NUMERICAL INVESTIGATION OF STEADY-STATE THERMAL BEHAVIOR OF AN INFRARED DETECTOR CRYO CHAMBER

by

Mayank SINGHAL^a, Gaurav SINGHAL^{b*}, Avinash C. VERMA^b, Sushil KUMAR^a, and Manmohan SINGH^a

^a Solid State Physics laboratory, Timarpur, Delhi, India ^b Laser Science and Technology Centre, Metcalfe House, Delhi, India

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An infrared (IR) detector is simply a transducer of radiant energy, converting radiant energy into a measurable form. Since radiation does not rely on visible light. it offers the possibility of seeing in the dark or through obscured conditions, by detecting the IR energy emitted by objects. One of the prime applications of IR detector systems for military use is in target acquisition and tracking of projectile systems. The IR detectors also have great potential in commercial market. Typically, IR detectors perform best when cooled to cryogenic temperatures in the range of nearly 120 K. However, the necessity to operate in such cryogenic regimes makes the application of IR detectors extremely complex. Further, prior to proceeding on to a full blown transient thermal analysis it is worthwhile to perform a steady-state numerical analysis for ascertaining the effect of variation in viz., material, gas conduction coefficient, h, emissivity, ε , on the temperature profile along the cryo chamber length. This would enable understanding the interaction between the cryo chamber and its environment. Hence, the present work focuses on the development of steady-state numerical models for thermal analysis of IR cryo chamber using MATLAB. The numerical results show that gas conduction coefficient has marked influence on the temperature profile of the cryo chamber whereas the emissivity has a weak effect. The experimental validation of numerical results has also been presented.

Key words: *infrared detector, cryo chamber, gaseous conduction, thermal analysis*

Introduction

The application of thermal sciences is not only limited to common engineering scenarios but is being potentially extended to realms which have not been envisaged earlier. One such modern application, which has come to the fore, is in the analysis and prediction of thermal behavior of IR detector cryo chamber. The IR detectors are devices, which are highly sensitive to temperature and require special cryo chambers, maintained under high vacuum, for housing them for their optimal functioning.

The IR detectors have been called the *eyes of the digital battlefield*. In general, military applications have spearheaded and dominated the requirements in this field akin to many other emerging fields. One of the prime applications of IR detector systems for military use is in

^{*} Corresponding author, e-mail: singhal_g@rediffmail.com

target acquisition and tracking of projectile systems [1]. The IR detectors also have great potential in commercial market with the market share expected to grow by over 70% in volume and 40% in value [2]. Non-military uses of IR detector systems include thermal efficiency analysis, remote temperature sensing, short-ranged wireless communication, spectroscopy, and weather forecasting.

In general, only scant amount of data [3-5] regarding thermal analysis of IR devices is available in open literature more so because of classified nature of its applications. An experimental study by Kang *et al.* [3] and an analytical study for steady-state cooling have been reported by Kim and Kang [4]. Another experimental study by Kim *et al.* [5] investigating cryo chamber operation at pressures in the region of 10^{-3} torr or below has been reported.

However, detailed works attempting rigorous thermal modeling of cryo chamber detector assembly are still lacking. The thrust of the paper, therefore, is to present a steady-state numerical analysis of cryo chamber detector assembly taking into account all heat loads in their original form.

Numerical simulation

The investigation of cooling characteristic of an IR detector cryo chamber is actually a relatively complex heat transfer problem. This occurs primarily because of the kind of heat transfer processes involved in the analysis. In a cryo chamber, conduction and radiation occur simultaneously and both steady and transient responses are important. Due to these seemingly complex heat transfer processes, very few modeling efforts [4] have been undertaken till date.

This section deals with the modeling of heat transfer processes and the analysis of effect of variation of parameters on temperature profile of cryo chamber in steady-state operating mode.

The thermal modeling for a generalized cryo chamber domain has been carried out using the typical configuration shown in fig. 1, which also shows the solid works view and typical hardware.



Figure 1. Typical cryo chamber: (a) solid works view, (b) hardware, and (c) modes of heat transfer

Basically, the cryo chamber has the following components viz., (1) vacuum vessel (outer cylinder), (2) a cold well (inner axi-symmetric thin walled structure), (3) a feed through unit, (4) the IR detector is located on the top of the cold well, and (5) the IR window *i. e.* the roof of the cylinder through which the detector receives the signal. The vacuum vessel is made of stainless steel material, whereas the cold finger is a thin cylindrical vessel made of glass or steel. It is evident that the thickness in case of glass will be greater than in steel on account of its mechanical integrity. The base is again made of stainless steel.

The space between the vacuum vessel and the cold well is maintained under vacuum, to minimize heat loss, by evacuation through a pump. The cold well shape is mostly cylindrical and the cooling is executed by an external cooler, which may be based on various refrigeration schemes (Stirling, Joule – Thompson, *etc.*).

In order to correctly solve the problem the foremost objective is to ascertain the possible modes of heat transfer. Simultaneously, it is also essential to identify those possible modes of heat transfer whose contribution is negligible. In the final analysis these latter modes of heat transfer will be neglected greatly simplifying the problem to obtain realistic results. The possible modes of heat transfer in the present case are:

- (1) Heat conducted from the base to the top of the cold well along the thin wall.
- (2) Radiative heat transfer occurs between the inner surface of the vacuum vessel and outer surface of the cold well.
- (3) Natural convection on to the cold well.
- (4) Although, domain is under vacuum, however, the remaining gas may participate in heat transfer on account of gaseous conduction.

Here, it is pertinent to mention that heat load due to natural convection, designated as (3) may be neglected. The reason is that due to small domain dimension, the Nusselt number determined on the basis of gap spacing for the given annular interior space (solved as a vertical large aspect ratio enclosure) is small [6]. Moreover, the Rayleigh number (Ra = $\Pr g \Delta T \delta^3 / T_{avg} v^2$), is much smaller (~1) than critical Rayleigh number, Ra_{crical}, owing to small annular spacing, δ (~5 mm) and high kinematic viscosity (5·10⁻³ m²/s) at low *T* and *P* negating the onset of natural convection due to Rayleigh-Bernard instability. This is because the Ra_{crical} is required to be >657 [7] for free boundaries, >1100 [8] for rigid-free case, and for rigid-rigid geometry >1708 [9] (present case). Further, gas conduction is dealt using a coefficient, *h*, units of Wm⁻²K⁻¹ [6, 10], apparently because *k/L* ~ *h* (are of the same order).

Assumption

- (1) Heat transfer due to convection may be neglected.
- (2) Since the cold well is a thin cylinder (typical thicknesses close to 1 mm), the conduction may be considered to be 1-D *i. e.* along the axis. If one were to calculate the Biot number, it would turn out be of the order of 0.02, indicating that transverse gradients are negligible.
- (3) The cold well finger inserted into the bore has the same temperature distribution as the cold well, thus neglecting heat transfer analysis at the internal surface of the cold well.
- (4) The thermal contact resistance between the metal (stainless steel) base and the glass bore is negligible since the glass is typically fused to the base. Therefore, T_b is the same as the temperature of the metal base, which is identical to the ambient temperature, T_{∞} .
- (5) At the other end detector array is bonded to the glass bore by an epoxy and it is assumed the temperature of the end of glass bore is same as the detector, T_d .
- (6) The shape factor, which in the present case is the fraction of energy leaving the vacuum vessel and intercepted by the cold well, for radiation heat transfer is unity.

The thermal modeling of the cryo chamber has been carried out for steady-state case. The basic energy balance for an elemental volume of the cold well in case of steady-state heat transfer is shown in fig. 2.

Hence, this can be mathematically represented:

$$Q_w + Q_c + Q_R = Q_e \tag{1}$$



Figure 2. Heat transfer in an elemental volume of cold well

The terms Q_w and Q_e represent the heat being conducted into the finite volume and away from it and can be expressed using the Fouriers law. The Q_c represents the term because of *gaseous conduction* and is expressed as function of heat transfer coefficient, temperature difference, and the cir-

cumferential area of the finite volume under consideration. The Q_R represents the radiative heat transfer which is mathematically expressed by means of the Stefan Boltzmann law. Thus, eq. (1) may be transformed as in eq. (2):

$$-kA\frac{\mathrm{d}T}{\mathrm{d}x} + P\delta xh(T_a - T_p) + P\delta x\sigma\varepsilon(T_a^4 - T_p^4) = -kA\frac{\mathrm{d}T}{\mathrm{d}x} + \frac{\mathrm{d}}{\mathrm{d}x}\left(-kA\frac{\mathrm{d}T}{\mathrm{d}x}\right)\delta x \tag{2}$$

Further, simplifying the equation we get:

$$kA\frac{d^{2}T}{dx^{2}} + Ph(T_{a} - T_{p}) + \sigma P\varepsilon(T_{a}^{4} - T_{p}^{4}) = 0$$
(3)

Equation (3) can further be modified:

$$\frac{d^{2}T}{dx^{2}} + \frac{Ph}{kA}(T_{a} - T_{p}) + \frac{\sigma P\varepsilon}{kA}(T_{a}^{4} - T_{p}^{4}) = 0$$
(4)

In order to solve the problem numerically, eq. (4) is to be integrated over the finite volume from control surface w to control surface e. Thus, eq. (4) may be expressed:

$$\int_{w}^{e} \frac{\mathrm{d}^{2}T}{\mathrm{d}x^{2}} \mathrm{d}x + \int_{w}^{e} \frac{Ph}{kA} \left(T_{a} - T_{p}\right) \mathrm{d}x + \int_{w}^{e} \frac{\sigma P\varepsilon}{kA} \left(T_{a}^{4} - T_{p}^{4}\right) \mathrm{d}x = 0$$
(5)

Term (I) Term (II) Term (III)

Term (I) and term (II) can be expressed for discretization quite easily. Hence, eq. (5) may be written:

$$\left(\frac{\mathrm{d}T}{\mathrm{d}x}\Big|_{e} - \frac{\mathrm{d}T}{\mathrm{d}x}\Big|_{w}\right) + \frac{Ph}{kA}\left(T_{a} - T_{p}\right)\Delta x + \frac{\sigma P\varepsilon}{kA}T_{a}^{4}\Delta x - \int_{w}^{e}\frac{\sigma P}{kA}\varepsilon T^{4}\mathrm{d}x = 0$$
(6)

However, a part of the term (III) is non-linear as is clearly evident in eq. (6) with finite volume representative temperature appearing as a power of 4. Hence, it needs to be linearized. The linearization procedure adopted is as suggested by Patankar [11].

The last term of eq. (6) *i. e.*:

$$\int_{w}^{e} \frac{\sigma P}{kA} \varepsilon T^{4} \mathrm{d}x$$

may be treated as a source term (S) and expressed:

$$S = S_c + S_p T_p = S^* + \left(\frac{\mathrm{d}S}{\mathrm{d}T}\right)^* \left(T - T^*\right) \tag{7}$$

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Here, *, values represent previous iteration values of the variable being sought, which in this case is T_p . It will be evident later when we discuss the operating algorithm, the solution of the eq. (5) would require iteration for non-linearity. The terms S_c and S_p turn out to be:

$$S_{c} = -3 \frac{\sigma P}{kA} \varepsilon T_{p}^{*4}$$
$$S_{p} = 4 \frac{\sigma P}{kA} \varepsilon T_{p}^{*3}$$

Thus, eq. (6) on discretization may be written:

$$\frac{T_{P+1} - 2T_P + T_{P-1}}{\Delta x} + \frac{Ph}{kA}T_a\Delta x - \frac{Ph}{kA}T_p\Delta x + \frac{\sigma P}{kA}\varepsilon T_a^4\Delta x - \frac{\sigma P\varepsilon}{kA}\Delta x \left(-3T_P^{*4} + 4T_P^{*3}T_P\right) = 0 \quad (8)$$

Thus, rearranging eq. (8) may be written in the generalized discretization form given by eq. (9):

$$a_p T_p = a_e T_e + a_w T_w + b \tag{9}$$

where

$$a_{P} = \frac{2}{\left(\Delta x\right)^{2}} + \frac{Ph}{kA} + \frac{\sigma P}{kA} \varepsilon 4T_{P}^{*3}$$
$$a_{e} = \frac{1}{\left(\Delta x\right)^{2}}$$
$$a_{w} = \frac{1}{\left(\Delta x\right)^{2}}$$
$$b = 3\frac{\sigma P}{kA} \varepsilon T_{P}^{*4} + \frac{Ph}{kA} T_{a} + \frac{\sigma P}{kA} \varepsilon T_{a}^{4}$$

Boundary conditions

- (1) The T_b is the same as the temperature of the metal base, which is identical to the ambient temperature T_{∞} *i. e.* 300 K.
- (2) The end of glass bore temperature is the one optimal for IR detector operation *i. e.* T_d which is 77 K in the present case.

Thus, the boundary conditions are case of Drichlet boundary with fixed temperatures being specified at both the boundaries.

Working algorithm

The basic algorithm for solution of steady-state cryo chamber heat transfer problem is as under:

- (1) Define the input conditions: material properties, boundary conditions, and geometry of the domain to be evaluated.
- (2) Define the number of control volumes.
- (3) Discretize the equation and calculate the coefficient a_{p} , a_{w} , a_{e} , and b from the input conditions for each control volume.

(4) Specify the applicable boundary conditions and calculate coefficients for boundary cells.

- (5) Solve the equations using standard tridiagonal matrix algorithm to generate the temperature field along the length of the cold well.
- (6) Iterate for non-linearity of the equation till convergence criterion of 1e⁻³ is met. The code has been developed in MATLAB adhering to the above stated algorithm.

Numerical results

The present section is devoted to the presentation of parametric results of thermal modeling of cryo chamber. The results for steady-state thermal numerical analysis are discussed.

Table 1. Dimension and propertiesof glass cryo chamber

Hollow glass cryo chamber (Case-I)	
Outer diameter, d_0	9.4 mm
Inner diameter, d_i	7.2 mm
Length, L	48 mm
Density, ρ	2640 kg/m ³
Gas conduction	$1.0 \text{ W/m}^2\text{K} (\equiv 0.9 \text{ Pa})$
coefficient, h	or 6.84·10 ⁻³ Torr)
Cold well thermal	0.8 W/mK
conductivity, k	
Cold well	800 J/kgK
specific heat, c_p	
Emissivity, ε	0.02
Base temperature, T_b	300 K
Detector	77 K
temperature, T_d	



Figure 3. Temperature profile along the length of the cryo chamber; three set of grids, 24, 48, and 96 have been used for grid independence study

A typical glass cryo chamber (Case-I) is considered having material and transport properties given in tab. 1. The temperature profile along the length of the cryo chamber is determined and is shown in fig 3. In order to check the efficacy of the numerical model developed the first task is to perform the grid independence check. The grid independence check is carried out by varying the number of grids. In the present case the problem has been solved employing 24 grids (size 2 mm), 48 grids (size 1 mm), and 96 grids (size 0.5 mm), and overlapping temperature profiles have been obtained, as shown in fig. 3. This proves that the profile is not changing with grid size and hence 48 grids have been typically used in the analysis.

Here, gas conduction coefficient, *h*, is a critical parameter and is determined from the theory of rarefied gas conduction [10] with our interest in low pressure regime of $5 \cdot 10^{-3}$ to 2 Torr. It is therefore, calculated using the following relations available in literature as function of pressure, *P* in Pa. Also, k_{air} thermal conductivity of air (14.8 \cdot 10^{-3} W/mK) and *D* is the enclosing vessel diameter (19.4 mm for Case I and 20 mm for Case II):

$$h = 1.48P \quad \text{for} \quad P < P_{fm} \tag{10}$$

$$h = \frac{1.48P}{1+0.34P}$$
 for $P_{fm} < P < 1 \text{ Torr}$ (11)

$$h = \frac{2k_{\text{air}}}{d_0 \ln\left(\frac{D}{d_0}\right)} = 4.35 \quad \text{for} \quad P > 1 \text{ Torr}$$
(12)

where P_{fm} is the pressure below which the free molecular gas conduction regime occurs and this limit is calculated considering that it occurs for cases where Knudsen number [defined as ratio of free molecular mean free path, λ , to the characteristics dimension, l_s , (which is gap spacing in our case)] is greater than 10. It is given:

$$P_{fm} = \frac{k_B T}{\mathrm{Kn}\sqrt{2\pi a^2 l_a}} = \frac{k_B T}{10\sqrt{2\pi a^2 l_a}}$$
(13)

Here, *a* is the molecular diameter of air (0.37 nm). The first expression is general and the latter is considering Kn =10. Thus, considering average operating temperature between 77 K and 300 K, the P_{fm} value turns out to be $2.72 \cdot 10^{-4}$ Torr. The gas conduction coefficient is hence essentially a function of pressure inside the cryo chamber, evident from eqs. (10)-(12). In present case, eq. (11) has been used for calculating *h* and for simulating pressure higher than 1 Torr, *h* value has been taken as 4.35, as shown in fig. 4, as cryo chamber geometry is fixed.

After, establishing the effectiveness of the model a parametric analysis for the same geometry is done by varying the gas conduction coefficient, h. The effect on temperature profile for various gas conduction coefficients has been shown in fig. 4.

Another, parameter that may vary owing to the extent of polishing of the cryo chamber surface is the emissivity, ε . Hence, the temperature profile has been generated by varying the emissivity and is shown in fig. 5. It is apparent that the effect of emissivity variation on overall temperature profile is marginal since the contribution of radiation heat transfer as compared to gas and thermal conduction is relatively small.



Another case analyzed was for a hollow metal cryo chamber (Case–II), the typical material and transport properties considered are given in tab. 2.

The effect of variation in gas conduction coefficient has also been studied. The results are shown in fig. 6 for gas conduction coefficient varying between 0 to 4.35. A comparative analysis of the profiles of glass and steel cryo chamber reveal that the temperature profile is much steeper in case of steel than in case of glass. It is understandable, since the major portion of heat transfer is through thermal conduction, and steel has sufficiently larger thermal conductivity as compared to that of glass.



Validation

The validation studies for the developed numerical model were carried out by performing experimental thermal testing of IR cryo chamber. The schematic and configuration of the experimental set-up used for the cryo chamber experiments for validation of the simulation results is shown in fig. 7.



Figure 7. Experimental set-up: (a) schematic, (b) photograph, and (c) 5 RTD mounted on cold well

The main components are: cryo chamber assembly with Joule-Thompson cooling provision, high pressure bottle, solenoid and regulation valves, and the filter assembly connected to data acquisition. Five numbers of resistance temperature device (RTD), type Pt-100 (Make: Omega, Model: PR-17) were mounted at, x = 0 mm, 12 mm, 24 mm, 36 mm, and 48 mm on the cryo chamber surface (shown in fig. 7) for determining the temperature profile along the length of the cryo chamber under steady-state condition.

The results for glass cryo chamber are shown in fig. 8. The effect of variation of gas conduction coefficient on account of change in cryo chamber pressure was studied for twin

pressures of 0.9 Pa, and 140 Pa for cryo chamber surface emissivity, $\varepsilon = 0.02$. The obtained experimental and numerical temperature profiles are also shown for comparison. It is evident that the experimental data is in close agreement with the numerically predicted profile. The maximum variation of nearly 9% occurs close to the maximum gradient region.

Similarly, the experimental data for the case of metallic chamber is shown in fig. 9. The experimental data compares reasonably well in qualitative terms with the numerical data (shown in fig. 6), showing a steep fall in temperature owing to high thermal conductivity of material of construction.



coefficient: h = 1.0, 4.35, emissivity, $\varepsilon = 0.02$ numerical (line), experimental (data point)

Figure 9. Experimental length profile of temperature for metallic cryo chamber for gas conduction coefficient: h = 1.0, 4.35, emissivity $\varepsilon = 0.074$

However, it is evident that in order to accurately capture the steep change in temperature quantitatively along metallic cryo chamber several more temperature sensing elements will be required. Due to practical constraints of proper mounting and corresponding feed through it was practically not feasible to mount additional such elements in the present system.

Conclusion

A suitable numerical model for performing thermal analysis of IR cryo chamber has been presented. The effects of variation in cryo chamber material, gas conduction coefficient (which in turn is a function of gas pressure), emissivity on thermal behavior are discussed. The model has been validated experimentally to prove its efficacy. It may be safely concluded that gas conduction coefficient greatly influences the temperature profile of the cryo chamber whereas emissivity has only a marginal effect on the same. The cryo chamber made of glass as the material of construction encounters a more gradual variation of temperature along the length for all gas conduction coefficients whereas for a steel cryo chamber the temperature variation is quite steep.

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