Modeling of multilayer vacuum insulation – complexity versus accuracy

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A thermodynamic analysis of different insulation systems used in low temperature installations is presented and the desired features of cryogenic insulation are derived. The most efficient and best matched to cryogenic conditions is a multilayer vacuum insulation (MLI). A simple mathematical model of heat transfer through MLI is proposed. Available experimental data and are compared with the model output. The limitations of the model applicability are defined and the possibilities of its upgrade discussed. A sensitivity analysis of heat flux variation with the selected physical properties of materials used to MLI production is done.

INTRODUCTION

Even the most efficient thermal insulation can not prevent parasitic heat flows to low temperature parts of any cryogenic system. To maintain a constant temperature of the cryostated object, a heat flux through thermal insulation must be compensated by cooling power generated by a refrigerator. The minimal specific input power $\dot{w}$ of the reference Carnot refrigerator, which compensates parasitic heat flux from ambient temperature T_H to cryostatic temperature T_C through the insulation with thermal conductivity k, can be described by equation 1:

$$\dot{w} = k\frac{(T_H - T_C)^2}{T_C} \tag{1}$$

It can be derived from engineering experience, that independently on a low temperature level, the refrigerator input power to compensate heat flows through 1m^2 of any insulation is of the order of 1W. Figure 1 presents typical thermal conductivity values and operation temperature ranges of cryogenic and refrigeration insulations. They lay along the line representing the value of thermal conductivity k calculated from equation (1) on the assumption that $\dot{w}$ =1 W. A vacuum insulation is characterized by the best thermal performance, however this insulation is very sensitive to the vacuum level and hence it is rather rarely used, except for special cases like very low temperature research systems. In most cases instead of vacuum insulation a multilayer vacuum insulation (MLI) is implemented. There is also a tendency observed, to replace powder vacuum insulation by MLI due to the cost reduction and industrial scale availability of the multilayer insulation.

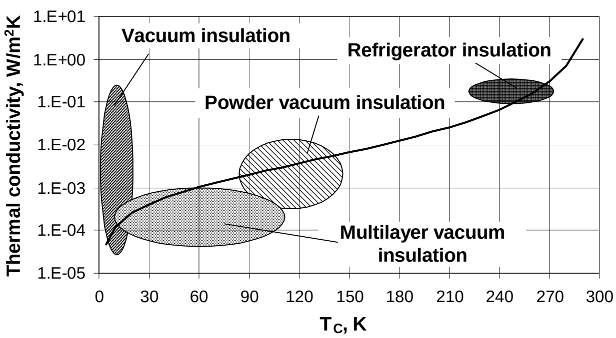

Figure 1. Typical thermal conductivity values and work temperature ranges of cryogenic and typical refrigerator insulation, line shows k calculated for w = 1 W.

MODELLING OF HEAT TRANSFER THROUGH MULTILAYER INSULATION

Multilayer insulation consists of a number of thin aluminum or metalized (aluminized or goldized) plastic foil radiation shields, alternated with a low-conductivity spacer material and placed in a vacuum space between the insulated boundary walls (see Figure 2a). A heat flux through multilayer insulation involves of three heat transfer modes: residual gas conduction, thermal radiation and solid conduction via the spacer material. Total heat flux between two adjacent layers is a sum of all three transfer modes, except for the outermost shield not thermally bridged with the boundary wall by the spacer. (see Figure 2b). To estimate numerically the heat flux through MLI, a mathematical model was developed and a dedicated computer code was created. The mathematical model is based on fundamental equations of heat transfer between two boundaries of T_1 and T_2 temperature, namely: residual gas conduction in both molecular and transient conditions (2), heat radiation (3) and solid conduction (4).

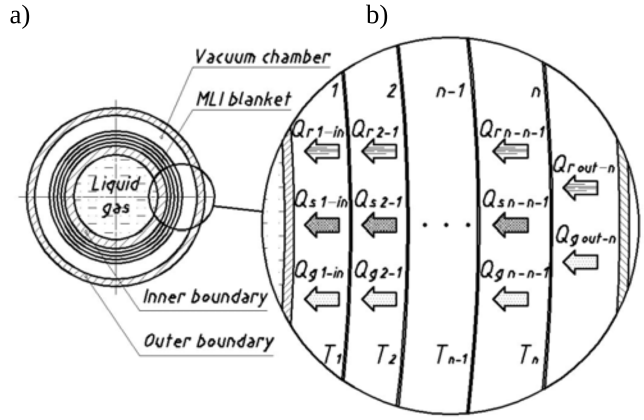

Figure 2. Structure and heat flux through multilayer insulation scheme

$$\dot{Q}_g = \frac{1}{N_a \cdot d_0^2}\left(\frac{R_u}{\pi}\right)^{1,5}\frac{9\kappa-5}{4(\kappa-1)}\left(\frac{T_0}{M_0}\right)^{0,5}\frac{F}{L+\frac{9\kappa-5}{\kappa+1}\frac{2-\alpha}{\alpha}\frac{k_B}{\pi\sqrt{2}d_0^2}\frac{T_0}{p}}(T_1-T_2) \tag{2}$$

$$\dot{Q}_r = F\frac{\sigma}{\frac{1}{\varepsilon_1}+\frac{1}{\varepsilon_2}-1}(T_1^4-T_2^4) \tag{3}$$

$$\dot{Q}_s = F\frac{\lambda}{g}(T_1-T_2) \tag{4}$$

where: d_0 - average gas molecule diameter, κ - adiabatic exponent, M_0 - gas molecular weight , α - accommodation factor, L - distance between boundaries, F - boundaries area, p - gas pressure, N_a - Avogadro number, R_u - universal gas constant, k_B - Boltzmann's constant, T_0 - average gas temperature, g - spacer thickness, λ - spacer thermal conductivity coefficient, σ - Stefan – Bolzmann constant, $\varepsilon_1/\varepsilon_2$ emissivity of hot/cold boundary.

The set of equations (2)-(4) was solved iteratively on the assumption that the sum of all heat transfer modes is constant in a steady state condition. It was also assumed, that the gas concentration in the vacuum space and between MLI layers (interstitial gas concentration) is constant, as well as the contact resistance between individual radiation shields and spacer for all the layers is the same.

Modeling results

Figures 3 and 4 show a comparison between the computed results and literature experimental data. In the calculations the inner and outer wall emissivities were equal to 0.16 for 300 K, 0.12 for 77.3 K and 0.074 for 4.2 K, the spacer material thermal conductivity coefficient was given by equation (5) and the foil shields emissivities were calculated according to (6) [1, 2].

$$\lambda = 8.823\cdot 10^{-6} + 1,04\cdot 10^{-7}\cdot T \tag{5}$$

$$\varepsilon = 0.0035\cdot T^{0.5} \tag{6}$$

Figure 3 presents the calculated and measured [3, 4] heat flux through multilayer insulation in a function of residual gas pressure for two boundary temperature ranges: 300 – 77.3 K and 77.3 – 4.2 K. The number of layers was N=10 in temperature range 77.3 K – 4.2 K, and N=30 in 300 K – 77.3 K temperature range. Figure 4 shows the measured [5] and computed results of heat transfer through multilayer insulation as a function of the number of layers for the temperature range 300 K – 77.3 K and a high vacuum i.e. for residual gas pressure lower than 10^{-3} Pa. A good agreement between the calculated and measured results is observed.

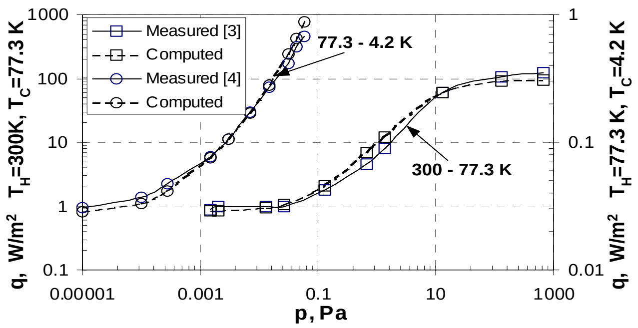

Figure 3. The MLI heat transfer variation with residual gas pressure

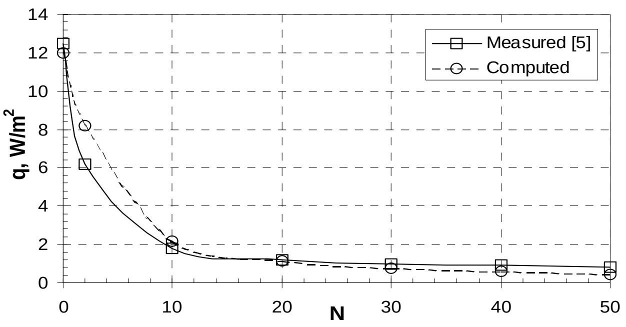

Figure 4. The MLI heat transfer variation with number of layers, 300 – 77.3 K case

Based on the model, the sensitive analysis of heat flux variation with selected physical properties of materials used to MLI production have been done, and the results are presented in Figure 5. The variated parameters were: ε_{hb} – hot boundary emissivity, ε_{cb} – cold boundary emissivity, ε_{sh} – shield surface emissivity, λ – spacer material thermal conductivity. Nominal heat fluxes are q_{nom}=0.9 W/m^2 and q_{nom}=0.03 W/m^2 for 300 – 77.3 K and 77.3 – 4.2 K temperature range adequately. Nominal values of the variated parameters were the same as given above.

a) b)

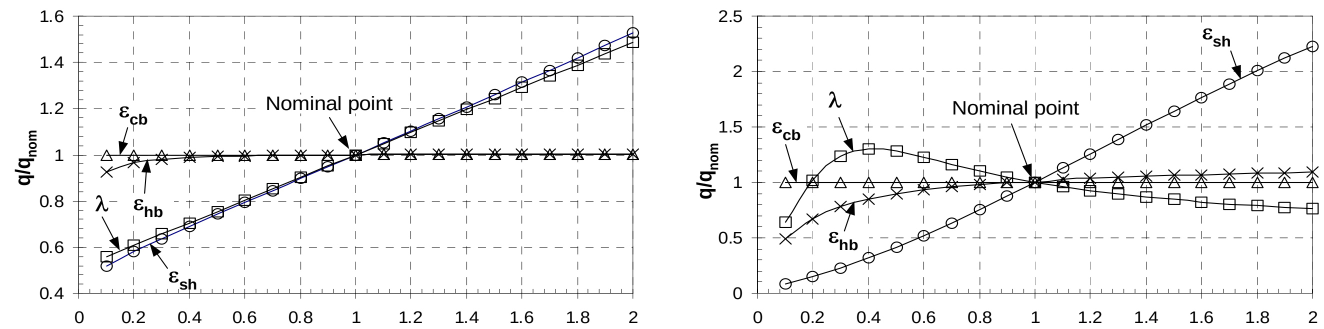

Figure 5. Heat flux variation with selected physical properties of materials used to MLI production. a) N=30 layers MLI, 300 K – 77.3 K case; b) N=10 layers MLI, 77.3 K – 4.2 K; ε_{hb} – hot boundary emissivity, ε_{cb} – cold boundary emissivity, ε_{sh} – shield surface emissivity, λ – spacer material thermal conductivity

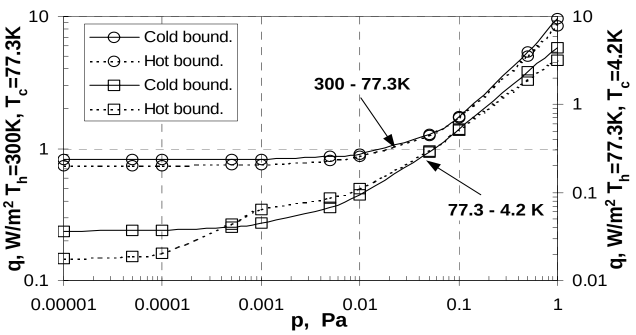

Figure 6. The influence of the location of radiation shields on the heat flux

We have also investigated the influence of the location of radiation shields on the heat flux. Due to technical reasons radiation shields are usually placed beside a cold wall. But in some cases, e.g. cold boxes, the shields can be located beside a warm boundary wall. The results calculated for MLI comprising 30 layers, for 77.3 K – 4.2 K and 300 K – 77.3 K temperature ranges are presented in Figure 6.

LIMITATIONS OF THE MODEL

S.L. Bepat et al. measured the dependence of the heat flux on the layer density expressed in layers/cm [6]. The calculated results are in the agreement with the measurements only if the layers density is below 20 – see Figure 7. It can be concluded that for higher layer densities some physical parameters having an influence on the heat transfer are missing in the model.

The dependence of heat flux on the layer density is shown schematically in Figure 8. If the density is low (left side region on Figure 8), there are only a few shield-spacer contact points and the contact thermal resistivity can be treated as infinite. Moreover, there are good conditions to evacuate residual gas from interlayer space during vacuum pumping process. This situation is corresponding to the presented above model assumptions.

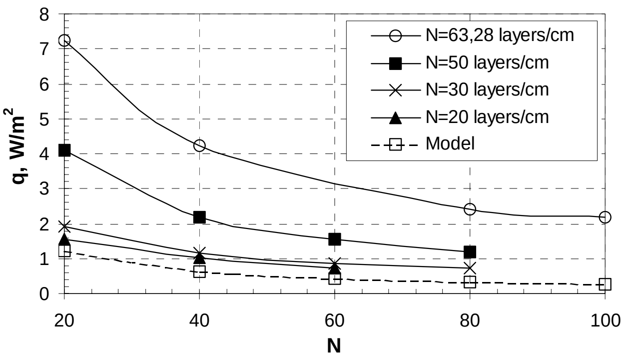

Figure 7. The MLI heat transfer variation with layers density for different layers number, 300 – 77.3 K case

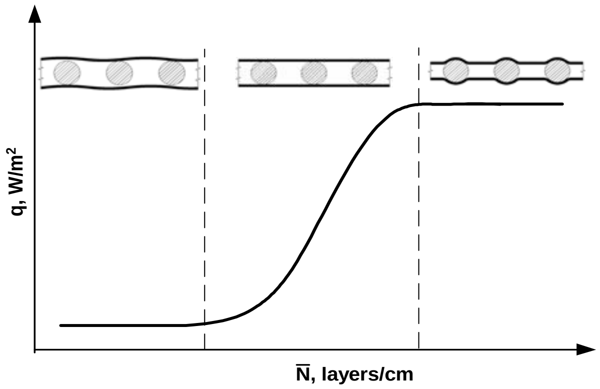

Figure 8. The influence of layer density variation on the heat flux expected character

To provide accurate results for layer density higher than 20 the following phenomena should be taken into account into the model: out gassing rate of radiation shield and spacer materials, the conductivity of radiation shield orifices, interlayer passage conductivity and contact thermal resistance in function of the layer density. Interstitial pressure in the *i-th* interlayer passage p_i can be described by:

$$p_i = 2 j_o \cdot L_p \cdot \left(\frac{1}{C_0} + \frac{L_p}{c} \right) \sum_{i=1}^{n} (n - i + 1) + p_{ch} \tag{7}$$

where: j_0 – specific outgassing rate, C_0 – orifice conductivity, L_p – length of passage, c – interlayer passage conductivity, n – layer amount, i – number of interlayer passage, p_{ch} – vacuum chamber pressure.

CONCLUSIONS

Thermal insulations used in refrigeration and cryogenics are characterized by a similar work demand to compensate unit area heat fluxes to low temperature parts of the installations.
It is sufficient to consider residual gas conduction, radiation and solid conduction when describing heat transfer through MLI when layer density is below 20 layer/cm. For higher layer densities it is necessary to take additional phenomena into account, namely: out gassing rate of radiation shield and spacer materials, the conductivity of radiation shield orifices, interlayer passage conductivity and contact thermal resistance.

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