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Application of metal foam heat exchangers for a high-performance liquefied natural gas regasification system

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ABSTRACT

The intermediate fluid vaporizer has wide applications in the regasification of LNG (liquefied natural gas). The heat exchanger performance is one of the main contributors to the thermodynamic and cost effectiveness of the entire LNG regasification system. Within the paper, the authors discuss a new concept for a compact heat exchanger with a micro-cellular structure medium to minimize volume and mass and to increase thermal efficiency. Numerical calculations have been conducted to design a metal-foam filled plate heat exchanger and a shell-and-tube heat exchanger using published experimental correlations. The geometry of both heat exchangers was optimized using the conditions of thermolators in LNG regasification systems. The heat transfer and pressure drop performance was predicted to compare the heat exchangers. The results show that the metal-foam plate heat exchanger has the best performance at different channel heights and mass flow rates of fluid. In the optimized configurations, the metal-foam plate heat exchanger has a higher heat transfer rate and lower pressure drop than the shell-and-tube heat exchanger as the mass flow rate of natural gas is increased.

1. Introduction

Gas heating with ambient heat sources is required in various cryogenic applications. After extracting cold exergy from cryogenic sources, the temperatures of the source fluids are still below zero degrees Celsius since specific applications do not fully utilize all exergy in cold sources. This low temperature could cause several technical problems. In the application of cold exergy in LNG (liquefied natural gas), solid methane hydrate could be generated at temperature below zero and adhere to pipe walls. As a result, pipeline pressure losses are increased, and in the worst case, the pipe is blocked with methane clathrate. Similarly, cold fluid could cool down the pipe to below zero, and frost layers could be formed outside the pipe. During pipeline transport, there are natural temperature drops from pressure losses caused by friction. In this case, the fluid is preheated to more than zero degrees Celsius before entering the local pipeline network to compensate for the temperature drops caused by the Joule–Thompson effect. To prevent these problems, a gas-heater is used in cryogenic applications [1–11].

Many have investigated the heat and mass transfer characteristics in the medium of compact heat exchangers, such as wire screen and metal-foam structures. Tian et al. [12] researched fluid flow and heat transfer of cellular metal lattice structures. Paek et al. [13] and Calmidi et al. [14,15] presented a comprehensive analytical and experimental investigation for determination of the effective thermal conductivity (k_e), permeability (K) and inertial coefficient (f) of high-porosity metal foams. Liu et al. [16] measured the pressure drop through various types of foam matrixes and developed empirical equations for the friction characteristics of a foam matrix. Lu et al. [17] presented an analytical study of the forced convection heat transfer characteristics in high-porosity open-cell metal foam-filled pipes. Tadrist et al. [18], Vafai et al. [19] and Kim et al. [20,21] investigated the impact of porous fins on the pressure drop and heat transfer characteristics in plate-fin heat exchangers.

This study presents a novel concept of a metal-foam plate heat exchanger for weight reduction and higher thermal efficiency. Based on empirical data and correlations of heat and mass transfer, heat exchangers were designed for the thermolators in LNG regasification systems. The exchangers were also compared to conventional shell-and-tube heat exchangers.

2. Physical model

2.1. LNG regasification systems

The IFV (intermediate fluid vaporizer) is a new kind of vaporizer for LNG regasification systems. A thermal model has been

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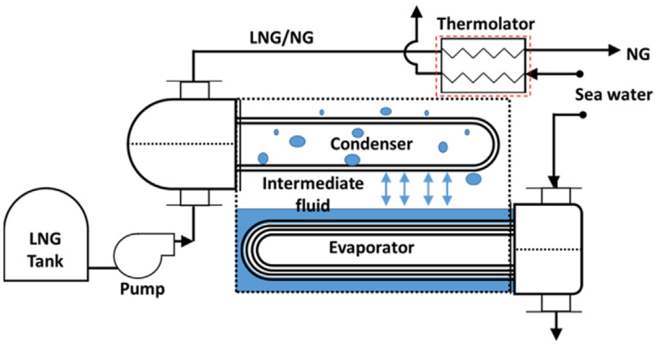

Fig. 1. Schematic of IFV with thermolator.

Table 1
Default values of the known parameters.

T_{Cold_in} (K)	T_{Hot_in} (K)	P_{Cold_in} (bar)	P_{Hot_in} (bar)	$\dot{m}_{Cold\_in}$ (kg/s)	$\dot{m}_{Hot\_in}$ (kg/s)
233	288	122	10	50	1000

developed on the energy balance of the three typical parts of IFV: the evaporator, condenser, and thermolator. A schematic of the system is shown in Fig. 1.

LNG vaporization is achieved using two main heat exchangers operating in series. The dotted box in Fig. 1 is a second medium cycle that is driven using the latent heat of propane condensation to partially heat the LNG. The IF (intermediate fluid) transfers its latent heat during vaporization and condensation using seawater as a heat source and LNG as a cooling source. The thermolator using seawater heats LNG further to the required temperature. LNG is pumped into the condenser by a pump, where it gasifies. The natural gas from the condenser is heated continuously in the thermolator by the seawater, and then goes into the gas-supplying system. Since LNG is pre-heated by a second fluid, it avoids direct contact with seawater and cryogenic LNG and prevents freezing of the seawater. For this reason, seawater close to freezing can be used in this configuration [22–26]. The default values of known fluid parameters are shown in Table 1.

2.2. Shell-and-tube heat exchangers

Shell-and-tube heat exchangers have many different configurations. This paper investigates the TEMA E type exchanger with one shell and two tube passes. Seawater flows on the tube side and the vaporized LNG flows on the shell side. The models for the pressure drop and heat transfer correlations are thus considered as a convective single-phase flow. The tubes in the bundle are in a 45° rotated-square arrangement. The geometric parameters are shown in Fig. 2.

Table 2
Dimensions of shell-and-tube heat exchanger.

Parameters	Dimension
Tube-side outside diameter, d_o(mm)	20
Tube-side inside diameter, d_i (mm)	16
Tube pitch, p_t (mm)	25
Tube bundle layout (°)	45°
Central baffle spacing, $L_{b,c}$ (m)	0.279
Outlet baffle spacing, $L_{b,o}$ (m)	0.318
Inlet baffle spacing, $L_{b,i}$ (m)	0.318
Tube length, L_t (m)	2–11
Shell side inside diameter, D_s (m)	0.78–1.78
Baffle cut, l_c (mm)	86.7
Number of sealing strip pairs, N_{ss}	1
Width of bypass lane, w_p (mm)	19.0
Number of tube passes, n_p	2
Number of pass partitions, N_p	2
Tube-to-baffle hole diametral clearance, δ_{tb} (mm)	0.794
Shell-to-baffle diametral clearance, δ_{sb} (mm)	2.946
Heat Transfer Area (m^2)	470
Total Volume (m^3)	4.98–5.24

The dimensions of the exchanger are shown in Table 2. To find the optimized geometry of the shell with the best heat and mass transfer performance, a simulation was conducted with different aspect ratios of the tube length and shell-side inner diameter with a fixed total heat transfer area.

2.3. Metal-foam plate heat exchangers

Plate heat exchangers can have different types of corrugations [27], such as chevron or herringbone corrugations. However, in this study, the inner shape of the plate heat exchanger is aluminum metal-foam instead of general corrugations. The conceptual design and geometrical parameters of the metal-foam plate heat exchanger are shown in Fig. 3. A one-pass plate heat exchanger with a channel filled with metal-foam is used to heat cryogenic NG with seawater on the other side.

To find the optimized plate configuration with the best heat and mass transfer performance, a simulation was conducted with different channel heights at a given total volume. Table 3 shows dimensions of the metal-foam plate.

The inner structure of metal-foam is shown in Fig. 4. Specimens with porosity of $\varepsilon = 0.92$ and pore density of 10PPI (pore per inch) were selected so that the porosity would be comparable to that of conventional fins used in heat exchangers. The permeability of the aluminum foams was determined from pressure drop data measured by Kim et al. [20,21]. The metal-foam parameters are shown in Table 4. The fiber diameter, d_f was 0.2 mm and the pore size, d_p was 1 mm.

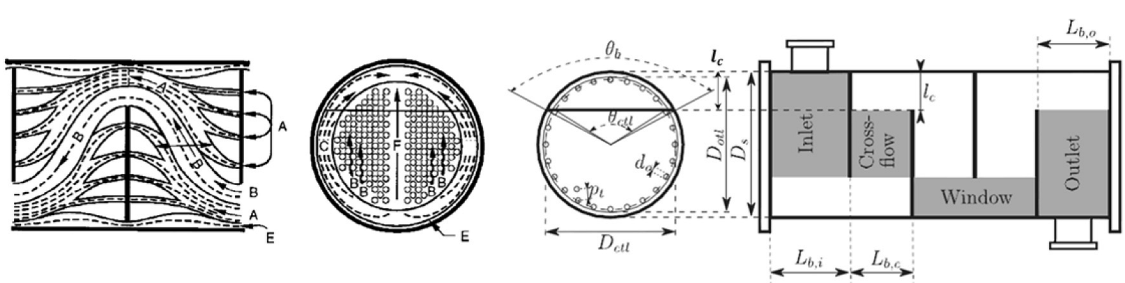

Fig. 2. Geometrical characteristics of a shell-and-tube heat exchanger. Figure adopted from Ref. [27].