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The role of expandable thermal systems in improving performance of thermal devices

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Abstract

In this work, heat transfer through various expandable thermal systems is analyzed theoretically. These systems include single layered expandable insulation system, expandable surfaces like balloons and microchannels supported by flexible seals including both soft seals and flexible complex seals. It is found that heat transfer in expandable thermal insulation decreases as the ratio of the thermal conductivity of the gas to its gas constant (k/R) decreases. Heat convection over expanding spherical surfaces is found to be proportional to $(\frac{T_S}{T_\infty})^{2/3}(T_S - T_\infty)$ rather than $(T_S - T_\infty)$ for rigid spherical surfaces and that flexible microchannels, especially those supported by flexible complex seals are preferred to be used at lower values of Reynolds number, Prandtl number and aspect ratio with uniform temperature at the inlet. Finally, expandable thermal systems can pave the way for a new class of thermal devices with favorable thermal characteristics.

Keywords: Microchannel; Insulation; Expandable surfaces; Heat transfer; Seals

1. Introduction

Electronic devices especially those with large integration density of chips in digital devices have increased current-voltage handling requirements. As such, they require removal of large amount of dissipated heat. Microchannel heat sinks are one of the proposed methods to remove this excessive heating [1–8].

One of the important drawbacks of microchannel heat sinks is that coolant temperature becomes very large as large amount of heat is carried out by a relatively small amount of coolant. As such, the new technologies developed in the work of Khaled and Vafai [9–12] provide new methodology for cooling of electronic components utilizing microchannel heat sinks. In summary, this new technology is based on utilizing flexible soft seals to separate the microchannel plates in contrary the use of rigid systems. Khaled and Vafai [9,11] demonstrated that additional cooling can be achieved if flexible thin films including flexible microchannel heat sinks are utilized. In this work, the expansion of the flexible thin film including flexible microchannel heat sink is directly related to the internal pressure. Khaled and Vafai [12] have also demonstrated that significant cooling inside flexible thin films including flexible microchannel heat sinks can be achieved if the supporting seals contain closed cavities which are in contact with the heated surface. They referred to this kind of sealing assembly as “flexible complex seals”. Moreover, Khaled and Vafai [12] demonstrated that flexible complex seals along with thin films can be used in different designs to achieve optimal flow and thermal characteristics thermal devices.

Expandable systems are used in the work of Khaled and Vafai [13] to improve the insulating properties of an insulator. Their methodology is built on taking advantage of the volumetric thermal expansion property of gases. Their results have paved the way for construction of compact insulating assemblies that enhance insulating properties especially at high operating temperatures. The proposed expandable systems constitutes a new technology which requires additional investigation in order to explore its advantages.

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Nomenclature

B	microchannel length	m
c_p	specific heat of the coolant	$\mathrm{J\,kg^{-1}\,K^{-1}}$
F_n	dimensionless expansion (or fixation) parameter	
F_T	dimensionless thermal expansion parameter	
H	microchannel thickness or insulation thickness	m
H_o	reference microchannel thickness	m
h, h_c	convective heat transfer coefficient	$\mathrm{W\,m^{-2}\,K}$
k	thermal conductivity of the fluid	$\mathrm{W\,m^{-1}\,K^{-1}}$
Nu	lower plate's Nusselt number	
Pr	Prandtl number, $\mu c_p/k$	
p	fluid pressure	$\mathrm{N\,m^{-2}}$
q	heat flux at the lower plate	$\mathrm{W\,m^{-2}}$
Re	Reynolds number, $\rho u_m H/\mu$	
Re_o	dimensionless pressure drop, $\rho u_m H_o/\mu$	
T, T_1	temperature in fluid and the inlet temperature	K
T_S	surface temperature	K
T_∞	free stream temperature	
U	dimensionless axial velocities, u/u_m	
u	dimensional axial velocities	$\mathrm{m\,s^{-1}}$
u_m	average axial velocity	$\mathrm{m\,s^{-1}}$
X	dimensionless axial coordinates, x/H	
x	dimensional axial coordinates	m
Y	dimensionless normal coordinates, y/H	
y	dimensional normal coordinates	m

Greek symbols

ε	perturbation parameter, H/B	
ε_o	reference perturbation parameter, H_o/B	
μ	dynamic viscosity of the fluid	
θ	dimensionless temperature, $(T-T_1)/(qH/k)$	
θ_m	dimensionless mean bulk temperature, $(T_m-T_1)/(qH/k)$	
θ_W	dimensionless temperature at the heated plate, $(T_W-T_1)/(qH/k)$	
ρ	density of the fluid	$\mathrm{kg\,m^{-3}}$

In this work, thermal characteristics are analyzed in three expandable devices: single layer insulator, convective systems with expanding spherical surfaces and microchannels supported by flexible seals including both soft and flexible complex seals. The governing equations for flow and energy fields are properly non-dimensionalized and reduced into simpler equations. The resulting equations are then solved. The controlling parameters are obtained and the appropriate range of operation of these expandable systems is established.

2. Problem formulations

2.1. Expandable insulating systems

Consider an expandable horizontal layer containing a gas as shown in Fig. 1(a). The temperature at the lower surface of the layer is T_1 and the upper surface of the layer is kept at a temperature T_2. The heat transfer through the layer is equal to

$$q = kA\left(\frac{T_1 - T_2}{H}\right) \tag{1}$$

where k, A and H are the thermal conductivity of the gas, surface area of the layer and the thickness of the gas layer, respectively. Utilizing Ideal gas formulation, the gas layer thickness is related to the average temperature in the gas layer through the following relation

$$H = \frac{mR(T_1 + T_2)}{2PA} \tag{2}$$

where m, R and P are the mass of the gas, the gas constant and the absolute pressure of the gas, respectively. As such, the heat transfer through the layer (Eq. (1)) can be written as

$$q = \left(\frac{k}{R}\right)\left(\frac{2PA^2}{m}\right)\left(\frac{[T_1/T_2]-1}{[T_1/T_2]+1}\right) \tag{3}$$

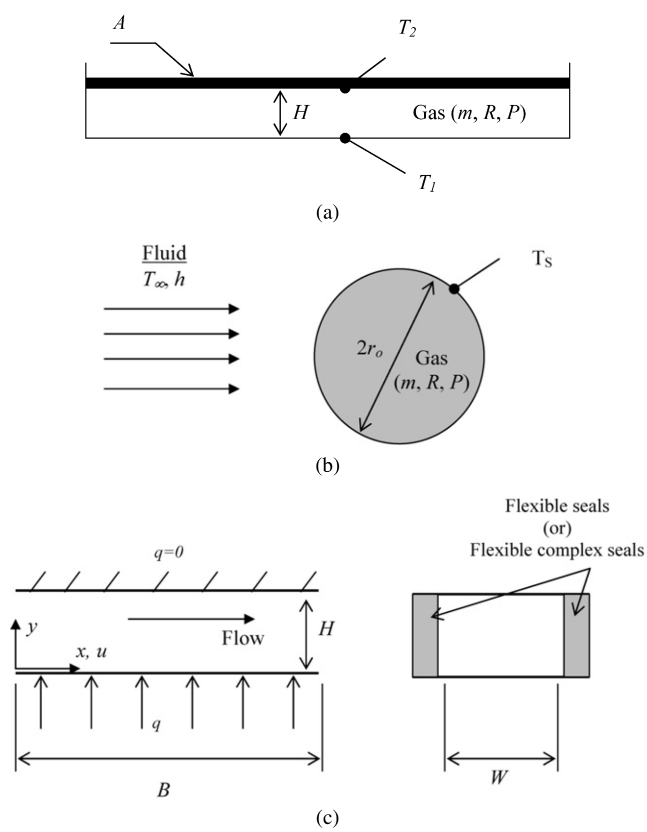

Fig. 1. (a) Expandable single gas layer insulation system. (b) Expandable heated balloon subject to convection heat transfer. (c) Expandable microchannel supported by flexible or flexible complex seals.

Insulation systems that have minimum k/R ratio are preferred to be used in expandable insulating systems. Helium and air are the most preferable gases from Table 1 as they possess the smallest k/R values.

Table 1
Various properties of proposed different gases at $T = 373$ K and $p = 1$ atm

Main gas	k [W m^{-1} K^{-1}]	ρ [kg m^{-3}]	R [J kg^{-1} K^{-1}]	k/R [kg m^{-1} s^{-1}]
Xenon	0.0068	4.3	64.05	1.06×10^{-4}
Krypton	0.011	2.75	99.78	1.102×10^{-4}
Helium	0.181	0.13	2077	8.714×10^{-5}
Neon	0.0556	0.66	412.1	1.35×10^{-4}
Argon	0.0212	1.3	209	1.014×10^{-4}
Air	0.028	1.2	287	9.76×10^{-5}

2.2. Expandable convective systems

2.2.1. Heat transfer from balloons

Consider a spherical balloon of outer radius r_o containing a gas as shown in Fig. 1(b). The balloon along with the gas has a temperature T_S. The surface of the balloon absorbs thermal radiations at a rate of q_s and it is subject to convection with the surrounding fluid with a convection coefficient h and free stream temperature of T_∞ . The convective heat transferred is equal to

$$q = h\left(4\pi r_o^2\right)(T_S - T_\infty) \tag{4}$$

Utilizing the Ideal gas formulation, the radius of the balloon is related to the balloon's temperature through the following relation

$$\frac{r_o}{r_o|_{T_S=T_\infty}} = \left[\frac{T_S}{T_\infty}\right]^{1/3} \tag{5}$$

under constant pressure expansion approximation. Therefore, the heat transfer can be written as

$$q = 4\pi(r_o|_{T_S=T_\infty})^2 h\left[\frac{T_S}{T_\infty}\right]^{2/3}(T_S - T_\infty) \tag{6}$$

2.2.2. Expandable microchannels

Consider flow inside a two-dimensional flexible microchannel heat sink with a height H and axial length B. The x-axis is aligned along the channel length while the y-axis is in the traverse direction as shown in Fig. 1(c). The fluid is taken to be Newtonian with constant average properties. The energy equation for this configuration can be written as

$$\rho c_p u\frac{\partial T}{\partial x} = \frac{\partial^2 T}{\partial y^2} \tag{7}$$

where T, ρ, c_p and k are the temperature, fluid density, fluid specific heat and thermal conductivity, respectively. The velocity field u in the channel is taken to be fully developed. As such, the velocity profile is expressed as

$$\frac{u}{u_m} = 6\left(\frac{H_o}{H}\right)\left(\left(\frac{y}{H_o}\right) - \left(\frac{y}{H_o}\right)^2\left(\frac{H_o}{H}\right)\right) \tag{8}$$

where u_m and H_o are the mean flow speed and a reference height of the microchannel. Non-dimensionalizing Eq. (7) with the following dimensionless variables:

$$X = \frac{x}{B}, \qquad Y = \frac{y}{H_o}, \qquad U = \frac{u}{u_m}, \qquad \theta = \frac{T - T_1}{qH_o/k} \tag{9}$$

leads to the following dimensionless energy equation:

$$Re\,Pr\,\varepsilon\left(\frac{H_o}{H}\right)^2 U\frac{\partial\theta}{\partial X} = \frac{\partial^2\theta}{\partial Y^2} \tag{10}$$

where q, T_1 and Re are the heat flux at the heated plate, the inlet temperature and the Reynolds number ($Re = (\rho u_m H)/\mu$), respectively. Pr and ε are the Prandtl number ($Pr = \upsilon/\alpha$) and the perturbation parameter ($\varepsilon = H/B$). The mean velocity is related to the pressure drop across the channel, Δp, through the following relation:

$$u_m = \frac{1}{12\mu}\frac{\Delta p}{B}H_o^2\left(\frac{H}{H_o}\right)^2 \tag{11}$$

where μ is the dynamic viscosity of the coolant.

For microchannel heat sinks supported by flexible soft seals or flexible complex seals, the separation between the microchannel's plates can be approximated according the following, respectively [12]:

$$\frac{H}{H_o} = 1 + F_n Re_o \tag{12}$$

$$\frac{H}{H_o} = 1 + F_T(\theta_w)_{AVG} \tag{13}$$

where F_n and F_T are the dimensionless expansion parameters for the flexible seal and the flexible complex seal, respectively. The elongation in the channel's height is considered to be proportional to the average dimensionless wall temperature when flexible complex seals are supporting the plates of the microchannel. The reference Reynolds number and reference perturbation parameters are defined as

$$Re_o = \frac{\rho}{\mu}\left(\frac{1}{12\mu}\frac{\Delta p}{B}H_o^2\right)H_o \tag{14}$$

$$\varepsilon_o = \frac{H_o}{B} \tag{15}$$

As such, the Reynolds number and the perturbation parameter can be expressed according to the following relations:

$$Re = Re_o\left(\frac{H}{H_o}\right)^3 \tag{16}$$

$$\varepsilon = \varepsilon_o\left(\frac{H}{H_o}\right) \tag{17}$$

As such Eq. (10) reduces to

$$Re_o Pr\,\varepsilon_o\left(\frac{H}{H_o}\right)^2 U\frac{\partial\theta}{\partial X} = \frac{\partial^2\theta}{\partial Y^2} \tag{18}$$

The dimensionless mean bulk temperature as obtained from the solution of integral form of Eq. (18) is

$$\theta_m(X) = \frac{X}{Pr\,Re_o\varepsilon_o}\left(\frac{H_o}{H}\right)^3 \tag{19}$$

Boundary conditions. The lower plate is assumed to have a uniform wall heat flux and the upper plate is considered to be insulated. As such the dimensionless boundary conditions can be written as

$$\theta(0, Y) = 0, \qquad \frac{\partial\theta(X, 0)}{\partial Y} = -1, \qquad \frac{\partial\theta(X, 1)}{\partial Y} = 0 \tag{20}$$

The Nusselt number is defined as

$$Nu = \frac{h_c H_o}{k} = \frac{1}{\theta_W(X) - \theta_m(X)} = \frac{1}{\theta(X,0) - \theta_m(X)} \tag{21}$$

where $\theta_W(X)$ is the heated plate dimensionless temperature.

2.2.2.1. Solutions for thermally fully developed conditions For thermally fully developed conditions, axial temperature gradients is equal to the gradient of the mean bulk temperature. As such, Eq. (18) reduces to

$$\left(\frac{H_o}{H}\right)U = \frac{\partial^2\theta}{\partial Y^2} \tag{22}$$

Eq. (22) has the following solution

$$\theta(X,Y) = 6\left(\frac{H_o}{H}\right)^2\left(\frac{Y^3}{6} - \frac{Y^4}{12}\left(\frac{H_o}{H}\right)\right) - Y + \theta_W(X) \tag{23}$$

The dimensionless mean bulk temperature is then equal to

$$\theta_m(X) = \frac{\int_0^{H/H_o} U\theta\,dY}{\int_0^{H/H_o} U\,dY} = -\frac{13}{35}\left(\frac{H}{H_o}\right) + \theta_W(X) \tag{24}$$

As such

$$(\theta_W)_{AVG} = \frac{0.5}{Pr\,Re_o\varepsilon_o}\left(\frac{H_o}{H}\right)^3 + \frac{13}{35}\left(\frac{H}{H_o}\right) \tag{25}$$

Under fully developed thermal conditions, Nusselt number has the following value:

$$Nu = \frac{h_c H_o}{k} = \frac{1}{\theta_W(X) - \theta_m(X)} = 2.69\left(\frac{H_o}{H}\right) \tag{26}$$

2.2.2.1.1. Microchannels supported by flexible seals When flexible seals support the plates of the microchannel, the minimum exit wall temperature occurs when the expansion parameter is

$$\left.\frac{d\theta_W(X=1)}{dF}\right|_{(F_n)_{\text{opt}}} = 0$$

$$\Longrightarrow (F_n)_{\text{opt}} = \frac{1.685}{(Re_o Pr\,\varepsilon_o)^{1/4} Re_o} - \frac{1}{Re_o} \tag{27}$$

The minimum dimensionless exit temperature is then equal to

$$\left[\theta_W(X=1)\right]_{\min} = \frac{0.835}{(Re_o Pr\,\varepsilon_o)^{1/4}} \tag{28}$$

And the ratio of $\theta_W(X=1)$ to $[\theta_W(X=1)]_{\min}$ is

$$M = \frac{\theta_W(X=1)}{[\theta_W(X=1)]_{\min}} = \frac{1.197}{(Pr\,Re_o\varepsilon_o)^{3/4}(1+F_nRe_o)^3} + 0.445(Re_o Pr\,\varepsilon_o)^{1/4}(1+F_nRe_o) \tag{29}$$

2.2.2.1.2. Microchannels supported by flexible complex seals When flexible complex seals support the plates of the microchannel, the dimensionless average temperature of the lower plate is calculated from the following expression, Eq. (25):

$$(\theta_W)_{AVG} = \frac{0.5}{Pr\,Re_o\varepsilon_o(1+F_T(\theta_W)_{AVG})^3} + \frac{13}{35}\left(1+F_T(\theta_W)_{AVG}\right) \tag{30}$$

3. Numerical analysis

Eq. (18) was descritized using three points central differencing in the transverse direction while backward differencing was utilized for the temperature gradient in the axial direction. The resulting tri-diagonal system of algebraic equations at $X = \Delta X$ was then solved using the well established Thomas algorithm (Blottner [14]). The same procedure was repeated for the consecutive X-values until X reached the value of unity.

4. Discussion of the results

Fig. 2 illustrates the effects of temperature ratio T_2/T_1 across the expandable single layered insulation system on the heat transfer. The conduction heat transfer increases as T_2/T_1 increases and it approaches the value of $q = (\frac{k}{R})(\frac{2PA^2}{m})$ for large values of T_2/T_1. Favorable insulating behavior for the proposed system is obtained when the layer is filled with larger masses of the gas at a lower pressure possessing a lower value of k/R. Fig. 3 shows that the convection heat transfer is proportional to $(T_S/T_\infty)^{2/3}((T_S/T_\infty)-1)$ rather than $((T_S/T_\infty)-1)$ when the balloon is rigid indicating that expandable surfaces may be utilized to augment heat transfer.

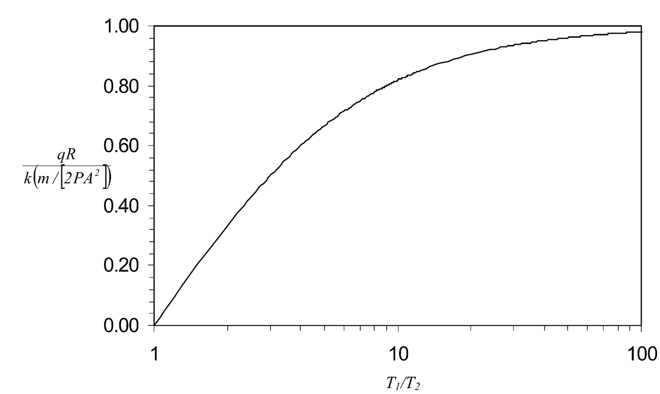

Fig. 2. Effect of temperature ratio across the expandable insulation layer T_1/T_2 on heat transfer q.

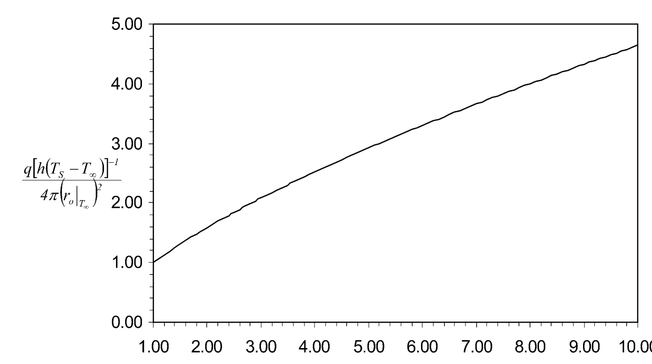

Fig. 3. Effect of the balloon temperature T_S on heat transfer q.

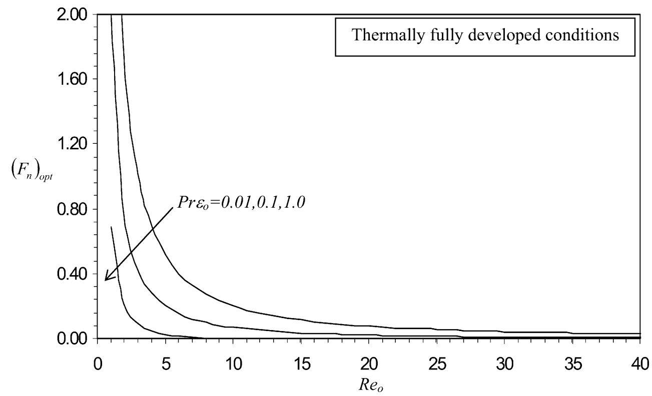

Fig. 4. Effect of Re_o and $Pr\,\varepsilon_o$ on the critical dimensionless expansion parameter F_n for thermally fully developed conditions.

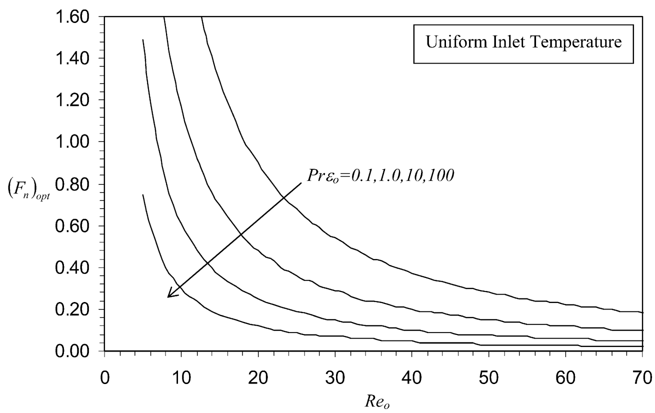

Fig. 5. Effect of Re_o and $Pr\,\varepsilon_o$ on the critical dimensionless expansion parameter F_n with uniform inlet temperature.

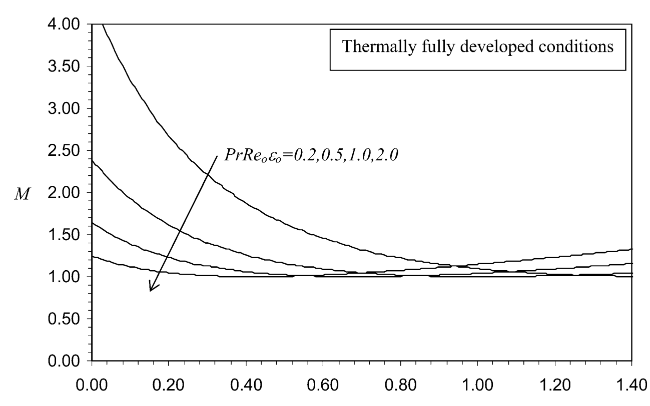

Fig. 6. Effect of the dimensionless expansion parameter F_n on the dimensionless lower plate temperature ratio at the exit wall for thermally fully developed conditions.

Figs. 4 and 5 illustrate the effects of the dimensionless pressure gradient Re_o, the Prandtl number and the reference aspect ratio ε_o on the critical dimensionless expansion parameter $(F_n)_{\text{opt}}$ for thermally fully developed flow and for a case where inlet thermal effects are considered, respectively. The average temperature of the lower plate decreases as F_n increases until F_n reaches the value of $(F_n)_{\text{opt}}$ after which the temperature starts to increase with an increase in F_n as shown in Figs. 6 and 7. The value of $(F_n)_{\text{opt}}$ decreases as Re_o, Pr and ε_o increases while inlet effects increase the value of $(F_n)_{\text{opt}}$. This indicates that cooling in microchannels supported by flexible soft seals can increase as seals become softer for a wider range of Re_o.

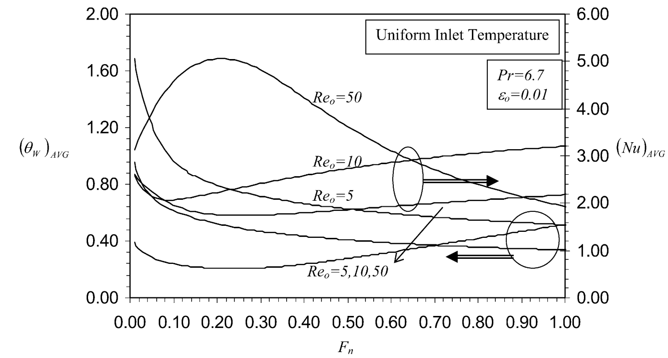

Fig. 7. Effect of F_n and Re_o on the dimensionless average lower plate temperature and average Nusselt number with uniform inlet temperature.

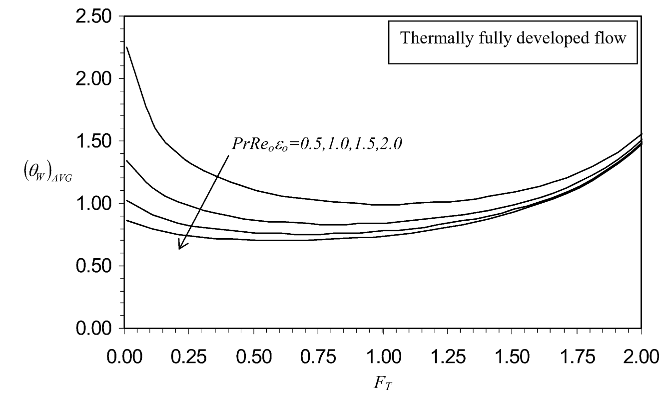

Fig. 8. Effect of the thermal dimensionless expansion parameter F_T on the dimensionless lower plate temperature for thermally fully developed conditions.

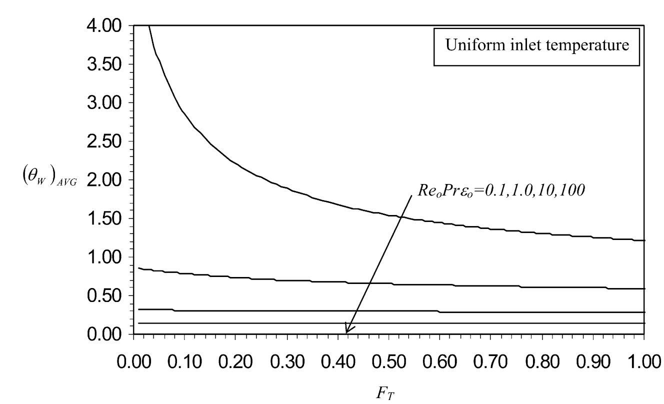

Fig. 9. Effect of the thermal dimensionless expansion parameter F_T on the dimensionless lower plate temperature ratio with uniform inlet temperature.

Fig. 8 illustrates the effects of the thermal expansion parameter F_T on the dimensionless average temperature of the heated plate for thermally fully developed flow starting from the inlet of the microchannel. The average temperature of the heated plate is shown to decrease as F_T increases until F_T reaches a critical value after which that temperature starts to increase.

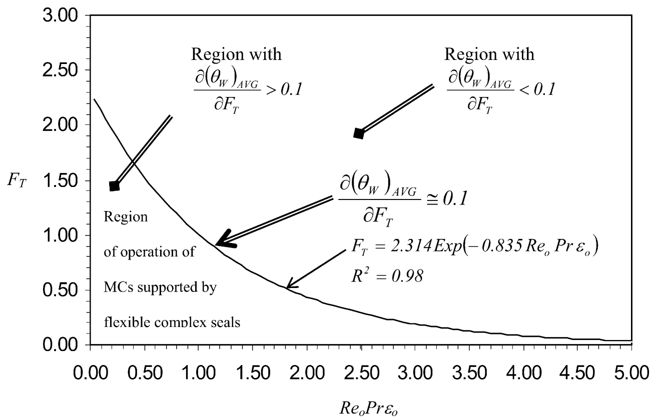

Fig. 10. Recommended values of $Re_oPr\varepsilon_o$ and F_T for the operation of microchannels (MCs) supported by flexible complex with uniform inlet temperature.

This phenomenon is not observable when the fluid enters the microchannel at a uniform temperature as shown in Fig. 9. This figure shows that the effect of thermal expansion parameter is significant at lower values of F_T and $Re_oPr\varepsilon_o$. As such, the operating conditions for microchannels are of the curve shown in Fig. 10.

5. Conclusions

Heat transfer through several expandable thermal systems were analyzed theoretically in this work. These were single layered expandable insulation system, expandable surfaces and microchannels supported by flexible seals and flexible complex seals. It was established that expandable thermal insulations reduce heat transfer as the ratio of the thermal conductivity of the gas to its gas constant (k/R) decreases and that convection over expanding spherical surfaces is proportional to $(\frac{T_S}{T_\infty})^{2/3}(T_S - T_\infty)$ rather than $(T_S - T_\infty)$ for rigid spherical surfaces. Finally, microchannels supported by soft and flexible complex seals were found to be preferable for use at lower values of Reynolds number, Prandtl number and aspect ratio with uniform inlet temperature. Finally, the finding that better thermal performance can be achieved by putting more freedom to change the configuration of thermal devices is in agreement with constructal theory, where "freedom is good for design" is one of the recurring trends [15].

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