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Optimization approach of insulation thickness of non-vacuum cryogenic storage tank

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Abstract

Cryogenic insulation systems, with proper materials selection and execution, can offer the highest levels of thermal performance. Insulations are listed in order of increasing performance and, generally, in order of increasing cost. The specific insulation to be used for a particular application is determined through a compromise between cost, ease of application and the effectiveness of the insulation. Consequently, materials, representative test conditions, and engineering approach for the particular application are crucial to achieve the optimum result. The present work is based on energy cost balance for optimizing the thickness of insulated chambers, using foamed or multi layered cryogenic shell. The considered insulation is a uniformly applied outer layer whose thickness varies with the initial and boundary conditions of the studied vessel under steady-state radial heat transfer. An expression of the optimal insulation thickness derived from the total cost function and depending on the geometrical parameters of the container is presented.

Keywords: cryogenics, tank diameter, insulation thickness, energy cost, optimization

Nomenclature

C	Cost, €/kg
D	Tank diameter, m
L	Length of the cylinder, m
n'	Mass unit cost of stored fluid, €/kg
N	Lining lifetime
$\dot{Q}$	Thermal flux, W
R_i	Inner radius, m
R_o	Outer radius, m
$\bar{R}$	Average radius of reference, m
S	Surface, m^2
$\bar{S}$	Thermal surface of reference, m^2
T_i	Inner surface temperature, K
T_o	Outer surface temperature, K
T_{ext}	External temperature, K
T_{liq}	Liquefaction temperature, K
V_{tk}	Tank volume, m^3
X_i	Insulation thickness, m
x_{opt}	Optimal thickness, m
x	Reduced thickness
λ_i	Insulator thermal conductivity, W/(m·K)
θ	Parameter of configuration
ΔH_v	Latent heat of vaporization, J/kg

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Subscripts

en	Energy
in	Insulation
sh	Shell
tk	Tank

1. INTRODUCTION

In designing and manufacturing cryogenic tanks for transport and storage of liquid nitrogen, oxygen, and argon, special attention is given to improving their technical characteristics and, in particular, to reducing the specific loss of the liquid caused by evaporation. However, this often contradicts the technical and economic considerations.

A technical and economic model of optimization was constructed [1] which takes into account both the manufacturing and service conditions of the tanks. Two compulsory requirements must be fulfilled in this case: Comparison of the tanks with insulation of various types should be carried out for the same holding capacity of the tanks to avoid the effect of the scale factors; optimum thickness must be determined for each type of insulation. The optimality criterion was represented by the corrected expenses consisting of the sum of net cost of the product and the proportion of capital investment taken into account by the norm factor.

Measurement results are reported for the heat transfer rate, as a function of the number of layers of different types of multilayer insulation, for temperature differences of 284 K to 77 K and 77 K to 4.2 K [2]. The insulation was found to reduce heat transfer between the higher temperatures but generally had a detrimental effect at the lower temperatures. An exception was noted in double-aluminium-coated NRC2, which also gave a slight reduction in heat transfer for up to ten layers at the lower temperatures.

The thermal performance of multilayer insulation (MLI), consisting of double aluminized Mylar radiation shields and nylon net thermal spacers, was evaluated using a double guarded cylindrical calorimeter and a tank calorimeter over the temperature range 300-77 K [3]. The degradation in the effective thermal conductivity of MLI was evaluated to be 1.68 using the calorimeters. The optimum number of layers for the MLI was 40 to 50 at a layer density of 25 layers cm^{-1}. The temperature profile and heat flux through the MLI were obtained as a function of vacuum level for different numbers of insulation layers.

A mathematical model has been developed to describe the heat flux through multilayer insulation (MLI) from 80 K to 4.2 K [4]. The total heat flux between the layers is the result of three distinct heat transfer modes: radiation, residual gas conduction and solid conduction. The mathematical model enables prediction of MLI behaviour with regard to different MLI parameters, such as gas insulation pressure, number of layers and boundary temperatures. The calculated values have been compared to the experimental measurements carried out at CERN.

The evaluation of optimal insulation has been presented for two layers insulated pipelines [5]. An efficient nonlinear optimization method was applied for the minimum cost design of insulated pipelines with constraints. The objective cost function includes the heat loss and the material costs of the insulating layers and the tube as well. The thicknesses of insulating layer and the minimum costs depending on the temperature of the surrounding air and flowing material are presented.

Perforated MLI blanket systems are targeted for large scale cryogenic facilities. Space applications and particle accelerators are two fields concerned with thermal shielding of cryogenic devices. Because radiation heat transfer varies with T^4, heat transfer in the range of 300 K to 77 K is dominant even for devices operating at temperatures as low as 2 K. The results of an experimental study of a perforated MLI blanket system using a steady state liquid nitrogen evaporation method are presented [6].

In order to evaluate the thermal performance of the MLI fabricated in the horizontal cryostats of superconducting magnets, it is important to investigate the contact pressure in the MLI [7]. At first, a single thin film wound around the horizontal cylinder was analyzed to evaluate the contact pressure acting on the cylinder. The analysis has been extended to the multiply wound film around horizontal cylinder, in order to investigate the distribution of contact pressure between adjacent layers. By using experimental data obtained with a flat panel calorimeter, the results of this analysis have been applied to evaluate the thermal performance of MLI around a horizontal cylinder.

Nast et al. [8] presents a summary of studies for National Aeronautics and Space Administration (NASA) focused on multilayer insulation MLI systems on the larger tankage. The sensitivity of boil off to MLI thermal conductivity is presented. A novel large tank simulator approach for MLI testing is presented along with recommendations for maturation of the MLI technology.

A novel cryogenic air separation process with LNG (liquefied natural gas) cold energy utilization that produces liquid nitrogen and oxygen is proposed and analyzed [9]. A power generation cycle is integrated with the process to completely utilize the cold energy of the LNG and to offset some of the consumed power. It is shown that the energy and exergy efficiencies increase by 59.4% and 67.1%, respectively.

A generalized layer by layer model has been proposed to predict the thermal performance of the Foam Variable Density Multilayer Insulation combination (FMLI) at different vacuum levels [10]. A combination of polyurethane foam and multilayer insulation is used for cryogenic propellant storage. The experimental data verified the validity of the model and indicated that the heat fluxes through the FMLI and the single VDMLI almost made no difference in vacuum of 10^{-3} Pa, which were both equal to 0.23 $W \cdot m^{-2}$ with the boundary temperatures of 77 and 293 K, respectively.

Jiang et al. [11] presents a theoretical model that considers three heat transfer mechanisms simultaneously within composite insulation composed of polyurethane foam, a variable density multilayer insulation and convective heat transfer inside the vapour-cooled shield, to predict and optimize the thermal performance of the insulation combination.

Kim et al. [12] investigates the economic feasibility of the additional boil-off gas (BOG) liquefaction facilities in the high pressure fuel supply system on the vessels. For the comparison of the fuel supply system and its variations with BOG liquefaction, they are optimized with respect to total annual cost (TAC) as the objective function. The optimization results show that the use of BOG liquefiers on LNG vessels reduces the TAC by at least 9.4% compared to the high pressure fuel supply system.

Deng et al. [13] studied the relation between the thermal performance of multilayer insulation MLI used for cryogenic transfer lines and the layer density, number of layer and material of reflectors and spacers theoretically and experimentally. An optimum combination of layer density and number of layer of MLI is proposed. Both theoretical and experimental results show that heat flux decreases with the increases of number of layer and the decrease of layer density.

A layered composite insulation system has been developed for non-vacuum applications and extreme environmental exposure conditions [14]. Layered composite insulation system for extreme conditions (or LCX) is particularly suited for complex piping or tank systems that are difficult or practically impossible to insulate by conventional means. Primary requirements for such non-vacuum thermal insulation systems include the combination of harsh conditions, including full weather exposure, vibration, and structural loads. The LCX system is suitable for temperatures from approximately 4 K to 400 K.

Many papers were devoted to the optimization of cryogenic vessels, exploring several configurations but related directly to the vacuum systems. The present study was intended to develop a concept for a cryogenic double-walled cylindrical tank, to evaluate the performance of the insulation in a near-operational environment. Foams and MLI are one by one used in the non-vacuum annulus of vessel to meet different thermal performance, cost, or mechanical objectives such as space and weight. Materials, representative conditions, and engineering approach must be considered for a particular application. Deficiency in one of these three areas can prevent optimum performance and lead to costly inefficiencies.

2. DETERMINATION OF OPTIMAL THICKNESS

Cryogenic liquid storage vessels are designed to minimize the transfer of heat into the tank. The heat which is transferred into the tank is absorbed by the cryogenic liquid in the tank and will cause some of the liquid to be vaporized into gas.

We seek to determine the optimal insulation thickness of a given shaped tank using geometrical, thermal and economical parameters. The studied tank is a cylindrical double-walled container with elliptical bottoms, made of stainless steel (Fig. 1). The space between the two walls is filled with insulation.

Tank walls are thin so the temperature drop due to conduction is negligible. If insulation layer adheres perfectly to the wall, the steady-state conduction heat flux through the cylinder:

$$\dot{Q} = \frac{2\pi\lambda_i L}{\ln\left(1 + \frac{2X_i}{D}\right)} (T_i - T_o) \quad (1)$$

An approximate equation [15] through an average surface of reference $\bar{S}$ is written:

$$\dot{Q} = \frac{\lambda_i \bar{S}}{\ln\left(1 + \frac{2X_i}{D}\right)} 0.9(T_{ext} - T_{liq}) \frac{1}{\bar{R}} \quad (2)$$

Where, T_{ext} = 293.15 K.

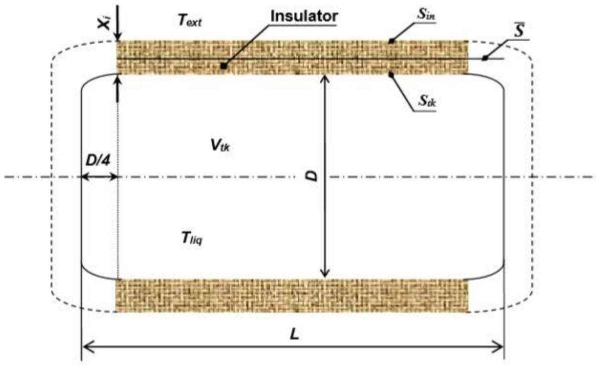


Fig. 1. Main geometrical characteristics of the tank.

Energy cost equations in terms of dimensionless geometrical ratios allow us to formulate an economical optimization of insulation thickness of the studied reservoir (Fig. 1).

According to Conte [15], an approximate geometric dimensions of a cylindrical tank with elliptical bottoms based on polynomial interpolation methods [16], are expressed bellow.

Dimensionless configuration parameter of the vessel:

$$\theta = \frac{L}{D} \quad (3)$$

Dimensionless reduced thickness:

$$x = \frac{X_i}{D} \quad (4)$$

Tank volume:

$$V_{tk} = (\theta - 0{,}1667)\frac{\pi D^3}{4} \quad (5)$$

Tank surface:

$$S_{tk} = (\theta + 0{,}19)\pi D^2 \quad (6)$$

Insulation volume:

$$V_{in} = \left[(\theta + 0{,}25)x + (\theta - 0{,}5)x^2 - \frac{x^3}{3}\right]\pi D^3 \quad (7)$$

External surface of insulation:

$$S_{in} = [(\theta + 0{,}19) + (2\theta + 2{,}76)x + 0{,}76x^2]\pi D^2 \quad (8)$$

Thermal surface of reference:

$$\bar{S} = [(\theta + 0{,}19) + (2\theta + 1{,}38)x + 0{,}19x^2]\pi D^2 \quad (9)$$

The total cost of the installation (C_T) is based on geometrical approximations of the container and depends on three necessary costs.
Insulation cost:

$$C_{in} = V_{in} \cdot C_{in/v} \tag{10}$$

Cost of shell (insulator):

$$C_{sh} = S_{in} \cdot C_{sh/s} \tag{11}$$

Operating energy cost:

$$C_{en} = \frac{\bar{S}}{xD} C_{en/L} \tag{12}$$

And, the energy cost per unit length of insulation:

$$C_{en/L} = \lambda_i \frac{3{,}156 \cdot 10^7}{\Delta H_v} n' N \, 0.9 (T_{ext} - T_{liq}) \tag{13}$$

The coating lifetime N is expressed in years and n' the unit cost of 1 kg of the stored fluid.
The total cost is expressed as:

$$C_T = C_{in} + C_{sh} + C_{en} \tag{14}$$

Hence, equations (10), (11) and (12) become:

$$C_{in} = \left[(\theta + 0{,}25)x + (\theta - 0{,}5)x^2 - \frac{x^3}{3}\right] \pi D^3 C_{in/v} \tag{15}$$

$$C_{sh} = [(\theta + 0{,}19) + (2\theta + 2{,}76)x + 0{,}76x^2] \pi D^2 C_{sh/s} \tag{16}$$

$$C_{en} = [(\theta + 0{,}19) + (2\theta + 1{,}38)x + 0{,}19x^2] \frac{\pi D C_{en/L}}{x} \tag{17}$$

Finally, the equation of the total cost:

$$\begin{aligned} C_T = &-\pi D^3 C_{in/v} \frac{x^3}{3} + \left[(\theta - 0{,}5)\pi D^3 C_{in/v} + 0{,}76 \pi D^2 C_{sh/s}\right] x^2 \\ &+ \left[(\theta + 0{,}25)\pi D^3 C_{in/v} + (2\theta + 2{,}76)\pi D^2 C_{sh/s} + 0{,}19 \pi D C_{en/L}\right] x \\ &+ (\theta + 0{,}19) \frac{\pi D C_{en/L}}{x} \\ &+ \left[(\theta + 0{,}19)\pi D^2 C_{sh/s} + (2\theta + 1{,}38)\pi D C_{en/L}\right] \end{aligned} \tag{18}$$

In order to obtain an analytical expression giving the optimal insulation thickness, the interpolation of the type below is defined with an approximation precision of 0.001.

$$x_{opt} = [x_0^{-n} + x_\infty^{-n}]^{\frac{-1}{n}}, \quad x_{opt} \in]0, 1[\tag{19}$$

n being a parameter that depends on the configuration factor (θ), x_o is estimated for $C_{en/L} \rightarrow 0$ and x_∞ estimated for $C_{en/L} \rightarrow \infty$.

$$\begin{aligned} \frac{dC_T}{dx} = &-x^4 + [(2\theta - 1) + 1{,}52b]x^3 \\ &+ [0{,}19a + (2\theta + 2{,}76)b + \theta \\ &+ 0{,}25]x^2 - (\theta + 0{,}19)a = 0 \end{aligned} \tag{20}$$

With,

$$a = \frac{C_{en/L}}{C_{in/v} D^2} \text{ and } b = \frac{C_{sh/s}}{C_{in/v} D}$$

The coefficient n is calculated as:

$$n = 0{,}54 \left[1 + \frac{1}{\sqrt{\theta}}\right] \tag{21}$$

If $X_i << D$, terms in x^3 and x^4 are negligible.

$$\lim_{a \to 0} \frac{dC_T}{dx} \Rightarrow x_0 = \sqrt{\frac{(\theta + 0{,}19)a}{0{,}19a + (2\theta + 2{,}76)b + \theta + 0{,}25}} \tag{22}$$

and,

$$\lim_{a \to \infty} \frac{dC_T}{dx} \Rightarrow x_\infty = \sqrt{1 + \frac{\theta}{0{,}19}} \tag{23}$$

Replacing x_o and x_∞ by their respective terms in equation (19), one obtains:

$$x_{opt} = \left\{ \left[\frac{3{,}19 \left(\frac{C_{en/L}}{C_{in/v} D^2}\right)}{0{,}19 \left(\frac{C_{en/L}}{C_{in/v} D^2}\right) + 8{,}76 \left(\frac{C_{sh/s}}{C_{in/v} D}\right) + 3{,}25} \right]^{-0{,}4259} + 0{,}3008 \right\}^{-1{,}1737} \tag{24}$$

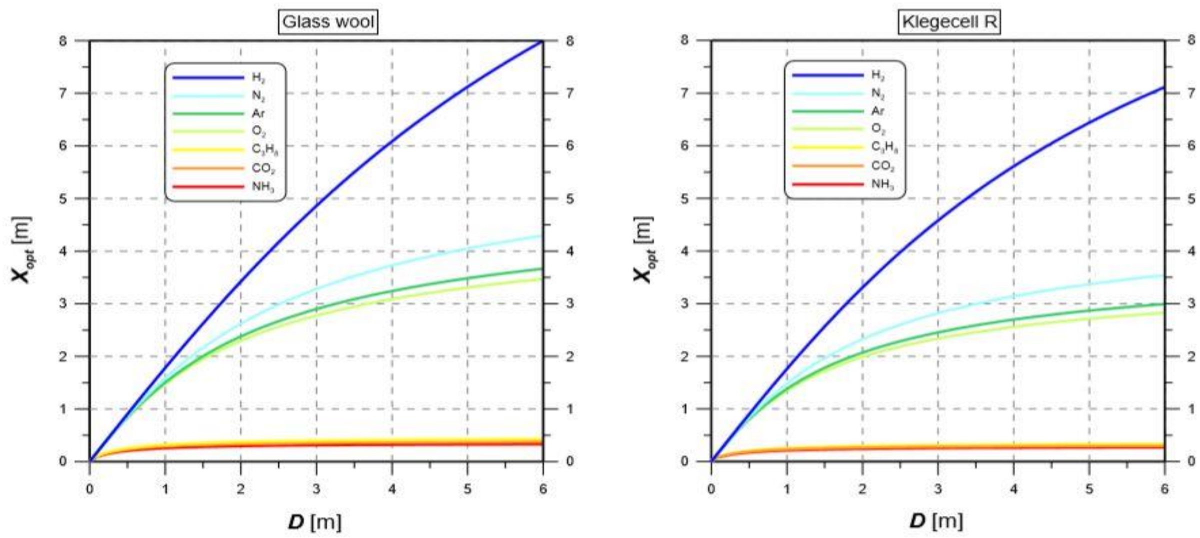
The ratio ($C_{en/L}/C_{in/v}$) has the dimension of a surface; it depends on the insulator price, its thermal conductivity and the fluid thermodynamic properties.
In addition to the thermophysical characteristics, other criteria must also be considered to choose the suitable insulation, namely the implementation, the lifetime, the reliability [17] and the cost of achieved insulation.

3. RESULTS AND INTERPRETATION

The application of equation (24) is performed for two insulation types, foam and MLI. One will assume the lining longevity, N = 10 years and the configuration factor, θ = 3. The mean cost of shell per unit of surface adopted in the calculation is $C_{sh/s}$ = 50 €/m^2.

TABLE I
CRYOGENICS DATA.

Gaz	NH_3	C_3H_8	CO_2	O_2	Ar	N_2	H_2
T_{liq} [K]	239.65	230.95	244.25	90.18	87	77.36	20.3
ΔH_v [kJ/kg]	1371.20	425.31	301.70	212.98	160.81	198.38	454.30
n' [€/kg]	0.064	0.025	0.016	0.400	0.340	0.560	10.000
Foam	*Glass wool*, λ_i = 0.040 W/(m.K)						
$C_{en/L}$ [€/m]	32.23	53.87	42.71	5651.78	6495.58	9889.1	82774.55
Foam	*Klegecell R*, λ_i = 0.031 W/(m.K)						
$C_{en/L}$ [€/m]	21.14	32.32	25.62	33391	3897.37	5933.45	49664.73
Foam	*Polyurethane*, λ_i = 0.020 W/(m.K)						
$C_{en/L}$ [€/m]	14.1	21.55	17.082	2260.71	2596.23	3955.63	33109.82
MLI	*Aluminized Mylar*, λ_i = 0.0002 W/(m.K)						
$C_{en/L}$ [€/m]	0.14	0.214	0.17	22.6	28.864	39.556	331.098
MLI	*Aluminized Polyester*, λ_i = 0.00003 W/(m.K)						
$C_{en/L}$ [€/m]	0.02112	0.03232	0.04362	3.39	4.328	5.932	49.664



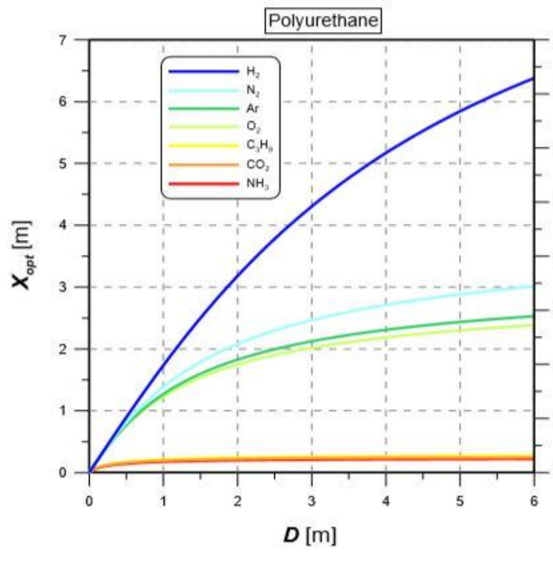


Fig. 2. Foams:Optimum thickness vs tank diameter for different cryogenic fluids stored.

Summary graphs of optimum thickness as a function of tank diameter for different cryogenic fluids stored (Tab. 1) are presented in figures 2 and 3. While calculating the optimum thickness the materials are applied separately. In the category of foams, glass wool, klegecell R and polyurethane were the first investigated insulators (Fig. 2), assuming an average cost per unit of volume $C_{in/v}$ = 250 €/m^3. The MLI materials used in this study consist of aluminized layers made of mylar and polyester (Fig. 3), the average cost per unit of volume is $C_{in/v}$ = 1200 €/m^3.

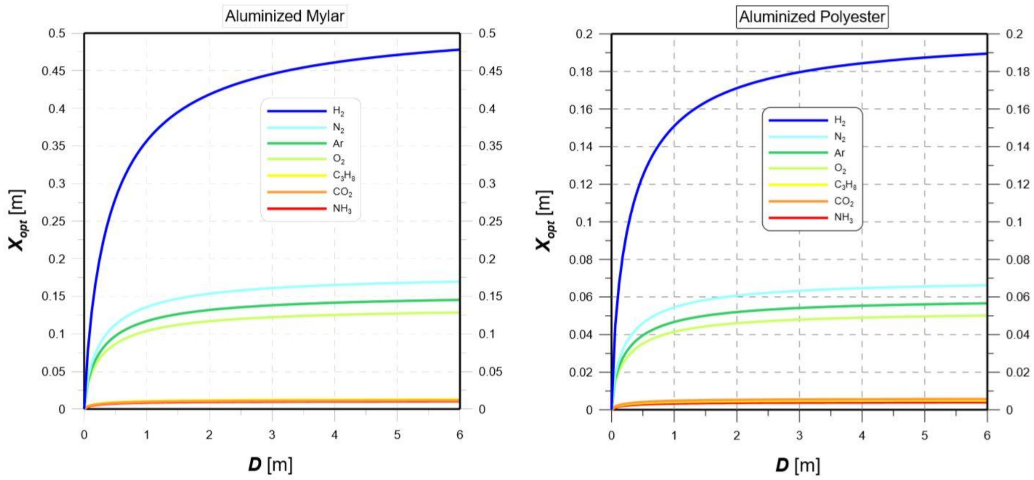


Fig. 3. MLI:Optimum thickness vs tank diameter for different cryogenic fluids stored.

Although some of the graphs may seem similar to the reader but they are different either in value or meaning. The graphical results reveal important information on thermally and economically appropriate insulation for a dimensioned tank intended for cryogenic fluid storage. For large scale storage at low temperatures, ranging from approximately 200 K to 273 K, it is preferable to use foams that offer good insulation efficiency at low price. However, it is possible to ensure cryogenic storage for very low temperatures in the vicinity of 100 K with polyurethane insulation but there should be an insulation thickness almost equal to the tank diameter, which means more space for more volume. For example, for the studied tank 2 m long with a diameter of 1 m and containing liquid oxygen, the optimal thickness would be X_{opt} = 1 m.

On the other hand, for extreme temperatures storage, from 20 K to 100 K, the curves obtained from aluminized mylar and aluminized polyester (Fig. 3) confirm the high efficiency of multilayer insulation (MLI), which are ideally suited for this type of storage. Liquid hydrogen whose boiling point is approximately 20.3 K stored in the tank in question with a diameter of 1 m, requires an insulation thickness of 35 cm if it is aluminized mylar insulator but only 15 cm for aluminized polyester. In large capacity cryogenic containers, MLI thickness does not exceed 50 cm if the insulation is in mylar and only 20 cm if the insulation is made of polyester.

Usually, foams are used for large cryogenic storage despite relatively high thermal conductivity. They are considered as cellular materials made essentially from plastic. Multilayer insulation consists of a succession of reflective screens in which low-conductive sheets are interposed. These insulators which have a very high thermal resistance are very expensive, and are especially reserved for space cryogenics.

4. FINAL CONCLUSIONS

The problem of optimal design has been formulated taking into account the total cost (Shell + insulation + exploitation) presented as a function of insulation thickness. Obviously, this is a limited simplification especially for vacuum multilayer insulation systems (MLI). In such systems the cost of insulation is a function of many parameters and requires a more complex optimization. Nevertheless, the obtained formulation with the adopted technico-economical criterion remains applicable for the case study of non-vacuum insulated tank and suitable for temperatures from approximately 80 K to 273 K.

To minimize heat leaks into storage tanks and transfer lines, high-performance materials are needed to provide high levels of thermal isolation. Complete knowledge of thermal insulation is a key part of enabling the development of efficient, low-maintenance cryogenic systems. The choice of insulation often depends on the tank size, in fact, for big containers it is possible to admit less efficient insulators, on the other hand, smaller is the tank and high must be the thermal characteristics. The complexity due to heat transfer processes and physico-mechanical constraints imposes a large number of laboratory and workshop tests before launching a product on the market.

Obviously, cryogenic thermal insulation systems that incorporate a vacuum environment can provide the lowest possible heat transfer but are used extensively for small laboratory-size Dewars. High performance can be achieved by the right combination of materials in a non-vacuum system like collated insulator layers and applied from a single roll separated by a micro-fiberglass paper spacer. This is one of our perspectives in future research.

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