An effect of axial conduction on liquid air storage tank for liquid air energy storage system coupled to Nuclear Power Plant

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1. Introduction

Unlike conventional nuclear power plant, the recent nuclear technologies pursue connection with various energies to increase the flexibility. For increasing flexibility of nuclear power plant, a liquid energy storage system is studied to further enhance load following capability of the nuclear power plant [1].

A liquid air energy storage system (LAES) is one of the large-scale energy storage systems using air as a storage medium. When electricity production can exceed the demand, the excess energy of nuclear power plant is used to compress air. The compressed air is cooled down by liquefaction process and expanded through cryo-turbine to produce liquid state air. Liquid air can be used for several ways: power generation, air conditioning, cooling, and refrigeration. The characteristic of liquid air storage tank determines the longevity of the stored energy, and it is substantially dependent of the heat ingress to the tank. The heat ingress occurs due to the low temperature of the stored liquid air (-196°C). The heat penetration via wall makes boil-off gas (BOG). BOG is not only the loss of energy, but it also increases pressure. Therefore, it is important to accurately evaluate external heat ingress.

Currently, many studies assumed the uniform heat flux or use the heat transfer coefficient calculated from experiment [2]. Some studies considered 2D axial conduction, however, those studies did not evaluate how much influence the axial conduction has. If an axial conduction effect is high, the estimated thermodynamic state of tank can be changed hugely. Therefore, in this study, the effect of axial conduction is investigated.

The purpose of this study is to investigate the effect of axial conduction on the thermodynamic state of liquid air tank. For evaluation, a simulation code is developed. The effects of axial conduction are investigated in terms of heat ingress, temperature, evaporation rate, and pressure of tank.

2. Methodology

2.1 Description of liquid air tank

Fig. 1 shows the schematic diagram of nuclear integrated liquid air energy storage system (LAES). When electricity price is low, steam is bypassed from steam Rankine cycle and operates steam turbine. The work of steam turbine is mechanically transferred to air compressor. Air is compressed by air compressor and liquefied by liquefaction process. The main research object of liquid air tank. The liquid air is stored in the cryogenic tank with multiple insulation layers to minimize the heat ingress. However, heat ingress is inevitable, and natural circulation is occurred along the side wall. The heated fluid rises due to the buoyancy effect and is accumulated at the interface which evaporates the liquid air.

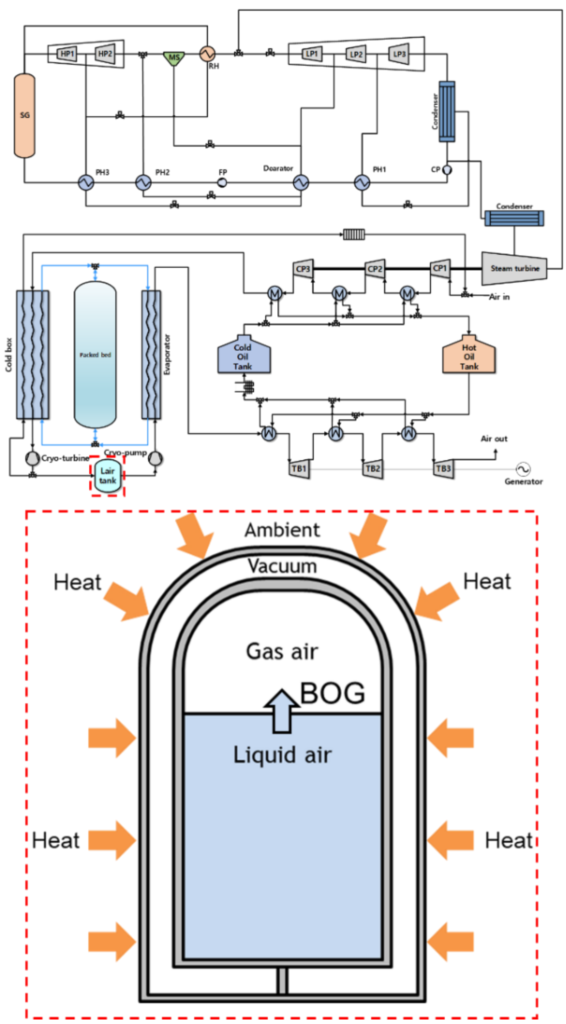

Fig. 1. Schematic diagram of liquid air energy storage system

2.2 Partial Equilibrium Model

Partial Equilibrium Model (PEM) is developed to simulate the thermodynamic state of liquid air tank. PEM is one of the methods which assumes the liquid and the gaseous parts are in thermal equilibrium, respectively [3]. PEM can simulate the thermodynamic performance of liquid air tank with low computational resources, but it is reported to give good accuracy.

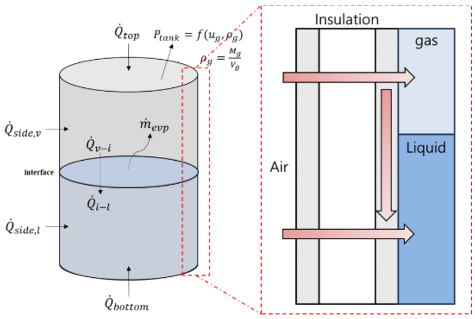

Fig. 2. Schematic diagram of liquid air tank and side wall.

To use PEM, the following assumptions are made as the following.

1. Tank is cylindrical geometry.
2. There are three phases: vapor, liquid, and interface.
3. Each phase has a uniform temperature distribution.

The governing equations for wall heat ingress are given as the following:

$$k\nabla^2 T = k\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) = 0 \cdots eq(1)$$

$$k\nabla^2 T = k\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + k\frac{\partial^2 T}{\partial z^2} = 0 \cdots eq(2)$$

$$BC: \frac{\partial T}{\partial z}(z = 0, H) = 0, T(r = 0) = T_{fluid}, T(r = R) = T_{air}$$

For side wall heat ingress, steady heat conduction equation is applied. The upper and lower boundaries of the wall are assumed as an insulated condition. The left boundary condition is air and right boundary condition is fluid. For non-axial conduction case, the axial heat conduction term is eliminated. For air side, a heat transfer coefficient is assumed as a constant. For fluid side, it is assumed that the natural convection occurs along the side wall. The heat transfer coefficient is given as follows [5]:

$$Nu_{wall} = \left(0.825 + \frac{0.387Ra^{\frac{1}{6}}}{\left(1+\frac{0.492}{Pr}\right)^{\frac{8}{27}}}\right)^2 \cdots eq(3)$$

where Ra is Rayleigh number, Pr is Prandtl number, and Nu is Nusselt number.

Interface heat transfer and evaporation rate are modelled by energy jump model. An energy jump model assumes that the evaporation rate is determined by the energy transfer across the interface. There are two heat transfer paths at the interface: vapor to interface (*v-int*) and interface to liquid (*int-l*). The net energy transfer between gas to interface and interface to liquid can be expressed with an evaporation energy [4]. The mathematical definition is as follows:

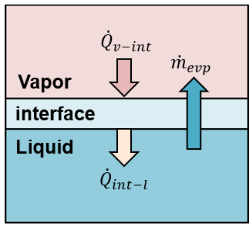

Fig. 3. Heat transfer at the interface.

$$Nu_{v-int} = 0.54Ra_v^{0.25} \cdots eq(4)$$

$$Nu_{int-l} = 0.27Ra_l^{0.25} \cdots eq(5)$$

$$\dot{m}_{evp} = \frac{\dot{Q}_{v-int} - \dot{Q}_{int-l}}{h_{fg}} \cdots eq(6)$$

where h_{fg} is latent heat, $\dot{m}_{evp}$ is evaporation rate.

2.3 Simulation conditions

To observe the effect of axial conduction, simulation condition is determined as follows. For a storage tank, the volume is assumed as $1m^3$ and aspect ratio is 4. The thickness of wall is assumed as 3cm and insulation thickness is assumed as 10cm. The effective thermal conductivity of insulation is assumed as 0.001W/m-K.

Table 1. Geometry conditions

Properties	Value
Tank volume	$1m^3$
Aspect ratio	4 (H = 2.73m, D = 0.68m)
Wall thickness	3cm
Wall material	SUS316L
Insulation thickness	10cm
Insulation effective conductivity	0.001W/m-K
Level fraction	80%

Table 2. Fluid conditions

Properties	Value
Fluid	Air
Fluid initial pressure	1bar
Fluid initial temperature	Saturated condition
Air temperature	25°C
Air heat transfer coefficient	$4W/m^2$-K

Table 3. Simulation conditions

Properties	Value
Time step	1sec
Simulation time	24hours

3. Results

In this section, the effect of axial conduction is investigated. The heat ingress and thermodynamic states are compared.

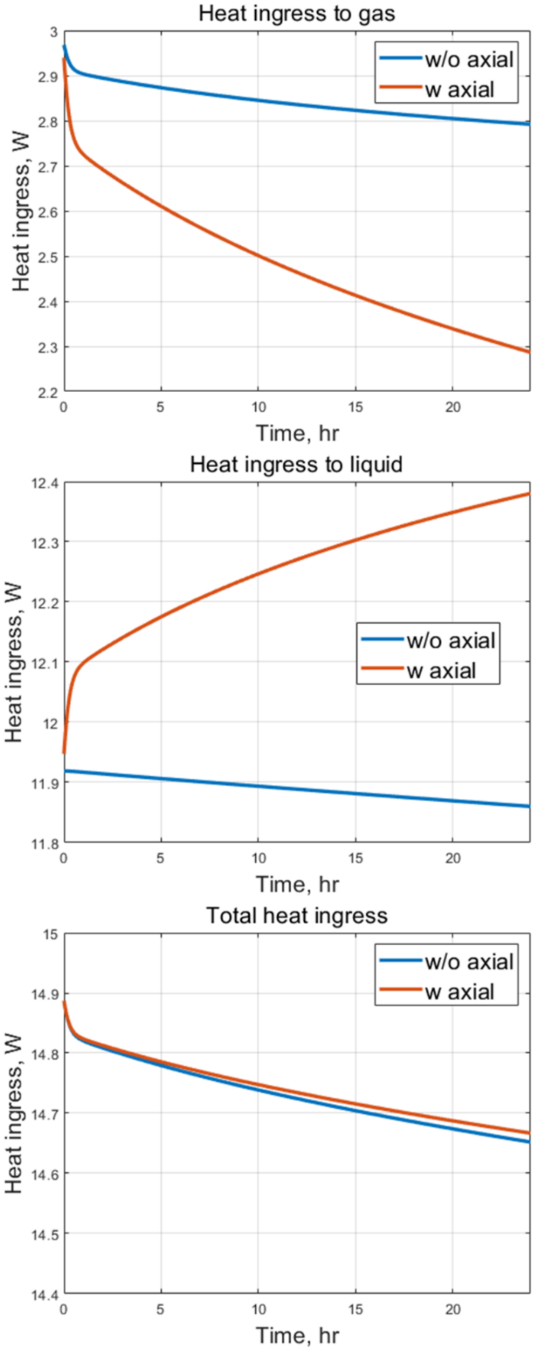

Fig. 4. Heat ingress to gas, liquid, and total.

Fig. 4 shows the heat ingress to gas and liquid region with respect to time. As seen in Fig. 4, the total heat ingress is almost the same for both cases. However, the heat ingress to gas is low when axial conduction is considered. The heat is decreased almost 20%p compared to non-axial conduction case. However, the heat ingress to liquid is increased. This means that some of the heat is bypassed through wall and transferred to liquid region. The reason for this is the difference of heat transfer coefficient of gas and liquid region. The calculated heat transfer coefficients are near 3.6W/m^{2}K and 91.6W/m^{2}K for gas and liquid regions, respectively. The low heat transfer coefficient of gas region acts as huge thermal resistance which makes heat bypassing through stainless steel which has high thermal conductivity.

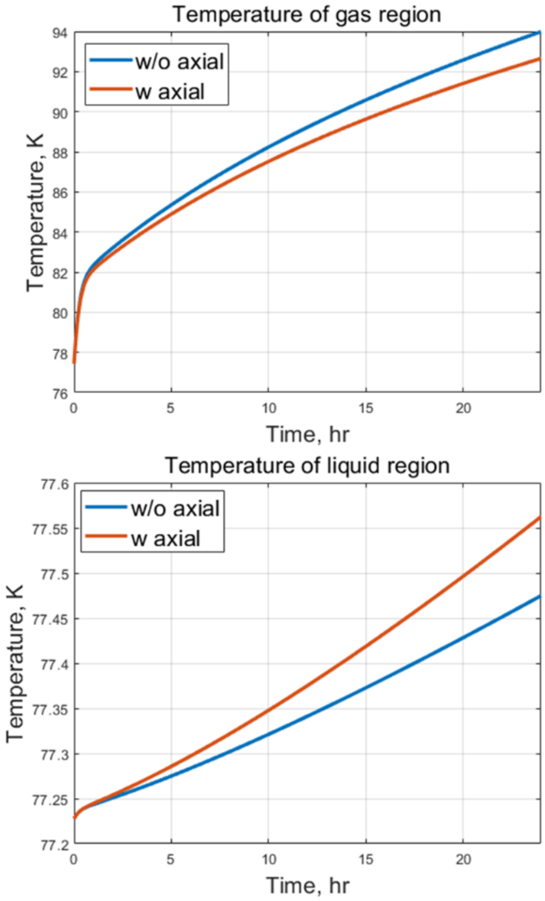

Fig. 5. Temperature of liquid and vapor region with respect to time.

Fig. 5 shows the temperature of liquid and vapor region with respect to time. The temperature of gas is higher when axial conduction is not considered due to high heat ingress. The absolute temperature difference is quite low, but the effect on the evaporation rate is remarkable.

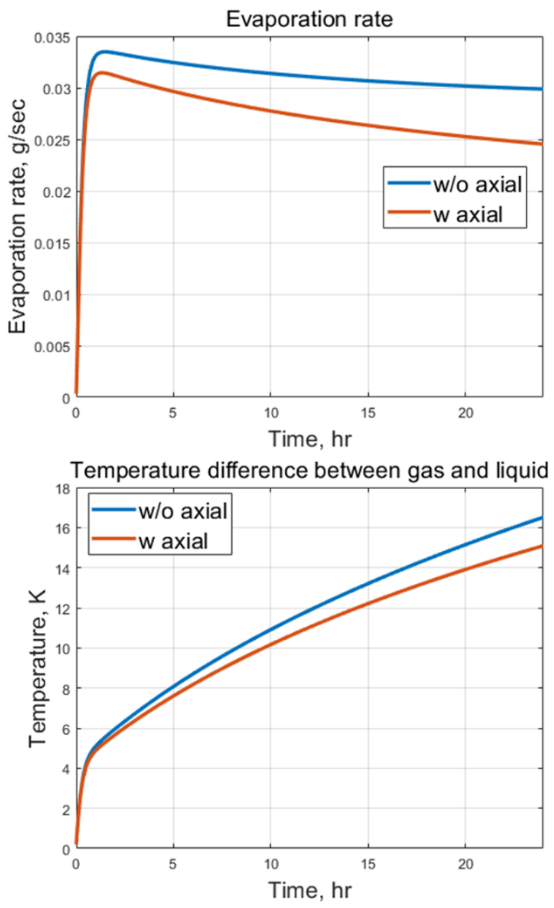

Fig. 6. Evaporation rate and temperature difference between gas and liquid with respect to time.

Fig. 6 shows the evaporation rate and temperature difference between gas and liquid with respect to time. Since high temperature gas at non-axial conduction case, the heat transfer rate to interface is higher than axial conduction case. Consequently, the net heat transfer rate to interface is increased and evaporation rate is also increased. Conversely, if axial conduction is not considered, the internal temperature and evaporation rate can be overestimated.

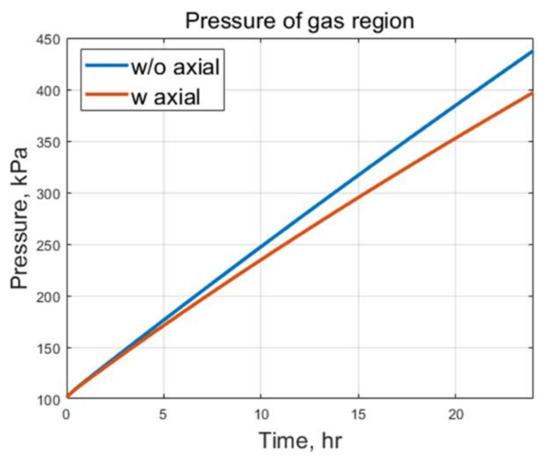

Fig. 7. Pressure of tank with respect to time

Fig. 7 shows the change of tank pressure with respect to time. The final pressure of tank differs by almost 50kPa. In this study, it is assumed that both phases are in saturated condition, but in real world, temperature of gas is higher than saturated temperature. Therefore, if considering the real conditions, the gap between axial and non-axial conduction is expected to become even larger.

4. Conclusions

In this study, the effect of axial conduction on the liquid air tank is investigated. For research, the partial equilibrium model is developed and applied. The thermodynamic changes of fluid inside of the tank are investigated with time. When axial conduction is considered, the heat ingress to gas is remarkably decreased by almost 20%p. Consequently, the temperature of gas decreases due to the influence of axial heat conduction. The decrease of gas temperature leads to less heat transfer to interface which makes less evaporation rate. Comprehensibly, when axial conduction is considered, the temperature, evaporation rate, and pressure of tank is decreased compared to non-axial conduction case. Conversely, non-axial conduction assumption or uniform heat distribution can overestimate the thermodynamic state of the tank. Therefore, the axial conduction should be considered for more accurate simulation. In the future, the effect of geometry, level fraction, and initial condition will be evaluated with developed simulation code.

5. Acknowledgment

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