Optimizing the Performance of the Domestic Wall Mounted Space Comfort Heater

J. Tangkeh and J Gryzagoridis

Abstract— The performance of a wall mounted space comfort heater has been studied with respect to the geometry of its mounting condition. Tests were conducted in a laboratory with the heater positioned at various heights from the floor and the channel that is created by the various gaps with the wall on which the heater was mounted. Tests were also performed with the heater mounted on the wall whose emissivity was adjusted to low, medium and high values as well as placing insulation material on the wall directly behind the heater. The experiments revealed an acceptable geometry of the heater's mounting at least 200mm above the floor, and 50 mm off-set from the wall. The heater's mounting against the wall caused a drop in performance of about 15% of it's maximum "benchmark" performance (with an efficiency of about 41%) achieved when the heater was freely standing on the floor of the laboratory. This efficiency, based on the convective heat transfer generated by the heater's warm/hot surfaces, is relative to the electrical energy input. The heat transfer by radiation from the heater's surface is treated as net loss to the walls of the room/enclosure.The performance of the heater when mounted against the wall improved almost to the benchmark value when the wall behind the heater was made refelective (low emissivity).

***Keywords* — Calorimetric measurements, Heat transfer modes, Natural Convection, Net Radiation, Grashof, Nusselt and Prandtl numbers.**

1 INTRODUCTION

Numerous researches on the natural convection mode of heat transfer (HT) from a heated vertical plate to surrounding air have been studied in great detail. Research combining experimental and numerical approaches to multi-mode heat transfer between vertical parallel plates was conducted [1], with the aim to determine the significance of radiation's contribution to the total heat transfer rate on a steel plate, in a relatively big channel gap, and the validity of the isothermal surface assumption. Geometry tests (variation of height from the floor and gap space from the wall) were performed for a vertical plate with insulation placed on the surface of the plate facing the wall [2], attempting to match the results with those obtained through a simulation using ANSYS. The results showed an increase in the heat transfer coefficient from the heated surface of the plate, when the height with respect to the floor was smaller than the gap space from the wall. An experiment was conducted on the numerical analysis of the flow and heat transfer characteristics [3] of buoyancy-driven air convection behind photovoltaic panels to validate the effect of different input heat fluxes and emissivity of the bounding surfaces on the heat transfer across the air layer. Similarly in another experiment [4], validation of the predicted results for the mass flow rate, velocity, temperature rise and location of neutral height (location where the pressure in the air gap is equal to the ambient pressure) behind solar cells located on vertical facades, was achieved by varying the geometry of the air gap and the location of the solar cell module.

In the present study, the heater's performance was evaluated at various geometries of its mounting and compared to its bench mark performance when positioned standing freely in the middle of the enclosure. This performance was also investigated with the heater mounted on the wall whose emissivity was adjusted to low, medium and high values as well as placing insulation material on the wall directly behind the heater.

The heat transferred from the heater's outer surface to the enclosure space was determined considering the air properties at film temperature (defined as the average temperature between the hot surface's and the bulk of the fluid's temperatures), through the dimensionless numbers presented respectively in [5] and [6], that enable the determination of the coefficient of convective heat transfer when combined in a relevant power law relationship:

The Prandtl number is a dimensionless parameter defined as the ratio of the fluid's hydrodynamic to its thermal boundary layer. The Prandtl number is expressed as:

$$Pr = \mu \times Cp / k \quad (1\text{-}1)$$

The Grashof number is a dimensionless parameter representing the ratio of the buoyancy forces to the viscous forces in the free convection flow system. The Grashof number is expressed as:

$$Gr_L = \rho^2 \times g \times \beta \times (T_S - T_\infty) \times L^3 / \mu^2 \quad (1\text{-}2)$$

The Rayleigh number is a dimensionless number obtained through the product (function) of Grashof and Prandtl numbers expressed as:

$$Ra_L = \text{Function}\ (Pr, Gr_L) \quad (1\text{-}3)$$

The Nusselt number is defined as the ratio of the convective to the conduction heat transfer in the boundary layer, expressed as:

$$Nu_L = h \times L / k \quad (1\text{-}4)$$

The equation shown below, as proposed by Churchill and Chu [7], is the power law relationship that matches the conditions of natural or free convection heat transfer from an (assumed) isothermal heated surface in air, was judged to match the conditions of this research

$$Nu_L = 0.68 + \frac{0.67 Ra_L^{\frac{1}{4}}}{\left[1+\left(\frac{0.492}{\Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}} \quad \text{for } Ra_L < 10^9 \qquad (1\text{-}5)$$

The Coefficient of free convective heat transfer h may be obtained using equation 1-4 above

The Convection heat transfer rate from the surfaces of the heater can be evaluated using the following equation:

$$Q_{conv} = h \times A \times (T_S - T_\infty) \qquad 1\text{-}6)$$

In the present study, the global behaviour of heat transfer through the vertical channel gap formed by the heater's inner surface and the wall that it is mounted on was dealt firstly with the empirical formula 1-5 that assumes the existence of a classical boundary layer on both surfaces of the heater. Because of the disparity on the heat balance on the heater by the sum of convective and radiative heat losses with the electrical energy input, a calorimetric approach for the measurement of heat through the channel was adopted. This entailed determining the amount of heat gained by the air stream whilst flowing through the channel, given by the relationship.

$$Q_{cal} = \dot{m}\, Cp_{inlet} (T_{outlet} - T_{inlet}) \qquad (1\text{-}7)$$

where,

$$\dot{m} = \rho_{inlet} \times v_{inlet} \times Ac \qquad (1\text{-}8)$$

The equation for the net radiation heat transfer between two infinitely parallel vertical surfaces and from a vertical surface that does not see any part of itself surrounded by an enclosure, was used for the evaluation of the net radiation heat transfer inside the channel and from the outer surface of the heater to the surrounding enclosure's walls respectively. It is given by:

$$Q_{1\text{-}2} = \sigma(T_1^4 - T_2^4)/((1-\varepsilon_1)/(A_1\varepsilon_1) + 1/(A_1F_{1\text{-}2}) + (1-\varepsilon_2)/(A_2\varepsilon_2)) \qquad (1\text{-}9)$$

The parameters of the various dimensionless numbers and equations 1-1 to 1-9 above are defined as follows:

A Surface area of the heater (m^2)
A_1,A_2 Surface areas 1 and 2 respectively (m^2)
Ac Cross sectional area of the channel (m^2)
β Coefficient of thermal expansion of the air (1/K)
Cp Heat capacity of air at a constant pressure (J/kg/°C)
Cp_{inlet} Specific heat capacity of air at channel entry (J/kg/°C)
ε_1, ε_2 Emissivity of surfaces 1 and 2 (dimensionless)
$F_{1\text{-}2}$ Form or view factor between surface 1 and surface 2 (dimensionless)
g Gravitational constant (m/s^2)
Gr_L Grashof number with respect to the characteristic length of the heater (dimensionless)
h Coefficient of free convective heat transfer $W/(m^2.K)$
k Coefficient of thermal conductivity of the air (W/(m.K))
L Characteristic length (height) of the heater (m)
$\dot{m}$ Mass flow rate of air going through the channel (kg/s)
μ Absolute viscosity of air (kg/(m.s)) or (Pa/s)
Nu_L Nusselt number with respect to the heater's characteristic length (dimensionless)
Pr Prandtl number (dimensionless)
Q_{conv} Convection HT from the heater's surface (s) (W).
Q_{cal} Calorimetric HT through the channel (W)
$Q_{1\text{-}2}$ Net radiation HT between surface 1 and surface 2 (W)
Ra_L Rayleigh number with respect to the characteristic length of the heater (dimensionless)
ρ_{inlet} Density of the air steam at inlet of the channel (kg/m^3)
σ Stefan Boltzmann constant 5.67×10^{-8} W / ($m^2.K^4$)
T_S Temperature of the heater's hot surface (K)
T_∞ Room's air temperature (K)
T_{inner} Temperature at the surface of the heater facing the wall
T_{outer} Temperature of the heater surface facing the rest of the enclosure (K)
T_{inlet} Temperature of the air stream at channel entry (k)
T_{outlet} Temperature of the air stream at channel exit (k)
v_{inlet} Velocity of the air steam at inlet of the channel (m/s)

2 EXPERIMENTAL APPARATUS AND PROTOCOL

The tests were performed on a H 400W square model heater with a side length L = 0.6m and a thickness of 10mm.

The following methodology was adopted:

i. The first part of the work was to evaluate the performance of the heater mounted at a free position inside the enclosure at a height of 200 mm supported by a wooden frame.
ii. The heater was mounted against a normal grey wall at a height of 200 mm (manufacturer's recommendation) from the floor and its off-set from the wall (S) was varied from 10 to 60mm
iii. The heater was mounted against a normal grey wall at a gap of 50 mm from the wall that was found to be the

optimal gap and its heights (H) were varied from 0 to 300 mm.

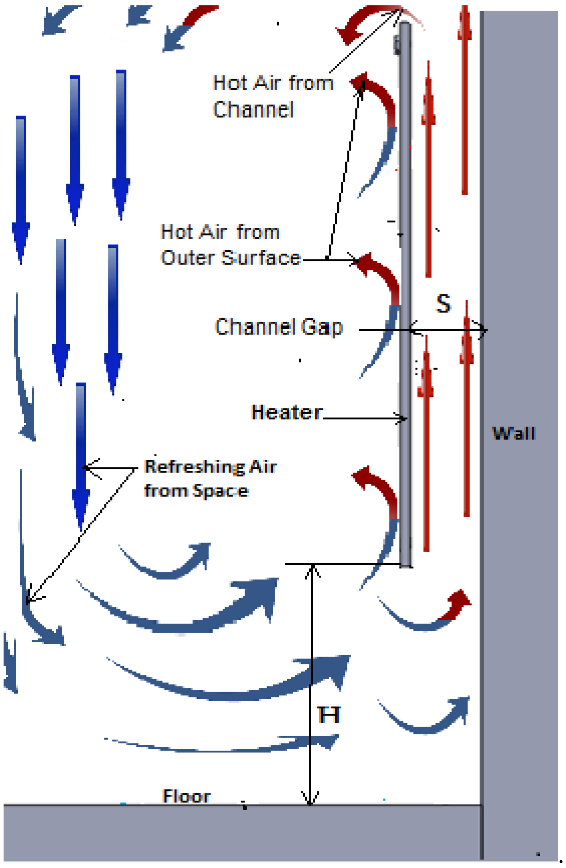

Fig. 1. Wall mounted space heater

iv. The heater was mounted on the wall whose emissivity was adjusted to low, medium and high values as well as placing insulation material on the wall directly behind the heater.

For each of the above experimental conditions, the heater was connected to an electrical supply with a normal AC voltage of 230V Assuming a unity power factor the input power was calculated (391 W). K-type thermocouples were used for all the temperature measurements on the heater and wall surfaces, whilst a digital-probe velocimeter was used for the measurement of velocity plus temperature of the air stream through the channel. The room's air temperature was measured with thermometers. Figure 1 depicts geometry of the heater mounted against the wall.

3 Results And Discussion

3.1 Experiment No 1: Heater operating freely inside the enclosure

When placed freely in the middle of the enclosure far from all walls, it was found that the heater delivered an efficiency of 41% in warming the enclosure's air, with respect to the total electrical energy consumed. This efficiency became the bench mark to which all other results were compared.

3.2 Experiment No 2: Heater mounted at various gaps from the wall.

Mounted against the wall, the evaluation of the heat transfer (HT) to the enclosure's space was made through the fundamental equations used for HT from the heater's outer surface and the calorimetric equation for the HT to the air stream through the channel between heater and wall.

The heater's performance at various gaps from the wall is presented in the Figure 2.

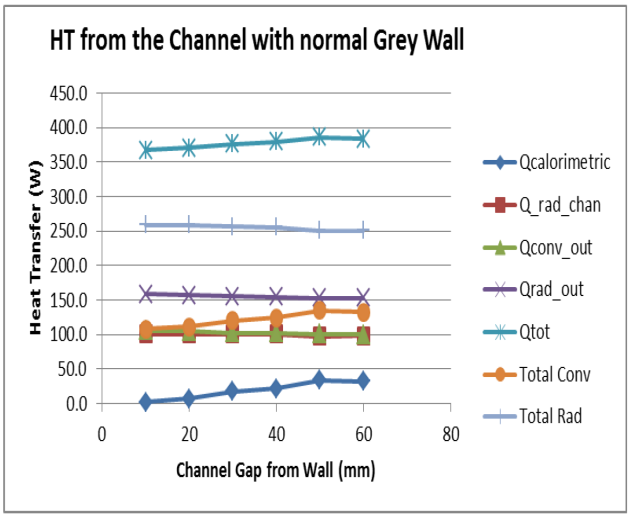

Fig. 2. Heat transfer components from the heater at various gaps from the wall.

The wall-mounting caused the heater a loss of almost 15 % of its efficiency with respect to the benchmark delivery. This resulted from sacrificing of almost 60% of the performance through the channel and a recovery of almost 30% of it from the outer surface.

The illustration of all the heat transfer components displayed in Figure 2 reveals that the increase in channel gap size between the heater and the normal grey wall behind it resulted in an increase of the overall performance of the heater, approaching a maximum value around the gap of 50mm. The calorimetric HT through the channel appeared to be the only quantity affecting the overall heater's performance, given that the convection at the outer surface and all the radiation components of HT remained constant with the gap variation. The heater's performance stays unchanged with further increase beyond the gap of 50mm. As mentioned above, the investigation of the temperature behaviour across the channel enabled a better understanding of the increase of the calorimetric heat transfer inside the channel with respect to the gap size, towards the optimal performance around the 50 mm gap, which is considerably higher than the 20 mm gap recommended by the manufacturer.

The graphs pertaining to the temperature's behaviour across the channel displayed a temperature profile that is similar to the findings of La Pica et al. (1991) [8], although their geometry was different from the geometry in this research.

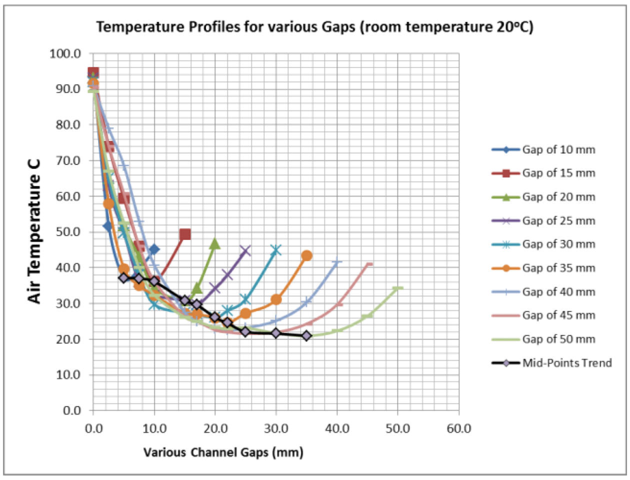

Fig. 3. Temperature profiles as a function of increasing the size of the channel's gap.

Figure 3 shows the behavior of the temperature profile with the increase of the gap size. It can be seen that the minimum value of the air's temperature profile approaches the room's air temperature (20°C), as the gap size increases

3.3 Experiment No 3: Heater mounted at various heights from the floor

The heater was mounted at a distance of 50 mm away from the wall (on the basis of the results from experiment no. 2) and further tests of its performance were conducted. The heater's position was varied from it sitting on the floor to 300 mm above it. Record of the temperatures of the channel's wall and the heater's surfaces were taken for all heights as well as the temperatures and velocity of the air stream through the channel. The subsequent analysis of these temperatures for the evaluation of the heater's performance is presented in Fig. 4. The graphs in Fig. 4 reveal that the calorimetric heat (Qcal) through the channel is as expected zero when the heater is placed on the floor. From this zero starting point, the graph increases gradually towards its peak values around the height of 200mm which incidentally is the height recommended by the manufacturer.

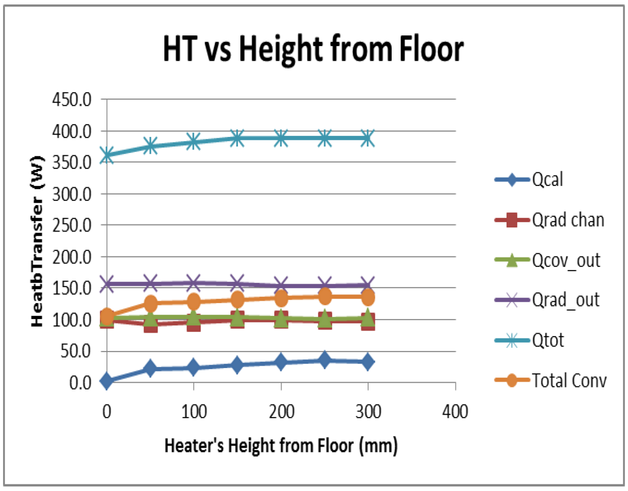

Fig. 4. HT from the heater mounted at various heights from floor.

3.4 Experiment No 4: The effect of changes of the surface conditions of the wall behind the heater

This experiment involved the change of emissivity of the wall behind the heater which would likely impact on the net radiation losses from the heater's inner surface and possibly affect the heat transfer to the air in the channel and the convection heat transfer from the heater's outer surface.

3.4.1 Black wall behind the heater

The heater was mounted at a height of 200 mm from the floor and at the optimum distance of 50 mm away from the wall (which was painted with mat black paint) and further tests of its performance were carried out. The black wall behind the heater absorbed the total radiation emitted from the inner surface facing it. Part of this absorption helped warm up the air of the channel which affected the calorimetric heat transfer through the channel. The amount of calorimetric HT through the channel, when compared to the condition of normal grey wall behind the heater, showed a small improvement (nearly 9%), however there was an overall decrease of more than 52% with respect to the maximum performance (freely-mounted) or benchmark condition. The heater's overall performance when mounted on the mat black wall is similar to that displayed when it was mounted on the normal grey wall.

3.4.2 Black Insulated Wall behind the Heater

The heater was mounted at an identical position as in experiment 3.4.1 (above) with an insulated black wall behind it and further tests of its performance were conducted. The outcome revealed that the channel's calorimetric HT reflects an improvement of more than 12% when compared to the normal grey wall mounted condition, but just about 4% with respect to the black-non-insulated wall condition. The overall performance remained almost unchanged compared to the condition of black-non-insulated wall behind the heater.

3.4.3 Shiny Wall

Aluminium foil (emissivity of 0.05) was used to make the wall behind the heater highly reflective. The heater was again mounted at its minimum/optimum height and gap.

The temperature of the air was measured as in previous tests at the channel's inlet and outlet. The results of this experiment revealed that the heater recovered nearly all its efficiency with respect to the benchmark value. With an improvement of almost 20% of the calorimetric HT through the channel and nearly 15% increase of natural convection heat transfer at the heater's outer surface, meaning an overall improvement of 16% of the heater's efficiency compared to the normal grey wall mounted condition. The 20% improvement inside the channel is probably explained by the drop in net radiation heat loss inside the channel, due to the reflection on the shiny wall. The reflection is captured back by the heater's inner surface that hence undergoes a temperature increase, which subsequently enhances the transfer of calorimetric heat to the air rising in the channel's space. Part of the reflection captured by the inner surface is transmitted via conduction through its thickness, to the outer surface of the heater which also experiences a temperature rise. This temperature increase at the outer surface subsequently enhances the convection HT from the outer surface (15%), however it generates an increase of radiation heat transfer loss to the enclosure's walls.

In summary, by making the wall behind the heater shiny,

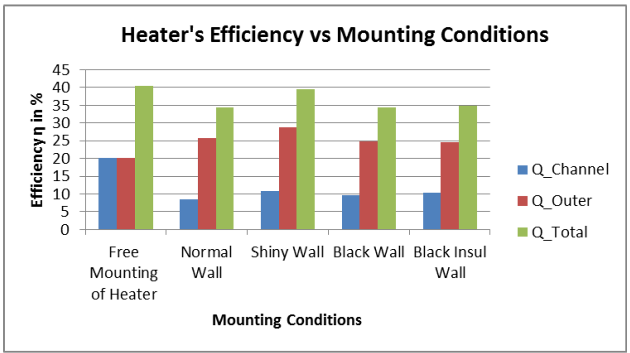

Fig. 5. Efficiency of the Heater with respect to the mounting wall's conditions.

the overall efficiency of the heater was shifted or increased to 40%, bringing it closer to the 41% which is the benchmark based on the freely standing condition within the enclosure.

The overall comparison of the heater's performance resulting from the various experiments, with respect to the benchmark or maximum performance is displayed in Figure 5. The graphs pertaining to the temperature's behaviour across the channel displayed a temperature profile that supported the findings of La Pica et al. (1991) [9], although their geometry was different from the geometry in this research.

4 CONCLUSION

The performance of the heater has been studied with respect to the geometry of its mounting. Gaps from the wall as well as the heights from the floor were the respective mounting geometry parameters that affected the heater's performance. It was found that gaps of at least 50 mm as well as heights above 200 mm appear to be the mounting geometry that provides the heater with a better performance. The temperature profiles in the channel revealed that a channel gap of at least 50 mm accommodates effectively the room temperature and hence establishes a better heat sink condition for improved/optimal heat transfer through the channel.

Further tests were conducted with the heater mounted at the optimal height and gap, against the high emissivity, insulated high emissivity and low emissivity walls.

In summary, the results of the various experiments revealed that the maximum performance of the heater was about 41%, achieved with the heater operating freely in the middle of the enclosure far from the walls. Mounting the heater against the wall causes its overall performance to fall to 34% of which only 30% of it is transferred through the channel. When mounted at optimum height and gap against the highly reflective wall (low emissivity), the heater recovers almost all of its performance.

It is recommended that further research should be undertaken to thoroughly investigate the "mode" of heat transfer, by the self-induced flow through the channel, in a more formal or scientific modelling approach, that would cater for the particular geometry of the channel.

REFERENCES

[1] Krishnan, A.S., Premachandran, B., Balaji, C. and Venkateshan, S.P., (2004, Dec.), Combined Experimental and Numerical Approaches to Multi-mode Heat Transfer between Vertical Parallel Plates. *Experimental Thermal and Fluid Science*, [Online]. 29(1), pp 75-86. Available: http://www.sciencedirect.com/science/article/pii/S089417770400038X.

[2] Comunelo, R. and Güths, S., (2005, Nov.), Natural Convection at Isothermal Vertical Plate: Neighborhood Influence, presented at the 18th International Congress of Mechanical Engineering, Ouro Preto, MG. [Online]. Available:: http://www.lmpt.ufsc.br/publicacao/86.pdf

[3] Sandberg, M. and Moshfegh, B., 2002, "Buoyancy-induced Air Flow in Photo-voltaic Facades, Effect of Geometry of the Air Gap and Location of Solar Cell Modules", *Building and Environment*, [Online]. 37, pp 211-218. Available: https://www.researchgate.net/publication/245145314_Buoyancy-induced_air_flow_in_photovoltaic_facades

[4] Moshfegh, B. and Sandberg, M., (1998, Apr.), Flow and heat transfer in the air gap behind photovoltaic panels, *Renewable and Sustainable Energy Reviews*, [Online]. 2, pp 287-301. Available: https://www.researchgate.net/publication/227421632_Flow_and_heat_transfer_in_the_air_gap_behind_photovoltaic_panels

[5] CENGEL, A.Y. and Ghajar A.J., Natural Convection, in *Heat and Mass Transfer: Fundamentals and Applications*, 5th ed., McGraw-Hill Education, New York, 2015, ch. 9, sec. 9-1 to 9-3, pp. 534-541.

[6] Gryzagoridis, J., (1971, Jan) Natural Convection from a Vertical Flat Plate in the Low Grashof Number Range, *International Journal of Hear and Mass Transfer*, [Online]. 14(1), pp 162-165. Available: http://www.worldpapercat.com/1747/Article4306972.htm

[7] Incropera, F.P., Dewitt, D.P., Bergman, T.L. and Lavine A.S., Radiation Exchange Between Opaque, Diffuse, Grey surfaces in an Enclosure, in *Fundamentals of Heat and Mass Transfer*. John Wiley & Sons Inc, New Jersey, USA., 6th ed. 2006, ch. 13, sec. 13.2, pp. 822- 839

[8] La Pica, A., Rodono, G. and Volpes, R. (1993), An Experimental Investigation on Natural Convection of Air in a Vertical Channel, *International Journal of Heat and Mass Transfer*. [Online]. 36(3), pp 611-616. Available: https://www.researchgate.net/file.PostFileLoader.html?id=530ede36d685ccdd2c8b4579&assetKey=AS%3A272440071000081%401441966182461.

Jerome Tangkeh Njofang holds a B-Tech Degree in Mech Eng and is graduating this Autumn 2017 for his M-Tech. in Mech Eng at the Cape Peninsula Univ. of Technology (CPUT) Cape Town South Africa, PH-0027219596546. The previous field of study of the aothor was crowned with a Bachelor Degree in Management Science in 1994 at the Univ. of Yaounde II Soa Cameroon. Some years of Industrial services (3years), teaching and part-time lecturing as well (7years), build up his professional experience. E-mail: tangkehnj@mail.com

Emeritus Professor Jasson Gryzagoridis is a retired academic from the University of Cape Town (UCT). Currently he is holding a position in the Mechanical Engineering Departments at UCT as a Senior Scholar and Researcher and with the Cape Peninsula University of Technology as an Adjunct Professor, supervising Postgraduate students. E-mail: gryzagoridisj@cput.ac.za