Design and Optimization of Cryogenic Storage Vessel

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***Abstract* - After a cryogenic fluid has been liquefied and purified to the desired level; it must then be stored and transported. Cryogenic fluid storage-vessel and transfer line design has progressed rapidly as a result of the growing use of cryogenic liquids in many areas of engineering and science. The development of the Dewar vessel represented such an improvement in cryogenic fluid storage vessels that it could be classed as a "break-through" in container design. The high performance storage vessels in use today are based on the concept of the Dewar design principle a double walled container with the space between the two vessels filled with an insulation and the evacuated from the space. The detailed conventional-cryogenic-fluid storage vessel design is covered in such standards as the American society of mechanical engineers (ASME) boiler and pressure vessel code, section VIII (1983), and British Standards Institution standards 1500 or 1515. Most users require that the vessels be designed, fabricated, and tested according to the code for sizes larger than about 250 dm3 i.e. 66 U.S. gallons, because of the proven safety code design. Cryogenic storage vessels are pressure vessels are used for storage cryogenic liquids with minimum heat in-leak into the vessel from the outside as far as possible. The challenge of design is to use such materials that do not lose their desirable properties at such a low temperature. Here the utmost care is taken to design a storage vessel satisfying both mechanical and thermal design. The results will be compared to the existing vessel of industry.**

***Keywords* - cryogenic; vacuum; inner & outer vessel; Boil-off rate**

I. INTRODUCTION

A pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. Pressure vessels are used in a variety of applications in both industry and the private sector. They appear in these sectors as industrial compressed air receivers and domestic hot water storage tanks. Other examples of pressure vessels are diving cylinders, recompression chambers, distillation towers, pressure reactors, autoclaves, and many other vessels in mining operations, oil refineries and petrochemical plants, nuclear reactor vessels, submarine and space ship habitats, pneumatic reservoirs, hydraulic reservoirs under pressure, rail vehicle airbrake reservoirs, road vehicle airbrake reservoirs, and storage vessels for liquefied gases such as ammonia, chlorine, propane, butane, and LPG.

II. MAIN TEXT

In 1892 Sir James Dewar developed the vacuum-insulated double walled vessel that bears his name today (Dewar 1898). The development of the Dewar vessel (which is the same type of container as the ordinary Thermos bottle used to store coffee, iced tea, etc.) represented such an improvement in cryogenic-fluid storage vessels that it could be classed as a "breakthrough" in container design. The performance storage vessels in use today are based on the concept of the Dewar design principle –a double-walled container with the space between the two vessels filled with insulation and the gas evacuated from the space. Improvements have been made in the insulation used between the two walls, but the Dewar vessel is still the starting point for high-performance cryogenic fluid vessel design. [2]

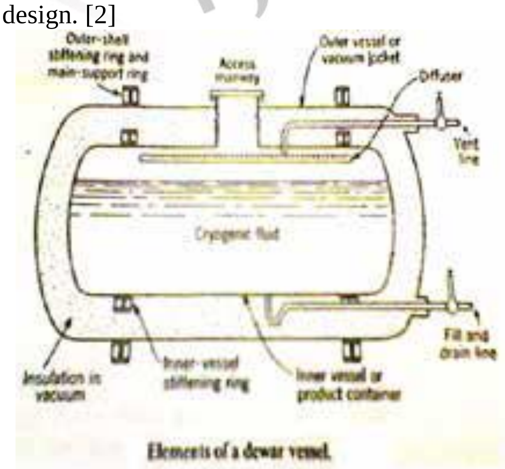

Fig.1 Cryogenic Storage Vessel (DEWAR)

The essential elements of a Dewar vessel are shown in fig 1. The storage vessel consists of an inner vessel called the product container, which encloses the cryogenic fluid to be stored. The inner vessel is enclosed by an outer vessel or vacuum jacket, which contains the high vacuum necessary for the effectiveness of the insulation and serves as a vapor barrier to prevent migration of water vapour or air (in the case of liquid hydrogen and liquid helium storage vessel) to the cold product container. The space between the two vessels is filled with insulation, and the gas in the space may be evacuated. In small laboratory Dewar's, the "insulation" consists of the silvered walls and high vacuum alone; however, insulation such as powders, fibrous materials, or multilayer insulations are used in larger vessels. Since the performance of the vessel depends to a great extent upon the effectiveness of the insulation, we shall devote a section to the discussion of insulations used in cryogenic – fluid storage and transfer systems.

III. Inner Vessel Design [2]

According to the ASME Code, Section VIII, the minimum thickness of the inner shell for a cylindrical vessel should be determined from

$$t = \frac{pD_{ii}}{2S_a e_w - 1.2p} \qquad (1)$$

Where,

t_i = minimum thickness of inner vessel
p = design internal pressure
D_{ii} = inside diameter of inner shell
S_a = allowable stress
e_w = weld efficiency

The minimum thickness for spherical shells, hemispherical heads, elliptical heads, or ASME torispherical heads is determined from

$$t_{hi} = \frac{pDK}{2S_a e_w - 0.2p} \qquad (2)$$

Where,

t_{hi}= minimum thickness of head for inner vessel
p = design internal absolute pressure
D = inside diameter of the head
Design of Outer Vessel

$$K = {}^{1}/_{6}\left[2 + ({}^{D_{ii}}/_{D_1})^2\right] \qquad (3)$$

Where,
D_{ii} = major diameter of the elliptical head
D_1 = minor diameter of the elliptical head

IV. Outer Vessel Design [2]

The outer vessel has to withstand only atmospheric pressure acting on it, so it would not fail because of excessive stress, but it would fail from the standpoint of elastic instability (collapsing or buckling). The performance of the insulation will deteriorate by the moisture condensing on the inner vessel. The outer vessel acts as a vapour barrier for the insulation. The outer vessel always remains at ambient temperature. The critical pressure according to the ASME section VIII is given as

$p_c = 4 \times p_a$ (4)

p_a = atmospheric pressure, kpa

The collapsing or critical pressure for a long cylinder exposed to external pressure is given by (Timoshenco and Gere 1961):

$$p_c = \frac{2E(t_o/D_{Oo})^3}{1-\mu^2} \qquad (5)$$

Where E = Young's modulus of shell material
t_o = minimum thickness of outer Shell
D_{Oo}= outside diameter of outer shell
μ = poisson's ratio for shell material

A "long" cylinder is defined as one for which the length-to-diameter ratio meets the following condition:

$h/Do_o > 1.140(1-\mu^2)^{1/4} (D_O/t_o)^{1/2}$ (6)

Where,

h = height of the cylinder
D_{Oo}= outside diameter of outer shell
t_o = minimum thickness of outer Shell
μ = poisson's ratio for shell material

If LHS>RHS then long cylinder otherwise short cylinder
The collapsing pressure for a 'short' cylinder subjected to external pressure may be calculated from the U.S. Experimental Modal Basin formula (windenburg and Trilling 1960):

$$P_c = \frac{2.42E\left({}^{t_o}/_{D_{O_o}}\right)^{5/2}}{(1-\mu^2)^{3/4}\left((h/D_{O_o}) - 0.45\left(t_o/D_{O_o}\right)^{1/2}\right)} \qquad (7)$$

Where,

E = Young's modulus of shell material
h = height of the vessel
D_O= outside diameter of outer shell
t_o = minimum thickness of outer Shell
μ = Poisson's ratio for shell material
p_c = critical pressure

The heads for the outer vessel must withstand the collapsing load of atmospheric pressure, and the mode of failure is elastic instability rather than rapture due to excessive stress. The critical pressure for a hemispherical, elliptical, or torispherical head (or for a spherical vessel) is given by (Timoshenco and Gere 1961):

$$P_c = \frac{0.5E(t_{h_o}/R_o)^2}{(3(1-\mu^2))^{1/2}} \qquad (8)$$

Where,

Ro = outside radius of the head of the vessel,
t_{h_o} = thickness of head for outer shell

V. Radiation Shield. [9]

Radiation shield is interposed between the hot and cold surfaces.
Heat transfer from surrounding to the shield = heat transfer from the shield to the inner vessel (3)

$$Q_{2-s} = Q_{s-1} = \frac{\sigma\,(T_2^4 - T_S^4)}{\frac{1}{e_2}+\frac{1}{e_S}-1} = \frac{\sigma\,(T_S^4 - T_1^4)}{\frac{1}{e_1}+\frac{1}{e_S}-1} \qquad (9)$$

T_2 = temperature of surrounding
T_1 = temperature of liquid nitrogen
e = emissivity factor
T_s = temperature of radiation shield

VI. Heat Transfer through Vacuum.[1,3,7]

The use of vacuum insulation essentially eliminates two components of heat transfer solid conduction gaseous convection. Heat is transferred across the annular space of a vacuum insulated vessel by radiation from the hot outer jacket to the cold inner vessel and by gaseous conduction through the residual gas within the annular space.

Heat Transfer Through Radiation: [3]

The radiant heat transfer rate between two surfaces is given by the modified Stefan-Boltzman equation,

$$Q_r = F_e F_{1-2}\,\sigma\,A_1\,(T_s^4 - T_1^4) \qquad (10)$$

Where Q_r = heat transfer by radiation
F_e = emissivity factor
F_{1-2} = configuration factor
σ = Stefan-Boltzman constant = 56.69 x 10^{-9} W/m^2-k^4
A_1 = surface area of inner vessel
T_s = temperature of the radiation shield
T_1 = temperature of inner vessel

Subscript 1 refers to inner vessel (i.e. enclosed surface) and 2 refer to outer vessel (enclosure). In addition, the emissivity factor for diffuse radiation for concentric spheres or cylinders is given by

$$\frac{1}{F_e} = \frac{1}{e_1} + \frac{A_1}{A_2}\left(\frac{1}{e_2} - 1\right) \qquad (11)$$

Radiant heat transfer can be reduced by interposing floating (thermally isolated) radiation shield between the hot and cold surfaces.

The emissivity factor with shield emissivity e_s is given by,

$$\frac{1}{F_e} = \frac{1}{e_1} + \frac{A_1}{A_S}\left(\frac{1}{e_S} - 1\right) + \frac{1}{e_S} + \frac{A_S}{A_2}\left(\frac{1}{e_2} - 1\right) \qquad (12)$$

Radiation through cylindrical portion:

$Q_{rl} = F_e F_{1\text{-}2}\, \sigma\, A_1 (T_s^4 - T_1^4)$ (13)

Radiation through Head Portion:

$Q_{r2} = F_e F_{1\text{-}2}\, \sigma\, A_1 (T_s^4 - T_1^4)$ (14)

Total heat transfer by radiation = $Q_r = Q_{rl} + Q_{r2}$

Heat transfer through support rods by conduction:

$Q_{rod} = (K_h - K_c)(A/l)$

Where,

K_h = Thermal conductivity integral at high temperature

K_c = Thermal conductivity integral at low temperature

A = cross-sectional area of the road

I = length of rod

Heat transfer by residual gas conduction: [8]

In addition to the heat transfer by radiation, energy is transmitted by gaseous conduction through the residual gas in the vacuum space. If the pressure of the gas is low enough that the mean free path of the gas molecules is greater than the distance between the two surfaces, the type of conduction differs from the usual continuum-type conduction at ambient pressure. For ordinary conduction with constant thermal conductivity, there is a linear temperature gradient within the medium transmitting heat. On the other hand, for free molecular conduction, the gas molecules rarely strike each other, thus an individual gas molecule travel across the space without transferring energy to the other gas molecules.

The degree of approach of the molecules to thermal equilibrium upon collision is expressed by the accommodation coefficient, defined by

$$a = \frac{actual\ energy\ transfer}{maxiximum\ possible\ \ energy\ transfer}$$

It depends upon the specific gas-surface combination in addition to the surface temperature. The accommodation coefficient factor for concentric sphere and cylinders is given by

$$\frac{1}{F_a} = \frac{1}{a_1} + \frac{A_1}{A_2}\left(\frac{1}{a_2} - 1\right) \quad (15)$$

The accommodation coefficient factor with shield is given by

$$\frac{1}{F_a} = \frac{1}{a_1} + \frac{A_1}{A_s}\left(\frac{1}{a_s} - 1\right) + \frac{1}{a_s} + \frac{A_s}{A_2}\left(\frac{1}{a_2} - 1\right) \quad (16)$$

The energy transfer rate by molecular conduction may now determine from:

$Q = G \times p \times A_1 \times (T_s - T_1)$ (17)

Where,

$$G = \frac{\gamma+1}{\gamma-1}\left(\frac{g_c R}{8\pi T}\right)^{1/2} F_a$$

R = specific gas constant

γ = specific-heat ratio = ${}^{c_p}/_{c_v}$

Boil-off rate (with radiation shield):

The total energy transfer to the vessel contents during one day is;

$E_d = Q\, t_d$, kJ/day

The total energy required to evaporate the contents of the vessel is

$E_t = p_f\, h_{fg}\, V$, kJ

Thus, the fraction of the full-vessel contents that is evaporated during one day is

E_d / E_t, % per day

VII. MATERIAL SELECTION:[2,3,5,6]

Important consideration in the selection of structural materials for liquid nitrogen vessel from cryogenic point of view include suitable mechanical and physical properties, compatibility with liquid nitrogen, fabricability, weldebility, cost and compliance with regulatory codes.

The following are required properties for liquid nitrogen storage applications:

- Tensile and shear moduli
- Thermal conductivity
- Surface emissivity
- Vacuum characteristics
- Low thermal contraction co-efficient
- Method of fabrication
- Hot and cold tensile and yield strengths
- Availability in standard shape and size
- Economical

TABLE I Allowable Stress for materials at room temperature or lower (Courtesy: ASME Code Section VIII, 1983) [2,5,6]

Material		Allowable Stress	
		Mpa	Psi
Carbon steel (for outer-shell only)	SA-285 Grade C	94.8	13750
	SA-299	129.2	18750
	SA-442 Grade 55	94.8	13750
	SA-516 Grade 60	103.4	15000
Low-alloy steel	SA-202 Grade B	146.5	21250
	SA-353-B(9% Ni)	163.7	23750
	SA-203 Grade E	120.6	17500
	SA-410	103.4	15000
Stainless steel	SA-240(304)	129.2	18750
	SA-240(304)	120.6	17500
	SA-240(316)	129.2	18750
	SA-240(410)	112.0	16250
Aluminum	SB-209(1100-0)	16.2	2350
	SB-209(3004-0)	37.9	5500
	SB-209(5083-0)	68.9	10000
	SB-209(6061-T4)	41.4	6000
Copper	SB-11	46.2	6700
	SB-169(annealed)	86.2	12500
Nickel-alloys (annealed)	SB-127(Monel)	128.2	18600
	SB-168	137.9	20000

VIII. RESULTS & DISCUSSIONS

This part of paper includes the discussion on the variation of heat transfer rate through cylindrical portion, head, total heat transfer rate, with respect to pressure. It also includes the discussion on boil-off rate with respect to pressure. All these discussions are done for two cases, i.e. with and without radiation shield. Further we also visualize that supporting rod dimensions also have significant effect on heat transfer rate. Although some of the graphs may seem similar to the reader but they are different either in value or meaning.

1. Variation of heat transfer through cylindrical portion with Vacuum

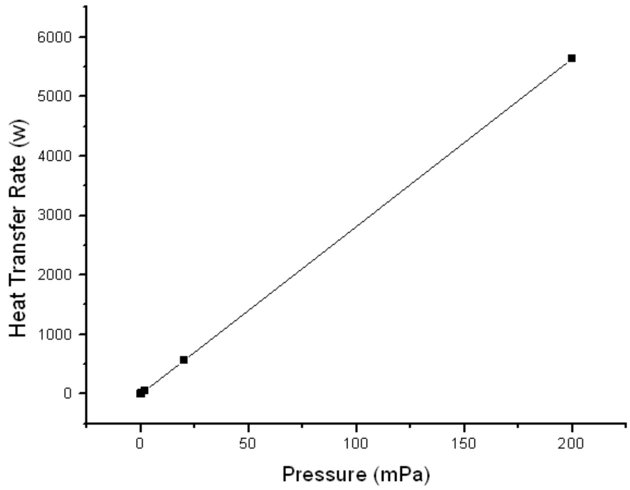

Fig.2 Pressure Vs Heat Transfer rate (With Radiation Shield)

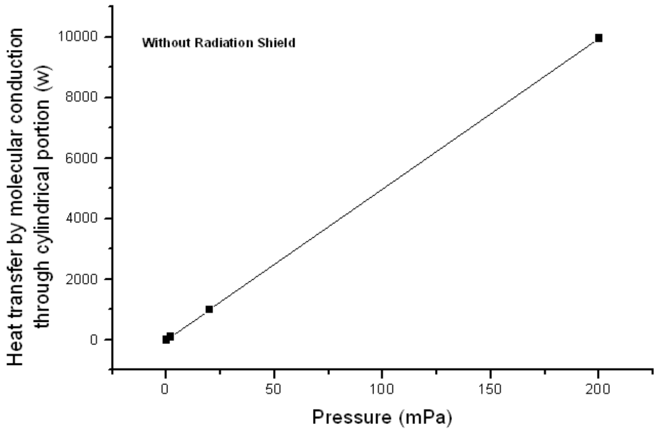

Fig.3 Pressure Vs Heat Transfer rate (Without Radiation Shield)

Figure 2 & 3 represents the variation of heat transfer rate through cylindrical portion in the presence and absecnce of radiation shield respectively.

2.Variation of heat transfer through head portion with Vacuum.

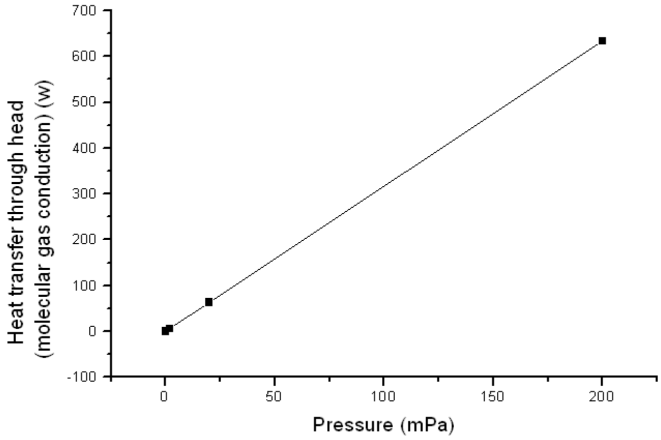

Fig.4 Pressure Vs Heat Transfer rate (With Radiation Shield)

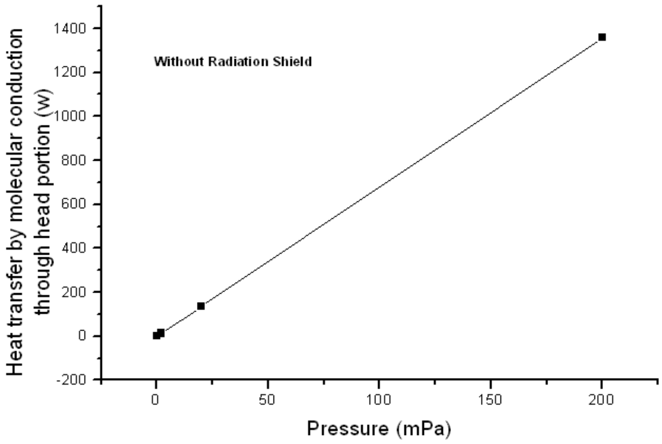

Fig.5 Pressure Vs Heat Transfer rate (Without Radiation Shield)

Figure 4 & 5 represents the variation of heat transfer rate through head portion in the presence and absecnce of radiation shield respectively.

3. Variation of total heat transfer with Vacuum keeping Q_r constant.

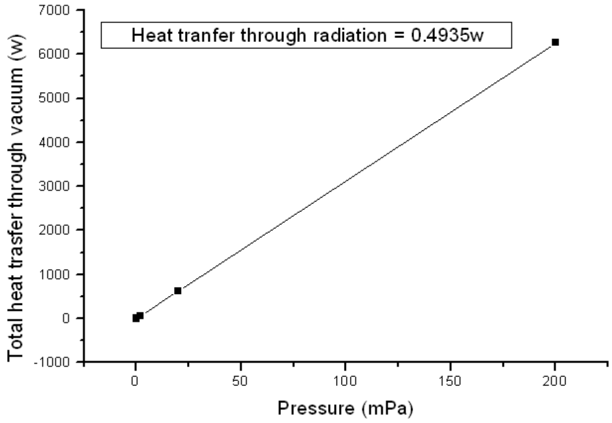

Fig.6 Pressure Vs Heat Transfer rate (With Radiation Shield)

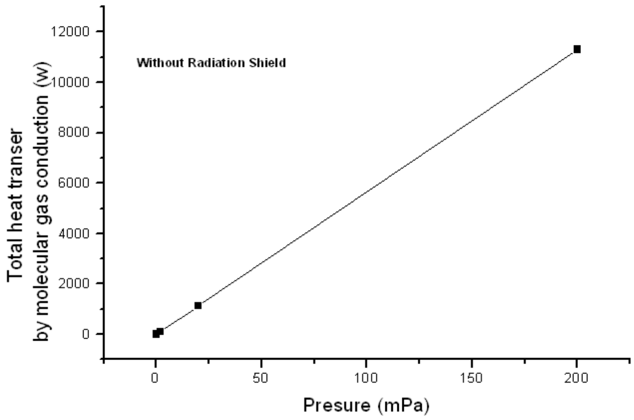

Fig.7 Pressure Vs Heat Transfer rate (Without Radiation Shield)

Figure 6 & 7 represents the variation of total heat transfer rate through vessel in the presence and absecnce of radiation shield respectively.

4. Variation of Boil-off rate with Vacuum

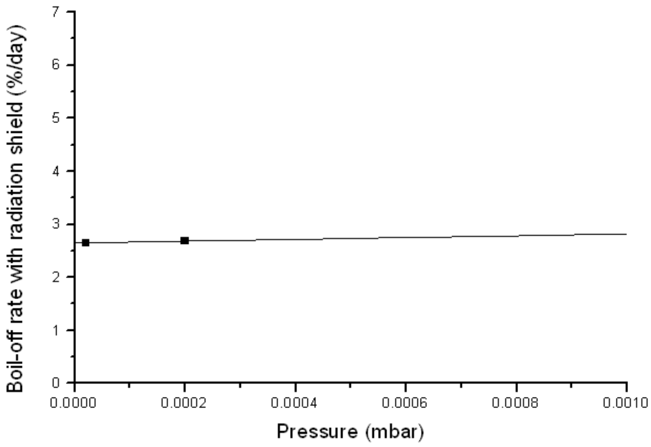

Fig.8 Boil-off rate Vs Pressure (With Radiation Shield)

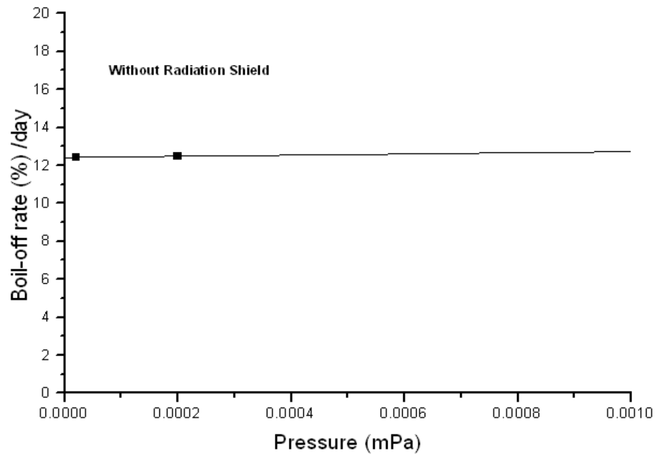

Fig.9 Boil-off rate Vs Pressure (Without Radiation Shield)

Figure 8 & 9 represents the variation of Boil-off rate through Dewar (storage vessel) in the presence and absecnce of radiation shield respectively.

5. Variation of Heat transfer rate with Radiation shield diameter.

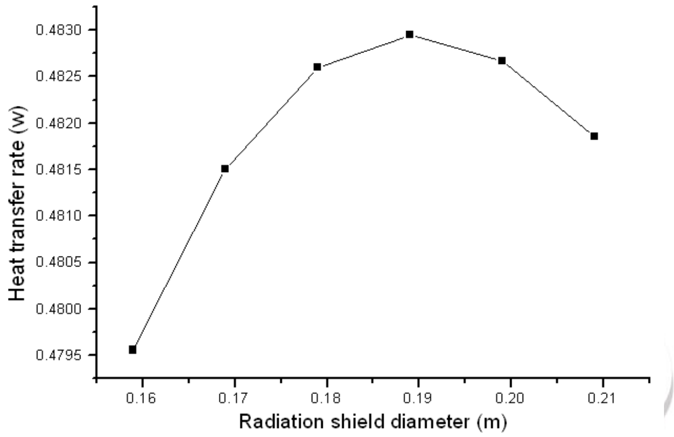

Fig.10 Heat transfer rate Vs Radiation shield diameter

From fig 10 we can observe that as the diameter of radiation shield is increasing the heat transfer rate is increasing, but up to certain limit, after this heat transfer rate starts decreasing. Here the selected value i.e. 0.189m gives the maximum heat transfer rate, among all values but since the radiation shield has to be kept between inner and outer vessel, so this value is a compromise between heat transfer rate and space availability and also the difference between heat transfer rate for all these diameters is not much, so the selection seems to be ok.

6. Variation of Heat transfer rate with supporting rod diameter for different length of rods.

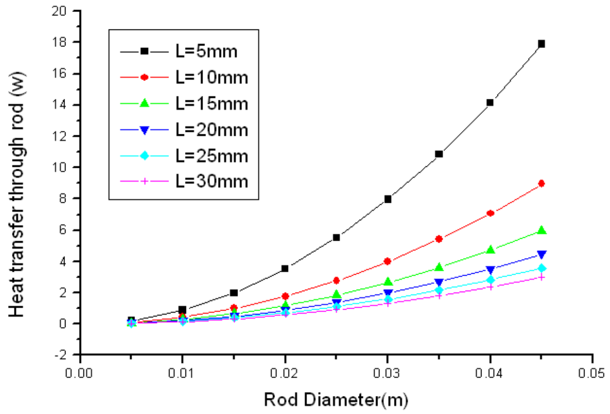

Fig 11 Heat transfer rate Vs Supporting rod diameter

From fig 11 we can see that the heat transfer rate is proportional to the supporting rod diameter, i.e. as the rod diameter is increased heat transfer rate is increased, but the selected value of diameter 20.35 mm and length 10 mm are giving heat transfer rates within acceptable limits as well as they are safe from strength point of view.

7. Variation of Heat transfer rate with supporting rod length for different diameter of rods.

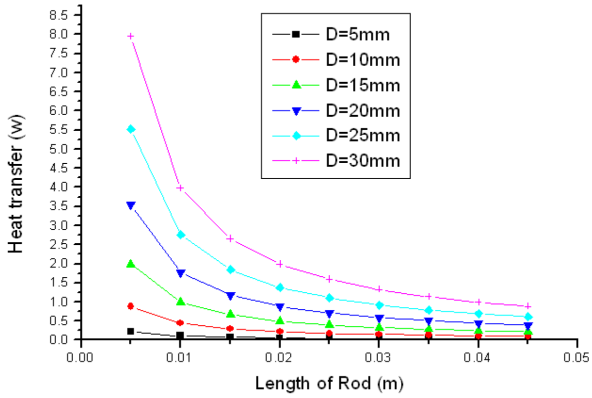

Fig 12 Heat transfer rate Vs Supporting rod length

From fig 12 we can see that the heat transfer rate is inversely proportional to the supporting rod diameter, i.e. as the rod length is increased heat transfer rate is decreased, but the selected value of diameter 20.35 mm and length 10 mm are giving heat transfer rates within acceptable limits as well as they are safe from strength point of view.

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