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Research Paper

Thermal design optimization analysis of an intermediate fluid vaporizer for liquefied natural gas

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HIGHLIGHTS

- Thermal design model for IFVs was established based on the DPM.
- The optimal intermediate fluid was screened by considering saturation parameters.
- Effects of operating parameters on heat transfer performance were analyzed.
- The heat load ratio was recommended to guide the IFV design.

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ABSTRACT

An intermediate fluid vaporizer (IFV) is the core heat transfer equipment in a liquefied natural gas (LNG) regasification system, particularly in an offshore floating LNG receiving terminal where more efforts are focused on improving the efficiency and structure size of the vaporizer for reducing the volume and weight. By considering the constraints of both the initial velocities of the working fluids and length of the heat transfer tubes, a new numerical model based on the distributed parameter method is developed to determine the heat transfer performance and required heat transfer area (HTA) of an IFV. The effects of the intermediate fluids and their saturation parameters, inlet temperature of the seawater, and temperature drop of the seawater in the thermolator are investigated. The results show that propylene exhibits the best heat transfer performance, but its higher saturation pressure would require an increase in the wall thickness of the IFVs and therefore, limit its application. The heat transfer performances of propane and dimethylether are better than the other intermediate fluids, and are promising to be used in IFVs. With increase in the saturation temperature of propane, the required total HTA of IFVs first decreases and then increases, and the optimal saturation temperature is in the range of 250–265 K. A higher seawater temperature is beneficial for reducing the HTA, and it is also indicative of a wider optimization saturation temperature range in which the required total HTA is not sensitive to the saturation temperatures. When the temperature drop of the seawater in the thermolator varies from 0.3 K to 0.8 K, the variation in the required area is not more than 5% compared to the lowest area, and the recommended range for the corresponding heat load ratio between the evaporator and condenser is recommended is 5–15.

1. Introduction

As a type of clean energy, natural gas (NG) is playing an increasingly important role in providing global energy supplies [1]. In the supply chain of NG, the consumer market is typically located far from the major gas field. Liquefying NG is an effective approach for large-quantity transportation via ocean shipping [2]. Liquefied natural gas (LNG) must be first vaporized to NG before it can be used as a fuel in the industry and as an urban gas. Therefore, efficient and reliable vaporizers are important for the LNG regasification process.

Various vaporizers have been developed for LNG receiving terminals such as the ambient air vaporizer (AAV), open-rack vaporizer (ORV), submerged combustion vaporizer (SCV), and intermediate fluid vaporizer (IFV). The selection of a suitable type of vaporizer is related to the climate condition, plant site location, throughput capacity, and demand fluctuation [3], and it will influence the

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Nomenclature

A	area (m^2)
C_p	isobaric heat capacity ($J\,K^{-1}$)
d	diameter (m)
g	gravitational acceleration (ms^{-2})
h	heat transfer coefficient ($Wm^{-2}\,K^{-1}$)
L	tube length (m)
M	molecular weight ($g\,mol^{-1}$)
m	mass flow rate ($kg\,s^{-1}$)
p	pressure (Pa)
P_t	tube pitch (m)
Q	heat transfer rate (W)
q	heat flux ($W\,m^{-2}$)
r	heat of vaporization ($J\,kg^{-1}$)
T	temperature (K)
U	overall heat transfer coefficient ($Wm^{-2}K^{-1}$)
u	flow velocity (ms^{-1})

Greek symbols

ΔT	logarithmic mean temperature difference (K)
ΔH	enthalpy difference ($J\,kg^{-1}$)
ρ	density ($kg\,m^{-3}$)
λ	thermal conductivity ($Wm^{-1}\,K^{-1}$)
φ	heat transfer capacity (W)
δ	thickness of the tube wall (m)
α	area percentage (%)
β	heat load ratio

Subscripts

0	initial value
1, 2, 3	based on inlet, intermediate, or outlet condition
b, f, w	based on bulk, fluid, and wall temperature
evp, cond, them	evaporator, condenser, and thermolator
i, o	inner or outer
j	element index of the heat exchanger
m	mean
ng	natural gas (NG) or liquefied natural gas (LNG)
sat	saturation
sw	seawater
w	based on wall temperature

capital cost, operating cost, and environmental impact of the LNG terminals. AAVs have a simple construction and low operating cost, but their heat transfer coefficients (HTCs) are relatively low and easily influenced by both the atmospheric conditions and operation parameters. In addition, moist air can frost on the surface of the fin and decrease the heat transfer performance of an AAV significantly during the heat transfer process of natural convection [4]. Therefore, an AAV is mainly used in peak shaving plants. In contrast, the ORV, SCV, and IFV are primarily employed in base load LNG regasification terminals. A conventional ORV is a type of heat exchanger that uses the direct heat transfer between cryogenic LNG and seawater, and it also faces a problem similar to an AAV. When LNG exchanges heat with the seawater through the heat tubes, an ice layer can be formed on the outside of the tube resulting in a tremendous deterioration of the heat transfer efficiency. The heat transfer efficiency could also be influenced by increasing the LNG flow rate and decreasing the seawater flow rate [5]. To reduce the thermal resistance of icing, a new type of ORV named SuperORV was developed by Osaka Gas and Kobe Steel that has a duplex tube configuration in the lower part of the tube panels. Compared with a conventional ORV, SuperORV was considered to increase the LNG vaporization rate per heat transfer tube by three to five times [6]. An SCV is generally used either as a temporary alternative of an ORV when the seawater temperature is lower than the design value, or to provide the required flexibility for satisfying the peak demands during cold seasons [7]. The major disadvantages of an SCV are that it consumes a fraction of 1.2–1.3% of LNG and the emission of combustion products can affect the ambient environment [8].

Compared with the above LNG vaporizers, an IFV is a compact shell-and-tube heat exchanger consisting of three parts, namely, an evaporator, a condenser, and a thermolator. The evaporator and condenser of an IFV are often arranged in one large shell, while the thermolator is installed either on the same shell or independently [9–11]. In the shell of an IFV, an intermediate fluid (IF) with a low boiling point is filled and the evaporator is submerged in it. The IF absorbs heat from the seawater and converts to a gas in the evaporator; it then releases the heat into the LNG and converts into a liquid again in the condenser. The indirect heat transfer between the seawater and cryogenic LNG avoids the seawater freezing, and improves the HTC and operation reliability. The excellent heat transfer performance and compact structural arrangement make the IFV particularly appropriate for offshore floating storage and regasification systems and LNG cold energy utilization [12].

Using the indirect heat transfer makes the thermal design of an IFV more complex with various types of heat transfer mechanisms compared with other vaporizers. In addition, its highly integrated configuration increases the difficulty of operating analysis. At present, increasingly, scholars are being attracted by this new type of vaporizer [13,14]. Pacio and Dorao [15] reviewed the thermal hydraulic models for cryogenic applications, of which the distributed parameter model (DPM) is commonly used in heat exchanger design with a large temperature difference. Bai et al. [16] developed a one-dimensional numerical method for calculating the heat transfer area (HTA) of an IFV, and the HTAs of the three parts, i.e., the evaporator, condenser, and thermolator were provided. Pu et al. [13] conducted the evaluation of an IFV by using a method similar to their previous work. The outlet temperatures of the NG and seawater, HTCs in the three parts, and propane saturation temperatures were calculated based on the given HTA and inlet parameters of both the LNG and seawater. Xu et al. [17] performed numerical calculations to compare the required HTA of IFVs utilizing the candidate refrigerants of propylene, propane, isobutane, butane, and dimethylether and operating under various saturation temperatures. They suggested that propylene and dimethylether are promising refrigerants for an IFV system, in addition to the widely reported use of propane. Furthermore, Xu et al. [18] recently, established a thermal design method for the IFV processing of subcritical LNG. The applicable HTC correlation in different sections and flow boiling zones were employed in their model. Kim et al. [19] conducted numerical calculations to design a metal foam filled plate heat exchanger and shell-and-tube heat exchanger, both of which are used as thermolators in an LNG regasification system. The results show that the metal-foam plate heat exchanger has twice the heat transfer rate and 20% lower pressure drop than the shell-and-tube heat exchanger.

In previous studies, the heat transfer model of an IFV has been established. The effects of the IF and its saturation condition on the IFV performance were also reported in a relatively narrow temperature range [13,14]. However, the optimal saturation condition and its sensitivity to the seawater temperature have been rarely discussed. A reasonable medium and working parameter would lead to a decrease in the volume and weight, but increase the invest-