

EVALUATING THERMAL RESISTANCE IN CRYOGENIC LNG CARGO CONTAINMENT SYSTEMS: A CASE STUDY OF THE TANGGUH-FSRU JAWA SATU TRANSPORT ROUTE

Rafi Adisasmito Syamsudin¹, Fayza Yulia^{1*}

¹Department of Mechanical Engineering, Faculty of Industrial Engineering, Universitas Pertamina

Abstract

This study evaluated the thermal resistance performance of spherical cryogenic storage tanks on liquefied natural gas carriers operating between the Tangguh refinery and the Jawa Satu Floating Storage Regasification Unit to address the operational challenge of cargo shrinkage. The natural evaporation of cryogenic fluids during maritime transportation represented a physical loss of energy supply and caused a significant financial burden on the national energy supply chain. To measure the thermal efficiency of the tanks, the theoretical heat transfer rate and the theoretical daily boil-off rate were calculated based on the thermal resistance of the composite wall structure. These theoretical baselines were then compared with actual volume shrinkage data from voyage reports to determine the efficiency deviation and the subsequent economic loss. The analytical model demonstrated that the theoretical heat transfer rate penetrating the storage system was 116.9 kW, which corresponded to an expected daily boil-off rate of 0.17%/day. However, the field performance data revealed an actual daily boil-off rate of 0.35%/day. This comparison identified a performance inefficiency deviation of 0.18%/day, which was attributed to the degradation of the insulation materials. Consequently, this thermal inefficiency caused a commercial energy supply failure of 31,351.46 MMBtu that evaporated during the voyage. This energy loss translated to a financial deficit of \$194,108.77 for a single voyage cycle, highlighting the critical necessity for strict insulation maintenance.

This is an open access article under the [CC BY-NC](#) license



Keywords:

Boil-off gas; thermal resistance; tangguh LNG; cryogenic insulation; FSRU Jawa Satu

Article History:

Received: July 24th, 2026

Revised: October 29th, 2026

Accepted: October 30th, 2026

Published: October 31st, 2026

Corresponding Author:

Fayza Yulia

Department of Mechanical
Engineering, Universitas Pertamina,
Indonesia

Email:

fayza.yulia@uiniversitaspertamina.ac.id

1. Introduction

One of Indonesia's energy company holds a strategic role in maintaining the reliability of the national energy supply chain, including the management of liquefied natural gas distribution for power plant requirements. One of the primary distribution routes within this supply chain is the shipment of cargo from the Tangguh refinery facility to the Jawa Satu Floating Storage Regasification Unit (FSRU) utilizing liquefied natural gas carriers. During this maritime transportation process, the phenomenon of cargo shrinkage caused by natural boil-off emerges as an operational challenge that directly affects energy distribution efficiency.

Based on thermodynamic principles, the evaporation of cryogenic fluids is triggered by heat flow from the external environment penetrating the extremely low-temperature storage tanks [1]. Spherical tanks commonly used on carriers feature a layered composite wall structure designed to provide maximum thermal resistance [2]. Nevertheless, in conjunction with the voyage duration and the potential degradation of insulation materials, a certain amount of thermal energy will continue to propagate inward, initiating a phase change of a portion of the

liquid cargo into gas [3]. This evaporated gas represents a physical loss of energy supply that should have been converted into electricity at the receiving facility [4].

A comprehensive analysis of the ship tank's thermal insulation performance is necessary to ensure that the evaporation rate occurring during the voyage remains within design tolerance limits. Modeling the thermal resistance of the spherical tank structure enables the theoretical calculation of the heat transfer rate. These theoretical calculation results can then be compared with actual field data on volume shrinkage to determine the deviation percentage. This technical evaluation is crucial for the company's procurement and logistics division because the discrepancy between actual and theoretical evaporation can be converted into an economic loss valuation. Therefore, this research is structured to measure the thermal efficiency of ship tanks on a specific shipping route and to calculate the financial impact of such inefficiencies on the company's supply chain cost burden.

The scope of the analysis in this report is confined to several specific conditions. The analysis focuses exclusively on carriers utilizing a spherical storage tank system. Additionally, the heat transfer calculations are assumed to occur under steady-state conditions throughout the duration of the voyage. The impact of fluid dynamics caused by ocean waves is disregarded, ensuring that all evaporation deviations are evaluated purely because of the tank's thermal resistance characteristics. Lastly, the calculation of economic losses employs the heating value standard applicable within the company's commercial agreements.

Based on the formulated problems, the objectives of this practical work report are defined accordingly. The primary objective is to calculate the heat transfer rate entering the tank utilizing the thermal resistance modeling of the spherical composite wall. Furthermore, this study aims to compare the actual boil-off rate obtained from the voyage with the theoretical boil-off rate to determine the percentage of deviation. The final objective is to analyze the financial impact by converting the volume difference of the wasted gas into commercial thermal energy units and monetary value.

2. Method

The methodology applied in analyzing the carrier's performance is based sequentially on the principles of energy conservation and heat transfer. The analytical flow is structured through the following working steps. First, the process begins by calculating the surface area of the liquid in direct contact with the tank wall. Due to the spherical shape of the ship's tank, this wetted surface area will continuously decrease as the cargo depletes. The calculation employs a sequential numerical approximation method to determine the average contact area throughout the voyage duration. Second, the process proceeds by modeling the capacity of the tank's composite wall layers to resist external heat. By incorporating the thickness and material properties of the insulation into the thermal conduction resistance equation, the total external heat energy penetrating the cryogenic tank is calculated. Third, this total incoming thermal energy is converted into the liquid evaporation rate. This step calculates the volume of liquefied natural gas projected to phase shift into gas due to the external heat intrusion. This figure serves as the theoretical baseline for the vessel's efficiency based on initial manufacturer specifications. Fourth, the discrepancy between the vessel's theoretical efficiency and its actual operational performance is calculated. This deviation percentage serves as the primary evidence of degradation in the vessel's thermal insulation capacity while operating along the company's distribution route.

2.1. Liquefied Natural Gas (LNG)

Liquefied Natural Gas (LNG) is the liquid phase state of natural gas cooled to a cryogenic temperature of approximately -160°C at atmospheric pressure. This cooling process causes a significant reduction in fluid volume, making inter-island transportation economically viable [4]. Phase change analysis in cryogenic systems requires a thorough understanding of the fluid's thermodynamic properties. Two properties essential in evaporation analysis are fluid density and the latent heat of vaporization [5]. The latent heat of vaporization

is the minimum amount of thermal energy required to change the liquid phase into gas at a constant temperature. This parameter serves as the fundamental basis for correlating the amount of external energy propagating into the system with the total volume of evaporated cargo.

2.2. Cryogenic Storage Tanks Geometric Modeling

The large-capacity storage of cryogenic fluids requires a tank geometry design capable of minimizing the external exposure surface area to reduce the heat transfer rate. One geometric design model implemented in maritime transportation is the spherical tank, commonly known as the Moss type. Based on spatial principles, a spherical shape possesses the smallest surface area to volume ratio compared to other geometric storage shapes [6]. In maritime transportation utilizing spherical tank designs, the overall carrying capacity is divided into several identical storage tanks. The maximum volume capacity for each tank is formulated as:

$$V_{per\ Tank} = \frac{Capacity}{No.\ of\ Tank} \quad (1)$$

Based on geometric modeling assumptions, the tank is projected as a perfect sphere, which can be formulated as:

$$V_{per\ Tank} = \frac{4\pi r_i^3}{3} \quad (2)$$

The inner radius of the tank can be determined mathematically by rearranging the spherical volume equation through the following formulation:

$$r_i = \sqrt[3]{\left(\frac{3V_{per\ Tank}}{4\pi}\right)} \quad (3)$$

Where:

r_i = Tank inner radius (m)

$V_{per\ tank}$ = Volume per tank (m³)

2.3. Overall Heat Transfer

The temperature difference between the external climate and the internal tank environment triggers a heat flow propagating towards the center of the tank [7]. This thermal flow penetrates the various constituent layers of the tank's insulation. Each component material provides a distinct thermal resistance value against conductive heat flow. Based on Fourier's law of one-dimensional heat conduction in spherical coordinates, the heat transfer rate is directly proportional to the temperature difference and inversely proportional to the total

thermal resistance of the system [2]. The theoretical heat transfer rate penetrating into each tank is calculated using equation:

$$\dot{Q} = \frac{-4\pi(T_{LNG}-T_{amb})}{\frac{1}{r_i^2 h_i} + \frac{1}{k_1} \left(\frac{1}{r_i} - \frac{1}{r_1} \right) + \frac{1}{k_2} \left(\frac{1}{r_1} - \frac{1}{r_2} \right) + \dots + \frac{1}{r_o^2 h_o}} \quad (4)$$

Where:

- $\dot{Q}$ = Overall heat transfer (W)
- T_{LNG} = Liquefied Natural Gas temperature (°C)
- T_{amb} = Ambient temperature (°C)
- r_1 = First layer radius (m)
- r_2 = Second layer radius (m)
- r_o = Outer layer radius (m)
- h_i = Inner heat transfer coefficient (W/m²°C)
- h_o = Outer heat transfer coefficient (W/m²°C)
- k_1 = First layer thermal conductivity (W/mk)
- k_2 = Second layer thermal conductivity (W/mk)

The degradation of material resistance over the operational lifespan of the equipment will trigger an increase in the heat transfer coefficient [3].

2.4. Boil-Off Rate

Surrounding thermal energy penetrating the insulation layer is absorbed by the cryogenic liquid, initiating a phase transition into gas. Based on the previously established energy equilibrium principles, the theoretical volume of gas formation per day is calculated by combining the heat transfer rate, latent heat of vaporization, and fluid density [4]. The theoretical BOG is calculated using equation:

$$BOG_{theo} = \frac{\dot{Q} \times 3600 \times 24}{h_{fg} \times \rho_{LNG}} \quad (5)$$

Where:

- BOG_{theo} = Theoretical boil-off gas (m³/day)
- h_{fg} = Latent heat of vaporization (kJ/kg)
- ρ_{LNG} = LNG density (kg/m³)

The international performance indicator for a vessel's thermal insulation system is expressed as a daily boil-off rate percentage [8]. Based on the theoretical volume of gas generated, the theoretical daily boil-off rate limit is determined using the following equation:

$$BOR_{theo} = \frac{BOG_{theo}}{V_{per\ tank}} \times 100\% \quad (6)$$

Where:

BOR_{theo} = Theoretical boil-off rate (%/day)

BOG_{theo} = Theoretical boil-off gas (m^3 /day)

$V_{per\ tank}$ = Volume per tank (m^3)

The actual cargo shrinkage performance must be measured from the difference in volume readings at the loading port and the discharge port [9]. The total volume of boil-off gas lost during the voyage is calculated using the difference between the initial volume and the final volume:

$$V_{BOG} = V_i - V_f \quad (7)$$

Where:

V_{BOG} = Boil off gas volume (m^3)

V_i = Initial Volume (m^3)

V_f = Final volume (m^3)

The actual average volume of evaporated gas per day in each tank during the voyage is formulated as:

$$BOG_{act} = \frac{V_{BOG}}{t_{voyage} \times No.of\ Tanks} \quad (8)$$

Where:

BOG_{act} = Actual boil-off gas (m^3 /day)

t_{voyage} = Voyage duration (day)

Subsequently, the actual daily boil-off rate percentage is calculated by dividing the actual gas volume by the tank volume:

$$BOR_{act} = \frac{BOG_{act}}{V_{per\ tank}} \times 100\% \quad (9)$$

Where:

BOR_{act} = Actual boil-off rate (%/day)

BOG_{act} = Actual boil-off gas (m³/day)

$V_{per\ tank}$ = Volume per tank (m³)

The deviation in voyage efficiency is evaluated based on the difference between the actual evaporation rate at sea and the evaporation rate derived from theoretical thermal resistance calculations. This evaporation deviation value is recorded as a system inefficiency and is expressed through the following equation:

$$\Delta BOR = BOR_{act} - BOR_{theo} \quad (10)$$

Where:

ΔBOR = Deviation between BOR_{act} and BOR_{theo} (%/day)

2.5. Economic Valuation and Financial Loss

Within the scope of global logistics and the transfer of primary energy commodities, the change in ownership status is evaluated based on the calculation of its heating value [4]. Evaporated gas from the tanks is strictly evaluated as a lost opportunity for energy supply to consumers.

To conduct the economic valuation, the deviation in evaporated gas volume must be converted into the commercial thermal energy unit applicable within the industry, namely million British thermal units (MMBtu), utilizing the standard heating value of the fluid [4]. The conversion formula for the lost energy is defined as follows:

$$E = \frac{V \times \rho_{LNG} \times HV_{LNG}}{1055.12} \quad (11)$$

Where:

E = Energy (MMBtu)

HV_{LNG} = Heating value (MJ/kg)

The final stage of this economic valuation involves multiplying the total energy lost by the agreed liquefied natural gas price applicable during the voyage period. The financial loss or opportunity cost incurred is calculated using the following equation:

$$Cost = E \times price_{LNG} \quad (12)$$

Where:

Cost = LNG value (\$)

price_{LNG} = LNG price per unit energy (\$/MMBtu)

3. Result and Discussion

3.1. Operational Data and Vessel Specification

This research utilizes historical data sourced from official voyage reports and technical specification documents of the liquefied natural gas carrier. Data collection focuses on the delivery route from the Tangguh production facility to the Jawa Satu Floating Storage Regasification Unit. Operational data, including cargo volume, ambient temperature, and voyage duration, are presented in detail in **Table 1**.

Table 1 Operational data and vessel

Parameter	Data
Capacity	135.000 m ³
No. of Tank	5
Inner Temperature	-160°C
Ambient Temperature	30°C
Voyage Duration	6 Days
Initial Volume	132.872 m ³
Final Volume	130.054 m ³
HV Loaded	55,0133 MJ/kg
HV Delivered	55,1298 MJ/kg
Density Loaded	431,42 kg/m ³
Density Delivered	431,70 kg/m ³
LNG Price	\$6/MMBtu
Energy Loaded	2.988.820 MMBtu
Energy Delivered	2.921.079 MMBtu

In addition to operational data, this analysis also requires the design specification data of the storage tanks. This data encompasses the tank capacity and the thermal conductivity properties of each material lining the vessel's tank walls. The details of these design parameters can be found in **Table 2**.

Table 2 Tank component characteristics and materials

Parameter	Value	Source
Latent Heat of Vaporization LNG	508,82 kJ/kg	[1]
Aluminum Shell Thermal Conductivity	250 W/mk	[2]

Aluminum Shell Thickness	0,04 m	[2]
Perlite Insulation Thermal Conductivity	0,031 W/mk	[2]
Perlite Insulation Thickness	0,2 m	[2]
Polyurethane Insulation Thermal Conductivity	0,28 W/mk	[2]
Polyurethane Insulation Thickness	0,2 m	[2]
Inner Convection Coefficient	50 W/m ² °C	[2]
Outer Convection Coefficient	80 W/m ² °C	[2]

To provide a clearer representation of the thermal resistance within the tank wall, the configuration of the insulation materials, from the innermost layer in direct contact with the fluid to the outermost layer exposed to the marine atmosphere, is illustrated in **Figure 1**.

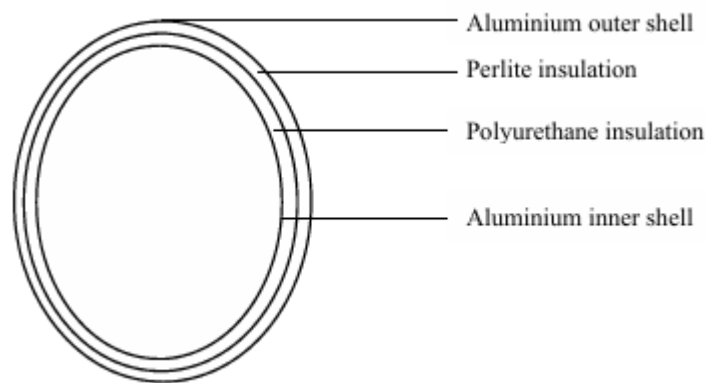


Figure 1 Spherical tank internal structure material arrangement

The arrangement of the tank wall layers and the heat resistance model are visualized through the thermal resistance in **Figure 2**. This structure illustrates the series of thermal resistances that environmental thermal energy must pass through before reaching the cargo inside the tank. This series includes the convection resistance at the inner surface, the conduction resistance across each layer of the composite insulation material, and the convection resistance at the outer surface exposed to the ambient air.

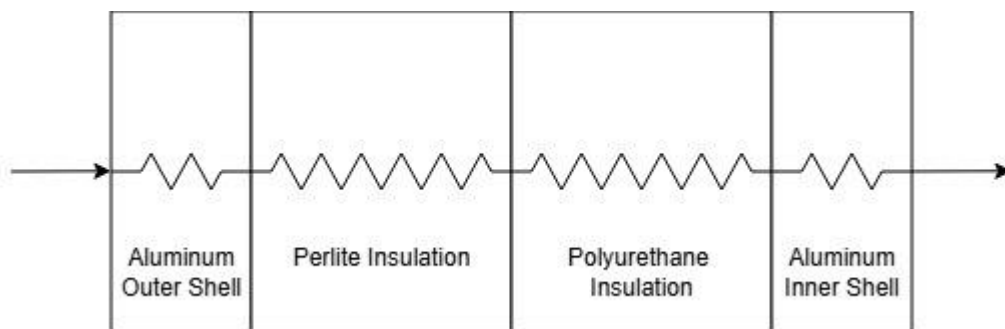


Figure 2 Thermal resistance model of the tank wall

The vessel utilized in the liquefied natural gas transportation process is illustrated in **Figure 3**. This vessel is a liquefied natural gas carrier equipped with a spherical or Moss-type storage tank system.



Figure 3 Liquefied natural gas carrier (LNGC)

5.2. Data Analysis

Table 3 Result analysis

Parameter	Value
Volume per Tank	27.000 m ³
Tank Radius	18,61 m
Overall Heat Transfer	116.852,15 W
BOG Theoretical	45.99 m ³ /day
BOR Theoretical	0.17 %/day
BOG Volume Theoretical	1379.77 m ³
BOG Volume Actual	2818 m ³
BOG Actual	93,93 m ³ /day
BOR Actual	0,35 %/day
Delta BOR	0,18 %/day
Energy Loss Theoretical	31.036,55 MMBtu
Energy Loss Actual	63.388,01 MMBtu
Cost Loss Theoretical	\$ 186.219,28
Cost Loss Actual	\$ 380.328,05
Financial Loss	\$ 194.108,77

Volume per Tank

$$V_{per\ Tank} = \frac{135.000\ m^3}{5} = 27.000\ m^3$$

Tank Radius

$$r_i = \sqrt[3]{\left(\frac{3 \times 27.000\ m^3}{4\pi}\right)} = 18,61\ m$$

Overall Heat Transfer

$$\dot{Q} = \frac{-4\pi(-160^{\circ}\text{C} - 30^{\circ}\text{C})}{\frac{1}{18,61^2 \times 50} + \frac{1}{250} \left(\frac{1}{18,61} - \frac{1}{18,65} \right) + \frac{1}{0,031} \left(\frac{1}{18,65} - \frac{1}{18,85} \right) + \frac{1}{0,28} \left(\frac{1}{18,85} - \frac{1}{19,05} \right) + \frac{1}{250} \left(\frac{1}{19,05} - \frac{1}{19,09} \right) + \frac{1}{19,09^2 \times 80}}$$

$$\dot{Q} = 116.852,15 \text{ W}$$

Theoretical BOG

$$BOG_{theo} = \frac{116.852,15 \text{ W} \times 3600 \times 24}{508.820 \text{ J/kg} \times 431,42 \text{ kg/m}^3} = 45,99 \text{ m}^3/\text{day}$$

Theoretical BOR

$$BOR_{theo} = \frac{45,99 \text{ m}^3/\text{day}}{27.000 \text{ m}^3} \times 100\% = 0,17\%/\text{day}$$

Theoretical BOG Volume

$$V_{BOG} = 45,99 \text{ m}^3/\text{day} \times 6 \text{ days} \times 5 \text{ tanks} = 1.379,77 \text{ m}^3$$

Actual BOG Volume

$$V_{BOG} = 132.872 \text{ m}^3 - 130.054 \text{ m}^3 = 2.818 \text{ m}^3$$

Actual BOG

$$BOG_{act} = \frac{2.818 \text{ m}^3}{6 \text{ days} \times 5 \text{ tanks}} = 93,93 \text{ m}^3/\text{day}$$

Actual BOR

$$BOR_{act} = \frac{93,93 \text{ m}^3/\text{day}}{27.000 \text{ m}^3} \times 100\% = 0,35\%/\text{day}$$

Delta BOR

$$\Delta BOR = 0,35\%/\text{day} - 0,17\%/\text{day} = 0,18\%/\text{day}$$

Theoretical Energi Loss

$$E = \frac{1.379,77 \text{ m}^3 \times 431,42 \text{ kg/m}^3 \times 55,0133 \text{ MJ/kg}}{1055,12} = 31.036,55 \text{ MMBtu}$$

Actual Energi Loss

$$E = \frac{2.818 \text{ m}^3 \times 431,42 \text{ kg/m}^3 \times 55,0133 \text{ MJ/kg}}{1055,12} = 63.388,01 \text{ MMBtu}$$

Theoretical Cost Loss

$$\text{Cost} = 31.036,55 \text{ MMBtu} \times \$6 = \$186.219,28$$

Actual Cost Loss

$$\text{Cost} = 63.388,01 \text{ MMBtu} \times \$6 = \$380.328,05$$

Financial Loss

$$\text{Loss} = \$380.328,05 - \$186.219,28 = \$194.108,77$$

The data analysis procedure begins by identifying the geometric parameters of the spherical tank based on the specification data in **Table 2**. The cargo capacity per tank is calculated using Equation (1) by dividing

the total vessel capacity by the number of tanks, resulting in a maximum volume of 27,000 m³ per tank. Based on this volumetric capacity, the inner radius of the tank is evaluated using Equation (2) and Equation (3), yielding a value of 18.61 m.

The subsequent analytical step involves evaluating the capacity of the tank's composite wall layer to resist heat propagation from the external environment. By substituting the material thickness and thermal conductivity data from **Table 2**, along with the temperature difference between the ambient air and the liquid from **Table 1**, the theoretical heat transfer rate is calculated using Equation (4). The calculation results in **Table 3** indicate that the ship's tank structure receives a continuous thermal energy flow of 116.9 kW penetrating into the cargo space.

This theoretical thermal energy flow triggers the evaporation of the cryogenic liquid. Through the application of Equation (5), the incoming thermal energy is converted into a projected theoretical gas volume of 45.99 m³ per day for each tank. This volume figure is then divided by the maximum tank capacity using Equation (6) to obtain a theoretical boil-off rate percentage of 0.17% per day.

The operational performance of the vessel at sea is then evaluated using the actual volume shrinkage data from **Table 1**. The total physical cargo volume lost during the voyage was recorded at 2,818 m³. Utilizing Equation (7) and Equation (8), the actual average of wasted gas is calculated by dividing the total shrinkage by the voyage duration and the number of tanks. The calculation results demonstrate that the actual evaporation rate reaches 93.93 m³ per day for each tank. This figure is equivalent to an actual field boil-off rate percentage of 0.35% per day, in accordance with the calculation results from Equation (9).

The thermal efficiency of the tank is critically analyzed by comparing the actual field performance with the theoretical performance on paper. Through Equation (10), the difference between the actual and theoretical boil-off rate percentages reveals an inefficiency of 0.18% per day. This deviation provides evidence that the cargo evaporation process has exceeded the capacity of the initial thermal insulation design.

The impact of this inefficiency is evaluated from a financial perspective. By using Equation (7), the difference in evaporation volume between the actual and theoretical conditions is converted into energy units using Equation (11) and the standard cargo heating value from **Table 1**, resulting in an unsupplied energy figure of 32,351.46 MMBtu. The final stage of the analysis employs Equation (12) to multiply this lost energy by the standard liquefied natural gas contract price [10]. This final calculation quantitatively proves that the deviation in the tank's thermal efficiency performance causes a financial loss of \$194,108.77 for a single voyage schedule.

5.3. Discussion

The calculation results from the data analysis stage reveal that the vessel's tank thermal resistance system does not operate optimally in accordance with theoretical calculations. The theoretical heat transfer rate calculation demonstrates that the designed thickness and type of the vessel's insulation material should be capable of maintaining the boil-off rate at 0.17% per day. However, actual evaporation data indicates that the boil-off rate increased to 0.35% per day. This actual figure, exceeding the tolerance limit, proves a decline in insulation efficiency during the voyage.

The thermal efficiency deviation of 0.18% per day can be attributed to several factors, such as material degradation and operational dynamics at sea. The most plausible primary technical factor is the degradation or decline in the thermal resistance capacity of the tank wall's insulation layer. As the vessel's operational age advances, the thermal insulation material may experience structural fatigue, causing external heat to propagate more rapidly into the cargo hold. In addition to material aging, oceanic wave movement during the voyage induces fluid sloshing within the spherical tank. This sloshing expands the contact area between the cryogenic

fluid and the warmer tank walls, thereby accelerating the heat absorption and evaporation processes compared to stationary conditions.

In the context of the company's supply chain resilience, the evaporation volume discrepancy exceeding theoretical calculations results in an energy loss of 32,351.46 MMBtu. The LNG that should have been delivered to the FSRU Jawa Satu receiving facility for conversion into power plant fuel is instead lost or consumed at sea due to the vessel's inadequate insulation quality.

The subsequent impact of this technical inefficiency culminates in substantial financial losses. The financial loss of \$194,108.77 for a single voyage constitutes an additional cost burden that will directly elevate the basic cost of electricity provision. Based on this deviation data, the company possesses a substantive rationale to demand accountability from the vessel service provider, to review the penalty clauses within the charter agreement, or to mandate the vessel owner to promptly execute comprehensive maintenance and repairs on the cargo tank insulation structure.

4. Conclusion

The evaluation of thermal resistance modeling proves that the composite layer structure of the spherical tank wall still permits thermal energy from the ambient air to propagate into the cargo. The theoretical heat transfer rate penetrating the vessel's storage system throughout the voyage is 116.9 kW.

The performance of the vessel's heat insulation capacity indicates a significant performance deviation or operational inefficiency. The theoretical daily boil-off rate percentage is calculated at 0.17% per day, while the actual daily boil-off rate percentage in the field reaches 0.35% per day. This confirms the existence of an inefficiency deviation of 0.18% per day, attributed to the suspected degradation of the insulation material quality.

This thermal performance deviation directly results in a commercial energy supply failure of 31,351.46 MMBtu, which evaporates during the voyage. This inefficiency burdens the supply chain resilience of the company, generating a financial loss of \$194,108.77 for every single voyage cycle on the specified route.

Acknowledgment

The authors express their sincere gratitude to the Department of Mechanical Engineering at Universitas Pertamina for their assistance and guidance as mentors during the internship program. Their continuous support was instrumental in the successful execution of this research. This practical work also fulfilled a mandatory academic requirement for the authors' university graduation.

References

- [1]. T. Włodek, "Analysis of boil-off rate problem in Liquefied Natural Gas (LNG) receiving terminals.," in *IOP Conference Series: Earth and Environmental Science*, Jan. 2019, vol. 214, no. 1. doi: 10.1088/1755-1315/214/1/012105.
- [2]. "Predicting the Produced Boil off Gas in the Moss Spherical Liquefied Natural Gas (LNG) Vessel," 2013.
- [3]. M. Miana, R. Legorburo, D. Díez, and Y. H. Hwang, "Calculation of Boil-Off Rate of Liquefied Natural Gas in Mark III tanks of ship carriers by numerical analysis," *Appl. Therm. Eng.*, vol. 93, pp. 279–296, Jan. 2016, doi: 10.1016/j.applthermaleng.2015.09.112.
- [4]. D. Dobrota, B. Lalić, and I. Komar, "Problem of Boil - off in LNG Supply Chain," *Transactions on Maritime Science*, vol. 2, no. 2, pp. 91–100, Oct. 2013, doi: 10.7225/toms.v02.n02.001.
- [5]. "TERMO - Moran M. J., Saphiro N. M. - [8th Ed, 2014] Fundamentals of Engineering Thermodynamics".
- [6]. M. Kalikatzarakis, G. Theotokatos, A. Coraddu, P. Sayan, and S. Y. Wong, "Model based analysis of the boil-off gas management and control for LNG fuelled vessels," *Energy*, vol. 251, Jul. 2022, doi: 10.1016/j.energy.2022.123872.

- [7]. “Theodore L. Bergman, Adrienne S. Lavine, Frank P. Incropera, David P. DeWitt - Fundamentals of Heat and Mass Transfer, 7th Ed”.
- [8]. I. M. Ariana, A. Z. M. Fathallah, H. A. Zaky, and T. A. Akbar, “Boil of Gas Formation Rate Analysis in LNG Storage Tank on LNG Fuelled Harbor Tug Vessel,” in *IOP Conference Series: Earth and Environmental Science*, 2024, vol. 1423, no. 1. doi: 10.1088/1755-1315/1423/1/012040.
- [9]. C. Migliore, C. Tubilleja, and V. Vesovic, “Weathering prediction model for stored liquefied natural gas (LNG),” *J. Nat. Gas Sci. Eng.*, vol. 26, pp. 570–580, Sep. 2015, doi: 10.1016/j.jngse.2015.06.056.
- [10]. Menteri Energi dan Sumber Daya Mineral Republik Indonesia, “Pedoman Penetapan Serta Evaluasi Pengguna dan Harga Gas Bumi Tertentu di Bidang Industri dan di Bidang Penyediaan Tenaga Listrik Bagi Kepentingan Umum,” 2022.

Biographies of Authors



Rafi Adisasmito Syamsudin An active undergraduate student of Mechanical Engineering at Universitas Pertamina, with academic interests focusing on the energy, heat transfer and industrial fields. I have organizational and teaching experience as the Academic Division Head of the Mechanical Engineering Student Association and as a Practicum Assistant. I have a commitment to utilizing my technical proficiencies backed by a strong academic foundation to contribute effectively to team-driven and problem-solving environments.



Dr. Fayza Yulia is an Assistant Professor in Mechanical Engineering at Universitas Pertamina, Indonesia, holding undergraduate through doctoral degrees from Universitas Indonesia. Her research focuses on adsorption-based carbon capture and storage (CCS) and sustainable building energy systems, with current work on Bio-MOF-based energy storage, energy system modelling and optimization, and HVAC-BIM-integrated thermal storage for low-carbon buildings.