch were obtained from the representative values and other parameters were calculated from the given initial parameters

The behavior of buoyancy head and pressure drop against the steam mass flow rate is shown in the figure 3. The result indicates very typical behavior of natural circulation and single steady point was obtained. However, it should be noted that in this calculation, same height and location between steam generator and seawater-cooling is assumed ($H_1 = H_4$). In other words, the first term in the equation (3) is not considered in this study. Since the significant amount of steam is condensed in the seawater-cooling part, the density difference, $\rho_1 - \rho_3$, will be large value affecting the natural circulation capability, depending on the height and location. Therefore, the effect of them should be investigated as further works.

Table II: Design parameters of the heat exchanger

Parameter	**Value**
$Q_{capacity}$, relative thermal load in air-cooling tower	20%
Foot print area	4 m^2
H_{tot}	10 m
Thermal load in air-cooling tower	200*0.01*0.2 = 0.4 MW
Heat transfer area	44 m^2
Tube length	2 m
Tube outside diameter	20 mm
Tube inside diameter	16 mm
Fin pitch	2.5 mm
Finned tube transverse pitch	70 mm
Finned tube longitudinal pitch	70 mm
Fin height	9 mm
Air Reynolds number	6900
Overall heat transfer coefficient	47 W/m^2K
Air ambient temperature	-3.4°C
Air outlet temperature	20°C
Seawater temperature	-1°C

In the figure 4, the steam mass flow rate in the loop is shown. It is indicated that in the low thermal load and H_{air} region, natural circulation is not formed due to not enough buoyancy head. Nevertheless, it is shown that natural circulation can be used in the most cases. The behavior of steam flow rate is clear. As the loop height and relative thermal load increase, the steam flow becomes larger as indicated in equation (3) (Larger height value and density difference).

The figure 5 represents the required height of the air-cooling tower. As expected, it is shown that the seawater-cooling can reduce the size of the air-cooling tower significantly as thermal load of seawater-cooling increases (low $Q_{capacity}$). In addition, it is observed that the height of the air tower is not almost affected by loop height. That is because that the air-side calculation mainly depends on its heat removal capacity $(Q = C_p\dot{m}\Delta T)$. Since the inlet and outlet temperature of the air are fixed, its flow is only dependent on thermal load. Furthermore, the effect of condensation heat transfer coefficient on U_o is almost negligible. Therefore, from given relative thermal load, the air-side pressure drop and tower height remain almost constant, regardless of loop height.

Lastly, the required heat transfer area in the seawater-cooling part to remove the rest of the decay heat is shown in the figure 6. As expected, the area is reduced as the thermal load in the air-cooling part increases. In addition, it is observed that the area is not sensitive to the loop height similar to the results from the cooling tower height. From the results, it is shown that the required heat transfer area is significant. Therefore, the finned tube heat exchanger might be necessary to satisfy the area.

4. Conclusion

In this study, the preliminary design of the new concept of PRHRS was conducted. The height of air-cooling tower and corresponding heat exchanger area in the seawater-cooling part were evaluated by varying loop height and relative thermal load. From the results, it is seen that the seawater-cooling can reduce the size of the tower significantly, so the system will be feasible for marine applications. As further works, the design of seawater-cooling part such as heat exchanger or location effect will be investigated.

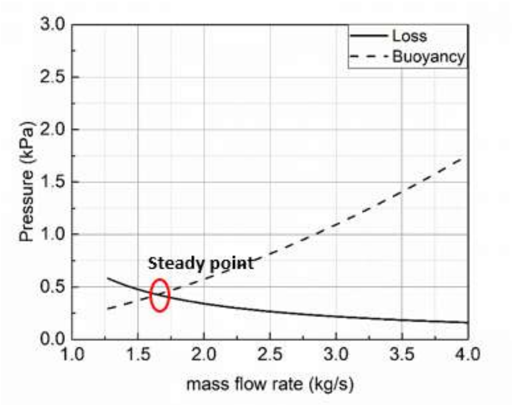

Fig. 3. Pressure head against steam flow ($A_{foot} = 4\ m^2, Q_{capacity} = 20\%$)

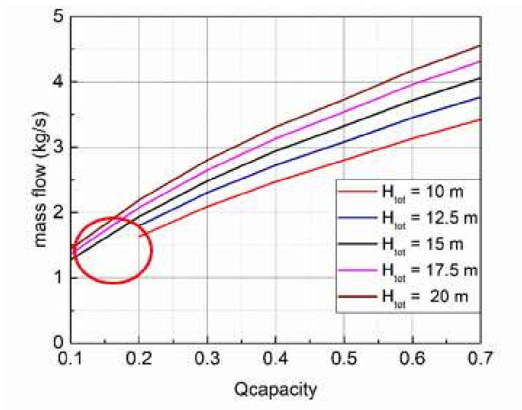

Fig. 4. Steam mass flow rate ($A_{foot} = 4\ m^2$)

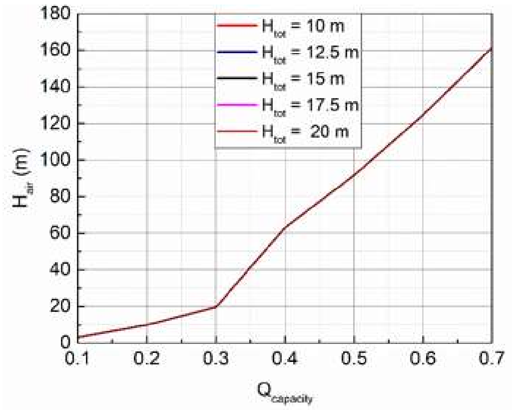

Fig. 5. Air-cooling tower height
($A_{foot} = 4\ m^2$)

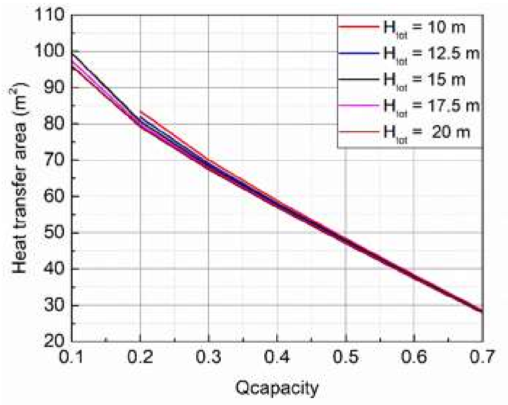

Fig. 6. Calculated heat transfer area in seawater-cooling part
($A_{foot} = 4\ m^2$)

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