2상 극저온 열전달 과정 계산에서의 CFD 응용

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Application of CFD to the Calculation of 2 Phase Cryogenic Heat Transfer Processes

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A two-phase numerical model for plate-fin heat exchangers with plain fins and wave fins is studied incorporating the thermodynamic properties and the characteristics of fluid flow. The numerical simulations for the two fins in cryogenic conditions are carried out by employing a homogenous two-phase flow model with the CFD code ANSYS CFX. The heat transfer coefficients and the friction factor for nitrogen saturated vapor condensation process inside two types of plate fin heat exchanger are evaluated including the effects of saturation temperature (pressure), mass flow rate and inlet vapor quantity. The convective heat transfer coefficients and friction factors will be used for design of plate-fin type heat exchangers operating under cryogenic conditions.

Keywords: Cryogenic heat transfer(극저온 열전달);Two-phase flow(이상유동);Plate-fin Heat Exchanger (판형 열교환기); CFD(전산유체)

1. Introduction

A plate-fin heat exchanger (PFHE) is a type of heat exchanger that uses plates and finned chambers to transfer heat between fluids. It is often categorized as a compact heat exchanger to emphasize its relatively high heat transfer surface area to volume ratio. The plate-fin heat exchanger is widely used in many industries, especially for LNG liquefaction under cryogenic condition. Three typical plate-fin heat exchanger models are shown in Fig.1. In this study, the first and second plain fin is selected and the third type (serrated fin) will be carried out in the future work.

The Computational Fluid Dynamics (CFD) technique can provide the flexibility to construct computational models that are easily adapted to physical conditions without the need to construct a large-scale prototype or expensive experimental models, and can directly provide results and detailed information on fluid flow and heat transfer at a relatively low cost and greatly reduce the volume of experimental work required.

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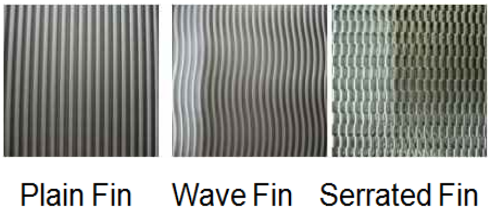

Fig.1 Three typical plate-fins.

Therefore, the CFD technique is a beneficial supplement to experimental study. For numerical studies on the performance of plate-fin heat exchangers, some investigations have been performed on the construction of numerical models and numerical simulations. For obtaining useful data for a heat exchanger design, a computational fluid dynamic (CFD) study of two-phase flow (condensation) in heat exchangers under cryogenic temperatures was carried out. Nitrogen in different saturated conditions was selected as the working fluids. Commercial CFD software ANSYS CFX was used for computational simulation to get the relevant data (heat transfer coefficients, friction factor, etc.) of specified case.

For such two phase flow fin heat exchanger design, it is assumed that the processes are steady and symmetric around the symmetry plane. The principal heat transfer direction is normal to the symmetric plane. The value of convective heat transfer coefficient for each flow is uniform and the fins do not have any multidimensional effect. The working process in a fin heat exchanger is illustrated in Fig.2. This study specially focuses on the two phase condensation process.

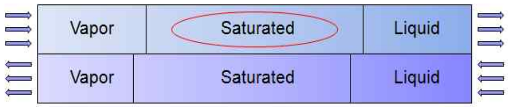

Fig.2 Schematic represenatation of transfer process in a fin heat exchanger in cryogenic range.

The objective of this paper is to develop a new simpler numerical modeling method for the numerical investigation of two phase flow in plate-fin heat exchangers by using the CFD technique. The three-dimensional model considers fluid flow and heat transfer incorporated with thermal conduction in the fins, and greatly reduces the degree of complexity of numerical model construction and numerical simulation. With the model, the pressure drop and heat transfer characteristics over plain fins in PFHE at low Reynolds number, Re, are investigated in detail.

2. Theoretical background

2.1 Homogeneous model

In homogeneous two-phase flow, a common flow field is shared by all fluids, as well as other relevant fields such as temperature and turbulence.

For a given transport process, the homogeneous model assumes that the transported quantities for that process are the same for all phases, that is,

$$\phi_\alpha = \phi \quad 1 \le \alpha \le N \tag{1}$$

Since transported quantities are shared in homogeneous multiphase flow, it is sufficient to solve for the shared fields using bulk transport equations rather than solving individual phasic transport equations. The bulk transport equations can be derived by summing the individual phasic transport equations over all phases to give a single transport equation for ϕ:

$$\frac{\partial}{\partial t}(\rho\phi) + \nabla \bullet (\rho U\phi - \Gamma\nabla\phi) = S \tag{2}$$

where:

$$\rho = \sum_{\alpha=1}^{N_p} r_\alpha \rho_\alpha$$

$$U = \frac{1}{\rho}\sum_{\alpha=1}^{N_p} r_\alpha \rho_\alpha U_\alpha$$

$$\Gamma = \sum_{\alpha=1}^{N_p} r_\alpha \Gamma_\alpha$$

Note that the homogeneous model does not need to be applied consistently to all equations.

2.2 Homogeneous model

The equilibrium phase change model is a single fluid, multicomponent phase change model. This model assumes that the mixture of the two phases is in local thermodynamic equilibrium. This means that the two phases have the same temperature and that the phase change occurs very rapidly.To determine the mass fraction of the vapor, the flow solver uses the lever rule:

$$x = \frac{h_{mix} - h_{sat,l}(p)}{h_{sat,v}(p) - h_{sat,l}(p)} \tag{3}$$

where h_{mix} is the mixture static enthalpy, and $h_{sat,l}(p)$ and $h_{sat,v}(p)$are the saturation enthalpies of the liquid and vapor respectively as a function of pressure. The following observation can be made about the quality:

When x<0, the mixture is 100% subcooled liquid so the liquid properties are selected.

When x>0, the mixture is 100% superheated vapor so the vapor properties are selected.

When 0≤x≤1,the mixture contains liquid and vapor. The bulk mixture properties are calculated using the lever rule. Using the mass fraction of vapor and the saturated liquid and vapor properties this gives:

$$\phi_{mix}(p) = (1 - x_v)\phi_{sat,l}(p) + x_v \phi_{sat,v}(p) \tag{4}$$

where ϕ is a property such as entropy, enthalpy, specific heat, thermal conductivity or dynamic viscosity. Since density is volumetric, saturated density is calculated using harmonic averaging instead:

$$\frac{1}{\rho_{mix}} = \frac{1-x_v}{\rho_{sat,l}} + \frac{x_v}{\rho_{sat,v}} \tag{5}$$

3. CFD solver setting

Commercial software ANSYS CFX is used for the numerical simulation of this study. It provides strong computational models for two phase condensation process. As discussed before, the Equilibrium Phase Change Model in CFX is used as the computational model for this CFD simulation.

A single rectangular-section (1mm×5mm) channel with a length of 50mm is carried out from a plane pin heat exchanger. It is a very simple geometry and easy for mesh generation. Because of the simple geometry, the CFX-Mesh is chosen for the mesh generation, and the mesh information is shown in Fig. 3.

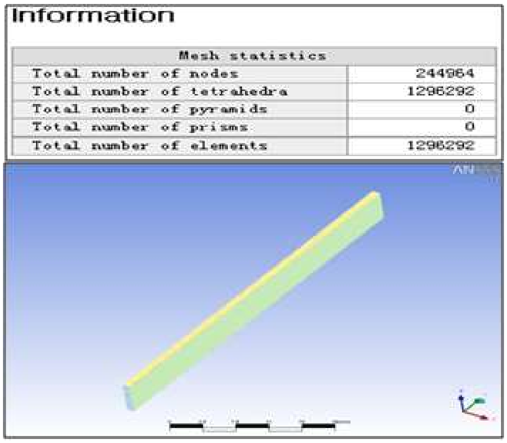

Fig. 3 Mesh information

The Equilibrium Phase Change model requires that consistent material properties be supplied for the two pure substance states involved in the phase change, as well as saturation curve. Although general material properties can be use with this model, in general, it is best practice to use this model with a real gas equation of state for liquid, vapor and saturation properties. Saturation properties are defined by using a Homogeneous Binary Mixture. A Homogeneous Binary Mixture is a mixture of two states of the same pure substance.

A general setup by incorporating two consistent materials (representing the two thermodynamic states) into a homogeneous binary mixture is used to set up saturation nitrogen properties.

This study focuses on a two phase condensation process of saturated nitrogen in a fin heat exchanger channel. A general working process is shown in Fig. 4.

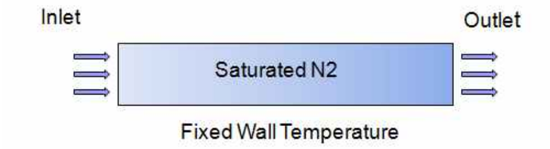

Fig.4 Boundary condition setting.

At inlet boundary, saturated pressure, temperature and mass fraction of vapor phase are specified, and the mass flow rate of the working fluid is set at outlet boundary. The boundary close to the boiling layer is set to a wall boundary with a fixed wall temperature which is lower than the saturated temperature, and other walls of the channel are set to be adiabatic.

For a two phase flow heat exchanger design, heat transfer coefficients in different saturated conditions are needed. The cases with following 5 different saturated conditions shown in Table 1 are calculated, and 5 cases with different fluid mass flow rate are also calculated in a specified saturated condition.

Table. 1 Saturated condition

$T_{sat}(k)$	100	105	110	115	120
$P_{sat}(MPa)$	0.78	1.08	1.47	1.94	2.51

4. Results and Discussions

Numerical results obtained by CFD method in CFX is presented here. In flow behaviour discussion, some special characteristics of two phase flow condensation process, such as velocity field, vapor mass fraction and pressure distribution, is illustrated by observing the contour graphics. The heat transfer coefficients and friction factors under different saturated conditions and mass flow rates are also presented and discussed. By using the numerical method, detailed information on the two phase flow and heat transfer in the channels of two types of plate-fin heat exchanger can be obtained, which can be used to improve and develop plate-fin heat exchangers.

4.1 Plane Fin Heat Exchanger

The fluid velocity in a variation along the flow direction in a plane fin channel is show in Fig 4. And Fig. 5 for $T_{sat}=110k$ and $\dot{m}=0.000135kg/s$. Unlike a single phase flow, the two phase flow velocity tends to be decreased along the flow direction. It is due to the increase of bulk density which caused by the condensation process.

Because of the restriction effect of the geometric wall boundary, the fluid velocity in the middle section of duct is higher, and decreases gradually from the middle section to surrounding boundary.

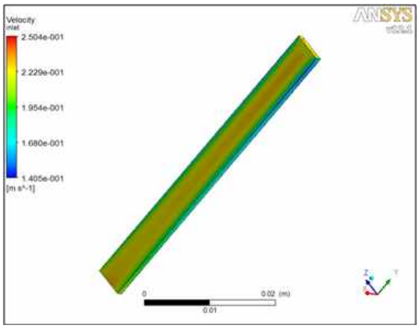

Fig.4 Contour of velocity (plane fin)

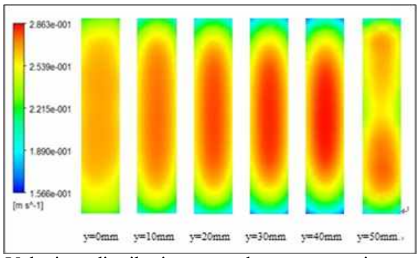

Fig.3 Velocity distribution on the cross-sections along the flow direction (plane fin)

The condensation process can be illustrated by contours of vapor mass fraction. From Fig. 6 and Fig. 7, the condensation of saturated nitrogen can be seen obviously especially in the region near to the wall which is fixed a temperature lower than Tsat.

As we know, pressure drop is the result of frictional forces on the fluid as it flows through the tube. The frictional forces are caused by a resistance to flow. The main factors impacting resistance to fluid flow are fluid velocity through the pipe and fluid viscosity.

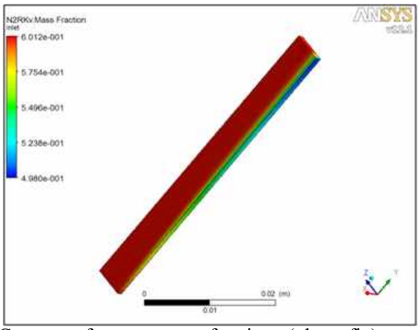

Fig.6 Contour of vapor mass fraction. (plane fin)

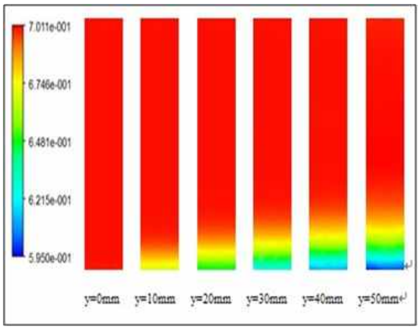

Fig.7 Vapor mass fraction on the cross-sections along the flow direction (plane fin)

The pressure distribution shown in Fig. 8 and Fig.9 show us an obvious decrease tendency along the flow direction and can be also coupled well with the velocity distribution. In a plane fin, both of the fluid velocity and pressure distribution appear as an O-shape on the cross-section of the tube due to the frictional forces on the fluid.

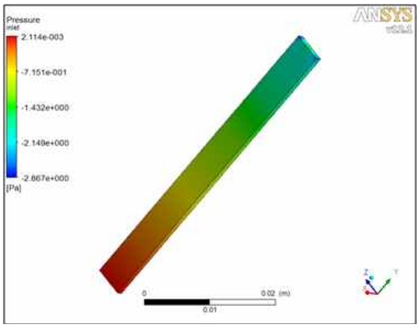

Fig8. Contour of pressure distribution. (plane fin)

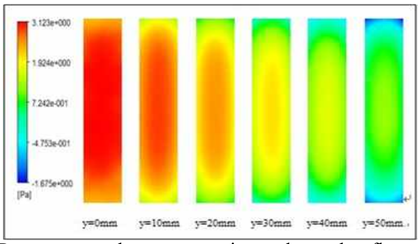

Fig.9 Pressure on the cross-sections along the flow direction. (plane fin)

Reynolds number of homogenous two phase flow can be calculated by:

$$Re = GD/\overline{\mu} \qquad (6)$$

where $G = \rho \overline{U}$ is mass velocity, D is the hydraulic diameter of the cross-section and $\overline{\mu}$ is the fluid dynamic viscosity which can be obtained by CFD calculation.

Fig. 10 plots the Reynolds numbers in different inlet vapor mass fraction under a fixed mass flow rate, saturated temperature and pressure. Since the mass flow rate is fixed, the Reynolds number of the fluid is just inversely proportional to the fluid dynamic viscosity in the cases with different inlet vapor mass fraction. The bulk dynamic viscosity can be obtained from Eq. (4). It is easy to know that the fluid with more vapor phase has a lower dynamic viscosity and a higher Reynolds number. If we fix the dynamic viscosity and density by specifying the inlet vapor mass flow rate, with different mass flow rate, the Reynolds number must present like shown in Fig.11.

Since the boundary setting of the internal surfaces of the tube is that one wall is set to be fixed temperature and three are adiabatic, the bulk heat transfer coefficient of the fluid can be represented as the wall heat transfer coefficient of the fixed temperature wall.

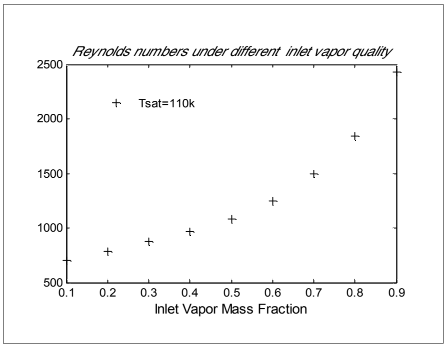

Fig.10 Reynolds numbers in different inlet vapor quality. (plane fin)

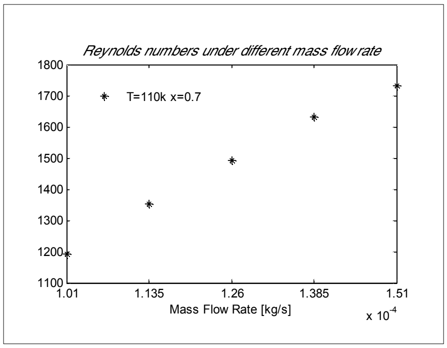

Fig.11 Reynolds numbers in different inlet vapor quality. (plane fin)

The thermal conductivity of liquid phase is much higher than vapor phase in a specified saturated temperature (pressure). So the wall heat transfer coefficient is great effected by the condensation rate of the fluid, especially in the region near to the fix temperature wall. Fig.12 and Fig.13 illustrate the wall heat transfer coefficients with different mass flow rates (fixed saturated temperature) and different saturated temperature (fixed mass flow rate). All of these attempts are with a 20K temperature difference to the fixed wall temperature. It is easy to image that the fluid with lower vapor quality must appear a higher heat coefficient. Although the thermal conductivity of fluid with lower saturated temperature and mass flow rate is lower, it is higher due to the higher condensation rate especially in near wall region.

For the homogeneous two phase flow, the friction factor f_{TP} can be expressed in terms of the Reynolds number by the Blasius equation :

$$f_{TP} = 0.079 Re^{-0.25} \qquad (7)$$

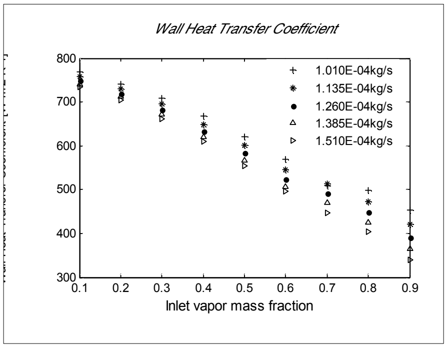

Fig.12 Wall heat transfer coefficient in different mass flow rate condition. (plane fin)

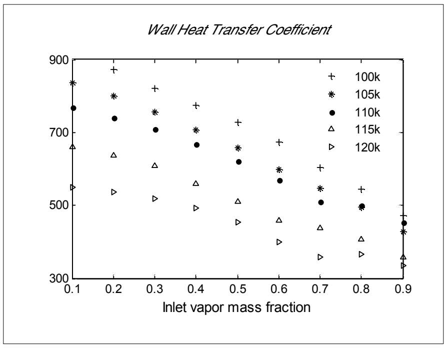

Fig.13 Wall heat transfer coefficient in different saturated temperature.(plane fin)

Fluid friction factors with different mass flow rates (fixed saturated temperature) and different saturated temperature (fixed mass flow rate) are presented in Fig. 14 and Fig. 15. It is can be seen that the largest values in two comparisons are appeared in the smallest mass flow rate case and the lowest saturated temperature case due to their lowest Reynolds numbers discussed before.

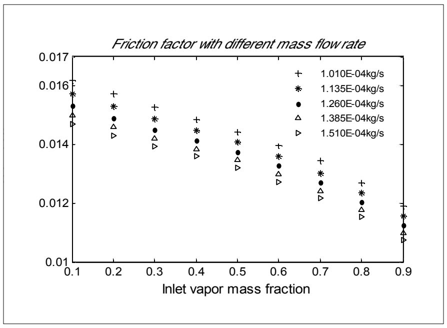

Fig.14 Friction factor with different mass flow rate. (plane fin, Tsat=110K)

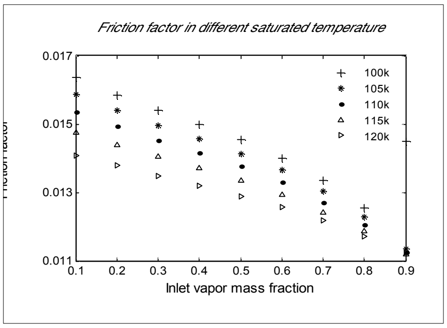

Fig. 15 Friction in different saturated temperature (plane fin $\dot{m}=0.000126kg/s$)

4.2 Wave Fin Heat Exchanger

The flow properties in a wave fin is different with a plane fin. Fig. 16 show us the velocity, pressure and vapor quality in a single channel which is taken from a wave fin. Since the geometry is not symmetric in Z direction, the O-shape fields are irregular and not obvious as in a plane fin in both of velocity and pressure distribution. The same phenomenon of condensation should be also found in the region near to the low temperature wall.

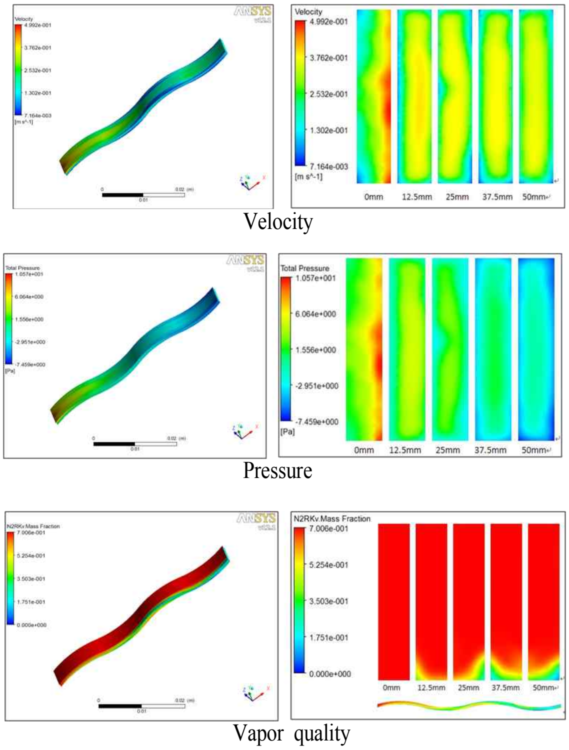

Fig.16 Flow properties in wave fin.

As same as in a plane fin discussed before, in a wave fin, the fluid with lower inlet mass flow rate and vapor quality in lower saturated temperature appeared higher values in both of heat transfer coefficient and friction factor. But the wave fin shows us a better heat transfer ability than a plane fin. As show in Fig. 17, the heat transfer coefficient in a wave fin is larger than in a plane fin under the same saturated and inlet conditions. Since the irregular shape, the friction factor in a wave fin is smaller than in a plane.

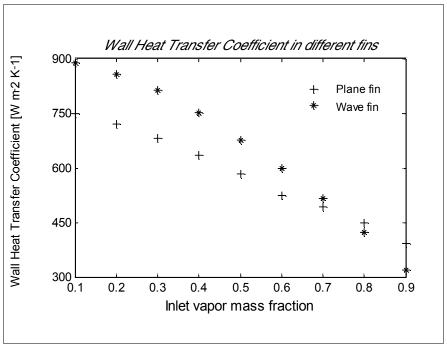

Fig.17 Wall Heat Transfer Coefficient in different fins.

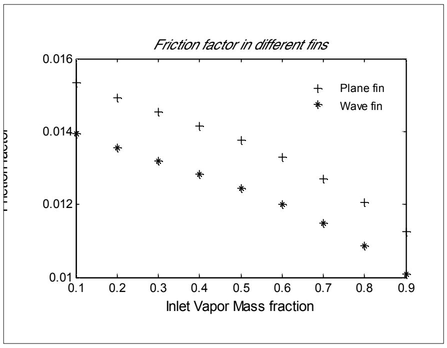

Fig.18 Friction factor in different fins.

5. Conclusion

In this paper, a simpler numerical model was presented for the numerical simulation of plate-fin heat exchangers, based on the characteristics of two phase condensation flow and heat transfer pressure. Numerical simulations of two types of plate-fin heat exchanger were carried out by using the numerical model in CFD code ANSYS CFX. The real gas properties of saturated nitrogen were defined by using equations of state and creating a Homogeneous Binary mixture. Equilibrium phase change model in CFX was introduced as a suitable computational model for homogeneous two phase condensation. It was shown that the model can be applied to simulate the characteristics of heat exchangers with higher accuracy and good convergence. The fluid velocity and pressure distribution in the duct on the cross-sections were obtained. The results presented the heat transfer coefficients and the friction factor measured during nitrogen saturated vapor condensation process inside a plate fin heat exchanger: the effects of saturation temperature (pressure), mass flow rate and inlet vapor quantity were investigated. The fluid with lower inlet mass flow rate and vapor quality in lower saturated temperature appeared higher values in both of heat transfer coefficient and friction factor. The heat transfer ability of wave fine is much better than plane fin. The method can also be used to study or optimize plate-fin heat exchangers with similar structures. The detailed characteristics of fluid flow and heat transfer in ducts and heat conduction in fins were analyzed. More attention should be paid to fins in the research and development of plate-fin heat exchangers. The conclusions of this paper are of great significance for the improvement and optimization of the design of plate-fin heat exchangers.

Acknowledgement

This research is supported by Korea Ministry of Land, Transport and Maritime Affairs through the Gas Plant R&D Project.

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