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# THERMOELECTRIC AIR CONDITIONING WITH WATER HEAT REJECTION

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## 1. SUMMARY

A brief review of thermoelectric air conditioning with water heat rejection is given.
Water vapour condensation can produce water droplet carry over which considerably reduces cooling performance.
A model to calculate condensation is described, calculations are presented in graphic form that give water film thickness on heat exchange surfaces and influence on cooling performances.
A new industrial design is presented that has two important features : a continuous water tube that is grounded and air heat exchangers that eliminate water as it condensates. Performances and specific volumes of subunits are given. The nomenclature includes conversion factors from Metric to U.S. units.

## 2. REVIEW OF THERMOELECTRIC AIR CONDITIONING

Thermoelectric air conditioning units with water heat rejection have been studied and especially for submarines, since the early nineteen sixties.
Several technologies have emerged through the years, there are two basic ones.

### 2.1. Heat exchangers electricially insulated

The heat exchangers are not in electrical contact with the electrical circuit, they are associated with preassembled thermoelectric modules see Fig. 1.

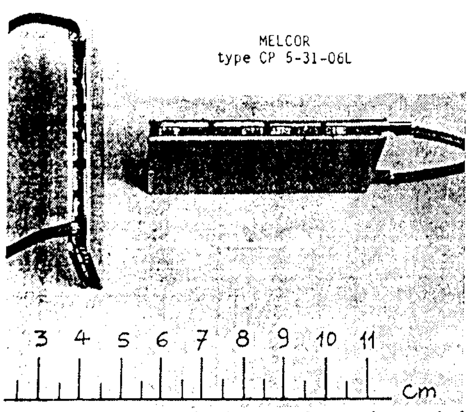


Fig. 1 - Photograph of thermoelectric module.

The main advantage is that it separates the "cold producing part" which is the thermoelectric module from the heat exchangers which can be of a relatively classic design. Two companies studied and built systems based on this technology, Carrier Corporation (1),(2) and R.C.A. (3),(4). The former company air conditioned offices for S.C. Johnson in Racine Wisconsin around 1965, a visit 8 years later confirmed that the system was still working well.

There are numerous disadvantages to this technology when dealing with large systems. The three most important are :

- stress problems because the thermoelectric material is vulnerable to shear stress,
- shock and vibrations can deteriorate the module unless enormous precautions are taken,
- cost of the preassembled thermoelectric module.

### 2.2. Heat exchangers electrically conducting

These technologies are characterized by the fact that at least part of the heat exchangers are used to conduct electricity through the thermoelectric material : see Fig. 2.

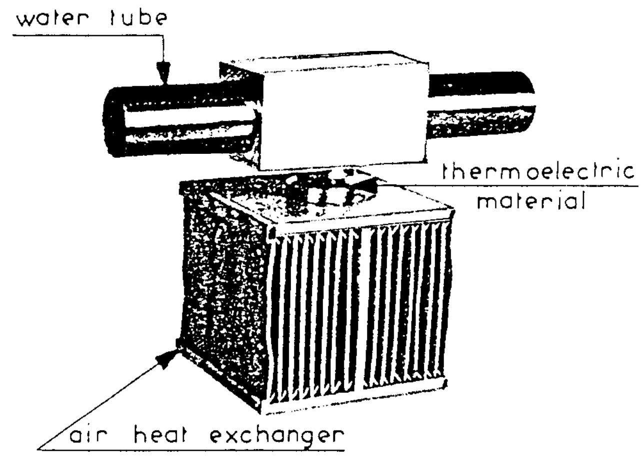


Fig. 2 - Photograph of thermoelectric material integrated with two heat exchangers.

To present these technologies it is necessary to consider separately air and water heat exchangers.

#### 2.2.1. Air heat exchangers- water heat exchangers

In air conditioning the air is moist and the heat exchangers are used to carry electrical current. Precautions against electro-corrosion must be taken and the voltages must not exceed 100 to 200 volts. The condensed moisture must be evacuated either inside or at the exit of the heat exchanger.

The design of water heat exchangers must satisfy the following requirements :

- continuity of the liquid circuit,
- electrical insulation along the circuit between 2 adjacent pieces of thermoelectric material,
- mechanical means of absorbing shear stress on thermoelement.

To satisfy these three conditions there are two industrial designs :

a) adjoining hollow pieces joined together by a seal with two functions : electrical insulation and absorbing small displacements.
b) continuous tube with two functions : partially or totally made of an electrically insulating material allowing small displacements.

#### 2.2.2. Industrial designs

Westinghouse Corporation has designed and built several systems (5),(6),(7) of which some have been in operation for many years. The technology uses electrically conducting air heat exchangers and the water circuit is based on hollow blocks joined together by bellows.
Air Industrie has developed thermoelectric systems where the air heat exchangers are electrically conducting (8) and the water circuit is a continuous tube (9).

## 3. THERMAL MODELING

Thermoelectric thermal modeling has been presented by the authors : for the water side (9) and for the air side (10). Associating the water side with an air side constitutes no problem, as the water temperature variations are small ; the water side can be characterized by an average water temperature. The only difficulty resides in condensation calculation which is presented in detail below, because condensation can have disastrous effects on performances when it is not properly evacuated.

The circuit to be cooled has been divided into 8 portions, each one corresponding to a small heat exchanger associated with a piece of thermoelectric material and a section of water tubing, a schematic is given below.

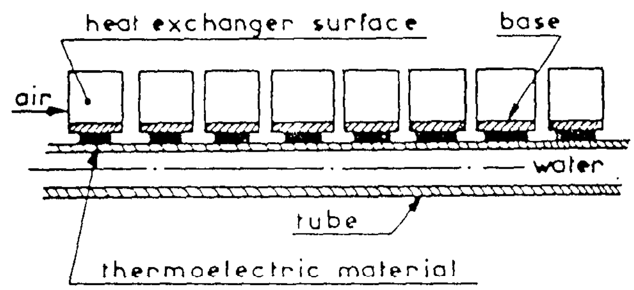


Fig. 3 - Schematic of air flow.

Water flow is chosen to be basically counter flow with the air flow, it is often cross-flow per air heat exchanger but the general flow as seen from outside a subunit is counter flow.
Overall calculations with counterflow require iterative procedures ; so the water temperature would be assumed to be known at the exit and the calculation would lead back to an inlet water temperature. In the following analyses where the interest lays in condensation calculations, the water temperature will be assumed to be known.
The following calculation is done for each heat exchanger :

### 3.1. Condensation theory

- Heat exchanger cooling power is equal to product of air flow rate by the difference in air enthalpy : $W_c = q\ (i_1 - i_o)$
- when there is condensation, air inside the boundary layer at $t_s$ is assumed to be saturated at the temperature $\theta s$ of heat exchanger base with an enthalpy $i_s$.(10)

The heat exchanger base is the coldest part of the exchanger so condensation starts on it before continuing on the heat exchanger surfaces.

As we know $W_c$ q and $i_o$ ; so we can calculate $i_1$.

The fundamental assumption is that :

The air exiting from the exchanger at $t_1$ is considered to be a mixture of air that has gone straight through without any change of its characteristics ($t_o$ and $i_o$) and air that has gone into the boundary layer which is saturated at the temperature $t_s = \theta_s$ of the surface of the exchanger (12),(13),(14).
This assumption is written :

$$\frac{t_1 - t_o}{i_1 - i_o} = \frac{t_o - t_s}{i_o - i_s}$$

examining the psychrometric chart of Fig. 4 :

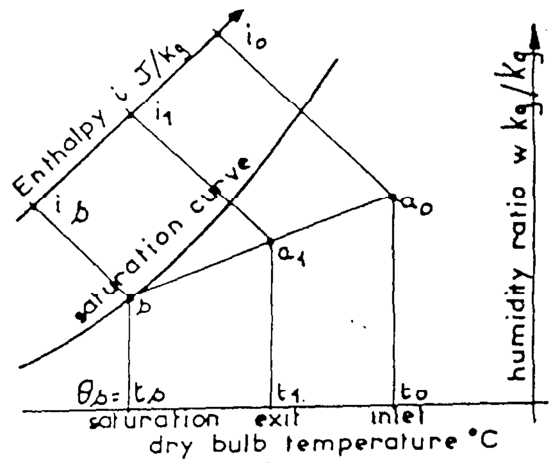


Fig. 4 - Psychrometric chart.

The above relation means that the 3 points :
- $a_o$ : inlet air
- s : saturated air at the base of exchanger
- $a_1$ : exit air

are in line.

### 3.2. Equations

The procedure consists in assuming a temperature for the base of the cold exchanger $\theta_s$ then calculating the 6 variables given below.

- $W_c$ : cooling power
- $W_h$ : heating power
- $\theta_{pc}$ : cold side temperature of thermoelement
- $\theta_{ph}$ : hot side temperature of thermoelement
- $\theta_{bh}$ : hot side exchanger temperature (base)
- $i_s$ : enthalpy of saturated air in cold exchanger boundary layer.

The equations are the following :

$$W_c = -\, aI\,(\theta_{pc} + 273) + \rho_c\, I^2 + C\,(\theta_{ph} - \theta_{pc}) + C_j\,(\theta_{bh} - \theta_s) \qquad (1)$$

$$W_h = +\, aI\,(\theta_{ph} + 273) + \rho_h\, I^2 - C\,(\theta_{ph} - \theta_{pc}) - C_j\,(\theta_{bh} - \theta_s) \qquad (2)$$

the heat flux between the thermoelement's cold sid and the base of cold exchanger at temperature $\theta_s$ i:

$$W_c = (\theta_{pc} - \theta_s)/R_{bc} \qquad (3)$$

heat flux between hot side of thermoelement and hot water :

$$W_h = (\theta_{ph} - t_h)/(R_{bh} + 1/(H.\sigma_h)) \qquad (4)$$

heat flux between cold exchanger base and average cold air of enthalpy $i_c$ :
(for the use of E Enthalpy transfer coefficient (12),(13),(14))

$$W_c = (i_s - i_c)\ E.\sigma_c$$

one can eliminate $i_c$ from equation below writing that $i_c$ corresponds to half the cooling power $W_c$

$$\frac{W_c}{2} = (i_c - i_{co}).\ q$$

hence

$$i_s = i_{co} + \left(\frac{1}{2.q} + \frac{1}{(E.\sigma_c)}\right) W_c \qquad (5)$$

also

$$W_h = (\theta_{bh} - t_h)\ H.\sigma_h \qquad (6)$$

The equations (1) to (6) are linear with respect to the six variables. Therefore one can calculate by matrix calculation the enthalpy $i_s$ of saturated air in the boundary layer as a function of the temperature of the exchanger base $t_s = \theta_s$
The temperatures are calculated using classical

moist air equations (11) that relate $i_s$ to $t_s$.

$$i_s = 1006\ t_s + r_s.(2501 + 1.83 t_s) \quad 1000 \qquad (\underline{7})$$

in which $r_s$ is the humidity ratio of saturated air at $t_s$. The relationship between $r_s$ and $t_s$ is obtained from the two following equations where $\Pi_s$ the saturation pressure in Pascals of water vapor in air at temperature $t_s$.

$$r_s = 0,622\ \Pi_s / (101325 - \Pi_s) \qquad (\underline{8})$$

$$\log 10\ \Pi_s = 2.7858 + t_s / (31.559 + 0.1354\ t_s) \quad (\underline{9})$$

By equalling equations ($\underline{5}$) and ($\underline{7}$), $i_s$ is eliminated and one solves for $\underline{t_s}$ by an iterative method (Newton). With $t_s$ one can work back using equations ($\underline{9}$),($\underline{8}$) and ($\underline{7}$) to obtain $i_s$.

The saturation air temperature $t_s$ is equal to the exchanger base temperature $\Theta_s$ One can calculate the inlet conditions : $t_o$, $i_o$ and $r_o$

exit conditions : $t_1$, $i_1$ and $r_1$

The amount of condensed water at each heat exchanger is $r_o - r_1$. This procedure is used in paragraph 4.

## 4. INFLUENCE OF CONDENSATION ON PERFORMANCES

First are presented some experimental work done on a large cooling unit of 5 KW, then calculations are done on a subunit designed to operate with water on the hot side.

### 4.1. Experimental study

The experimental laboratory study was done on 5 KW units built with different fin configurations with an air flow rate of 0.3 kg/s and a total exchange area between 50 and 80 m2 ; the object was to eliminate water carry over (water droplets that are in exit air) and minimize water retention. Thermal and mass balances were done and graphs such as Fig. 5 shown below were drawn.

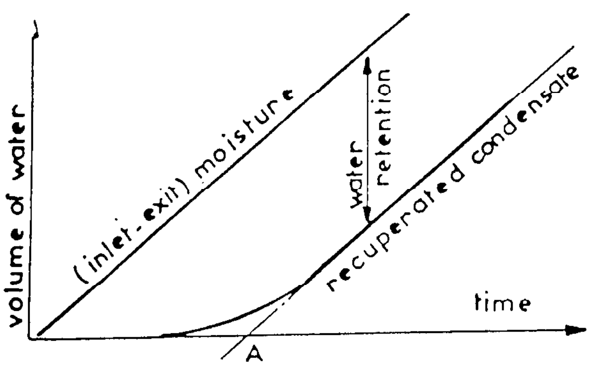


Fig. 5 - Water retention.

The worst design had a water retention which expressed in average film thickness was in excess of 0.3 mm. Point A was greater than several hours. For the best design the average film thickness was of 0.03 mm and point A varied between 30 and 90 minutes depending on the residual condensate at start up.

The best design had a spacing between exchanger surfaces of 2.5 mm there was no water carry over and no difference in cooling power was observed between start up conditions with an energy balance and after prolonged condensation. The condensation aspect has been extensively studied by us because many of our designs must operate in the cooling and in the heating mode. This requirement is very demanding because all the water that is retained is immediatly evaporated in the heating mode and creates abnormally high humidity that is not acceptable.

### 4.2. Theoretical analysis

With the previous experimental background, a detailed theoretical analyses is done on a water-air subunit designed with a favorable fin spacing that goes from 4 mm to 3.3 mm between top and bottom of adjacent fins. The general characteristics are given below.

- water circuit
  - counter flow to air circuit
  - inlet temperature .........20.3 °C
  - exit temperature ......... 24.6 °C
  - average water temperature 22.5 °C
  - flow rate 0.278 kg/s
- air circuit
  - lenght of air circuit 408 mm (8 exchangers)
  - overall inlet air cross section 6.70 dm2 (13 x 4 exchangers)
  - air flow rate per exchanger 2.54 g/s
  - heat exchanger area per exchanger 131 cm2 (8 fins)
  - total air flow rate 132 g/s
  - variable inlet temperature and humidity conditions.

The condensate will be examined for two different air inlet conditions.

| curve reference | Temperature | Humidity Relative |
|---|---|---|
| a | 37,5 °C | 50 % |
| b | 30,0 °C | 50 % |

The evolution of the air's characteristics are shown on the psychrometric chart of Fig. 6.

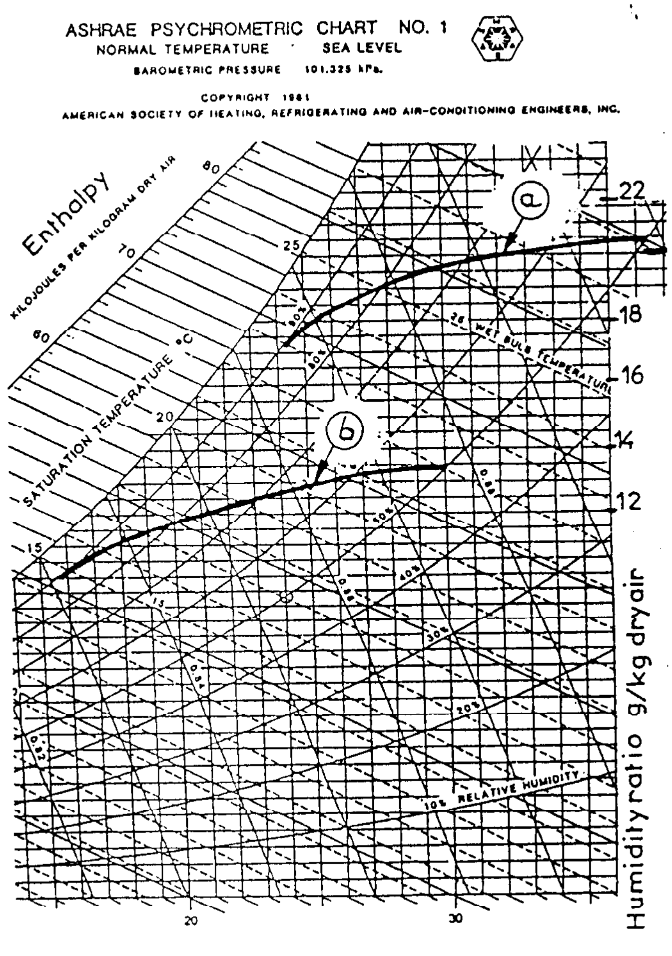


Fig. 6 - Psychrometric variation of characteristics

The exit air has approximately 90 % relative humidity, all our experiments in condensation with thermoelectric systems, when there is no water carry over confirm this value within a few percent.

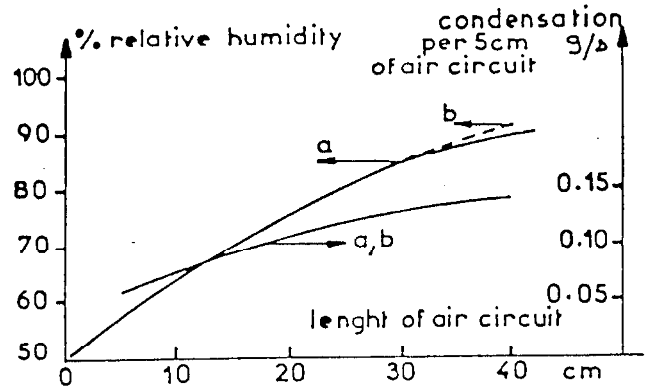


Fig. 7 - Variation of relative humidity and of condensation through cooling unit.

The above figure shows that the curves, for cases (a) 37,5 °C inlet temperature and (b) 30 °C inlet temperature, are nearly identical. The condensed water vapor must be evacuated as it is produced otherwise there is a water film build up which can lead to clogging up between adjacent exchanger surfaces. See Fig. 8 below.

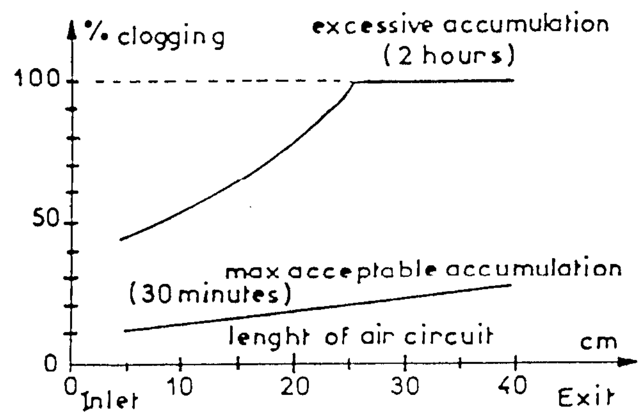


Fig. 8 - Film build up.

The above figure shows the curve for a normal wetting of exchanger surface and an excessive value than can lead to major water carry over problems and clogging up.

Fig. 9 gives the performances of 2 similar subunits that have the same general characteristics except one is designed to eliminate the condensation as it is produced and the other is not.

The general characteristics are :

- average water temperature 17.5 °C
- lenght of air circuit 408 cm
- total air exchanger area 5,45 m2
- air inlet temperature 30 °C
- air inlet relative humidity 50 %
- thermoelectric material 416 pieces of 1.5 cm2 x 1.5 mm.

The coordinates are cooling power as a function of COP.

Each point on each curve corresponds to a different operating voltage, the higher the voltage, the greater the cooling power and lower the COP. The average film thickness are indicated but the performandes were calculated using the film thickness

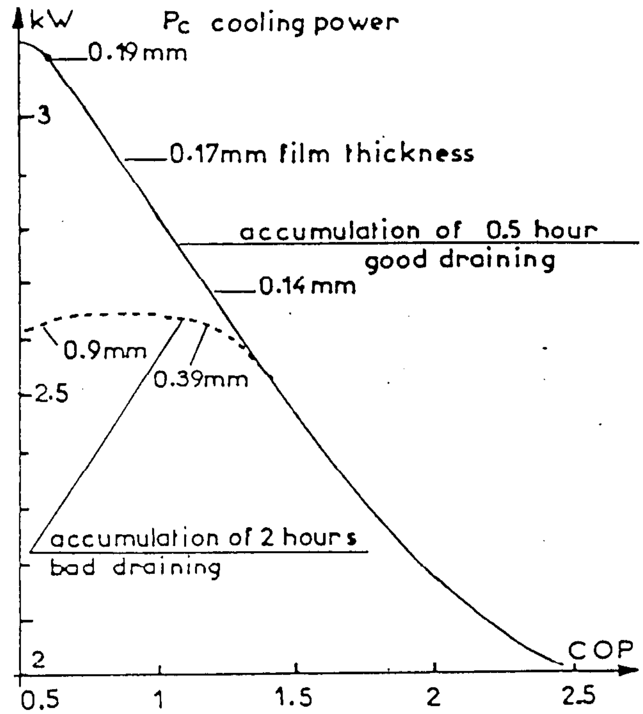


Fig. 9 - Cooling power with film build up.

calculated for each air heat exchanger along the air circuit. One sees how drastically the cooling power drops when the subunit operates with high levels of condensation and bad draining.

5. DESIGN TECHNOLOGIES

Many technologies are possible, each choice is a compromise between conflicting parameters. A survey is given below concerning the fluid circuits, water and air heat exchangers.

5.1. Fluid circuits

- The water circuit requires much less space then an air circuit per unit of thermal flux because water has a thermal capacity 4000 times greater than air.

The major problem in the water circuit is technology because of the need to protect the thermoelectric material from shear stress, our technology uses a partially electrically insulating continuous water tube which has been proven out over prolonged endurance tests.

- The air circuit requires considerable care because of the condensation problem which was examined in paragraph 4. The design we have developed and had operating for several years on a railway coach, is based on :

- horizontal air flow
- evacuation of condensate by gravity inside the air flow

When the condensate is not evacuated at each heat exchanger, there can be water droplet carry over and a water separator is necessary. A separator requires a lenght of 100 mm or more which increases the volume of the subunit by 25 %. It also has a pressure drop generally of the order of 30 Pa which increases the pressure drop through the subunit by more than 10 %.

The separator though it ensures that no droplets exit the system does not solve the problem of the water film that accumulates on the fins and decreases the cooling power.

Systems having air heat exchangers that evacuate the condensation as it is formed are more performing than those with separators because they require less fan power and also have a lower specific volume.

### 5.2. Heat exchangers

Two materials having sufficiently good thermal and electrical characteristics are copper and aluminum, the table below gives comparative values.

| | Thermal Conductivity | Electrical Resistivity | Specific Mass |
|---|---|---|---|
| Units | W/(m.k) | $\mu\Omega$.m | kg/dm3 |
| pur copper | 360 | $1.8.10^{-2}$ | 8,9 |
| Aluminum 99,5 % ASTM1050 | 200 | $3.5.10^{-2}$ | 2,7 |
| Cast Aluminum ASTM6063 (1) | 130 | $7.3.10^{-2}$ | 2,7 |

(1) In house measurements give considerably different values than those from the litterature our results take into consideration the porisity found in castings.

#### 5.2.1. Water heat exchanger

The water heat exchanger consists of a continuous tube such as shown in Fig. 2 which can be made out of various materials such as copper, stainless steel etc ....
The tube is grounded and is electrically insulated from the thermoelectric material. A block of metal conducts the thermal flux from the thermoelectric material to the tube. The block can be made out of copper of aluminum, the latter material saves weight and reduces cost but can reduce slightly performances.

#### 5.2.2. Air heat exchanger

An example of an air heat exchanger that evacuates the condensation as it is produced is shown in Fig. 10.

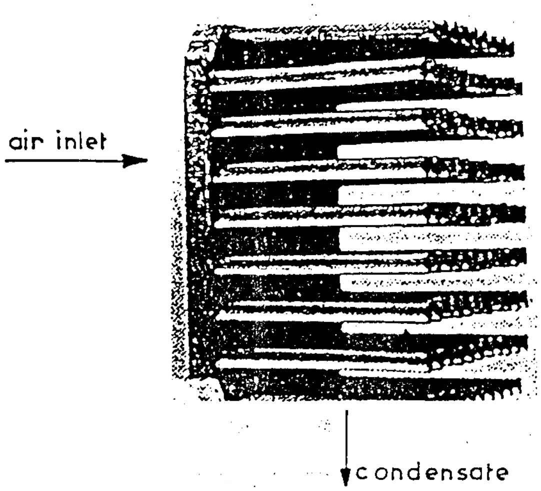


Fig. 10 - Heat exchanger with condensate evacuation

Air heat exchangers are constituted of a base which distributes the thermal flux from the thermoelectric material to one end of the heat exchange surfaces. The dimensions and shapes should satisfy the following constraints :

- thermal and electric characteristics of the material,
- evacuation of condensate
- cost for mass production

And for certain applications :

- minimum mass
- minimum volume

Copper has the most favorable properties but is heavy and also expensive fins can be thin and are easily soldered to a base plate. Detailed cost analyses have led us to often retain aluminum as the best material to satisfy the five constraints indicated above.
Heat exchangers made out of aluminum with a purity of 99.5 % have high efficiencies when fins of thickness 0. 4 mm are welded to a base plate of 3 or 4mm thick. Exchangers made out of drawn aluminum require thick fins for the drawing process, are economic but are not compact. Aluminum casted exchangers are economic but also require thick fins and clearance shapes, the performances are also less than those with thin fins welded to a base.

## 6. INDUSTRIAL DESIGN

Thermoelectric systems are designed on a modular bases and consist of an assembly of subunits. A subunit is discribed, a unit can be composed of 10 subunits placed one above the other to constitute a cabinet the overall dimensions of which would be

- width 600 mm
- height 1800
- depth 800

### 6.1. Subunit description

The overall dimensions of each subunit are :
- width : 408 mm
- height : 148 mm
- depth : it varies depending on the number of fins per air heat exchanger, for 8 fins it is 543mm

A drawing is given in Fig. 11 below.

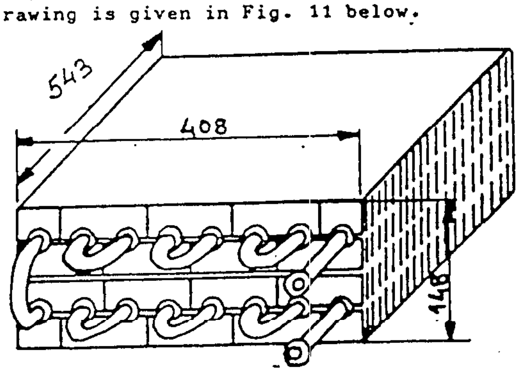


Fig.11 - Drawing of subunit.

It consists of 16 water tubes in series and contains 416 pieces of thermoelectric material of area 1.5 cm2. The blocks between the thermoelectric material and the water tubes are made out of copper. In one design EA403 the air heat exchangers are made out of cast aluminum. Each air heat exchanger is associated with 2 pieces of thermoelectric material ;the fins are perpendicular to the tubes, pins of pitch 5mm, the spacing from top to bottom of fin varies from 4 to 3 mm, the fin thickness is 1.7 mm at the bottom and 1 at the top.

### 6.2. Influence of fin area on cooling power

The object is to show how cooling power increases with fin area (and total volume).
The only variable is the number of fins per air heat exchanger, as the fins and their pitch remain the same it is the length of the heat exchanger along the tube axes that increases hence the depth of the subunit. The number of fins considered is 6,8 and 10 fins per air exchanger. The exchanger characteristics are :

| fins per TE | Number | 6 | 8 | 10 |
|---|---|---|---|---|
| heat transfer area per exchanger | cm2 | 98 | 131 | 161 |
| convection coefficient | w/(m2K) | 63.5 | 53.6 | 47 |
| Inlet dimensions depth | dm | 4,13 | 5,43 | 6,73 |
| height | dm | 1,48 | 1,48 | 1,48 |

The operating conditions are :
- average water temperature 22.5 °C
- inlet air : temperature 30 °C relative humidity 50 %
- air flow rate 0.132 kg/s is kept constant so the air velocity decreases as the number of fins increases.

The cooling powers are given in Fig. 12 below for 3 different numbers of fins per exchanger.

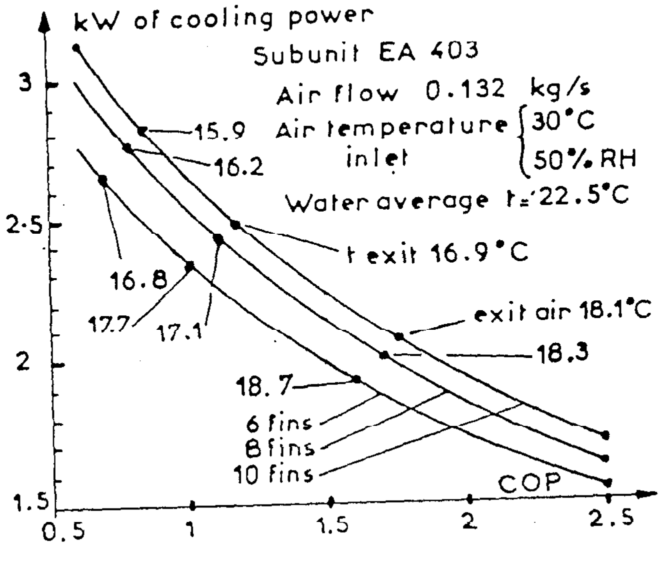


Fig. 12 - Subunit EA403 cooling power as a function of C.O.P.
Exit air temperature are given on the curves in °C.

6.3. Specific volume of subunit

The volume of a subunit is taken to be that of the active part which means the box excluding the tube elbows and water inlets. Fig. 13 below gives the specific volume in dm3/kW as a function of COP.

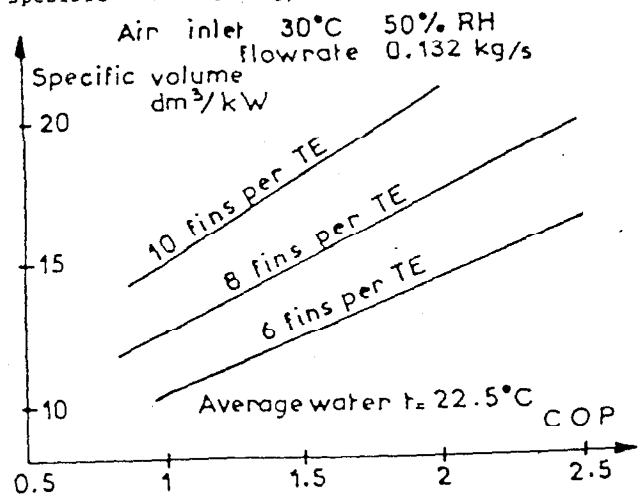


Fig. 13 - Specific volume of unit versus C.O.P.

The inlet conditions are :
- air 30 °C and 50 % RH
  flow rate 0.132 kg/s
- water average temperature 22.5 °C.

The most compact subunit for a given COP is the one that has the smallest heat exchanger (with 6 fins), the decrease in volume from 8 to 6 fins is at COP of 2 is greater than 15 %.

One sees that with the least number of fins the specific volumes goes from about 10 to 15 dm3/kw when the COP goes from 1 to about 2.

6.4. Cooling power function of water temperature

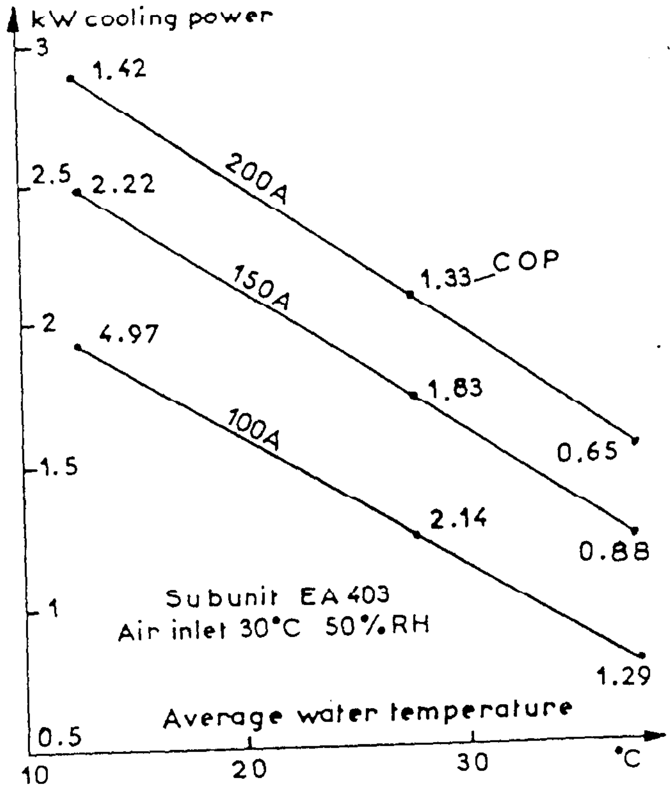


Fig. 14 - Cooling power versus average water temperature.

The cooling power for subunit EA403-6 is plotted in Fig. 14 as a function af average water temperature for inlet conditions of 30 °C, 50 % RH and a mass flow rate of 0.132 kg/s. The graph is plotted for 3 currents 100, 150 and 200 amperes, the COP are indicated against the curves.

6.5. Unit performances

The unit (cabinet) is made up of 10 subunits. The table below gives the cooling powers. The air inlet conditions are maintened constant : 30 °C, 50 % RH flow rate 1.32 kg/s (2.45 Ctm at 86 °F and 50 % RH). COP values can be taken from Fig. 14 for operating points with the same water temperature and electric current.

| Water temperature | °C | 12.5 | 17.5 | 27.5 | 37.5 |
|---|---|---|---|---|---|
| Cooling power for | I=100A | 19 | 17 | 12.5 | 8 |
| cabinet | I=150A | 25 | 22 | 17 | 12.5 |
| KW | I=200A | 29 | 26 | 21 | 15.5 |

Operating voltages of the cabinet are deduced from the cooling power, the COP and the electrical current.

V = Pc/(I.COP)

## 7. CONCLUSIONS

Thermoelectric air conditioning with water heat rejection made a big step forward in the nineteen sixties with the introduction by Westinghouse Corp. of : Direct transfer thermoelectric cooling.
The new concept presented (that can operate with corrosive water) here with a grounded continuous tube for the water circuit and with air heat exchangers designed to eliminate condensation as it is formed, constitutes a major step towards robust and industrially manufacturable systems.
The advantages are : suppression of compressors and of chlorofluorocarbons - modular conception with great flexibility in cooling powers - heating mode obtained by reversing electrical polarity - very high reliability.(15)
Cost is about three times that of traditional cooling systems with compression cycles but the advantages of thermoelectricity can in certain applications outway the disadvantage of a high cost. The performances given are based on experimental results obtained separately for the air and water circuit with thermoelectric materials, so they represent a sound basis for evaluation of thermoelectric air conditioning with water heat rejection using today's commercially available thermoelectric materials. Thermoelectric material selection will improve performances.

## NOMENCLATURE

| Symbol | Units | Designation |
|---|---|---|
| a | V/K | Seebeck coefficient |
| C | W/K | thermal conductance of TE material |
| E | kg/(m2s) | enthalpy transfer coef. on cold base area |
| H | W/m2K) | heat transfer coef. between hot base and hot water |
| i | J/kg | enthalpy of moist air |
| R | K/W | thermal resistance |
| $R_{bc}$ | K/W | " " between TE and base cold side |
| I | A | Electrical current |
| RH | % | relative humidity |
| r | kg/kg | humidity ratio per kg of dry air |
| t | °C | air or water temperature |
| TE | | abbrevation for thermoelectric material |
| W | W | cooling or heating power |
| q | kg/s | air flow rate |
| θ | °C | surface temperature of base or TE material |
| $\sigma$ | m2 | base area of heat exchanger |
| $\Pi_s$ | Pa | saturation pressure of water vapor at $t_s$ |
| $\rho$ | $\Omega$ | electrical resistance |

Indices

- b relative to base of exchangers (temperature θ and thermal resistance R)
- c cold side
- h hot side
- j relative to seal between hot and cold sides
- p relative to pellet of thermoelectric material (TE)
- s saturation
- o inlet conditions of a given heat exchanger
- 1 exit conditions.

Conversion factors :
1 kW of cooling = 3412 Btu/h = 0.284 tons of refrigeration.
Air at 30 °C and 50 % RH 1 kg has a volume of 0.877 litre.
air flow : 1kg/s (30 °C and 50 % RH) = 1.86 cfm (86 °F)
Specific volume 1dm3/kw = $10.35.10^{-6} ft^3/(Btu/h)$
= $0.124\ ft^3$/ton refrig.

## BIBLIOGRAPHY


1. HUDELSON. G.D., Thermoelectric air conditioning of totally enclosed environments. Electrical engineering, June 1960, p. 460-468.
2. HUDELSON G.D., GABLE GK, BECK A.A., Development of a thermoelectric air conditioner for submarine application ASHRAE Semiannual meeting Jan. 1964 New Orleans La.
3. CROUTHAMEL M.S., PANAS J.F., SHELPUK B, Nineton thermoelectric air conditioning system. ASHRAE Semiannual meeting Jan. 1964 New Orleans.
4. ANDERSON J.R., WRIGHT P.E., Performance estimation of a thermoelectric air conditioner.ASHRAE Semiannual Meeting Jan. 1964 New Orleans.
5. MOLE C.J., MEENAN D.F., Direct transfer thermoelectric cooling. ASHRAE Semiannual Meeting Jan. 1964. New Orleans La.
6. MOLE C.J., FOSTER D.V., FERANCHAK R.A. Thermo cooling technology. IEEE Transactions on industry applications vol. 1A-8, N° 2 March/April 1972.
7. VARLJEN T.C., Thermoelectric environmental conditioning technology, Intersociety conference on environmental systems. San Diego Ca. July 1973. ASME Publication.
8. STOCKHOLM J.G., PUJOL-SOULET L., STERNAT P., Prototype thermoelectric air-conditioning of a passenger railway coach 4th. Int.Conf. on Thermoelectric Energy Conversion, Arlington, Texas, March 1982.
9. BUFFET J.P., STOCKHOLM J.G., Industrial Thermoelectric water cooling 18th Intersociety Energy Conversion Engineering Conference, Orlando, Florida Aug. 1983.
10. STOCKHOLM J.G., BUFFET J.P., Transient response of a railroad driver's cab cooled by a thermoelectric heat pump. Proceedings of the 16th Intersociety Energy Engineering Conference, Vol. 2 p. 202-2029. Atlanta, Georgie 1981.
11. CADIERGUES R., Propriétés de l'air humide, Promoclim E. Etudes thermiques et aérauliques Tome 8 E N° 2 Mars 1977 p. 70-88 Tome 9 E N° 1 Jan. 1978 p. 21-23.
12. N GUYEN VAN HUONG, Transfert de chaleur et de masse dans une batterie à ailettes. Thermique et aéraulique 02. Oct. - Déc. 1970, p.113-162.
13. VEZILIER J.C., Le calcul des flux de chaleur et de masse dans une batterie à ailettes alimentée en eau glacée ou glycolée. Thermique et Aéraulique T2. N° 5 Juin 1973, P. 55-78, N° 6 Oct.1973, p. 21-49.
14. POMPIDOR G., Influence du pas d'ailettes sur la déshumidification dans la pompe à chaleur air-eau. To be submitted to AICVF Revue Chauffage Ventilation Conditionnement d'air.
15. BUFFET J.P., STOCKHOLM J.G., L'Effet Peltier appliqué au conditionnement d'air - Avantages - 16 è Congrès Int. du Froid - Paris Sept. 1983 - Commission E1, p. 181-184.