4 - Transient response of a railroad driver' s cab cooled by a thermoelectric heat pump. J. G. Stockholm, J. P. Buffet. 16 th Intersociety Energy Conversion Engineering Conference. Atlanta August 1981 (American Society of Mechanical Engineers; New York N.Y. paper N° 819826 pages 2022-2029).

819826

TRANSIENT RESPONSE OF A RAILROAD DRIVER'S CAB COOLED BY A THERMOELECTRIC HEAT PUMP

J.G. STOCKHOLM
AIR INDUSTRIE
16, rue Moulin des Bruyères
92400 COURBEVOIE, FRANCE

J.P. BUFFET
2, Avenue Dode de la Brunerie
75016 PARIS, FRANCE

ABSTRACT

The transient energy equations of enclosures cooled by thermoelectric heat pumps (air conditioning units) are presented. The model assumes cooling losses through an insulated wall with heat capacity, glass panes and air renewal. Discrete objects inside must also be cooled. Calculations and results presented as graphs are given using non dimensional and reduced parameters. This model is used for railroad driver's cabs and is verified experimentally with a freon compressor cooling unit where the input cooling was constant. Thermoelectric cooling is examined where the electrical current through the unit decreases with time. Cooling times for a railroad driver's cabs to reach a given inside air temperature with a thermoelectric unit are reduced by a factor of 2 compared to the time required with a freon compressor unit of equal cooling power at enclosure equilibrium.

NOMENCLATURE

Symbol	Units	
a	V/K	Seebeck coefficient
C	W/K	Thermal conductance
C	J/(kg.K)	Specific heat with indice p
E	m	Thickness of wall 1
H	W/(m2K)	Heat transfer coefficient at base of heat exchanger
h	W/(m2K)	Heat transfer coefficient at interface with fluid
I	A	Electrical current through thermoelectrical material
i	J/kg	Enthalpy of moist air
J	A/cm2	Electrical current density per cm2 of thermoelectric material
K	W/K	Cooling losses (indices 1, 2, a)
m	m2	Area of thermoelectric material
M	kg	Mass of discrete objects
Q	kg/s	Air flow rates for system
q	kg/s	Air flow rate through subunit
R	K/W	Thermal resistance
r	Ω	Electrical resistance of thermoelectrical material
S	m2	Area of wall (indices 1, 2)
T	°C	Temperature
t	s	Time
w	W	Cooling and heating powers of subunit
W	W	Cooling and heating powers of unit
x	m	Thickness of slice in wall 1
λ	W/(m.k)	Thermal conductivity of wall 1
ϖ	kg/kg	Water content per kg of dry air
π	Pa	Partial gas pressure
ρ	kg/m3	Specific mass
σ	m2	Base area of subunit heat exchanger

Superscripts

*	Non dimensional e.g. : $T^* = (T_o - T_i)/(T_o - T_f)$
—	Reduced parameters are divided by area S_1 of wall 1 e.g. $\overline{W}_c$.
n	Time increment

Subscripts

Symbol	Indices relative to :
a	Air renewal (Q_a, W_a) associated with i and o
b	Base of heat exchanger associated with c and h
bsat	Saturation corresponding to temperature of cold base
c	Cold side
d	Discrete objects
e	Thermoelectric material (Ce, ...)
el	Electrical power
f	Equilibrium temperature
h	Hot side
i	Inside of enclosure (T, h) Between bases of heat exchangers excluding thermoelectric material (C)
o	Outside air
p	Heat capacity of air associated with c and h
pm	Heat capacity of discrete objects c_{pm}
p1	Heat capacity relative to wall 1
s1	Sun on wall 1
s2	Sun on wall 2
t	Surface of thermoelectric material associated with c and h e.g. R_{th} = thermal resistance between hot face of thermoelectric material and hot air
ø	Inlet conditions to subunit
1	Outlet conditions to subunit
1	Wall 1
1e	Exterior of wall 1
1i	Interior of wall 1
2	Wall 2

INTRODUCTION

The driver's cabs of locomotives and suburban trains require, in Summer and especially when they have been in the sun, some precooling before they can be driven. A very simple mathematical model is presented, that consists of an enclosure and a cooling unit :

1.1. Enclosure

- Wall 1 of area S_1, of thickness E 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 is glass panes of area S_2 that are assumed to have no heat capacity and are characterized by an overall thermal heat transfer coefficien K_2.
- Discrete objects inside the enclosure are all the fixtures of total mass M and specific heat C_{pm}. They are assumed to have a high surface to volume ratio so that their temperature follows that of the inside air temperature T_i.
- Air renewal of flow rate Q_a.
- Sun on wall 1. The heat flux W_{1s} from the sun through wall 1 is assumed to be constant in time.
- Sun on glass panes 2. The flux W_{2s} through the panes is a proportion of the incident flux and constant in time.

1.2. Cooling unit

The cooling unit is a cross flow thermoelectric heat pump that is characterized by an amount of thermoelectric material, an air flow rate on the hot side and an air flow rate on the cooling side. The unit is operated at various electrical current densities J (A/cm2 of thermoelectric material). Ref. (1).

1.3. System : Enclosure With Cooling Unit

The air circuits are given below.

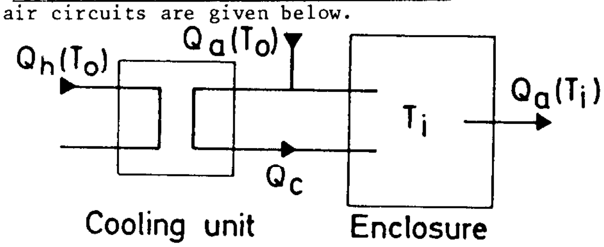

Fig. 1 - Fluid circuits between cooling unit and enclosure.

The cooling and heating fluxes are shown below.

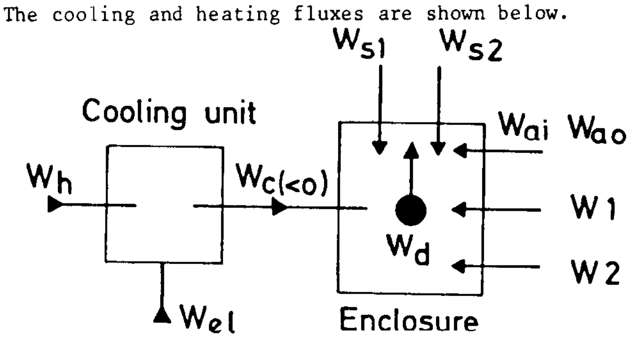

Fig. 2 - Heating and cooling fluxes.

The initial condition is that the whole system : enclosure and cooling unit are at a uniform temperature which is equal or above the outside temperature depending on the amount of heating from the sun. Then, the thermoelectric cooling unit and fans are turned on. The equations given enable the calculation of the inside temperature of the enclosure versus time. Thermoelectric cooling units present the advantage over freon compressor cooling units in that the electrical current through the thermoelectric unit can be varied and the cooling time can be considerably reduced compared to the cooling time required by a freon cooling unit having the same nominal cooling power.

2 THERMOELECTRIC COOLING UNIT

The air-air thermoelectric unit is composed of a number of subunits. The thermoelectric energy equations are solved for each subunit, so the inlet conditions to the next subunit are known, this enables a step by step calculation.

2.1. Equations of Subunit

The two thermoelectric equations are :

$$w_c = - aI(T_{tc} + 273.15) + 1/2\, rI^2 + C_e\,(T_{th} - T_{tc}) + C_\ell\,(T_{bh} - T_{bc}) \qquad (1)$$

$$w_h = + aI\,(T_{th} + 273.15) + 1/2\, rI^2 + - C_e\,(T_{th} - T_{tc}) - C_\ell\,(T_{bh} - T_{bc}) \qquad (2)$$

After the air has gone through on the sold side, a certain number of heat exchangers, the air temperature will reach its dew point hence it will condense some of its water vapor. The condensation leads to more complicated equations than without (1). Nevertheless the case of condensation is examined first so that the case without can be presented in an analogous way.

2.1.1. Subunit with condensation

The first step is to eliminate the terms T_{tc}, T_{th} temperatures of cold and hot faces of the thermoelement and T_{bh} temperature of base on hot side, and to keep in the equations the term T_{bc} which is the temperature of the base of the heat exchanger on the cold side. This term is important because depending whether it is above or below the dew point of the air, there is not, or there is condensation.

$$T_{th} = T_h + R_h\, w_h \qquad (3)$$

T_h is the temperature of the air on the hot side :

$$T_h = T_{h\emptyset} + w_h\, /\, (2q_h\, C_{ph}) \qquad (4)$$

As there is no condensation on the hot side

$$T_{th} = T_{h\emptyset} + w_h \left[R_{th} + 1\, /\, (2q_h\, C_{ph})\right] \qquad (5)$$

To simplify the writing

$$k_h = 1\, /\, (2q_h\, C_{ph}) \qquad (6)$$

and : $T_{th} = T_{h\emptyset} + w_h\,(R_{th} + K_h)$ (7)

In the same way :

$$T_{bh} = T_h + w_h\, /\, (H_h\, \sigma_h) \qquad (8)$$

$$T_{bh} = T_{h\emptyset} + w_h \left[k_h + 1\, /\, (H_h\, \sigma_h)\right] \qquad (9)$$

also :

$$T_{tc} = T_{bc} + w_c\, R_{tc} \qquad (10)$$

Replacing T_{tc}, T_{th} and T_{bh} in equations (1) and (2) one obtains :

$$w_c = - aI\left[T_{bc} + w_c\, R_{tc} + 273.15\right] + r_c\, I^2 + C_e\left[T_{h\emptyset} + (k_h + R_{th})\, w_h - T_{bc} - R_{tc}\, w_c\right] + C_\ell\left[T_{h\emptyset} + \left[(k_h + 1/(H_h\, \sigma_h)\right] w_h - T_{bc}\right] \qquad (11)$$

$$w_h = aI\left[T_{h\emptyset} + w_h\,(k_h + R_{th}) + 273.15\right] + r_h\, I^2 - C_e\left[T_{h\emptyset} + w_h\,(k_h + R_{th}) - T_{bc} - R_{tc}\, w_c\right] - C_\ell\left[T_{h\emptyset} + \left[k_h + 1\, /\, (H_h\, \sigma_h)\right] w_h - T_{bc}\right] \qquad (12)$$

The variables are : w_c, w_h and T_{bc} so that equations (11) and (12) can be written :

$$\alpha_1\, w_c + \beta_1\, w_h + \gamma_1\, T_{bc} = \delta_1 \qquad (13)$$

$$\alpha_2\, w_c + \beta_2\, w_h + \gamma_2\, T_{bc} = \delta_2 \qquad (14)$$

The term w_h can be eliminated and one obtains the following equation between w_c and T_{bc} :

$$\beta_2\delta_1 - \beta_1\delta_2 = (\alpha_1\beta_2 - \alpha_2\beta_1)\, w_c + + (\beta_2\gamma_1 - \beta_1\gamma_2)\, T_{bc} \qquad (15)$$

To simplify, we write :

$\alpha_1 \ \beta_2 - \beta_1 \alpha_2 = \overline{A}$

$\gamma_1 \ \beta_2 - \beta_1 \gamma_2 = \overline{B}$

$\delta_1 \ \beta_2 - \beta_1 \delta_2 = \overline{C}$

Equation (15) becomes :

$$\overline{A}\, w_c + \overline{B}\, T_{bc} = \overline{C} \quad (16)$$

A second equation between w_c and T_{bc} is obtained by calculating the enthalpy of the air on the cold side

$$w_c = (i_{bc} - i_c)\, H_i\ \sigma_c \quad (17)$$

This can be written :

$$i_{bc} = i_c + W_c / (H_i\ \sigma_c) \quad \text{but} \quad (18)$$

$$i_c = i_{c\emptyset} + w_c / 2q_c \quad (19)$$

Introducing an intermediate term k_{ci} which presents a certain analogy to k_h

$$K_{ci} = 1/2\ q_c \ \text{hence} \quad (20)$$

$$i_{bc} = i_{c\emptyset} + w_c \left[k_{ci} + 1 / (H_i\ \sigma_c)\right] \quad (21)$$

To simplify : $\alpha_3 = k_{ci} + 1 / (H_i\ \sigma_c)$ (22)

$$i_{bc} = i_{c\emptyset} + \alpha_3\, w_c \quad (23)$$

Between equations (16) and (21) one can eliminate the term w_c to obtain :

$$\overline{A}\, i_{bc} + \alpha_3\, \overline{B}\, T_{bc} = \overline{A}\, i_{c\emptyset} + \alpha_3\, \overline{C} \quad (24)$$

There is a relation between i_{bc} and T_{bc} which is of an implicit form. First there is a relation which introduces ϖ_{bsat} which is the amount of water in saturated air at the temperature T_{bc} of the cold base.

$$i_{bc} = 1006\ T_{bc} + \varpi_{bsat}\ (2501 + 1.83\ T_{bc})\ 10^3 \quad (25)$$

The term ϖ_{bsat} is related to T_{bc} by solving the two following simultaneous equations :

$$\varpi_{bsat} = 0.622\ \pi_{bsat} / (101325 - \pi_{bsat})\ 10^3 \quad (26)$$

$$\log_{10} \pi_{bsat} = 2.7858 + T_{bc} / (31.559 + 0.1354\ T_{bc}) \quad (27)$$

Equations (26) and (27) can be solved by using Newton's methode to obtain an equation of the form :

$$f_1\ (\varpi_{bsat} / T_{bc}) = 0 \quad (28)$$

and equation (25) becomes one of the form :

$$f_2\ (i_{bc},\ T_{bc}) = 0 \quad (29)$$

The resolution of equation (24) using the implicit equation(29) enables one to calculate T_{bc} from an equation of the form :

$$f_3\ (T_{bc},\ i_{c\emptyset}) = 0 \quad (30)$$

To calculate the exit temperature T_{c1} from the subunit one writes that the air coming out from the subunit is a mixture of saturated air that has been in contact with the cold base, which is characterized by (T_{bc} and i_{bc}) and air entering the subunit characterized by $t_{c\emptyset}$ and $i_{c\emptyset}$). This is shown in Fig. 3.

$$T_{c1} = T_{c\emptyset} + (i_{c1} - i_{c\emptyset})\ \frac{T_{c\emptyset} - T_{bc}}{i_{c\emptyset} - i_{bc}} \quad (31)$$

From equation (16) one obtains w_c

$$w_c = (\overline{C} - \overline{B}\ T_{bc}) / \overline{A} \quad (32)$$

and from equation (13)

$$w_h = (\delta_1 - \gamma_1\ T_{bc} - \alpha_1\ W_c) / \beta_1 \quad (33)$$

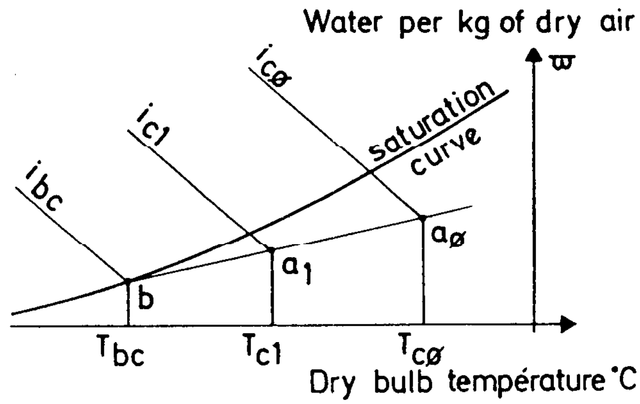

Fig. 3 - Psychometric chart.

2.1.2. Subunit With No Condensation

The following procedure is kept intentionally analogous to the case where there is condensation. All the equations will not be written, but references to the equations with condensation will be given.

Equations (3) through (9) can be written by replacing the indice h by the indice c (there is no condensation on the hot side). Equation (6) becomes :

$k_c = 1/2q_c\ C_{pc}$ so equation (9) becomes :

$$T_{bc} = T_{c\emptyset} + w_c \left[k_c + 1 / (H_c\ \sigma_c)\right] \quad (34)$$

Equations (10) through (21) can be written with indice c. Then, equation (22) becomes : $\alpha'_3 = k_c + 1/(H_c\ \sigma_c)$ (35)

and equation (34) is : $T_{bc} = T_{c\emptyset} + \alpha'_3\ w_c$ (36)

Equation (36) is analogous to equation (23), but where enthalpies have been replaced by temperatures.Comparing equation (36) with equation (16) which is still valid with or without condensation, one obtains a system of 2 linear simultaneous equations in w_c and T_{bc} which are easily solved.

$$\overline{A} w_c + \overline{B}\ T_{bc} = \overline{C} \quad (37)$$

Equations (36) and (37) lead to T_{bc}

$$T_{bc} = (\overline{A} + T_{c\emptyset} + \overline{C}\ \alpha'_3) / (\overline{A} + \alpha'_3\ \overline{B}) \quad (38)$$

Equation (36) can be written :

$$w_c = (T_{bc} - T_{c\emptyset}) / \alpha'_3 \quad (39)$$

and using equation (13) :

$$w_h = (\delta_1 - \gamma_1 T_{bc} - \alpha_1\ w_c) / \beta_1 \quad (40)$$

With the set of equations (38) (39) et (40), one has the cooling powers as a function of inlet conditions.

2.2. Calculation Of Overall Thermoelectric Cooling Unit

In paragraph 2.1.1. and 2.1.2. the outlet conditions of each subunit have been calculated.

Heat exchangers on the hot side do not have condensation so paragraph 2.1.2. is valid.

Heat exchangers on the cold side do not or do have condensation, so when calculating each subunit the cold base temperature is compared to the dew point of the inlet air and if it is lower than the dew point, there is condensation and the calculations of paragraph 2.1.1. are used.

The unit is shown schematically in Fig. 4.

There are 960 subunits in the unit, but it is only necessary to calculate 96 subunits because the operating conditions in the 10 layers cf Fig. 4 are identical. Each subunit is composed 1.5 cm2 of thermoelectric material that has a $Z = 2.58\ 10^{-3}\ K^{-1}$.

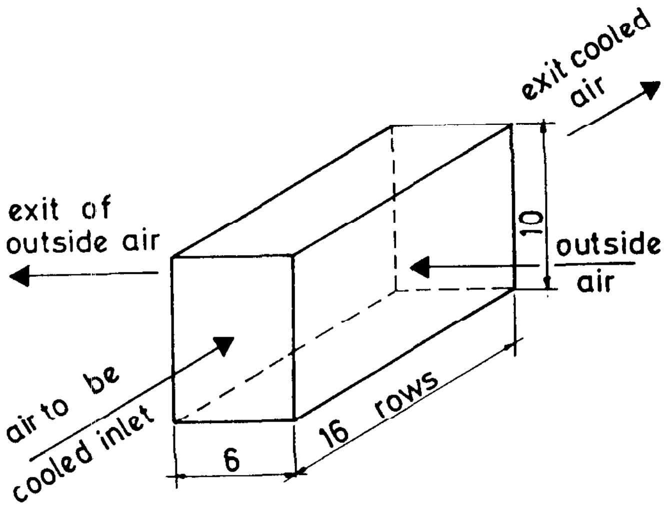

Fig. 4 - Schematic drawing of cross flow thermoelectric cooling unit.

A photograph of such a unit is shown below.

Fig. 5 - Photograph of air to air cross flow thermoelectric cooling unit.

The heat exchangers on either side are finned and made out of aluminium. Their characteristics are :

- Cooling side 242 cm2 of fin area with an average heat transfer coefficient of 52.4 W/(m2 K).
- Outside air circuit 192 cm2 of fin area with an average heat transfer coefficient of 65 W/(m2 K).

The dimensions of the unit excluding fans are 1000 x 600 x 450 mm. The weight is 180 kg.

The flow rates are :

- Cooled air flow 800 m3/h
- Outside air flow 2167 m3/h

3 ENCLOSURE

The enclosure calculation that is examined is based on a method developped by Jacq[2] and used by Veron[3] for transient heat losses in buildings.

The enclosure is assumed to have the following one dimensional heat flows. Details are given in paragraph 1.1

W_c cold entering the enclosure through cooled air

W_1 heat entering through a wall of area S_1

W_2 heat entering through glass of area S_2

W_{1s} solar heat entering through wall 1

W_{2s} solar heat flux going through the glass panes

W_d cold absorbed by discrete objects in the enclosure, they have a high surface to volume ratio so that their temperature follows the inside enclosure air temperature with a time lag equal to the difference in time between two steps in the transient calculation.

ΔW_a heat gained through moist air renewal is divided into two parts.

ΔW_{da} heat gained through the dry air renewal.

ΔW_{wa} heat gained through the difference in water content of entering and exiting air renewal.

The whole system : enclosure and cooling unit is assumed to be at a uniform temperature T_i (t = o) which due to the sun is greater than the outside temperature T_o.

Then the enclosure is cooled by cold air from the cooling unit. With the above assumptions the transient temperature calculation of the inside temperature T_i of the enclosure is relatively simple .

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

$$W_c = W_1 + W_2 + W_{1s} + W_{2s} + W_d + W_{da} + W_{wa} \quad (41)$$

With :

$$W_1 (t) = h_{1i} S_1 \left[T_i (t) - T_{1i} (t)\right] \quad (42)$$

$$W_2 (t) = K_2 S_2 \left[T_i (t) - T_o\right] \quad (43)$$

$$W_d = \frac{MC_{pm}}{\Delta t} \left[T_i (t) - T_i (t - \Delta t)\right] \quad (44)$$

$$\Delta W_{da} = QaC_p \left[T_i (t) - T_o\right] \quad (45)$$

$$\Delta W_{wa} = Q_a (\varpi_i - \varpi_o) C_{pv} \quad (46)$$

4 RESOLUTION OF THE COOLING UNIT-ENCLOSURE SYSTEM

The general equation is obtained by adding all the terms of equation (41). It is advantageous to group together the 3 terms W_c, W_{1s}, W_{2s} and W_{wa} hence we can write :

$$W_{cs} = W_c - W_{1s} - W_{2s} - \Delta W_{wa} \quad (47)$$

In this way one can write the following equation (48) and the terms on the right hand side of the equal sign are those of the "enclosure model" of Ref. (2).

$$W_{cs} = h_{1i} S_1 \left[T_i (t) - T_{1i} (t)\right] + K_2 S_2 \left[T_i (t) - T_o\right] + \frac{MC_{pm}}{\Delta t} \left[T_i (t) - T_i (t - \Delta t)\right] + Q_a C_p \left[T_i (t) - T_o\right] \quad (48)$$

W_{cs} is obtained from equation (47) and W_c from equation (39), this enables one to calculate T_i and T_{1i} (which are related) from equation (48), the time increment used is $\Delta t = x^2 / 2\alpha$.

The W_c of the thermoelectric unit is only calculated every time the wall temperature T_{1i} increases by 0.5° C. During the corresponding time the term ΔW_{wa} is assumed constant and is calculated at the same time as the thermoelectric unit.

5 COMPARISON WITH EXPERIMENT, CONSTANT COOLING INTO ENCLOSURE

Experimental tests have been done at the vehicle testing station at Vienna-Arsenal, Austria, on the driver's cab of the new French fast train T.G.V. The cab has a traditional air conditioning system with a freon compressor, during cooling down tests, measurements showed that the cooling input into the enclosure was constant within a fer percent. Hence the transient response of the cab is used to validate the model when the cooling input is constant in time. The test conditions were the following :

- Outside temperature 35° C and 54 % relative humidity.
- Initial temperature of wall 1 : 40° C.

- Initial temperature of discrete objects : 40° C.
- Simulated incident sunshine w_{1s} = 400 w ; w_{2s} = 700 w.
- Air renewal : 100 m3/h.
- Cooling power : 3500 watts.
- Air flow rate from cooling unit : 500 m3/h.

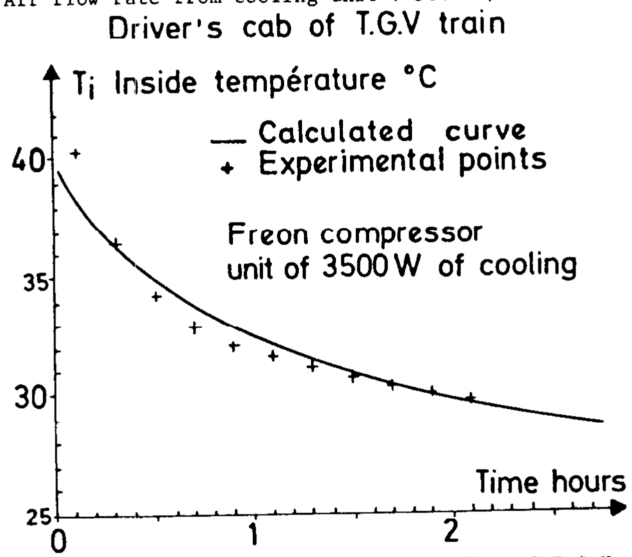

Fig. 6 - Enclosure temperature versus time of T.G.V. train driver's cab comparison between calculation and experiment.

Fig.6 shows the calculated curve (solid line) and the experimental points. At start up the temperature of the experimental points is higher than the calculated one, but the difference between the experimental and calculated never exceeds one degree Celsius. The nearness is considered sufficient to validate the mathematical model of the enclosure. Dimensionless and reduced parameters are used, they are presented in the next paragraph.

6 DIMENSIONLESS AND REDUCED PARAMETERS

The object of introducing non dimensional and reduced parameters is to enable the estimation of transient response of enclosures, cooled by a thermoelectric air to air unit, when the enclosure parameters are in the range of those on the graphs.

6.1. Non Dimensional Parameters

The non dimensional enclosure temperature T^* is defined :

$$T^* = (T_o - T_i) / (T_o - T_f) \tag{49}$$

T^* is the enclosure temperature and T^*_r is the enclosure control temperature defined later

$$T^*_r = (T_o - T_r) / (T_o - T_f)$$

T_o is the outside temperature, T_i is the inside temperature of the enclosure characterized by $\overline{K}$, α and $\overline{M}$ (defined further on) and a given cooling input $\overline{W}_c$ from the thermoelectric unit at equilibrium, T_f is the enclosure's equilibrium temperature.

$$T_f - T_o = \frac{W_c - W_{1s} - W_{2s}}{K} = \frac{W'_c}{K} \tag{50}$$

As the terms W_{1s} and W_{2s} are constant in time, it is useful when characterizing a graph to specify W'_c which is a sort of effective cooling power equal to the thermoelectric cooling power minus the sun's constant contribution.

The non dimensional time t^* is defined in the classic way using the thermal diffusivity of wall 1: α

$$t^* = \frac{\alpha}{E^2} \cdot t \quad \text{where : } \alpha = \frac{\lambda}{\rho C_{p1}} \tag{51}$$

E is the thickness and C_{p1} the heat capacity of wall 1
t is time in seconds.

6.2. Reduced Parameters

They are obtained by dividing by the area S_1 of wall 1 and are written with a bar over the symbol.

- Effective cooling power W'_c becomes :

$$\overline{W'}_c = W'_c / S_1 \tag{52}$$

- Overall cooling loss K of enclosure at equilibrium temperature T_f defined by equation (50) can be written

$$T_f - T_o = \overline{W'}_c / \overline{K}$$

Where $K = K_1 S_1 + K_2 S_2 + Q_a C_p$ and $\overline{K} = K/S_1$

and also the non dimensional parameters.

$$K^*_1 = K_1 S_1 / K \;;\; K^*_2 = K_2 S_2 / K \;;\; K^*_a = Q_a C_{p(m)} / K$$

These three starred parameters represent at equilibrium the proportion of heat entering the enclosure respectively wall 1, wall 2 and the air renewal.

- Thermal mass of discrete objects $M \; C_{pm}$

$$\overline{M} = M \; C_{pm}/S_1$$

- Thermoelectric unit parameters

The main characteristics of cross flow thermoelectric air-air units are :

. Quality of thermoelectric material coefficient of merit Z
. Amount of thermoelectric material m per unit of cooled air flow, the thermoelectric material is characterized by the total area perpendicular to the electrical current = area per piece multiplied by total number of pieces. m = m2 of thermoelectric material per m3/h flow of cooled air.
. Electrical current density $J = A/cm^2$.
. Ratio of outside air to inside air Q_h/Q_c.
. Cooling power of a unit W_c can be reduced by dividing by one of the two following parameters amount of electric material or by cooled flow rate, the latter is used here. Thermoelectric equipments having the same values for Z, m, Q_h/Q_c and J, operating under the same inlet temperature conditions can be compared. Differences in cooling powers W_c or W_c/m2 of material will be the result of design and technology. Thermoelectric units with a Z and m are designed to operate for a given Q_h/Q_c and J can be only within certain limits.

The Fig.7 gives the cooling power W_c and COP for a unit characterized by :

$$Z = 2.58 \; 10^{-3} \; K^{-1}$$
$$S = 193 \; \mu \; V/K \;;\; \rho = 10 \; 10^{-6} \; \Omega.m \;;\; \lambda = 1.44 \; W / (mk)$$

The other parameters are indicated on the graph given on next page. It corresponds to one set of temperature conditions during the transient response.

7 GIVEN ENCLOSURE VARIABLE COOLING

Using the above non dimensional and reduced parameters the temperature T^* versus time t^* is compared for different means of cooling.

7.1. Cooling Power and COP

A thermoelectric unit can be operated at different electrical current densities J, which in the case of transient response is extremely interesting. It was found very advantageous to vary J in a linear way between an initial value J_i at start up and J_r when the inside temperature T reaches a given value T_r.

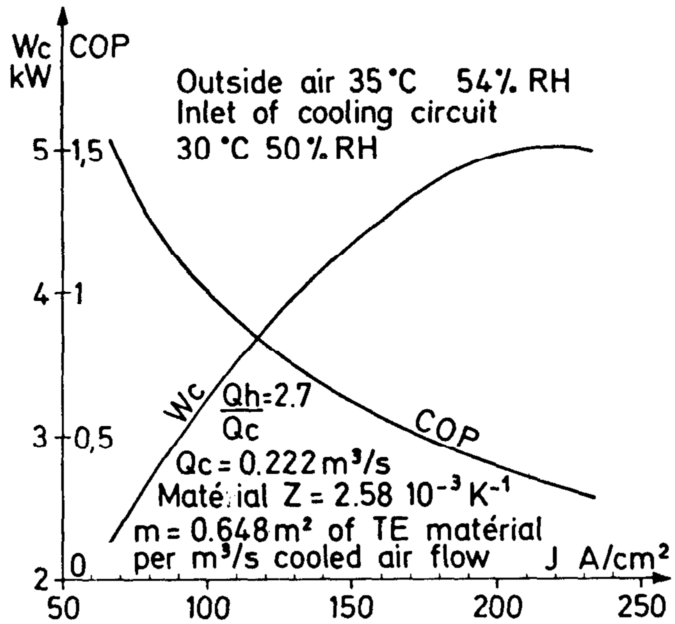

Fig. 7 - Cooling power and COP versus electrical current density J.

The relationship is :

$$J = J_r + (J_i - J_r) \frac{\left[T_i\ (t) - T_r\right]}{\left[T_i\ (t = o) - T_r\right]}$$

All the graphs indicate J_i and J_r meaning that the current density starts at J_i and decreases following the above relationship to J_r.

Fig. 8 shows how $\overline{W}$ and COP vary as a function of time t^* for 3 sets of J_i and J_r.

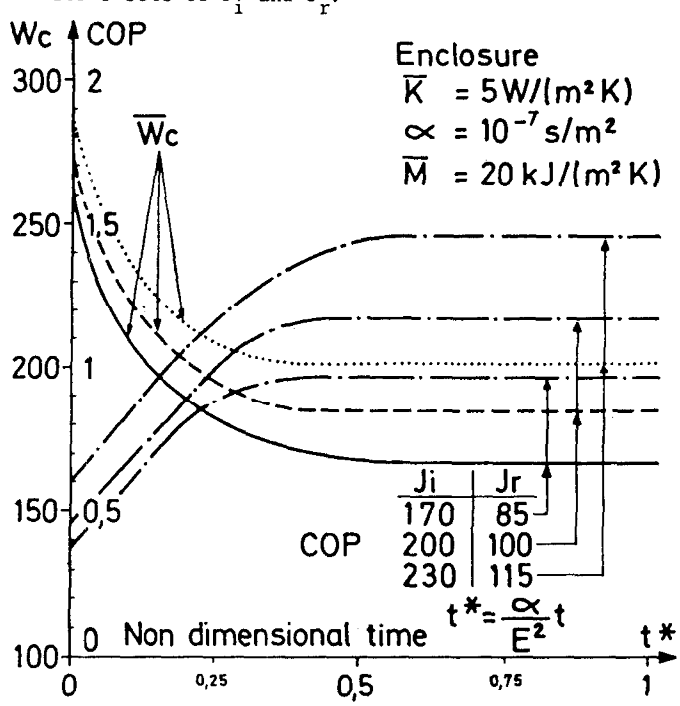

Fig. 8 - Reduced cooling power $\overline{W}_c$ and COP versus time t^* for a given enclosure and for 3 sets of J's.

The next graph Fig. 9 shows for the same enclosure how the electrical power varies as a function of time.

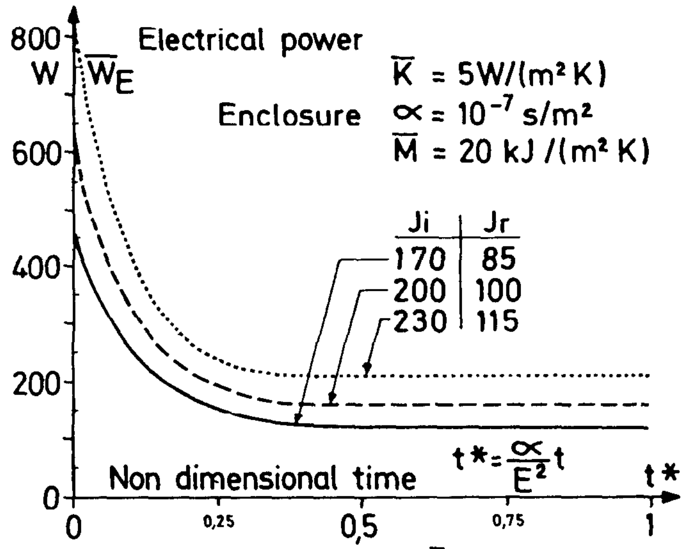

Fig. 9 - Reduced electrical power $\overline{W}_c$ versus reduced time t^*

The thermoelectric unit described accepts these electrical powers and the advantage of using start up electrical densities J_i that are higher than the value at equilibrium is shown in the next graph.

7.2. Comparison Between Thermoelectric And Constant Cooling

It is of interest to show the difference between the temperature T^* versus time t^* curves for a thermoelectric unit and for a constant cooling power unit. The thermoelectric unit is operated at 3 sets of J, the 3 corresponding thermoelectric equilibrium colling powers are the same as those of the constant cooling power units. The following graph shows how much cooling time can be gained by using thermoelectric units with increased starting electrical current densities.

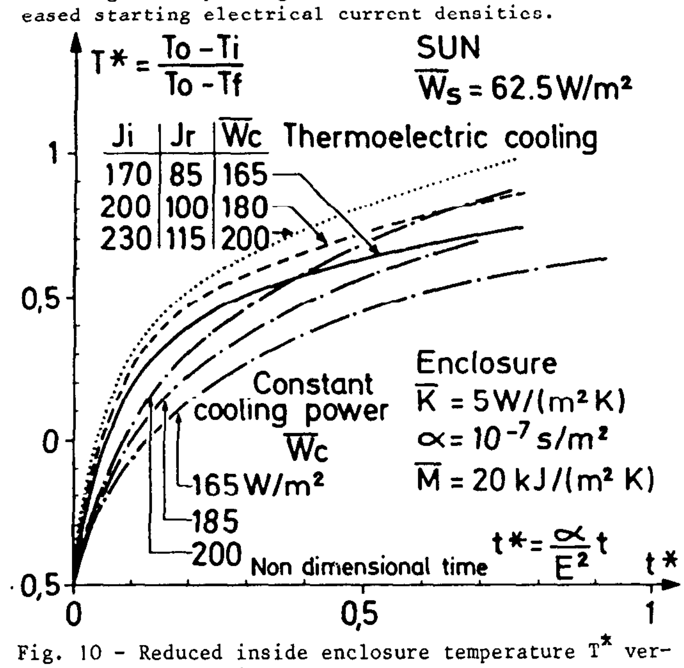

Fig. 10 - Reduced inside enclosure temperature T^* versus reduced time t^*. Comparison between 3 constant cooling powers and a thermoelectric unit operated under 3 sets of current densities.

This figure shows the amount of time that can be gained by cooling with a thermoelectric unit. For the given enclosure $\overline{K}, \alpha$, $\overline{M}$ a cooling unit of constant reduced power $\overline{W}_c$ = 165 W/m2 requires to reach $T^* = 0.5$; a time $t^* = 0.6$, while a thermoelectric unit of the same power at equilibrium $\overline{W}$ = 165 W/m2 that is operated with J : between 170 and 85 A/cm2 only takes a time $t^* = 0.3$. Therefore the use, of a thermoelectric unit operated at a double electrical current density at start up, which is gradually reduced with the enclosure temperature enables a gain in cooling time of around 50 %.

7.3. Influence On Enclosure Temperature Of Current J

In Fig. 10, three sets of J are already plotted, obviously the higher the values of J, faster the temperature T^*, increases towards 1.

The influence of changing J while keeping J_r constant is given in Fig. 11 :

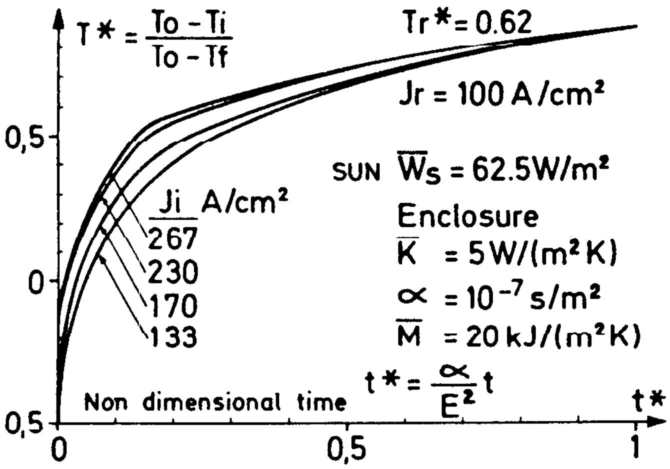

Fig. 11 - Influence of starting current J_i on enclosure temperature T^* versus time t^*.

The above graph shows that appreciable gains in time to reach a given temperature are obtained for reduced temperatures T^* less than 0.6. When the initial current J_i goes from 133 A/cm2 to 230 A/cm2, the time to reach $T^* = 0.6$ is reduced by about 30 % but this requires more electrical energy. Fig. 9 gives the starting up electrical powers for initial currenty of 170 and 230 A/cm2.

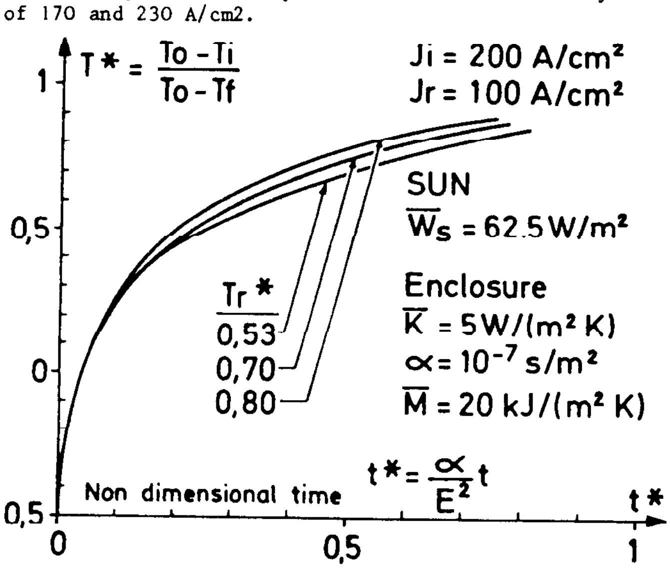

Fig. 12 - Influence of reduced control temperature T_r^* on enclosure T^*.

Influence of control temperature T_r^*. It is shown in Fig. 12, when T_r^* goes from 0.53 to 0.80 at $T^* = 0.6$ there is a time gain of about 25 %.

8 DIFFERENT ENCLOSURES

The thermoelectric cooling unit previously defined in Fig. 7 is operated in the same way with J_i = 200 A/cm2 and J_r = 100 A/cm2 at $T_r^* = 0.62$.

The thermoelectric cooling unit at the enclosures equilibrium temperature (with J = 100 A/cm2) has a cooling power $\overline{W}_c$ of 180 W/m2.

The enclosure is characterized by the three parameters
- $\overline{K}$ reduced overall heat loss coefficient of enclosure.
- α diffusivity of wall 1.
- $\overline{M}$ reduced thermal mass of discrete objects in enclosure.

The influence of the three parameters $\overline{K}$, α and $\overline{M}$, is examined.

8.1. Influence of $\overline{K}$

This parameter represents the heat losses through the walls. As $\overline{K}$ increases the greater the heat losses, hence the temperature T^* changes more slowly.

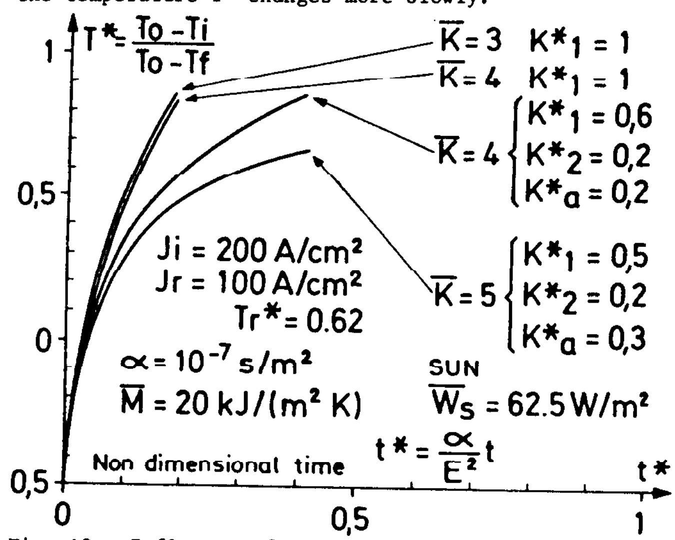

Fig. 13 - Influence of overall heat loss coefficient K on enclosure temperature T^* versus time t^*.

8.2. Influence Of Thermal Diffusivity

The graph below shows the influence of the thermal diffusivity of wall 1 on the enclosure temperature T^* versus time t^*.

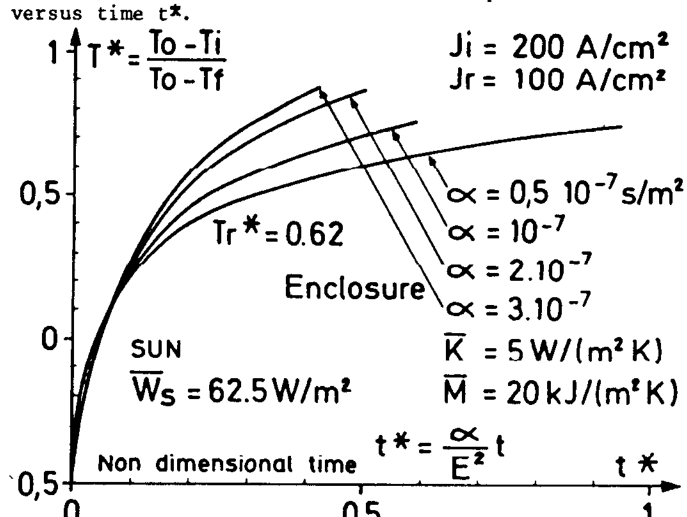

Fig. 14 - Influence of thermal diffusivity of wall 1 on enclosure temperature T^* versus time t^*

9.3. Influence Of Thermal Mass Of Discrete Objects

The parameter $\overline{M} = MC_{pm} / S_l$ which represents the cold that can be absorbed by discrete objects inside the enclosure. As can be seen from Fig. 15, it is an important factor. When the reduced thermal mass of discrete objects $\overline{M}$ varies from 0 to 30 the time to reach $T^* = 0.6$ increases over 2.5 times. This is the most important factor, often neglected in transient temperature calculations of enclosures.

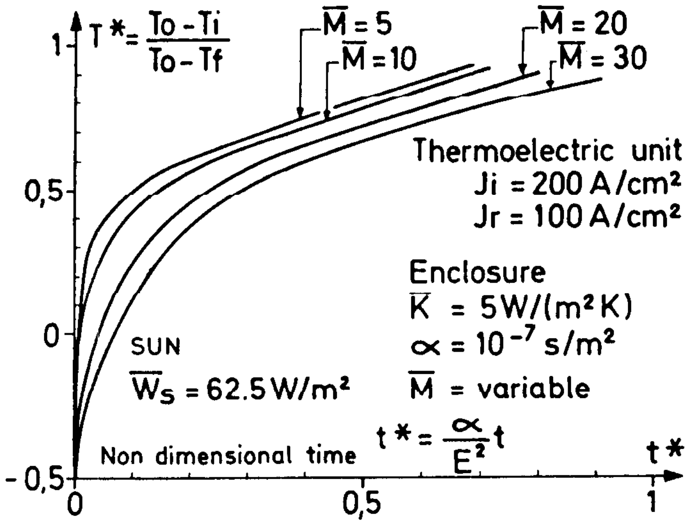

Fig. 15 - Influence of reduced thermal mass $\overline{M}$ of discrete objects on enclosure temperature T^* versus time t^*.

9 CONCLUSIONS

The cooling of enclosures by a thermoelectric cooling unit is examined. The enclosure consists of two walls receiving or not receiving heat from the sun both have cooling losses, one has a thermal capacity, the other which can be a window is assumed to have no thermal capacity. The enclosure contains discrete objects and has a constant air renewal. This type of enclosure describes that of a locomotive's driver's cab.

A thermoelectric unit is used to cool such enclosures. The advantage of a thermoelectric cooling unit over a conventional freon compressor unit is that a thermoelectric unit can be operated at start up with an electrical current density (Amperes per unit area of thermoelectric material) greater than the appropriate value at thermal equilibrium conditions of the enclosure.

Graphs are given with non-dimensional and reduced parameters that enable similar enclosures to be examined, under slightly different temperature conditions.

It is found advantageous to start up with an electrical current density that is double the value of the final value reached at a predetermined control temperature T_r.

The gain in cooling time with a thermoelectric unit is of the order of 50 %.

REFERENCES

1. Stockholm J.C. ; Despres J.P. Large Scale Thermoelectric Cooling. International Conference on Thermoelectric Energy Conversion, Arlington, Texas, March 1978.

2. Jacq J. Contribution à l'étude graphique des régimes thermiques variables. Comptes rendus de l'Académie des Sciences de Paris. 1951 p. 1292 - 1294 ; 1951 p. 2292 - 2295 and 1951 p. 1537 - 1539.

3. Veron Marcel. Champs thermiques et flux calorifiques Bulletin Technique de la Société Française des Constructions Babcok et Wilcox. Vol. 23 oct. 1950 et Vol. 24 nov. 1951.

4. Cadiergues R. Propriétés de l'air humide. Promoclim E Etudes thermiques et aérauliques. Tome 8 E N° 2 Mars 1977, p. 70 - p. 88, Tome 9 E N° 1 Janvier 1978 p. 21 - p. 33.

reprinted from

PROCEEDINGS OF THE 16TH INTERSOCIETY
ENERGY CONVERSION ENGINEERING CONFERENCE
VOLUME 2

published by

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
345 East 47th Street, New York, N. Y. 10017
Printed in U.S.A.