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TRANSIENT RESPONSE OF ENCLOSURES HEATED BY A THERMOELECTRIC HEAT PUMP

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Summary

The transient energy equations of enclosures heated by a thermoelectric heat pump are presented. The enclosures examined lose heat through two types of walls, through air renewal and also absorb heat in one of the walls and in discrete objects located inside the enclosure. The transient model of the enclosure is verified experimentally in the case of a constant heat input. Calculations are given using non dimensional and reduced parameters. Thermoelectric heating is compared to several constant heat input values, one of the values is equal to that of the heat pump when it has reached equilibrium conditions. The gain in time to reach a given temperature reaches 45%.

1. - Introduction

Enclosures with surface areas of several square meters or more are encountered in many applications. Often, it is necessary to preheat the enclosures before they can be in opera tion. In certain cases, air renewal is required especially if people are inside the enclosure. These types of enclosures can often be characterised in the following way :

- wall 1 of area S_1 with specific mass ρ thermal conductivity λ heat capacity C_{p1} with inside convection coefficient h_{1i} and outside convection coefficient h_{1e}
- wall 2 of area S_2 with no heat capacity, that can be thermally characterised by an overall heat transfer coefficient K_2
- discrete objects inside the enclosure of mass M and specific heat C_{pm} that have a high surface to volume ratio so that their temperature follows the inside temperature.
- air renewal of flow rate Q_a, the air is assumed to have a constant heat capacity C_{ph}.

The initial condition is that the whole system : enclosure and heat pump are at a uniform temperature equal to the outside temperature. The the thermoelectric heat pump and fans are turned on. The equations given enable the calculation of the inside temperature of the enclosure versus time.

Thermoelectric heat pumps present the advantage over constant heating power units in that they give out more heat in the transient mode than under the final equilibrium conditions.

2. - Description of the system

The system is composed of :
- a thermoelectric heat pump
- an enclosure

The fluid circuits are shown schematically in Fig.1 and the powers involved are shown in Fig. 2.

3. - Thermoelectric heat pump

The heat pump is composed of a number of subunits.

3.1. - Equations of subunit

The two thermoelectric heat pump equations are :

$$W_h = aI\,(T_{th} + 273.15) + \frac{1}{2} r\, I^2 - C_e\,(T_{th} - T_{tc}) - C_L\,(T_{bh} - T_{bc}) \qquad (1)$$

$$W_c = -aI\,(T_{tc} + 273.15) + \frac{1}{2} r\, I^2 + C_e\,(T_{th} - T_{tc}) + C_L\,(T_{bh} - T_{bc}) \qquad (2)$$

The following terms T_{th}, T_{tc}, T_{bh} and T_{bc} can be expressed as function of

T_h, T_c, W_h, and W_c.

These temperatures are shown in Fig 3 for a subunit.

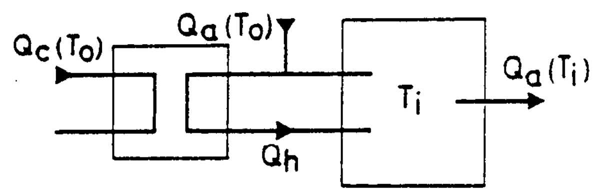

Fig. 1. - Fluid circuits.

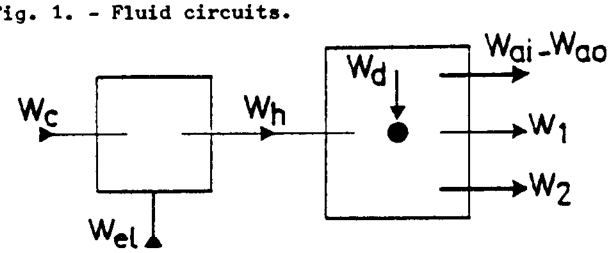

Fig. 2. - Heat fluxes in system.

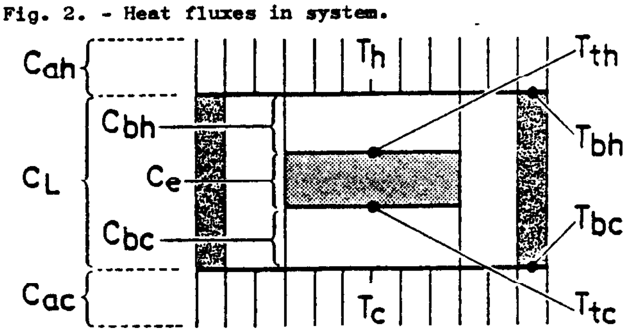

Fig. 3. - Detail of thermoelectric heat pump subunit.

The terms W_c and W_h can both be written in two ways :

$$W_c = C_{ac}(T_{bc} - T_c) = C_{bc}(T_{tc} - T_{bc}) \qquad (3)$$

$$W_c = \left(\frac{1}{C_{bc}} + \frac{1}{C_{ac}}\right)^{-1} (T_{tc} - T_c) = C_c(T_{tc} - T_c) \qquad (4)$$

$$W_h = C_{ah}(T_{th} - T_h) = C_{bh}(T_{th} - T_{bh}) \qquad (5)$$

$$W_h = \left(\frac{1}{C_{bh}} + \frac{1}{C_{ah}}\right)^{-1} (T_{th} - T_h) = C_h(T_{th} - T_h) \qquad (6)$$

With equations (3) , (4), (5) and (6), one can write

$$T_{tc} = T_c + \frac{W_c}{C_c} \quad ; \quad T_{th} = T_h + \frac{W_h}{C_h}$$

C_{ah} and C_{ac} can be expressed :

$$C_{ah} = H_h \sigma_h \quad ; \quad C_{ac} = H_c \sigma_c \qquad (7)$$

$$T_{bh} = T_h + \frac{W_h}{H_h \sigma_h} \quad ; \quad T_{bc} = T_c + \frac{W_c}{H_c \sigma_c} \qquad (8)$$

Equations (1) and (2) become :

$$W_c = -aI\left(\frac{W_c}{C_c} + T_c + 273.15\right) + \frac{1}{2} rI^2 + C_e\left(\frac{W_h}{C_h} - \frac{W_c}{C_c} + T_h - T_c\right) + C_L\left(\frac{W_h}{H_h\sigma_h} - \frac{W_c}{H_c\sigma_c} + T_h - T_c\right) \qquad (9)$$

$$W_h = aI\left(\frac{W_h}{C_h} + T_h + 273.15\right) + \frac{1}{2} rI^2 - C_e\left(\frac{W_h}{C_h} - \frac{W_c}{C_c}\right) + (T_h - T_c) - C_L\left(\frac{W_h}{H_h\sigma_h} - \frac{W_c}{H_c\sigma_c}\right) + (T_h - T_c) \qquad (10)$$

The third energy equation is :

$$W_{el} = W_h + W_c \qquad (11)$$

The calculations are done for a given value of electrical current I because the system of equations (9) and (10) is linear with two unknowns, one can write :

$$\alpha_1 W_c + \beta_1 W_h = \gamma_1 \qquad (12)$$

$$\alpha_2 W_c + \beta_2 W_h = \gamma_2$$

and the solutions are of the form :

$$W_c = \frac{\beta_2 \gamma_1 - \beta_1 \gamma_2}{\alpha_1 \beta_2 - \alpha_2 \beta_1} \qquad (13)$$

$$W_h = \frac{\alpha_1 \gamma_2 - \alpha_2 \gamma_1}{\alpha_1 \beta_2 - \alpha_2 \beta_1}$$

The values of α_1 α_2 β_1 and γ_2 are easily obtained from equations (9) and 10.

3.2. Calculation of overall heat pump

For each subunit, one can write for the heat exchanges on either side of the thermoelectric heat pump, the following equations :

$$T_c = T_{co} + \frac{W_c}{2Q_c . C_{pc}} \quad ; \quad T_h = T_{ho} + \frac{W_h}{2Q_h . C_{ph}} \qquad (14)$$

Where T_c and T_h are the average fluid temperatures in the subunit and T_{co} and T_{ho} the inlet temperatures into the subunit.

The exit conditions from the subunit are :

$$T_{c1} = T_{co} + \frac{W_c}{Q_c . C_{pc}} \quad ; \quad T_{h1} = T_{ho} + \frac{W_h}{Q_h . C_{ph}} \qquad (15)$$

This calculation is done for each subunit so that the exit temperature T_h from the whole heat pump is known. This temperature is used to calculate the heat input W_h into the enclosure.

$$W_h = Q_h . C_{ph} \, (T_h - T_i) \qquad (16)$$

4. - Enclosure

The enclosure calculation that is examined is a method developped by Jacq[1] and used by Veron[2] for transient heat losses in buildings.

The assumptions are that the enclosure has the following one dimensional heat flows :

W_h heat entering the enclosure by heated air.

$(W_{ai} - W_{ao})$ heat lost through air removal. Where W_{ai} heat leaving enclosure through exit, air at T_i. W_{ao} heat entering the enclosure through air at outside temperature T_o. The air is considered to have a constant heat capacity C_{ph}.

W_1 heat losses through a wall or area S_1, of thickness ε, of thermal conductivity λ with convection coefficients on the inside of h_{i1} and on the exterior of h_{e1}, thermal heat capacities C_{p1}.

W_2 heat losses through a wall area S_2 that has a negligeable heat capacity and that can be defined by its overall heat transfer coefficient K_2.

W_d heat absorbed by discrete objects in the enclosure. It is assumed that these objects have a high surface to volume ratio so that the temperature of the objects follows the inside enclosure temperature with a time lag equal to the difference in time between two steps in the transient calculation.

The whole system initially is at the outside temperature T_o then the enclosure is heated by hot air from the heat pump that is operated at constant current.

With the above assumptions the transient temperature calculation of the inside temperature T_i of the enclosure is relatively simple and is at a given instant presented in reference 1.

The heat equation for the enclosure is the sum of the terms previously defined.

$$W_h = W_1 + W_2 + (W_{ai} - W_{ao}) + W_d \qquad (17)$$

With :

$$W_1 (t) = h_{1i} . S_1 \left[T_i(t) - T_{1i}(t)\right] \qquad (18)$$

$$W_2 (t) = k_2 . S_2 \left[T_i(t) - T_o\right] \qquad (19)$$

$$W_{ai}(t) - W_{ao} = Q_a \, C_{ph}\left[T_i (t) - T_o\right] \qquad (20)$$

$$W_d = \frac{M . C_{pm}}{\Delta t} \left[T_i(t) - T_i (t - \Delta t)\right] \qquad (21)$$

The time increment used is $\Delta t = \dfrac{\Delta x^2}{2\alpha}$ (22)

5. - Resolution of the heat pump enclosure system

The mathematical model for the thermoelectric heat pump is solved numerically to obtain W_h from equation 16, so the overall equation to be solved is :

$$W_h(t) = h_{1i}.S_1\left[T_i(t)-T_{1i}(t)\right] + K_2.S_2\left[T_i(t) - T_o\right] + Q_a.C_{ph}\left[T_i(t)-T_o\right] + \frac{M.C_{pm}}{\Delta t}\left[T_i(t)-T_i(t-\Delta t)\right] \quad (23)$$

Knowing $W_h(t)$ one calculates from equation (23) T_i & $T_{1i}(t)$ as the two latter temperatures are related using a finite element calculation, the procedure adopted is given in reference 2.

The equations given to calculate W_h of the heat pump correspond to steady state. To calculate the transient enclosure conditions, the procedure used is :

The time increment is chosen equal to $\dfrac{\Delta x^2}{2\alpha}$ where Δx is the thickness of the slice in the wall. The number of slices in the wall was chosen equal to 5.

For the first increment Δt it is assumed that the enclosure has no heat losses and absorbs not heat. For this period, the heat generated by the thermoelectric heat pump is calculated. The experimental results which are examined in paragraph 6 lead us to reduce the heat output from the thermoelectric heat pump by 30% during the first time increment. $\Delta t^* = 0.02$. The reduction corresponds to heat absorbed in duct and fans between the heat pump and the enclosure. Having the effective W_h coming into the enclosure, one can calculate its temperature T_i ($t = \Delta t$). Then the calculation procedes in the same way, but with no coefficient reducing W_h. The W_h of the heat pump is calculated each time that the average wall temperature increases by 0.5°c. ($\Delta T^* = 0.017$).

6. - Comparaison with experiment at constant heat input into enclosure

A comparaison between a calculation and an experiment is done in the simplified case of a constant heat input such as that obtained by Joule heating. Fig. 4 shows the calculated curve (dashed) and the experimental curve (solid line).

At start up, the experimental curve is slightly below the calculated one, then exceeds it to finally pass slightly under it, around $t^* = 0.5$.

The nearness of the two curves is considered sufficient to validate the mathematical model of the enclosure.

Dimensionless and reduced parameters are used, they are presented in paragraph 7.

7. - Dimensionless and reduced parameters

The parameters : enclosure temperature T_i and time t from starting up of heat pump have both been made non dimensional.

The non dimensional temperature T^* is defined as

$$T^* = \frac{T_i - T_o}{T_F - T_o}$$

where T_o is the outside temperature and equal to the initial temperature of the whole system.

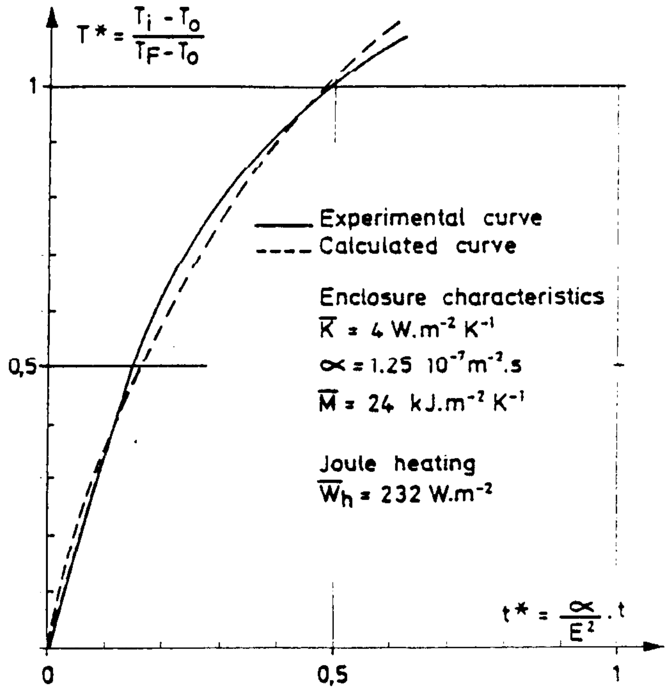

Fig. 4 - Enclosure temperature T* versus time t* comparaison between experiment and calculation.

T_F is the equilibrium temperature of the enclosure for an enclosure characterised by $\overline{K}$, α and $\overline{M}$ (defined further on.) and a given heat input from thermoelectric heat pump at equilibrium.

$\overline{W}_H = 120 \ W.m^{-2}$ as $T_F - T_o = \dfrac{\overline{W}_h}{\overline{K}} = 30°c.$

All the pump calculations are done with :

$T_F - T_o = 30°c.$

The non dimensional time t* is defined in the classic way using the thermal diffusivity α of wall 1; wall 2 has a thermal diffusivity equal to zero.

$$t^* = \frac{\alpha}{E^2}.t \quad \text{where } \alpha = \frac{\lambda}{\rho C_{p1}}$$

and E is the thickness of wall 1.

The following parameters are reduced by dividing them by the area S_1 of wall 1.

1. Heat input into the enclosure W_H

$$\overline{W}_h = \frac{W_h}{S_1}$$

2. Overall heat loss coefficient K of the enclosure at equilibrium is obtained from the equilibrium heat balance.

$$T_{eq} - T_o = \frac{W_h}{\left[\frac{1}{h_{1i}} + \frac{1}{h_{ie}} + \frac{E}{\lambda}\right]^{-1} .S_1 + K_2S_2 + Q_aC_{ph}}$$

The denominateur K is defined :

$$K = K_1.S_1 + K_2S_2 + Q_aC_{ph}$$

We defined $\overline{K}$ as : $\overline{K} = \dfrac{K}{S_1}$

and also the non dimensional parameters

$$K_1^* = \frac{K_1 S_1}{K} \; ; \; K_2^* = \frac{K_2 S_2}{K} \; ; \; K_a^* = \frac{Q_a C_{ph}}{K}$$

These three parameters represent at equilibrium the proportion of heat leaving the enclosure respectively through wall S_1, wall S_2 and the renewed air

The thermal mass of the discrete objects $M.C_{pm}$ is reduced by dividing the area of wall 1.

$$\overline{M} = \frac{M.C._{pm}}{S_1}$$

Two basic cases are examined :

- one enclosure, variable thermoelectric heat pump operating conditions ;
- constant thermoelectric heat pump operating condition with different enclosures.

8. - One enclosure variable heat pump conditions

The enclosure has the following reduced characteristics :

$\propto = 1.25 \; 10^{-7} \; m^{-2}.s$; $\overline{K} = 4 \; W.m^{-2} k^{-1}$

$\overline{M} = 24 \; k \; J.m^2 k^{-1}$

$K_1^* = 0.49$; $K_2^* = 0.23$; $K_a^* = 0.28$

The thermoelectric heat pump has the following characteristics :

W_h is produced by a heat pump having 60 cm^2 of thermoelectric material with a $Z = 2.86 \; 10^{-3} K^{-1}$. The properties of the material vary as function of the average material temperature, the above value is the peak value

8.1.- Heating power and C.O.P.

The heat pump is operated at 3 values of electrical current density J : 95 ; 110 and 125 $A.cm^{-2}$. Fig. 5 shows how W_h and C.O.P vary as a function of time t*.

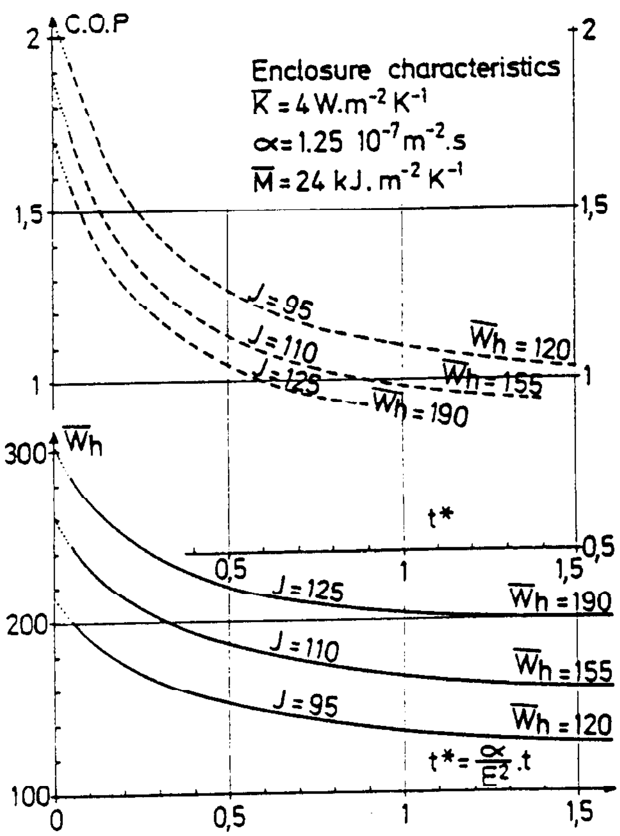

Fig. 5. - Heating power and C.O.P versus time t*.

The bottom graph shows that $\overline{W}_h$ at the beginning has a value far superior to the equilibrium value. The values corresponding to $t^*<0.05$ are extrapoleted which corresponds to no inertia in the thermoelectric heat pump.

The upper graph gives the C.O.P where the values are also quite high at start up. The C.O.P drops with time t* because the temperature difference $(T_i - T_o)$ between the two sides of the heat pump increases, when T* reaches 1 which corresponds to $T_i - T_o = 30°c$, it is normal that for current densities above a certain value, the C.O.P drops below 1. Such conditions are not interesting industrially because Joule heating has a C.O.P of 1.

8.2.- Enclosure temperature

The heat pump is operated at the same 3 different electrical current densities as previously. Fig. 6 gives the enclosure temperature T* versus time t*.

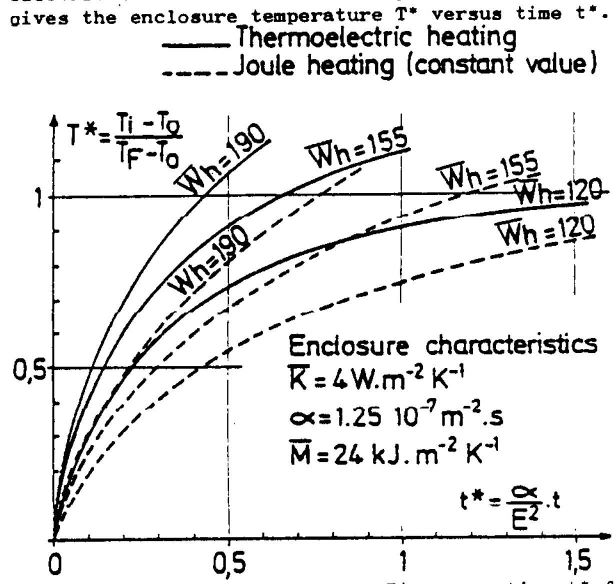

Fig. 6.- Enclosure temperature T* versus time t* for thermoelectric heating and Joule heating.

The solid lines correspond to thermoelectric heating and the dashed lines to Joule heating meaning a constant heat input into the enclosure versus time.

The $\overline{W}_h$ values indicated correspond to equilibrium conditions. The 3 values of Joule heating correspond to the 3 equilibrium values of the heat pump.

The following remarks can be made :

T* = 1 corresponds to equilibrium with $\overline{W}_h$ = 120 so obviously in the graph which is limited time wise to t* = 1.5 neither the Joule heater or the heat pump of the same heating power at equilibrium can reach T* = 1

When the heating power $\overline{W}_h$ is increased 22% from 120 $W.m^{-1}$ to 155 $W.m^{-2}$, the Joule heater requires t* = 1.18 to reach T* = 1, while the equivalent thermoelectric heater requires only 0.66 ; this represents a gain in heating time of 44%. The gain is the same for $\overline{W}_h$ = 190. $W.m^{-2}$

9. - Different enclosures with same heat pump

The enclosure can be characterised by the three parameters :

- $\overline{K}$ reduced overall heat loss coefficient of enclosure
- $\propto$ Diffusivity of wall 1
- $\overline{M}$ Reduced thermal mass of discrete objects in enclosure.

The influence of each of these 3 parameters is examined in the following paragraph.

9.1. - Influence of $\overline{K}$

The parameter $\overline{K}$ is varied from $\overline{K}$ = 2 to $\overline{K}$ = 5 as shown in table 1 below.

$\overline{K}$	K^*_1	K^*_2	K^*_a	Type of line on graph
2	1	0	0	————
3	0.68	0.30	0	————
4	0.49	0.51	0	————
4	0.49	0.37	0.14	—·——·—
4	0.49	0.23	0.28	--------
4	0.49	0	0.51	....
5	0.40	0.37	0.28	————

Table 1 values of $\overline{K}$ and K* of Fig. 7.

The heat pump is operated at a $\overline{W}_h$ = 155, the temperatures T* versus time t* are plotted in Fig 7 for the various values of K given in Table 1.

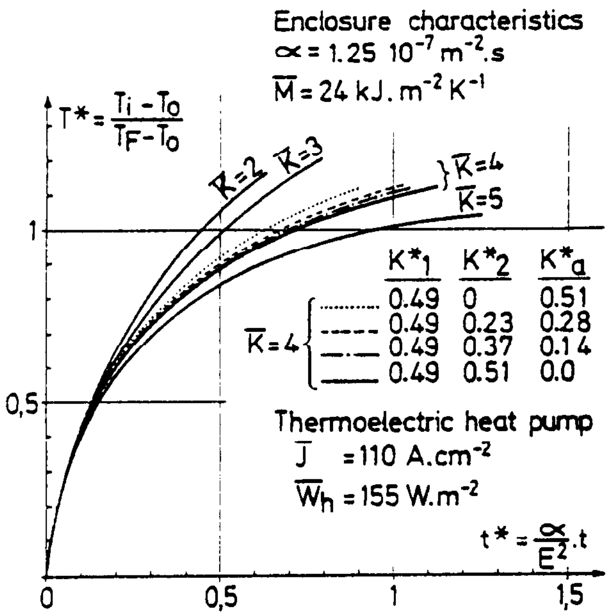

Fig. 7.- Influence of overall heat loss coefficient K on temperature T* versus time t*.

The Fig. 7 confirms that the changes in $\overline{K}$ have more influence than changes in the different values of K*.

To conclude the analysis of Fig. 7, the parameter $\overline{K}$ enables a first approach of the necessary time to reach a given value of T* ; for more precision it is necessary to examine the values of the different K* terms.

9.2. - Influence of thermal diffusivity

The three following values are used for thermal diffusivity of wall 1 :

$\propto$ = 0.675 x 10^{-7} m^{-2} s

$\propto$ = 1.25 x 10^{-7} m^{-2} s

$\propto$ = 2.50 x 10^{-7} m^{-2} s

Other basic parameters of the enclosure are :

$\overline{K}$ = 4. $W.m^{-2}$ K^{1}

$\overline{M}$ = 25. kJ. m^{-2} K^{-1}

The heat pump parameter $\overline{W}_h$ is equal to :

$\overline{W}_h$ = 155. $W.m^{-2}$

The curves T* versus time t* for the three above values of $\propto$ are given in Fig. 8 below

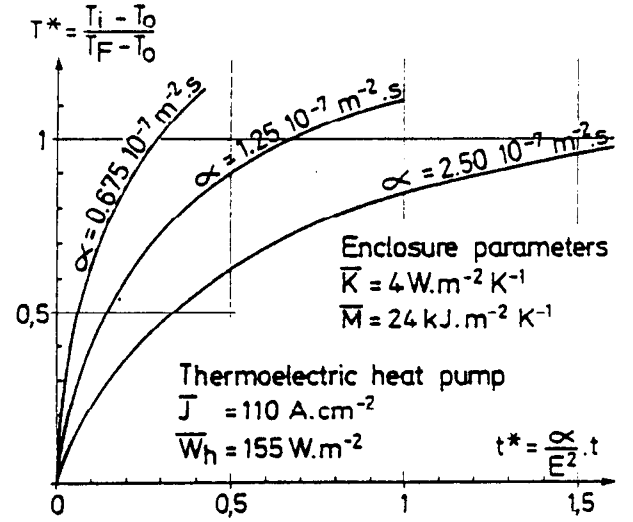

Fig. 8.- Influence of wall diffusivity on enclosure temperature T* versus time t*.

9.3.- Influence of thermal mass of discrete objects

The parameter $\overline{M} = \frac{M.C._{pm}}{S1}$ which represents the heat that can be absorbed by the discrete objects inside the enclosure is a very important factor : Fig. 9 gives T* versus for t* for the following enclosure and heat pump operation :

$\overline{K}$ = 4. $W.m^{-2}K^{-1}$

$\propto$ = 1.25 $10^{-7}m^{-2}K^{-1}$

$\overline{W}_h$ = 155. $W.m^{-2}$

Fig. 9 is drawn with 4 values of M : from 0.5 to 36.$kJ.m^{-2}K^{-1}$.

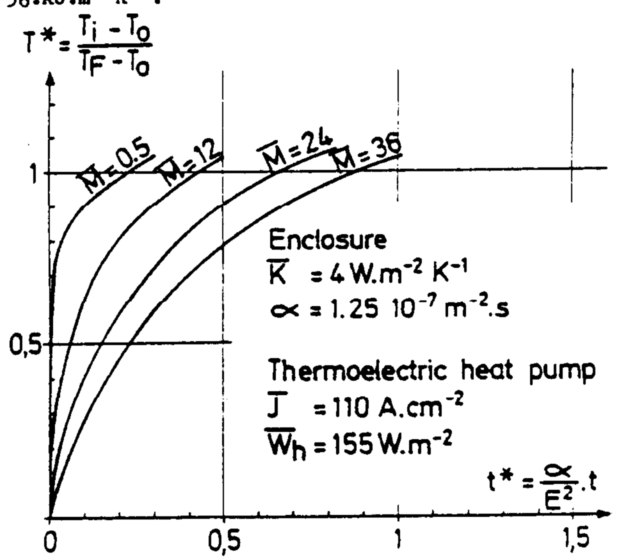

Fig. 9.- Influence of thermal mass of discrete objects on enclosure temperature T* versus time t*.

It can be seen from Fig. 9 that the parameter $\overline{M}$ is very important. This parameter is often underestimated or even omitted in transient heat calculations of enclosures; these curves show how important it is on the time response of the system.

10. - CONCLUSIONS

Enclosures that can be modelised by one wall with thermal heat capacity and heat losses, one wall with no heat capacity but with heat losses, renewal of air and containing discrete objects, can be characterised by reduced parameters :

$\overline{K}$ reduced overall heat loss coefficient of enclosure

α thermal diffusivity of wall 1

$\overline{M}$ reduced thermal mass of discrete objects.

When only the three parameters $\overline{K}$ α and $\overline{M}$ are taken into account, the accuracy of the curves is to within $\pm$ 10%. To obtain accuracy of several percent it is necessary to introduce the parameters K*. It must be stated that the above model assumes uniform temperatures in the enclosure and uniform heat transfer coefficients over the wall surfaces etc ; so, it is felt that the simplified approach is often quite sufficient

The thermal mass of the discrete objects plays a very important part in the time response of the enclosure.

The thermoelectric heat pump permits a considerable reduction in the time necessary to reach a given temperature when compared with Joule heating operating at constant output equal to the output of the thermoelectric heat pump at equilibrium. For the systems calculated reduction in the time of 45% are found.

From the energy point of view, a thermoelectric heat pump operating at a C.O.P greater than one obviously represents an energy savings compared to Joule heating but also the tansient mode increases the energy savings.

NOMENCLATURE

Symbol	Units	Designation
a	$W.K.^{-1}$	Seebeck coefficient
C	$W.K.^{-1}$	Overall thermal conductance, indices c : cold side, h : hot side
C_a	$W.K.^{-1}$	Thermal conductance between air and base of fins, second indices : c : cold side, h : hot side
C_b	$W.K.^{-1}$	Thermal conductance of bases : second indices c : cold, h : hot
C_e	$W.K.^{-1}$	Thermal conductance of thermoelectric material
C_L	$W.K.^{-1}$	Thermal conductance between bases excluding thermoelectric material
C_p	$J.kg.K^{-1}$	Specific heat of air, indices c : cold side h : hot side
C_{pm}	$J.kg.K^{-1}$	Specific heat of discrete objects
C_{p1}	$J.kg.K^{-1}$	Specific heat of wall 1
E	m	Thickness of wall 1
H	$W.m^{-2}K^{-1}$	Convection coefficient of heat pump per unit area of base, indice c: cold side, h : hot side
h_{1e}	$W.m^{-2}K^{-1}$	Convection coefficient on outside of wall 1
h_{1i}	$W.m^{-2}K^{-1}$	Convection coefficient on inside of wall 1
I	A	Electrical current through heat pump
J	$A.cm^{-2}$	Electrical current density through heat pump
$\overline{K}$	$W.m^{-2}K^{-1}$	Reduced overall heat loss coefficient of enclosure
K	$W.K.^{-1}$	Overall heat loss coefficient of enclosure
K_1^*	non dim	Proportion of heat through wall 1
K_2^*	non dim	Proportion of heat through wall 2
K_a^*	non dim	Proportion of heat in air renewal
K_1 ; K_2	$W.K^{-1}$	Overall conduction term through wall indices 1 & 2 for walls 1 & 2
M	kg	Mass of discrete objects
$\overline{M}$	$kJ.m^{-2}K^{-1}$	Reduced heat capacity of discrete objects $\overline{M} = \frac{M.C_{pm}}{S_1}$
Q	$kg.\ s^{-1}$	Air flow rates indices : a : flow of renewed air c : flow through cold side of heat pump, h : flow through hot side of heat pump
r	Ω	Electrical resistence of thermoelectric material
S_1	m2	Area of wall 1
S_2	m2	Area of wall 2
T	K	Air temperature indices : c : cold side of heat pump h : hot side of heat pump
T_b	K	Temperature of fin base of heat pump, second indices c : cold side, h : hot side
T_t	K	Temperature at interface of thermoelectric material second indices : c : cold side, h : hot side
T_i	K	Interior temperature of enclosure
T_{1i}		Temperature of wall 1 on the inside
T_o	K	Outside temperature, also initial temperature of the whole system.
T_F	K	Equilibrium temperature corresponding to the operating condition of enclosure with $\overline{W}_H$ = 120
t*	non dim	time $t^* = \frac{\alpha}{E^2}.t$
t	s	time
W	W	Heat is positive ; cold is negative
W	W	Heat fluxe : Indices c : generated on cold side of heat pump h : generated on hot side of heat pump 1 heat losses through wall 1 2 heat losses through wall 2 d heat absorbed by discrete objects.
W_{el}	W	Electrical power entering heat pump
W_a	W	Heat contained in renewed air ; second indices : i : air from inside o : air from outside
α	$m^{-2}.s$	Thermal diffusivity of wall 1
ρ	$kg.m^{-3}$	Specific mass of wall 1
σ	m2	Area of base
Δx	m	Thickness of slices in wall 1 used for thermal calculations
λ	$W.m^{-1}K^{-1}$	Thermal conductivity of wall 1

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