INVESTIGATION OF THE PERFORMANCE OF COOLING PANELS: "CEILING AND FLOOR PANElS.

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1.0 ABSTRACT:

New techniques in air conditioning by using cooling panels were put into investigation in the last decade or so. The cooling panels are cooled surfaces by chilled water carried to flow in pipes. The pipes are fitted to the panel back in a way to cool the panel surface.

Many obstacles face the use of the cooling panels in air conditioning inside rooms and buildings, namely: low surface temperature; condensation that may occur on the panel surfaces or tube surfaces and high cooling loads required.

Heat transfer mechanism and modes were investigated by many. The position of the panel found to affect its capacity considerably. Convection and radiation heat transfer percentage of the capacity differs and are functions of both position of the panel and panel surface temperature.

Panels considered were ceiling panels and floor panels.

Mathematical modeling was validated by experimental results and used for the comparison between ceiling panels performance and that of the floor panels.

Giacomini experimental lab was used for the experimental part of this study.

2.0 INTRODUCTION:

Inside surfaces of rooms and other spaces can be suitable places to accommodate cooling or heating panels for air conditioning purposes. These surfaces may be the ceilings, the floors or the walls.

This work is intended to investigate the effectiveness of cooling panels that are placed in both the ceiling and the floor. The panel effectiveness depends on the effective panel temperature, (tp); the area available for the panels, (A); and panel position.

The effective panel temperature is controlled by two factors:

a- Dew point of the inside space considered the lowest limit to avoid condensation formation on the surface. And.

b- Human discomfort floor temperature of about 16° C is also treated as lower limit.

This limit may result in large floor area required for the cooling floor panel to over come the cooling load.[1]

Two modes of heat transfer are existed in this cooling process: convection heat transfer and radiation heat transfer.

This study consists of two parts: Theoretical part and experimental part. The theoretical study depended on mathematical modeling and computer simulation. The experimental study was used as validation grounds for the theoretical one. A comparison between the heat fluxes received by the floor panel and that by the ceiling panel was almost the major conclusion of this study.

Methods of calculating the cooling loads of buildings and heat flux of radiant panels can be found in many references in the literature, [1, 2 and 3].

The experimental study was carried out in side what is called Giacomini laboratory, it is a lab within the mechanical engineering labs. This lab is fixed by metal radiant panels in the ceiling attached to copper water flow tubes, and floor single panel made of concrete slabs with plastic water flow tubes embedded in the slabs, [4 and 5].

3.0 THE LABORATORY UNDER CONSIDERATION:

Figure 1 shows the architectural plan of the laboratory space. It is a lecturing room of 50.5 m^2 floor area, located within the mechanical engineering labs, at roof, refs. [4 and 5]. The design cooling load required was calculated and found to be about 5.2 kW, and the area of unused surface temperature (AUST) found to equal $(1.167 + T_i)^o$ C. Where T_i is the room inside temperature. Calculations were conducted assuming design conditions as follows:
- Inside design conditions are: temperature, $T_i = 24^o$ C and Relative humidity, ϕ_i= 50%, and
- Outside design conditions are: temperature, $T_o = 33^o$ C, Relative humidity, ϕ_o = 45% and wind velocity, V_o = 0.7 m/s.

The floor area is an activity area but the ceiling panels are away from the space users, difference in construction and surface temperatures are concluded.

3.1 The Ceiling Panels.

Figure 2 shows a typical ceiling panel. The panel considered consists of an Aluminum surface plate of 0.30 X 0.60 m, used as fins for the bank of the copper tubes attached above the plate. The tubes of all the panels perform the net work of the cold water carriers around the whole area of the ceiling. The chilled water circulates inside these tubes. The chilled water temperature reaches as low as 3^o C, producing plate effective temperature as low as 5^o C. ASHRAE book of systems and applications, (2000), reference 6 stated the following two equations for calculating the heat flux supplied to the ceiling panels during cooling processes:

1- Natural convection heat flux, q_c, in kJ/ m^2

$$q_c = 2.13 \, [T_p - T_a]^{0.31} \, (T_p - T_a) \qquad (1)$$

2- Radiation heat flux, q_r, in kJ/ m^2

$$q_r = 5 \, 10^{-8} \, [\, (T_p + 273)^4 - (\, AUST + 273 \,)^4 \,] \qquad (2)$$

Where, T_p is the panel effective temperature, T_a is the average room air temperature, and AUST is the Area of Unused Surfaces Temperature.

3.2 The Floor Panel:

Floor consists of two concrete slabs sandwiching the plastic tube cooling net work between the as embedded inside. The lower slab sets over a reinforced concrete base, and the higher slab is covered by the floor covering material. The tube net work is lying on a sheet of insulation of polystyrene. Single panel covers all over the floor. Fig. 3 shows the floor panel details.

The effective panel temperature must not be higher than 16^o C for human purposes that keep the chilled water temperature higher than 12^o C.

ASHRAE book systems and applications, (200), ref. 6 stated the following two equations for calculating the heat flux supplied to the floor panel.

1- Natural convection heat flux, q_c, in kJ/ m^2

$$q_c = 0.87 \, [T_p - T_a]^{0.25} \, (T_p - T_a) \qquad (3)$$

2- Radiation heat flux, q_r, in kJ/ m^2

$$q_r = 5 \, 10^{-8} \, [\, (T_p + 273)^4 - (\, AUST + 273 \,)^4 \,] \qquad (4)$$

The radiation heat transfer flux is not affected by panel position, this lead to the fact that equations 2 and 4 are similar.

4.0 THE MATHEMATICAL MODEL AND COMPUTER SIMULATION:

Theoretical analysis to the system leads to the mathematical model. The mathematical model was simulated by a computer algorithm using MATLAB codes. Relations between different variables resulted in a performance representation

Independent variables supplied to the simulation program were: Chilled water inlet temperature, T_{win}, (^{o} C), water rate of flow, m_w, (kg/s).Average room air temperature, T_a, (^{o} C), Panels total area, A_T, (m^2) and total cooling load, Q_L, (kW).

Dependent variables as simulation out put resulted by solving the mathematical model are: mean water temperature, T_m, (^{o} C), effective panel surface temperature, T_p, (^{o} C), convection and radiation heat fluxes, q_c and q_r respectively, kW/m^2. and panel area required to satisfy the cooling load, A, (m^2).

The following two mathematical models were used:

4.1 Ceiling panels:

a- Heat flux for both convection and radiation can be calculated using the previous equations 1 and 2 consequently. The total heat flux, q can be calculated by the following equation:

$$q = (q_r + q_c)\ A_T \qquad (5)$$

b- Water mean temperature is:

$$T_m = T_{win} + q/(m_w C_p) \qquad (6)$$

Where C_p is the water specific heat.
c- Panels area required to satisfy the cooling load is:

$$A = Q_L / (q_c + q_r) \qquad (7)$$

d- Effective panel temperature can be calculated using fin theory conducted by Conry and Mumma, (2001) [7] and Jeong and Mumma, (2004), [8].
Depending on Fig. 4, the fin is represented by the Aluminum cover attached to the copper tubes:

$$T_p = T_{win} + m_w C_p\ (T_{wo} - T_{win})/(A_T F_R U_p) \qquad (8)$$

Where T_{wo} is the water outlet temperature and equals:

$$T_{wo} = T_{win} + q/\ m_w C_p \qquad (9)$$

The panel heat removal factor, F_R is:

$$F_R = m_w C_p\ (T_{wo} - T_{win})/A_T U_p\ (T_a - T_{win}) \qquad (10)$$

And over all panel heat transfer coefficient U_p is:

$$U_p = q/\ [F_R (T_p - T_{win})] \qquad (11)$$

While the room inside over all heat transfer coefficient, U_R can be extracted from equations 1 and 2 as:

$$U_R = 5\ 10^{-8}\ [(T_p + 273)^2 + (AUST+273)^2]\ (T_p + 273) + (AUST+273)$$
$$+ 2.13(T_p - T_a)^{0.31} \qquad (11a)$$

Where: $AUST = \Sigma(Ts\ As)/\ \Sigma As \qquad (12)$

Where As is the area of the un used surface, and Ts is its temperature.

e- Total heat flux is $q_T = A_T\ q \qquad (13)$

f- Computer simulation:
Figure 5a shows a flow chart of the computer simulation program which exhibits a solution sequence of the mathematical model presented by the equations that are related to the ceiling cooling panels in equations 1 to 13 shown above.
Lists of computer run results will be exhibited in Figs 7 to 12 shown at the end of the paper. These results were subjected to analysis and comparison with those for the floor panel computer simulation as will be discussed later.

4.2 Floor Panel:

a- Heat flux in this case is presented by both equations 3 and 4 previously mentioned and found in ASHRAE book of systems and equipment, (2000), [6]. The total heat flux, q can be calculated by:

$$q = q_r + q_c \qquad (14)$$

b- Water mean temperature T_m and area required to satisfy the cooling load, A can be calculated using same equation 6 and 7 respectively.
c- The method of ASHRAE, [6] for embedded tubes in slabs was adopted to calculate the effective panel temperature, T_p. This method uses the following equations:

$$T_p = T_m + q/U_p \qquad (15)$$
$$q = U_p\ (T_p - T_m) \qquad (16)$$

and $U_p = 1/R_T \qquad (17)$

Where R_T is the total resistance of the slab – water connection.

$$R_T = S\, r_t + r_p + r_c \qquad (18)$$

Where S is the tube pitch distance, Fig. 6, and tube resistance r_t is:

$$r_t = Ln\ (D_o/D_i) / 2\pi\, K_t \qquad (19)$$

Where D_o and D_i are tube outside and inside diameters and K_t is the tube thermal conductivity.
Cover panel resistance, $r_c = X_c / K_c$ (20)
Where X_c and K_c are the cover slab thickness and thermal conductivity respectively. Tube panel resistance is $r_p = (X_p - D_o/2\ )/K_p$ (21)
Where X_p and K_p are the panel thickness and thermal conductivity respectively.

d- Total heat flux $q_T = A_T\, q$ (22)

e- Computer simulation
Figure 5b shows a flow chart of the computer simulation program which exhibits a solution sequence of the mathematical model presented by the equations that are related to the floor cooling panel in equations 14 to 22 shown above.
Lists of computer run results will be exhibited in Figs 7 to 12 shown at the end of the paper. These results were subjected to analysis and comparison with those for the ceiling panels computer simulation as will be discussed later.

5.0 EXPERIMENTAL WORK:

Different experiments were carried out to test the ceiling panels as cooling equipment in summer. Chilled water maximum of 16^o C was circulated through the network of pipes attached to the panel plate. A local made chiller of 2.5 tons of cooling capacity was installed within the system,[9]. Water inlet and outlet temperatures were monitored, (T_{win} and T_{wo}). Panel temperature at 3 different points were listed, (T_1, T_2, and T_3), and room air temperature, T_a was also measured in four different points, (T_4, T_5, T_6 and T_7)
Calculations of the results were:
- Panel mean effective temperature was calculated as:

$$T_p = (T_1 + T_2 + T_3) / 3 \qquad (22)$$

- and

$$T_a = (T_1 + T_2 + T_3 + T_4) /4 \qquad (23)$$

Table 2 lists results of a typical experiment.

Table 2. typical experiment results

Temperature	Value, C.
T_a	21.0
T_o	26.0
T_p	18.8
T_{win}	16.1
T_{wo}	19.0

A relationship was found between T_p and water outlet temperature, T_{wo} in the form of:

$$T_p = 0.41\, T_{wo} + 11.2 \qquad (24)$$

The experimental results shown in Table 1 were comparable to those of the theoretical results with a difference of around 5.7 %.

6.0 ANALYSIS:

For the sake of comparison between the performance of the ceiling panels and that of the floor panel, the dependent variables values resulted from the computer simulations results were shown by the same figure. The Figures from 7 to 13 showed both results.
Figure 7 shows that the radiation heat flux for both panels are equal. There is a clear difference in the value of the heat flux for convection heat transfer. This is due to the effect of the difference in panel position and the fact that colder air has higher density and so stays at lower levels. This resulted in

lower convection heat flux for floor panels. The figure shows increase of loss in convection heat flux as effective panel temperature decreases. The loss reached at lowest panel temperature to about 72%.
The effect exhibited in Fig. 7 was repeated in Fig. 8 where heat flux of both modes are added to be q for both panels are presented. The heat flux of the floor panel is short than that of the ceiling heat flux by a value that increases as panel temperatures decreases and reaches a value of about 40% of the ceiling flux at lowest temperature of 0.0^{o} C.
This loss of heat flux by the floor panel limit the use of the panel and affects the design of the panels and requires more area to satisfy the cooling load. This effect is critical due to the limited panel temperature caused by human comfort factors.
The relation between effective panel temperature and panel area with the required cooling load can be understood by analyzing Fig. 9. It is shown that using the whole floor or the ceiling area to satisfy the cooling load requires a panel temperature of 6.5^{o} C and 14.5^{o} C respectively. While it is easy to reach a ceiling panel temperature of 14.5^{o} C it is restricted to go below 16^{o} C for floor panel temperature for human comfort reasons. And it is also clear that an area of 90 m^{2} is required for floor area to satisfy cooling load properly, this area is not available in this case.
The embedded tubes inside the concrete slabs in the floor panel give higher effectiveness for heat transfer between the panel surface and the chilled water flowing inside the tubes than the case of the ceiling panels where the tubes are attached to the upper surface of the panels. This is clear from Fig. 10 which shows the relation between the mean fluid temperature, T_m and the effective panel temperature, T_p. The difference is low but reasonable.
Figures 11 to 13 shows the effect of chilled water temperature at system inlet, T_{win} on several other panel parameters.
Fig 11 shows the relation between T_{win} and water outlet temperature, $T_{wo}$for different rates of flow for both positions. This relation is allready mentioned in equation 9 where a linear direct proportional relation is existing. The effect of water mass flow rate, m_w is clear to have an inversely proportional relation.
The combined heat flux of Eqs. 5 and 22 are slightly affected by T_{win} as shown in Fig 12, but highly affected by m_w as clear by the figure and by the equation.
Equation 11 showes the panel overall heat transfer coefficient for the ceiling panel, U_p as a function of T_{win} and m_w . This relation is exhibited in Fig. 13.As T_{win} is a major independent variable and affects all other variables shown in Eq. 11, so appears a minimum value for U_p at T_{win} ranged between 14^{o} C and 16.5^{o} C, this range can be described as the worst for panel performance for the case under consideration.
The panel mean temperature, (T_{pm}) is dependent on both the water inlet temperature, (T_{win}) and the rate of flow of the water, (m_w), this is presented in equation 8 for the
ceiling panel. A direct proportional relation for T_{pm} with m_w and T_{win} for the range of temperatures considered and the case studied. This is shown in Fig. 14.

7.0 CONCLUSIONS:

1- The mathematical model and the computer simulation used in this study gave results that showed conformity with that of the experimental results with difference of about 5.7%.
2- The ceiling panels are better in performance and much suitable to be used for cooling purposes compared to the poor performance of the floor panel.
3- Embedded tubes in the panel slabs gave better heat transfer performance than the case of attached tubes to the panel surface.

8.0 ACKNOWLEDGEMENT:

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Figures captions:

1- Fig. 1. Plan view of Giacomini lab.
2- Fig. 2. Typical ceiling radiant panel.
3- Fig. 3. Typical floor radiant panel.
4- Fig. 4. Cross section in the ceiling panel.
5- Fig. 5.a Flow chart of ceiling panels simulation program.
Fig. 5.b Flow chart of floor panel simulation program.
6- Fig. 6. Cross section in the floor panel.
7- Fig. 7. Heat flux Vs panel temperature.
8- Fig. 8. Combined heat flux Vs panel temperature.
9- Fig. 9. Required area Vs panel temperature.
10- Fig. 10. water mean temperatures Vs panel temperature.
11- Fig. 11. water outlet temperature Vs water inlet temperature.
12- Fig. 12. Combined heat flux Vs water inlet temperature.
13- Fig. 13. Panel over all heat transfer coefficient Vs water inlet temperature.
14- Fig. 14. Panel mean temperature

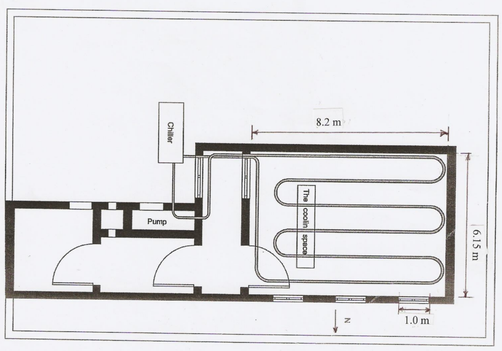

Fig. 1. Plan view of Giacomini lab.

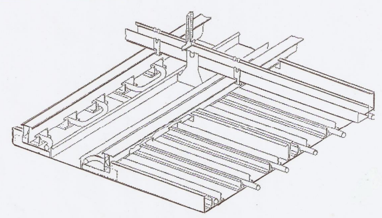
Fig. 2. Typical ceiling radiant panel.

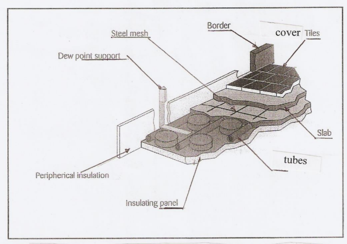

Fig. 3. Typical floor radiant panel.

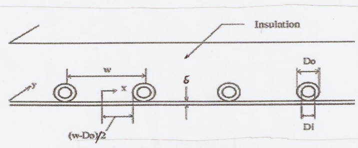

Fig. 4. Cross section in the ceiling panel.

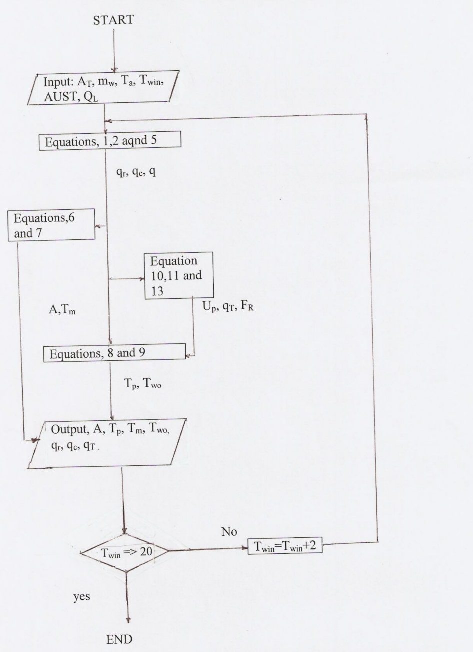

Fig. 5a. Flow chart of ceiling panels simulation program.

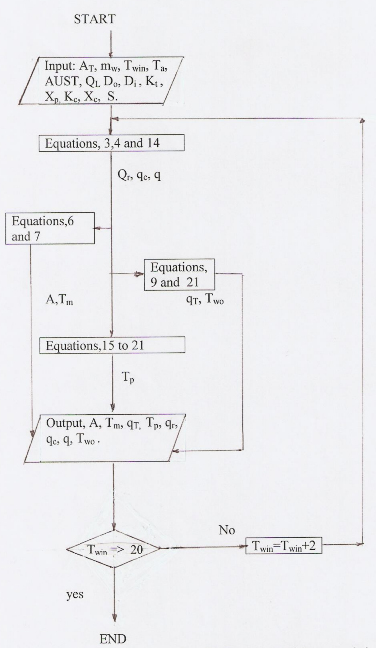

Fig. 5b. Flow chart of floor panel simulation program.

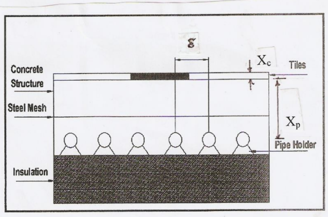

Fig. 6. Cross section in the floor panel.

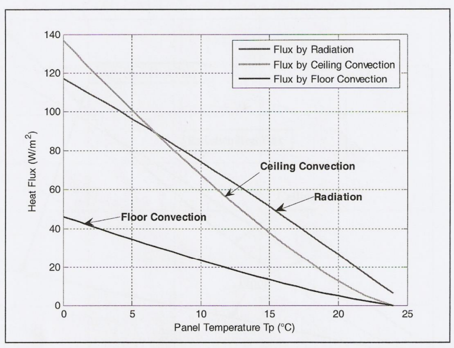

Fig. 7. Heat flux Vs panel temperature.

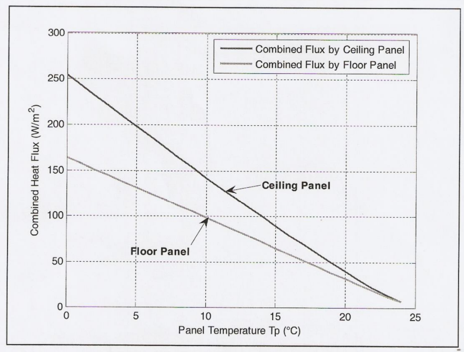

Fig. 8. Combined heat flux Vs panel temperature.

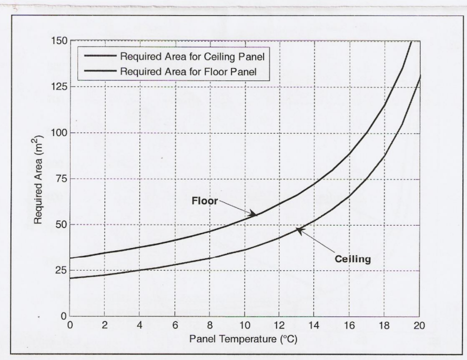

Fig. 9. Required area Vs panel temperature.

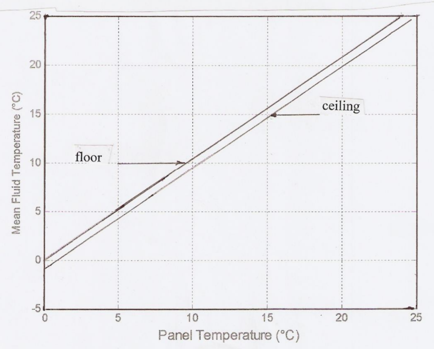

Fig. 10. water mean temperatures Vs panel temperature.

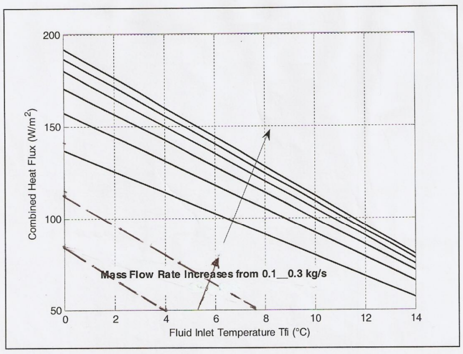

Fig. 12. Combined heat flux Vs water inlet temperature.
(____ ceiling, ----- floor)

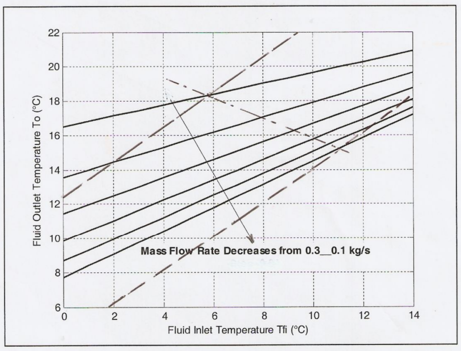

Fig. 11. water outlet temperature Vs water inlet temperature.
(____ ceiling, ----- floor)

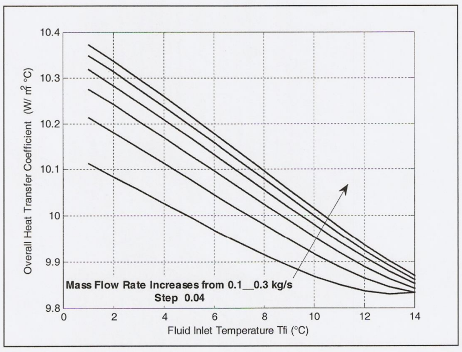

Fig. 13. panel over all heat transfer coefficient Vs water inlet temperatutre.
(ceiling)

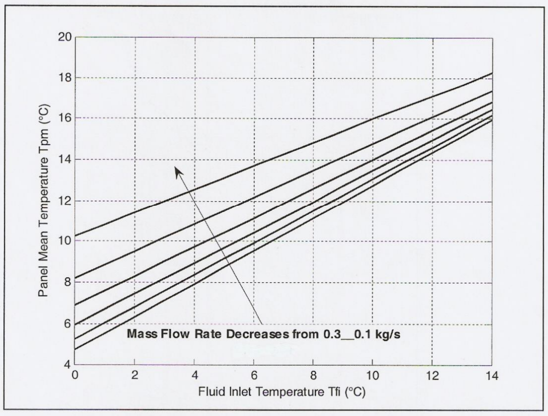

Fig. 14. Panel mean temperature Vs water inlet temperature.
(ceiling)