

Numerical Study and Comparison of Various Types of Insulation Systems in Terms of Controlling Heat in Cryogenic Tanks

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Abstract: This study aimed to numerically compare the thermal performance of various insulation systems and also to maximally reduce the ambient heat transferred to the cryogenic fluid stored in a tank. For this, we use such techniques as vacuum insulation, cryogenic cold shell around the main tank, multilayer insulation, two-sided insulation around the external tank, polyurethane insulation and anti-radiation coatings. The findings suggest that the thermal emissivity and vacuum pressure as well as the vacuum thickness are critical parameters affecting the inlet thermal emission to the main tank. The use of a cryogenic cold shell in a vacuum space is shown to cause a significant drop in the inlet thermal emission to the tank. Considering the separator material inside the various layers of different thicknesses, the multi-layer insulation is also modeled. In this situation, a layer density of 26 cm^{-1} has been regarded as the best layer density. Finally, while modeling several examples of composite insulation, the intra-tank evaporation rate for each insulation and the feasibility of using the mentioned insulation in Iran are provided.

Keywords: Vacuum insulation, Cold vapor shell, Multilayer Insulation, Polyurethane Insulation, Anti-Radiation Coatings, Intra-tank Evaporation Rate

1. Introduction

Today, controlling heat in tanks containing cryogenic fluid is very much significant. Cryogenic fluids are referred to fluids with very low boiling points, meaning that these fluids can experience great volume variations at ambient temperatures. To store fluids of very low boiling temperatures, a specifically-insulated tank is required and care should be taken in this case [1]. It is known the lower the heat transferred to the pressure tank and the cryogenic fluid, the lower the evaporation rate. A tank with this feature is considered to be a reliable tank to store cryogenic fluids [2]. The heat transferred to the cryogenic fluid and the creation of a biphasic fluid are performed by three mechanisms: conduction, convection, and radiation. It is common to use vacuum insulation systems to reduce conductive and convective heat transfer [3] while thermal shields are the best option to reduce radiative heat transfer [4-5]. A double-walled vacuum tank was first used by James Doyer in 1898 to store liquid hydrogen [6-8]. It is important not to disregard the radiative heat transfer in any way when analyzing and selecting the type of insulation, for it may cause significant errors in the design and selection of the best insulation. Therefore, it is advised to focus on all three heat transfer mechanisms when examining and selecting the type of insulation. Another method to reduce conductive and convective heat transfer is to use multilayer insulation to store energy. Mehdi Ma'reft and Ismaili demonstrated in a study that to have a desirable design, separators in the cold boundary should be selected from materials of low thermal conductivity, and it was necessary to increase the layer density in the vicinity of the warm boundary [9]. Theoretical and experimental works to reduce the conductive and radiative heat transfer is led by researchers such as Clair [10].

In most of these studies, high optical thickness approximation has been used to model radiation [11, 12]. In another study by Mehdi Ma'reft and Arash Ismaili, the two-flux approximation was applied to simulate radiation, which is valid for all ranges of light thickness with radiative properties of solid walls being taken into account in it [9]. In an experimental work by Mills, water vapor and other residual gases were removed between two internal and external tanks by a vacuum pump. For a high vacuum zone to be created, the space between the two walls was first washed once with nitrogen gas and again with helium gas, and then, a vacuuming operation was carried out.

The vacuum is 10^{-4} Pa before the tank is filled with liquid nitrogen and 10^{-5} Pa after being filled with liquid nitrogen. When pumping starts, the amount of thermal leakage from the insulation is calculated. At the onset, the amount of leakage is high due to the presence of gases between the two walls, with the thermal leakage gradually decreasing. In a steady state, when the vacuum is complete, the thermal leakage, after a while, reaches a constant value of $0.227 \frac{\text{W}}{\text{m}^2}$ [13].

Saberi Moghadam et al. [14] examined the performance of various insulation systems in storage tanks of cryogenic propulsion systems. They concluded that in an atmospheric state, the conductivity coefficient of helium was higher than that in air, but in high vacuum conditions, the conductive heat transfer of helium was lower. Therefore, under high vacuum conditions, the conductive heat transfer mechanism is different from that in atmospheric conditions. In a study, Nassett et al. [15] examined multilayer insulation of hydrogen tanks. This study provides some suggestions for insulating these tanks. In an experimental work, Dai et al. [16] designed and tested multilayer insulation of space tubes. This study also provides a strong insulation system for space tubes. A new layer of multilayer insulation has been introduced for space missions. This study was performed by Takashi

Miyamoto et al. [17]. In this research, a new type of multilayer insulation with new separators is demonstrated. In this research, separator material is discussed. Other recent studies on the insulation of tanks containing cryogenic fluids includes those done by Wang Li et al. [18]. They demonstrated the pressure behaviors and performance of the cryogenic tank on both sides of the insulation around the main tank when discharging the liquid. In this research, temperature distribution and energy distribution inside the tank were measured when the tank was discharged. They concluded that the internal insulation layer technique was an effective approach to improve the compression impacts of the discharge process under pressure; also, a relatively thin layer of foam was found to significantly reduce the heat transfer to the tank. As the tank temperature rises, the internal insulation layer can play a more efficient role.

Previously, most research focused on a type of insulation and no integrated system was applied. Moreover, in the insulation systems studied, the features of insulation (i.e., the impacts of vacuum pressure, vacuum thickness, range of thermal radiation and thermal conductivity) are not addressed. No other references clearly mention a combined state of cryogenic cold shell along with vacuum and other tank insulations in a larger scale. Also, the fluid evaporation rate in tanks for each insulation and the feasibility of using the mentioned insulation systems in Iran have been examined; therefore, in this research, the work performed by others will be expanded with a different approach for a combined state along with varying conditions and diversity.

2- Geometry and boundary conditions under study

Since the geometries under study are myriad, we just suffice to discuss the basics in here. Ten insulation systems have been examined to reduce heat emission to the main tank. The schemas of the problems under study and relevant conditions are provided in Tables 1 to 10 and Figures 1 to 4. Insulation models are corrected and selected in a way that a non-insulated tank is first modeled and the conductive, radiative, and convective heat transfer is specified. Next, proposals are made to reduce the conductivity of the vacuum insulation and it is noted that the thermal emission drastically decreases but there is still thermal radiation. New and combined insulations are modeled to reduce thermal radiation along with vacuum insulation. Modeling is also performed for both constant temperature and ambient conditions. The radius of the main tank (spherical and cylindrical) is 0.49m under all conditions. Concerning all conditions, the internal tank temperature was 77° K, the internal tank is made of 304 stainless steel, the thickness of the main tank wall is 0.02 m (thin wall), the thickness of the external tank is 0.01 m with the external tank material made of 304 stainless steel. In the present article, modeling has been demonstrated for two spherical (constant temperature conditions) and cylindrical tank (ambient conditions) states.

2-1- Constant temperature conditions (spherical tank)

Table 1: Features and conditions of vacuum-insulated spherical tank (constant temperature conditions)

Main tank material	Insulation thickness (m)	External tank temperature (K)	Vacuum pressure (Pa)
Aluminum-coated stainless steel	Variable	293	10 ⁻⁵

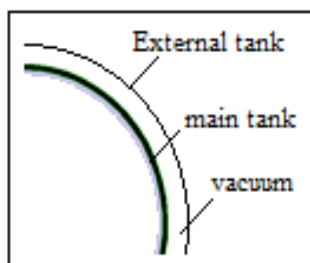


Figure 1: Schematic vacuum-insulated tank

Table 2: Specifications of a cryogenic cold shell-insulated and vacuum-insulated tank

Size of each vacuum layer (m)	Cold shell (m) thickness	temperature of the inlet vapors to the shell (K)	Inlet flow rate vapor to the shell (kg/s)	Vacuum pressure (Pa)
0.03	0.05	85	0.00002	10 ⁻⁵

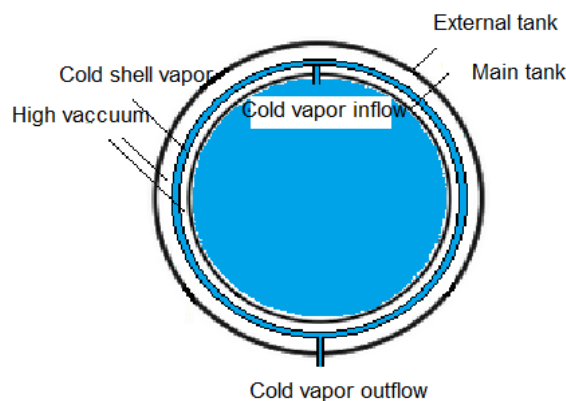


Figure 2: Schematic of a cold vapor shell tank

Table 3: Common conditions with details of ten-layer insulation

Thickness of each insulation (mm) layer	Total thickness of the (m) insulation	Separator material	Material of insulation layers	Temperature of the last insulation layer (K)	Separator Diameter (mm)
11.11	.01	Ceramic	AL	293	4.25

Note that the value of the vacuum pressure between the layers in a multilayer insulation in Table 3 is $10^{-5} Pa$, and the appropriate thickness for each insulation layer according to the reference [17] is $0.8 \mu m$.

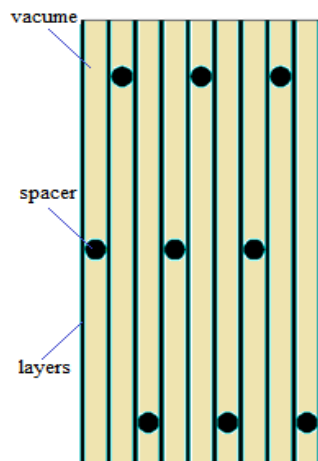


Figure 3: Schematic of ten-layer insulation

Table 4: Impacts of number of layers (variable thickness) on insulation performance

Number of insulation layers	Size of each insulation layer (mm)	Separator material	Temperature of the last insulation layer (K)	Separator diameter (mm)	Insulation layer material
Variable	0.1	Ceramic	293	0.43	AL

As in Table 4, the vacuum pressure between the layers of the multilayer insulation is $10^{-5} Pa$ and the appropriate thickness for each insulation layer consistent with reference [17] is $0.8 \mu m$.

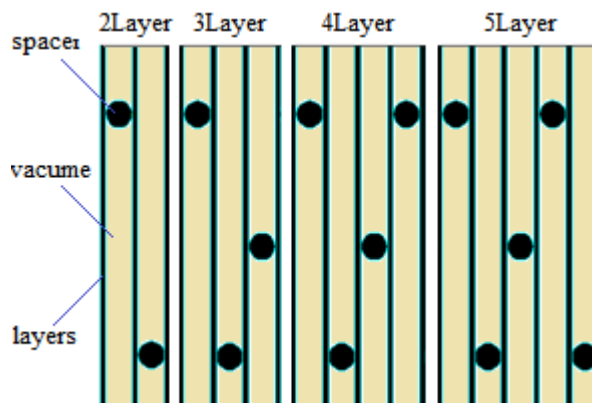


Figure 4: Model geometry for Table 4

Table 5: Impacts of the number of layers (constant thickness) on insulation performance

Number of layers in a constant thickness (m)	Total insulation size (m)	Separator material	Temperature of the last layer of insulation (K)	Separator diameter (m)	Material of insulation layers
Variable	.01	Ceramic	293	.00043	AL

2-2 Ambient conditions (cylindrical tank)

Reference [14] demonstrates the emissivity of 304 stainless steel and aluminum at different temperatures. In Tables 6, 7 and 8, the material of the main and the external tank is 304 stainless steel. The vacuum pressure used is also $10^{-4} Pa$.

Table 6: Features and boundary conditions of the vacuum-insulated tank

Ambient temperature (K)	Ambient convective heat transfer ($\frac{W}{K.m^2}$) coefficient	Internal wall absorption coefficient	External wall absorption coefficient	Vacuum insulation (m) thickness
293	10	.023	.017	Variable

Table 7: Use of polyurethane and anti-radiation insulations around the external tank

Reflective insulation absorption coefficient	Thermal conductivity coefficient of the last layer ($\frac{W}{m.K}$)	Size of each vacuum insulation (m) layer	Ambient temperature (K)	Ambient convective heat transfer coefficient ($\frac{W}{K.m^2}$)	Internal wall absorption coefficient	External wall absorption coefficient	Thickness of the last insulation (m) layer
.05	.03	.09	293	10	.13	.17	.01

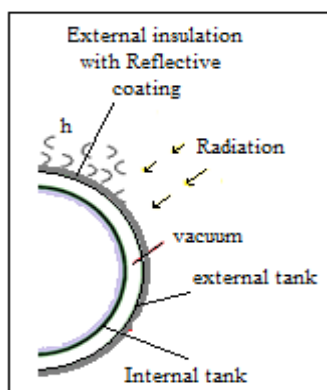


Figure 5: Geometry of insulation used in Table 7

Table 8: Use of combined vacuum, cold shell vapor vacuum and of two-sided insulation techniques around the main tank

Thickness of each vacuum insulation layer (m)	Anti-radiation absorption coefficient	Absorption coefficient of external foam with reflective coating	Thermal conductivity coefficient of the internal foam layer $\frac{w}{m.K}$	Thermal conductivity coefficient of the external foam layer $\frac{w}{m.K}$	Thickness of the internal foam layer(m)	Thickness of the external foam layer (m)	Thickness of cold vapor shell (m)
.03	.05	.023	.03	.04	.01	.015	.03

The convective heat transfer coefficient and ambient temperature are provided as in Table 8. It is also assumed that the temperature of the inlet vapors to the shell is 85°K and the inlet flow rate is 0.00002 kg/s. A temperature of 85 °K is used consistent with reference [14].

3- Numerical solution method

The available equations are solved numerically using Ansys Fluent software based on finite volume method. The second-order forward equations are explicitly used to solve the energy equation. Symmetry conditions have been used to reduce the computational costs. The output results for thermal emission take the form of $\frac{w}{m^2}$ which can be independently used of the tank geometry. In this study, the convergence criterion is shown to be 10^{-6} . The surface-to-surface model is used to model the radiation and the Boussinesq’s model to apply the free (natural) convection outside the tank.

4- Grid independence

Several elements are studied to examine the grid independence.

Table 6: Grid independence results for vacuum insulation with a thickness of .09m

	Grid	Total inlet heat to the main tank ($\frac{w}{m^2}$)
1	29943	14.951
2	48808	15.451
4	65326	16.951

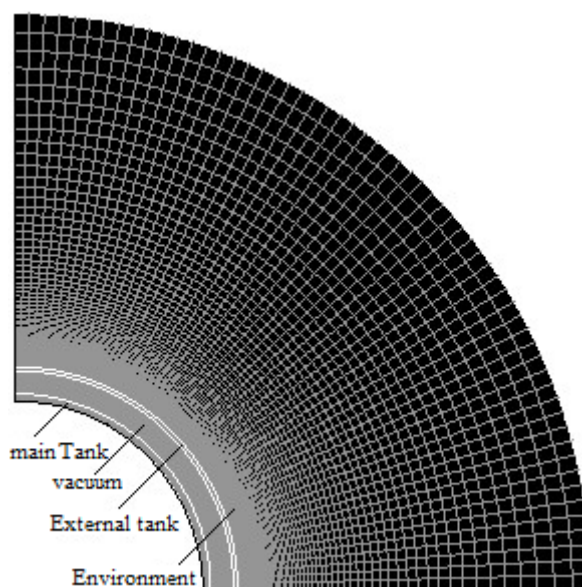


Figure 5: Meshing of vacuum insulation in ambient conditions

The grid used in this section pertains to ambient conditions and vacuum insulation, whose geometry and conditions are given in Table 6. It is seen that as the number of nodes increases from 48804 to 65326, no concrete

changes in the results are noted. Therefore, to avoid wasting time and reduce costs, 48,804 nodes are considered to be the final analysis (Table 11).

5- Results validation

5-1: Validation of non-insulated tank results

To validate the inlet thermal emission to the internal wall, we use the Fourier heat conduction law for the conditions in Table 1. In the meantime, the external wall temperature of 293°K and the internal wall temperature of 77°K were validated. the Fourier heat conduction law for a spherical tank is according to $q = \frac{4\pi k\Delta T}{(\frac{1}{r_1} - \frac{1}{r_2})}$. Fourier heat conduction law yields a thermal emission of 559220.52 W to the tank.

Table 12: Results validation for non-insulated tank (Specifications are provided in Table 1)

Exte mal wall tem pera ture (K)	Inte mal wall tem pera ture (K)	Mo del ing res ults (W)	Fou rier res ult (K)	E rr or (%)
293	77	60 02 67. 1	559 220 .55	6. 8

5-2: Validation of vacuum insulation results

At cryogenic temperatures, even if a vacuum system is available, heat is again transferred by the radiative mechanism and is significant. The rate of radiative heat transfer between the surfaces of two concentric bodies without an interference of any material is obtained from Equation 12 [14].

$$Q_r = .53 * 10^{-8} \left(\frac{w}{k.ft^2} \right) * \frac{e_1 e_2}{e_2 + \frac{A_1}{A_2} (1 - e_2) e_1} \times (T_2^4 - T_1^4) A_1$$

In equation 12, Q_r is the radiative heat transfer rate in watts, e_1 the internal surface emissivity, e_2 is the external surface emissivity whose value varies according to the material and surface temperature, T_1 is the internal surface temperature, T_2 is the external surface temperature in Kelvin, A_1 internal tank area and A_2 external tank area in terms of ft². Table 13 provides the results for insulation with a thickness of 8 cm (thermal radiation imparted to the external wall of the internal tank).

Table 13: Validation of results for 8cm thick insulation

Internal tank material	Internal tank (K) temperature	Main tank (K) temperature	Radiation for (W) simulation	Radiation for equation 12(W)	error (%)
Aluminum-coated steel	293	77	27.5	29.01	5.1

In reference [14], the emissivity of 304 stainless steel and aluminum at different temperatures are given. Internal and external tank radii are adopted as .51m and .59m, respectively.

5-3: Validation of multilayer-insulated tank results

In the first stage, modeling of the aluminum-made ten-layer insulation with a total thickness of one centimeter has been used for validation as demonstrated in Table 4. We consider a ten-layer insulation with the conditions in Table 4, the geometry of which is illustrated in Figure 2. In this section, the results of ten-layer insulation are validated by laboratory findings. Note that the diameter of retaining material and the thickness of the layers is used in line with the standards used in credible articles. Table 14 compares the simulation results with the laboratory findings found by Peng Li and Huier Cheng [19]. Figure 6 illustrates the temperature distribution in this ten-layer insulation.

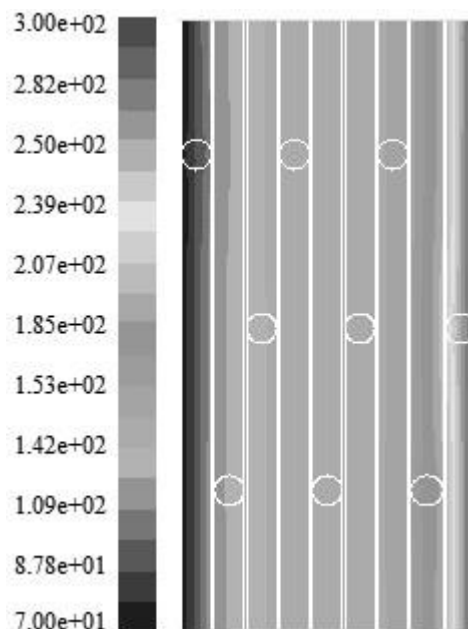


Figure 6: Temperature distribution for ten-layer insulation

Table 14: Comparison of simulation results with laboratory findings (Peng Li and Huier Cheng) for ten-layer insulation

Thermal conductivity for simulation (w)	Thermal conductivity for laboratory results (W)	Error (%)
0.118	0.11	6.7

6- Discussing the results

6-1- Spherical tank

6-1-1- Results for non-insulated tank

The thermal emission imparted to the internal wall of the tank by thermal conductivity is 600267.1 watts. Note that we used constant temperature conditions for analysis with the convective heat transfer not considered. The boundary conditions are provided in Table 1. Because the temperature of vapors and liquids inside the tank are equal, it provides symmetrical and constant temperature conditions for the wall and constant temperature conditions can yield a good approximation.

6-2: Vacuum insulation to reduce conductive heat transfer to the tank

Citing Table 2 and Figure 1, the results for vacuum insulation are provided. Figure 7 illustrates the heat emission to the internal tank. Both tanks in this state are made of 304 stainless steel. Increased vacuum thickness can reduce the conductive heat transfer. Because the surface of the two tanks has increased along with increased vacuum insulation thickness, the amount of radiative heat transfer increases also. Figure 8 demonstrates the thermal emission of vacuum insulation to the internal tank of aluminum-coated 304 stainless steel and the external tank of 304 stainless steel. Adding a coating of aluminum with a lower emissivity causes a more significant drop in radiation. Increased thickness of the vacuum insulation (increased surface area of the external tank) reduces the conductive heat transfer, on the one hand, and increases the radiative heat transfer, on the other. For an internal tank of stainless steel, the total heat transfer decreases and then increases as the thickness increases. However, this

does not hold true for aluminum-coated internal tanks. For the aluminum-coated internal tank, one can say that as thickness increases (increased area of the external tank), the radiative heat transfer increases, followed by decreased conductive transfer. Considering the higher emissivity of steel compared to aluminum, it is concluded that in a range of thicknesses, the rate of radiative volume is greater than the rate of conductivity, and the decrease in thermal conductivity from some certain thickness onwards cannot offset the increase in thermal radiation, which causes thermal emission to decrease first and then increase as thickness increases. For aluminum-coated tanks, because the emissivity is low, the amount of increased radiation is not enough to offset the decrease in thermal conductivity, while increased thickness now leads to a decrease in thermal emission.

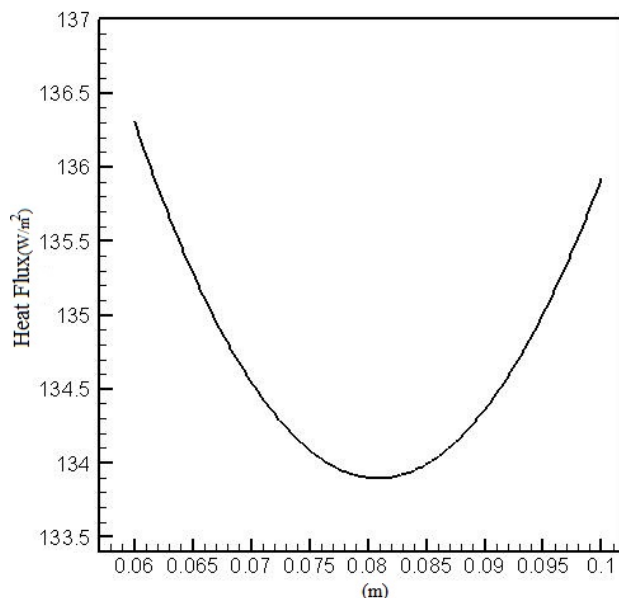


Figure 7: Total thermal emission variations in thickness for 304 stainless steel internal tank

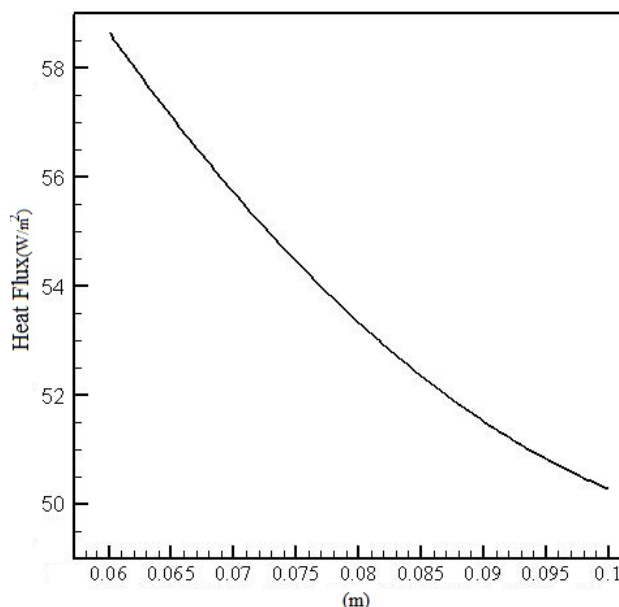


Figure 8: Total thermal emission variations in thickness for aluminum-coated 304 stainless steel internal tank

6-3: Impacts of vacuum pressure on the thermal emission to the external wall of the internal tank

It is natural that the lower the vacuum pressure, the less the distance between the gas molecules and the lower the thermal conductivity. On the other hand, as the remaining gases in the chamber are discharged and a high vacuum is created, the path for a smoother radiation is provided, causing an increase in radiation. The question is: Does high vacuum reduce heat transfer?

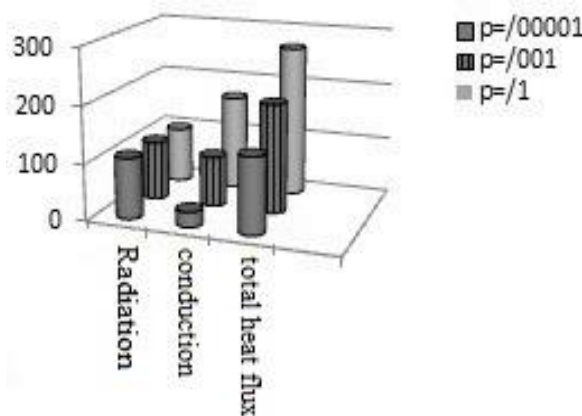


Figure 9: Radiative and conductive thermal emission imparted to the external wall of the internal tank in watts for various vacuum pressures

Figure 9 illustrates the radiative and conductive thermal emission for three different pressures. It is clear that as the vacuum pressure decreases, the conductive heat transfer decreases while the radiative heat transfer increases slightly. It is important to note that the rate of decrease in thermal conductivity is higher than the increase in radiation, and generally, a decrease in vacuum pressure is followed by a decrease in heat transfer imparted to the tank.

6-4: Cryogenic cold shell insulation around the main tank to reduce radiative heat transfer

Here, in this stage, the main tank, instead of exchanging heat in a direct exchange with the ambient, does this via a cold shell. By coating the main tank with shell nitrogen and liquid oxygen, the insulated walls are cooled for about 85°K, reducing radiative heat transfer. Using this type of insulation system causes almost all the heat at room temperature emitted from the surface to evaporate the liquid oxygen, reducing the total heat emission imparted to the tank by 245 w/m². The conditions stated in Table 3 and Figure 1 are used for insulation. Two vacuum insulation layers are used to reduce conductive and convective heat transfer, and using liquid oxygen shell is to reduce radiative heat transfer. In this method, we suppose that the cryogenic cold shell temperature remains constant. For this end, cryogenic cold vapors inside the tank are conducted into the cold shell through tubes as this cold vapor takes some radiative and conductive energy which is sent to a chamber for liquefaction by a pump. After liquefaction, oxygen will be returned to the main tank.

6-5: Multi-layer thermal insulations to reduce conductive and radiative heat transfer

6-5-1: Impacts of number of layers (Variable thickness) on insulation performance

The main goal is to examine the number of layers on the thermal flux and temperature profile. The boundary conditions and data taken for the simulation are provided in Table 5 and Figure 3. Increased number of layers causes the radiation imparted to the cold part of the tank to come down; on the other hand, as the number of layers increase, heat arises outside the tank in areas farther from the cold area, contributing to reducing heat transfer through conductivity. Table 15 illustrates the thermal emissions imparted to the cold parts for the insulation mentioned. It is noted that as the number of layers increase, the total thermal emission imparted to the tank decreases while the insulation thickness increases.

Table 15: Thermal emissions to the cold part of the tank

Number of layers	Total insulation thickness (M)	Main tank thermal emission ($\frac{w}{m^2}$)
3	0.0002242	10.25
4	0.0003363	7.15
5	0.0004484	5.45
6	0.0005605	3.21
7	0.0006726	2.61

6-5-2: Impacts of number of layers (Constant thickness) on insulation performance

Diagrams in Figures 10 and 11 demonstrate conductive and radiative thermal emissions by layer density. It is noted that as layer density increases, the passing radiative layer decreases, but conversely, thermal conductivity increases. An increase in the number of layers helps these layers act as heat shields against radiation and thus reduce radiative heat emission.

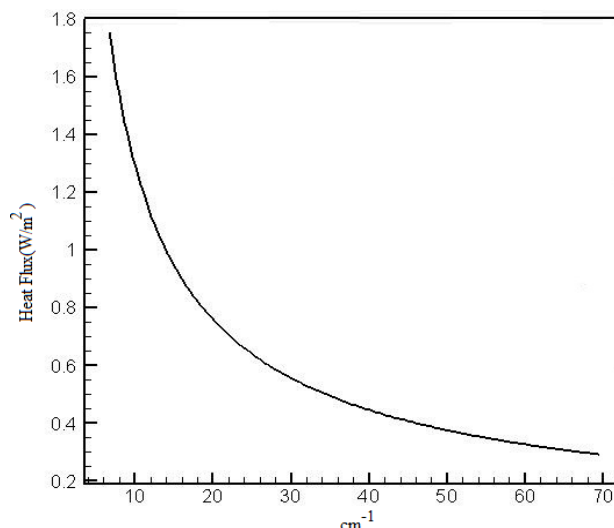


Figure 10: Radiative thermal emission in terms of layer density

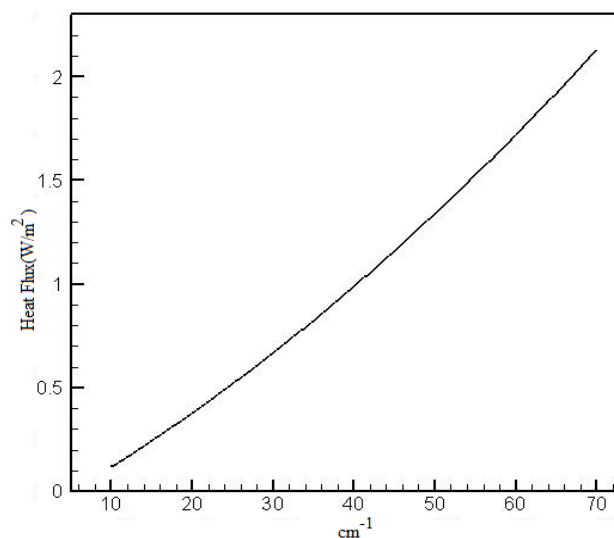


Figure 11: Conductive thermal emission by layer density

It is noted in the layer density that the thermal radiation has its maximum value. Excessive number of layers does not create significant changes in radiative heat transfer; hence, excessive number of layers in terms of radiation cannot produce good results. As the layers are made of aluminum and some retaining material is used between the layers, the thermal conductivity increases as the layers increase. This leads to a rise in the temperature of the tank via conductivity as the layer density increases. Figure 12 illustrates a diagram of the total heat flux in terms of insulation density. The total thermal emission is assigned the lowest value for a certain number of layers. If the number of layers is found to be less or more than the desired value, the passing heat flux will rise. As the number of layers increases, the radiative heat flux decreases and the thermal conductivity increases. The results from interactive increase and decrease are illustrated in Figure 12, which yields a desired value at a layer density of 30 cm⁻¹.

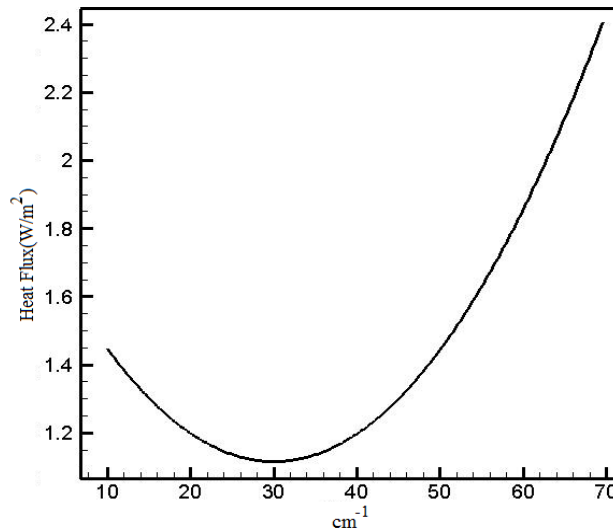


Figure 12: Total thermal emission by layer density

6-5-3: Impacts of emissivity on insulation performance

Figure 13 demonstrates the variations of total thermal flux in terms of emissivity for different layer densities.

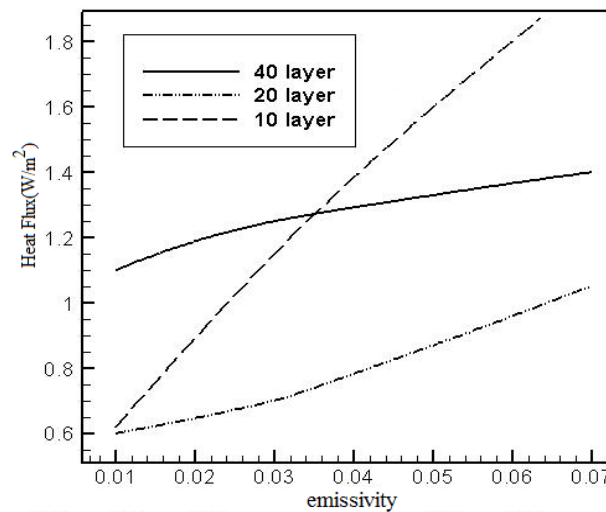


Figure 13: Impacts of emissivity on thermal flux imparted to the cold tank

It is noted that the total thermal flux increases as emissivity increases. One should note that the slope of the diagrams is lower for lower layer densities. The reason for this is that the emissivity only affects the thermal radiation and decreases as the number of layers increase. According to the results in Figure 13, materials with low emissivity should be used as much as possible, and the lower the emissivity of the selected materials, the better the total insulation performance.

7: Impact of ambient conditions on insulation performance (cylindrical tank)

7-1: Results from non-insulated tank

As noted in Figures 14 and 15, as temperature and h rise, the amount of radiative and convective thermal emission increase. In these two mechanisms, as clear, the rate of convective heat transfer is greater. Table 7 is used to show the results.

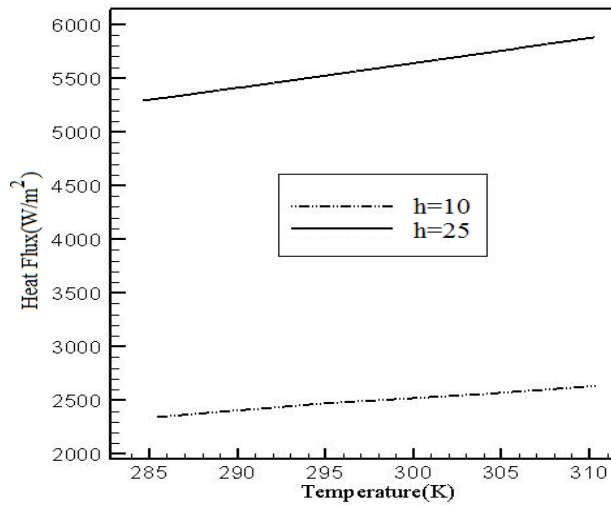


Figure 14: Variations of convective heat transfer imparted to the internal wall of a non-insulated tank by temperature variations at different h states

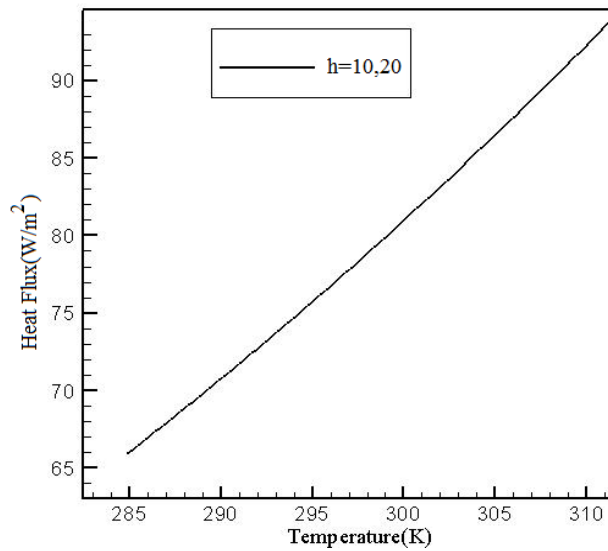


Figure 15: Variations of radiative heat transfer imparted to the internal wall of a non-insulated tank in terms of temperature variations at different h states

7-2: Vacuum-insulated tank results

Figure 16 demonstrates the temperature distribution of a .09m thick vacuum insulation. The ambient temperature, as seen in the figure, has a maximum value of about 293°K.

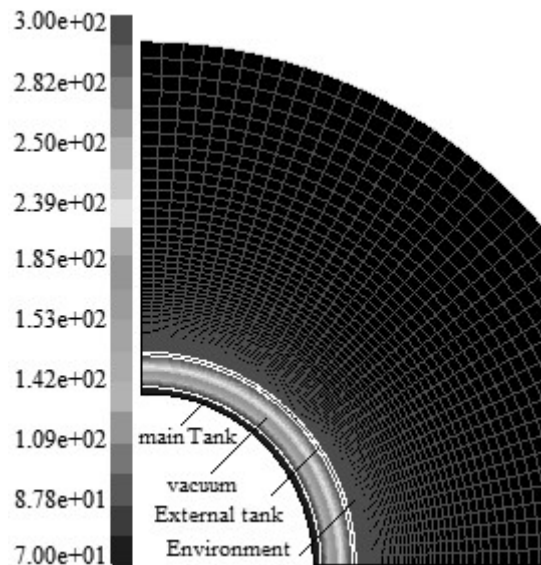


Figure 16: Temperature distribution of .09m thick vacuum insulation

Figure 16 illustrates that the temperature decreases as we move towards the main tank, so that temperature on the internal wall of the main tank is 77 °K. As the vacuum thickness increases (increased surface area of the external tank), the amount of radiative and convective heat outside the tank that imparts to the surface of the external tank increases. On the other hand, increased thickness of the vacuum reduces thermal conductivity. An interaction of this increase and decrease in thermal emission causes a decrease in thermal emission, because the decreased thermal conductivity will be greater than the radiative and convective increase outside the tank. Table 16 also exhibits that increased vacuum thickness causes thermal radiation in the vacuum space to rise. Wholly, the total heat emission imparted to the internal wall of the main tank is reduced. Here, by radiation in a vacuum space, it is meant the convective and radiative heat that imparts to the external surface of the tank and from there flows into the main tank via radiation in a vacuum space.

Table 16: Thermal emissions imparted for vacuum insulation in ambient conditions

Vacuum thickness (m)	Convective heat ($\frac{w}{m^2}$)	Radiative heat ($\frac{w}{m^2}$)	Radiation in vacuum space ($\frac{w}{m^2}$)	Total thermal emission ($\frac{w}{m^2}$)
.03	15.39	14.29	3.077	32.757
.06	10.57	9.09	3.141	22.801
.09	7.55	5.84	3.201	16.591

7-3: Use of polyurethane (PUR) and anti-radiation insulations around the external tank

Polyurethane (*PUR* and *PU*) is a polymer composed of organic units joined by carbamate polyols, and methylene diphenyl diisocyanate (MDI) allowed *polyurethane* rigid foams to be used as high-performance *insulation* materials. Polyurethane insulation, or PUR, refer to closed and rigid cellular insulation falling in the category of cold insulation. Polyurethane is available in prefabricated slabs or can be injected or sprayed in bulk at the insulation place. Spraying can be performed in several passes to meet the optimal thickness. Figure 17 illustrates the way polyurethane insulation is sprayed. When polyurethane is sprayed (Spray applied Polyurethane Foam), SPF requires special equipment and it is necessary to have proficient force to gain the best result. All the features of polyurethane insulation are provided in reference [20].

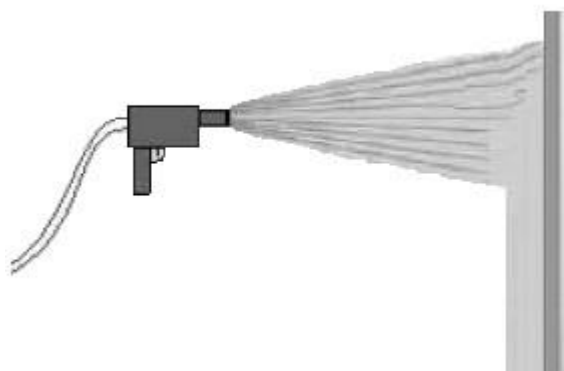


Figure 17: Polyurethane insulation can be sprayed at the site [19]

A very thin layer of anti-radiation reflective insulation on polyurethane is used to reduce the radiative heat transfer outside the tank. The radiation coefficient of common radiation insulation is given in reference [19]. If the insulation with the specifications of Table 9 and Figure 4 is used, the results will be as shown in Table 17. This insulation reduces the three heat transfer mechanisms. However, as it is clear, there is still a substantial amount of heat emission imparted to the main tank. Figure 18 demonstrates the temperature distribution of this insulation.

Table 17: Thermal emissions imparted to the internal wall of the main tank

Convective heat ($\frac{w}{m^2}$)	Radiative heat ($\frac{w}{m^2}$)	Radiation in vacuum space ($\frac{w}{m^2}$)	Total heat imparted to ($\frac{w}{m^2}$) the main tank
5.79	4.74	1.65	11.91

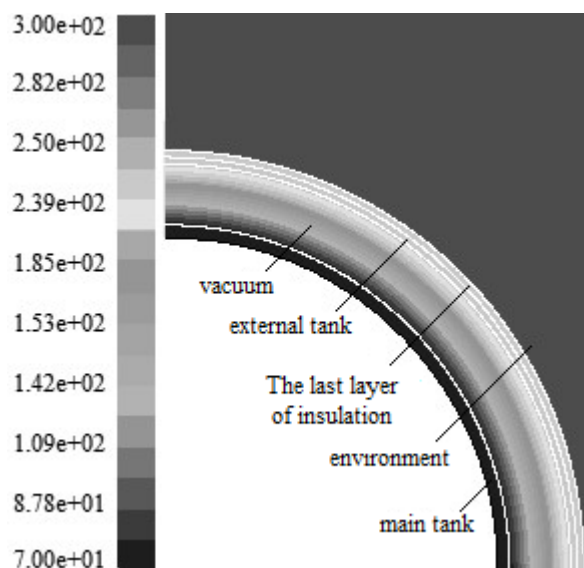


Figure 18: Temperature distribution in vacuum insulation with a non-conductive anti-radiation shell

7-4: Using combined vacuum, cold shell vapor insulation and the two-side-insulated technique around the main tank

One type of foam insulation is elastomeric foam or elastomeric insulation, which is usually applied in cooling equipment. When these foams are used on the two sides of the main tank along with vacuum insulation with cold shell vapor, the heat emission imparted to the cryogenic fluid inside the tank can be sharply reduced. The features of the combined insulation system to reduce the heating mechanisms are provided in Table 10. Note that the heat transfer of retaining material in this section is discarded. If the mentioned insulation is used, the total thermal emission decreases to 145 W/m², which is very desirable.

8- Calculation of evaporation rate inside the cryogenic tank

Using the cryogenic fluids-specific latent heat (enthalpy) of vaporization, the evaporation rate inside the tank can be easily calculated. Consider for example that the evaporation rate for a cryogenic fluid is \mathcal{Q} in which case the evaporation rate of the cryogenic fluid in the tank is calculated according to Equation 12.

$$\text{evaporation rate} = \frac{Q(W)}{\vartheta\left(\frac{J}{kg}\right)} \tag{12}$$

Using the unit conversion $W = \frac{J}{s}$, Equation 12 becomes $\frac{kg}{s}$, suggesting that how many kilograms of cryogenic fluid is converted to vapor each second. To convert to $\frac{kg}{day}$, it suffices to multiply the resulting expression by $3600*24$ to calculate the evaporation rate in the cryogenic tank. Thus, we use Equation 13 to calculate the evaporation rate of cryogenic tank by $\frac{kg}{day}$.

$$\text{evaporation rate} = \frac{Q(W)}{\vartheta\left(\frac{J}{kg}\right)} * 3600 * 24 \tag{13}$$

Generally speaking, the amount and performance of an appropriate insulation system in cryogenic applications are determined based on the percentage of evaporation of cryogenic material stored in the day. It is clear that in an equal tank volume, the lower the percentage of evaporation rate of cryogenic material, the higher the performance of the cryogenic insulation. Tables 23 and 24 demonstrate the impacts of various insulation systems on the evaporation rate of different cryogenic fluids. Cold shell vapor insulation combined with other insulations yield lower evaporation rate relative to other methods. Multilayer insulation also produces a small percentage of vapor each day, suggesting its high performance. The evaporation rate for liquid helium is greater than those of other cryogenic fluids and care should be taken when insulating these tanks. The reason for this is the low latent heat of evaporation of this liquid compared to other cryogenic fluids.

Table 23: Comparison of evaporation rate in spherical tank made of various insulations (constant temperature conditions)

Insulation	Helium $\left(\frac{kg}{day}\right)$	Hydrogen $\left(\frac{kg}{day}\right)$	Nitrogen $\left(\frac{kg}{day}\right)$	Oxygen $\left(\frac{kg}{day}\right)$
Non-insulated tank	2469658.1-	116545.6-	259314.05-	243487.3-
high vacuum	176.9	8.3	18.5	17.44
Vacuum with cold vapor shield	14.4	.68	1.51	1.41
Multi-layer with high vacuum	21.6	1.01	2.27	2.12

Table 23: Comparison of evaporation rate in a cylindrical tank of various insulations (ambient conditions)

Insulation	Helium $\left(\frac{kg}{day}\right)$	Hydrogen $\left(\frac{kg}{day}\right)$	Nitrogen $\left(\frac{kg}{day}\right)$	Oxygen $\left(\frac{kg}{day}\right)$
Non-insulated tank	216645.5-	1022.3	2274.7	2135.9
high vacuum	67.8	3.2	7.1	6.7
Vacuum with cold vapor shield	9.91	.46	1.1	0.97
Multi-layer with high vacuum	48.9	2.3	5.1	4.8

9: Conclusion

The results of the present study are generally summarized as follows.

- In vacuum insulation, increased thickness on the one hand reduces the conductive heat transfer, and on the other hand, increases the radiative heat transfer. For the main stainless-steel tank, as thickness increases, the total heat transfer decreases and then increases. This is while, with regards to the main aluminum-coated tank, since the emissivity is low, the rate of radiation is not enough to yield an unexpected outcome, and we always see reduced heat transfer to the main tank as thickness increases.
- In vacuum-insulated systems, the lower the vacuum pressure, the less the distance between the gas molecules and the lower the thermal conductivity.
- Applying vapor from liquid evaporation inside the tank and conducting it through tubes around the main tank is another technique to reduce radiative heat transfer in tanks containing cryogenic fluids. In this situation, the main tank, instead of exchanging heat in a direct exchange with the ambient, does this via a cold shell.
- A very thin layer of anti-radiation reflective insulation on polyurethane was been used to reduce the radiative heat transfer outside the tank.

- Using multiple layers in a certain thickness will be very useful to reduce radiative and conductive heat. As the layer density increases, the passing radiation decreases; however, the thermal conductivity increases. The interactive increase and decrease are in the form of Figure (14-4), resulting in an optimal value in the number of layers.
- As an important conclusion in a specific emissivity, two multi-layer insulations with several numbers of layers may yield similar functions to each other. In this case, insulation with a low number of layers is selected to reduce costs.
- Cold shell vapor insulation combined with other insulation systems has a lower evaporation rate relative to other methods. Multilayer insulation also causes a low percentage of vapor per day, indicating a greater performance.
- Evaporation rate of helium fluid is greater than those of other cryogenic fluids and care should be taken when insulating these tanks. The reason for this is the low latent heat of evaporation of this fluid relative to other cryogenic fluids.
- Concerning multi-layer and cold shell vapor insulation, using cutting edge technology and experienced workforce is critical. Looking at the studies done and the existing capacities in the field of multilayer insulation, it is not out of expectation to see this type of insulation being mass produced. If we manage to bring this type of insulation to the production stage, high-vacuum multi-layer insulation can be proposed as the best insulation system for cryogenic tanks.

10. Indices

Pressure (kgm ⁻¹ s ⁻²)	<i>P</i>
Area ft ²	<i>A</i> ₁
Accommodation coefficient	α_1
Accommodation coefficient	α_2
Mean temp. (K)	<i>T</i>
Specific heat capacity (W/kg.K)	<i>C</i> _{<i>P</i>}
Specific heat capacity (W/kg.K)	<i>C</i> _{<i>v</i>}
Internal surface temp(K)	<i>T</i> ₁
External surface temp(K)	<i>T</i> ₂
temperature difference (K)	ΔT
Radiative heat transfer (W)	<i>Q</i> _{<i>r</i>}
Emissivity	<i>e</i> ₁
Emissivity	<i>e</i> ₂
Greek sign	
$\frac{C_p}{C_v}$ ratio	γ

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