Mar 7th, 8:00 AM

Optimization of Crew Comfort System

John R. Malcolm
The Boeing Company

Roland K. Moir
The Boeing Company

Follow this and additional works at: https://commons.erau.edu/space-congress-proceedings

Scholarly Commons Citation
https://commons.erau.edu/space-congress-proceedings/proceedings-1966-3rd/session-3/4

This Event is brought to you for free and open access by the Conferences at Scholarly Commons. It has been accepted for inclusion in The Space Congress® Proceedings by an authorized administrator of Scholarly Commons. For more information, please contact commons@erau.edu, wolfe.309@erau.edu.
Summary

This paper presents an engineering evaluation of the environmental parameters that affect man's comfort during shirtsleeve operation under conditions of weightlessness. To obtain a minimum weight system, the penalty for providing convective, radiative, and evaporative cooling was established. Mathematical expressions were developed to relate how the total metabolic heat generated by a crew member is divided among radiation, convection, and evaporation. These expressions included the vehicle design parameters — air temperature, relative humidity, air velocity, and mean radiant temperature (MRT), and the crew-oriented parameters of clothing thermal resistance and effective wetted surface area. A basic premise was that the system be designed so that the crew member's effective wetted skin is 10 percent of the total area, and the crew member is comfortable under these conditions. For fixed values of the MRT and clothing thermal resistance, the velocity required to provide sufficient convection and evaporation was found as a function of compartment air temperature. The equipment required to dehumidify the compartment and provide air circulation is affected by the relative amounts of heat lost by convection, radiation, and evaporation. Equipment weight and power penalties were established for each mode of heat transfer for fixed values of MRT and clothing thermal resistance and as a function of compartment air temperature. The total vehicle penalty was then obtained.

Before the system design point could be chosen, an examination of the system off-design performance was necessary. This was done by examining how much the effective wetted area increases as the metabolic load increases. The design metabolic loads examined were for maintenance activities and for exercising.

The sensitivity of the optimum design values to changes in crew clothing were investigated by establishing how they would change if the crew were to wear a minimum-thermal-resistance garment. Decreasing the clothing thermal resistance allows the use of lower design air velocities and higher MRT and results in lower vehicle weight penalties. Savings were obtained at the expense of flexibility in operating at off-design conditions.

This study demonstrates that one can find an optimum combination of design parameters of air velocity, air temperature, clothing thermal resistance; and MRT for a wide range of crew activities. Additional work is required to verify the predicted heat and mass transfer coefficients in space vehicles.

Introduction

The first vehicles that took man into space did not provide a shirtsleeve environment for crew comfort. The primary method of cooling was through use of ventilated pressure suits, with a shirtsleeve environment furnished as a backup and as an experiment on the recent Gemini flight. As we move into the area of prolonged flight duration, a shirtsleeve environment becomes more desirable. The crew comfort system becomes more complex and more closely associated with items such as power system, wall heat leak, type of clothing, humidity control, mean radiant temperature (MRT), air temperature, and air circulation.

Comfort conditions for sea-level operation have been well defined through considerable testing and experience. Factors generally considered for determination of sea level comfort are:

1. Air temperature equal to MRT.
2. Considerable convective cooling provided by natural convection.
3. Natural convection largely independent of body orientation.
4. Atmosphere composition of 21% oxygen and 79% nitrogen at 14.7 psia.
5. Clothing easily adjustable.

These factors, however, are not necessarily applicable to zero-g comfort. A factor of utmost importance to maximum comfort during zero-g, but of less concern to sea level comfort, is system weight. The weight of equipment, and the weight associated with the power system for providing air circulation and humidity control, is of major concern on manned spacecraft.

Techniques to optimize crew comfort considerations for environmental control systems that must operate in zero g are needed. This paper presents an analytical procedure to optimize the environmental parameters to obtain the least vehicle weight penalty while providing a high degree of crew comfort.

Optimization Technique

The parameters of MRT, air temperature, relative humidity, clothing thermal resistance, and air circulation may be combined in various ways to pro-
duce a comfortable crew environment. Because of the penalties associated with power-using components and cooling-equipment weights, the optimum combination of parameters for a zero-g shirtsleeve environment may differ from the combination of parameters considered standard for sea level, 1-g comfort.

The procedure followed to optimize the crew comfort system for a zero-g shirtsleeve environment was:

1. Define the requirements that act as constraints or guides for the study.

2. Construct a mathematical model to represent the heat transfer from crew members by radiation, convection, and evaporation (skin and lungs).

3. Using the mathematical model, solve for the velocity required to provide sufficient convection and evaporation to satisfy the heat balance at an average metabolic rate.

4. Establish air-flow requirements of the humidity control system as a function of air temperature and MRT.

5. Calculate the weight associated with the power required for humidity control system air flow to remove the evaporated water.

6. Determine the weight associated with power required for air circulation to determine the convective heat rejection rates (combination of Steps 5 and 6 provides optimum design points for average metabolic rates).

7. Examine the design points for off-design metabolic rates.

**System Requirements**

Study constraints were defined as follows:

1. A balance is required between heat production and heat rejection at a normal body temperature (defined as a constant mean skin temperature of 91°F).

2. An average heat rejection capability is 520 Btu per manhour with peaks of 780 Btu per hour for 1.5 hours to perform maintenance and 1400 Btu per hour for 20 minutes during exercise periods.

3. The relative humidity must be maintained between 40 and 60 percent.

4. Cabin atmosphere is 7 psia (O₂-N₂).

5. Body surface area is 20 square feet.

6. Evaporative capacities between 10 and 25 percent are comfortable; between 25 and 70 are tolerable, and over 70 definitely unpleasant.

7. All internal spacecraft surface temperatures must be maintained above the dew-point temperature.

8. Vehicle contains 1000 cubic feet of free volume and a fuel cell power system; crew is two men; mission time is 30 days.

9. Clothing thermal resistance is 0.1 and 0.5 clo (clo is defined as a resistivity of $0.88°F\cdot ft^2\cdot hr/Btu$).

The type of clothing to be worn in shirtsleeve environments has not yet been defined. Figure 1 illustrates a reasonable range of values for the clothing insulation. Three materials were examined: two cotton materials (one typical of shirtling, and one typical of cotton undergarments) and one wool material similar to medium-weight dress trousers. The figure shows the influence the dead air space has on clothing thermal resistance. The least resistance occurs when the dead air space is zero (as would occur with a garment similar to tights). The plots show that a reasonable minimum value to consider is about 0.1 clo or a thermal resistance of 0.088 °F-ft²·hr/Btu. This would correspond to a tight-fitting cotton undergarment and a tight-fitting cotton overgarment. The curves suggest that 0.5 clo is a reasonable estimate of an average value of the clothing thermal resistance.

**Mathematical Model**

Crew members will lose heat by radiation, convection, and evaporation from both their skin and lungs. There are different fixed weight and power penalties associated with each of these modes of heat transfer. As an initial step in the optimization of crew comfort system, a heat-transfer model was prepared. The heat balance is shown schematically in Figure 2. The model was used to calculate the heat transfer by each mode for a range of values of MRT, air velocity, air temperature, effective wetted surface area of man, and two values of clothing thermal resistance.

The total heat transferred per unit body area can be expressed by:

$$Q_m = q_r + q_c + q_e + q_l$$

(q storage = 0)

where:

$$q_r = \text{radiative heat transfer to the surroundings;}$$

$$q_c = \text{convective heat transfer;}$$

$$q_e = \text{evaporative loss from the skin;}$$

$$q_l = \text{heat removed by the lungs.}$$
Radiative heat transfer can be expressed by:

\[ q_r = \epsilon f \sigma \left[ T_{cl}^4 - (MRT)^4 \right] \text{Btu/hr-ft}^2 \]

where:

\[ \epsilon = 0.95 = \text{body or clothing surface emissivity}; \]
\[ f = 0.75 = \text{radiation area factor}; \]
\[ \sigma = 0.1713 \times 10^{-8} \text{Btu/hr-ft}^2 \cdot \degree R^4 = \text{Stefan-Boltzmann constant}; \]
\[ T_{cl} = \text{average clothing surface temperature (\degree R)}; \]
\[ MRT = \text{mean radiant temperature (\degree R)}. \]

Clothing surface temperature can be expressed by:

\[ T_{cl} = T_s - R_{cl} H \]

where:

\[ T_s = \text{mean skin thermal temperature (\degree F)}; \]
\[ R_{cl} = \text{clothing thermal resistance (\degree F-ft}^2\text{-hr/Btu)}; \]
\[ H = \text{total sensible heat exchange at } T_s \text{ (Btu/ft}^2\text{-hr)}. \]

The combined value of air and clothing insulation may be expressed by:

\[ R_T = R_{cl} + \left[ \frac{1}{R_c} + \frac{1}{R_r} \right] \]

where:

\[ R_c = \text{convective resistance} = \frac{1}{h_c} \degree F \cdot \text{ft}^2 \cdot \text{hr/Btu}; \]
\[ R_r = \text{radiative resistance} = \frac{1}{h_r} \degree F \cdot \text{ft}^2 \cdot \text{hr/Btu}; \]
\[ T_a = \text{air temperature (\degree F)}. \]

Convective heat transfer per unit body area can be expressed by:

\[ q_c = h_c \left[ T_{cl} - T_a \right] \text{Btu/hr-ft}^2 \]

where:

\[ h_c = \text{convective heat transfer coefficient (Btu/hr-ft}^2 \cdot \degree F). \]

The convective heat transfer coefficient can be expressed by:

\[ h_c = 0.0197 V^5 \left( \frac{\rho_{act}}{\rho_{std}} \right)^5 \text{Btu/hr-ft}^2 \cdot \degree F \]

where:

\[ V = \text{air velocity (ft/hr)}; \]
\[ \rho_{act} = \text{air density at actual conditions (lb/ft}^3); \]
\[ \rho_{std} = \text{air density at standard conditions (lb/ft}^3). \]

Skin evaporative heat loss can be expressed by:

\[ q_e = f_s h_m (P_{H_2O,s} - P_{H_2O,a}) h_{fg} \text{Btu/hr-ft}^2 \]

where:

\[ f_s = \text{fraction of wetted area (percent)}; \]
\[ h_m = \text{mass-transfer coefficient (lb H}_2\text{O/ft}^2\cdot\text{hr-mm Hg)}; \]
\[ P_{H_2O,s} = \text{vapor pressure of water on skin (mm Hg)}; \]
\[ P_{H_2O,a} = \text{vapor pressure of water in air (mm Hg)}; \]
\[ h_{fg} = \text{heat of vaporization of water at skin temperature (Btu/lb H}_2\text{O)}. \]

The mass transfer coefficient can be expressed by:

\[ h_m = \frac{1.05 \times 10^{-4} G}{(P_{act}/P_{std})^{3.5}} \text{lb H}_2\text{O/hr-ft}^2\cdot\text{mm Hg} \]

where:

\[ G = \text{superficial mass flow rate (lb/ft}^2\cdot\text{hr)}; \]
\[ P_{act} = \text{pressure at actual conditions} \]
\[ P_{std} = \text{pressure at one atmosphere} \]

Evaporative lung losses or respiratory water loss denotes water transferred from the body during respiration. Recent work by AiResearch Manufacturing Company provided respiratory water loss data for subjects at various pressures, work rates, humidities, and drybulb temperatures. Evaporative lung losses are small — 35 Btu per hour at the average metabolic rate of 520 Btu per hour — and are roughly double with doubled metabolic rate.

Note: Theory predicts that the mass-transfer coefficient is inversely proportional to the compart-
humidity-control systems: one for shirtsleeve oper-
ation, and the other for suited operation. An alternate approach would require two
packages. Cabin air flows through a debris trap,
operator, CC>2 removal components, catalytic burner,
and a reheat heat exchanger. In addition, this ar-
range allows operation during either shirtsleeve
or pressure suit conditions. An alternate approach
required to provide the air circulation to achieve the
thermal balance.

The water evaporated from the man must be
removed by the humidity-control system. Figure 4
plots the air-flow requirements of the humidity-
control system as a function of air temperature and
MRT. The large increase in air-flow requirements
with decreasing air temperature is because the spec-
ific humidity of the cabin air approaches the specific
humidity of the air leaving the condenser. For this
study, a condensing temperature of 45°F was assum-
ed. Flow requirements decrease with a decrease in
MRT due to an increase in heat rejection by radiation
and a decrease by evaporation to maintain a thermal
balance.

Figure 5 shows the power required to provide the
necessary air flow through the humidity-control sys-
tem to remove the evaporated water, and the power
required to provide the air circulation to achieve the
necessary convective heat-rejection rates. Blower
power requirements for humidity control are based
on air-flow requirements and are assumed to oper-
ate at an overall efficiency of 30 percent at a pressure
rise of 8 inches of water. In a typical environmental-
control system, the air-circulation requirements for
humidity control sizes the complete air-revitalization
package. Cabin air flows through a debris trap,
activated charcoal canister, condenser, water separa-
ator, CO₂ removal components, catalytic burner,
and a reheat heat exchanger. In addition, this ar-
range allows operation during either shirtsleeve
pressure suit conditions. An alternate approach
isolating the humidity-control system and thereby
reducing the blower-pressure rise requirements was
not investigated. This approach would require two
humidity-control systems: one for shirtsleeve oper-
ation, and the other for suited operation.

The air-circulation system consists of two fans
and a single fan/heat exchanger combination. One fan
is directed over each crew member at his duty station
to provide the air velocity required for cooling. The
air flow rates are determined for a given air velocity
and a flow area of 2.2 square feet per man. The fans
are assumed to provide 5 cubic feet per minute of air
flow for each watt of power consumed, a value typical
of the fans considered for this application.

The cabin ventilation fan/heat exchanger unit
provides overall air circulation in the cabin and main-
tains the air temperature at the desired level. The
heat exchanger removes the heat transferred by con-
vection to the air from the crew members and the heat
dissipated by the two fans. The air flow rate through
the fan/heat exchanger unit is determined by allowing
a temperature drop of 10°F across the unit.

A single-fan circulation system was also consid-
ered as a substitute for the three-fan system. Prelim-
inary scale model tests (using water) at Boeing have
shown that circulating flow pattern can be obtained
by employing a centrally located ceiling inlet with a
single outlet located on the opposite face near the out-
er wall. The air from the inlet flows along the ceiling
at a relatively high velocity, entraining flow from the
center of the cabin. The resulting circulation pattern
is maintained and is not destroyed by the outflow
when the outlet is located near the outer wall opposite the inlet.

The combined power penalty is shown in Figure 6
as a function of air temperature and MRT. In addition,
the power weight penalty (1.2 pounds per watt) in the case of a fuel-cell power system for a 30-day
mission is presented. As noted, an air temperature
of 70°F to 75°F results in the lowest power penalty
range of MRT and clothing thermal resistance
considered. Equipment such as heat exchangers,
ducts, water separators, fans, and motors will in-
crease in size with an increase in flow rate. How-
ever, near the optimum point, the variation in air-
flow rate with MRT is small and the associated vari-
ation in equipment weight may safely be neglected.
Before selecting a final temperature, an appropriate
MRT must be determined.

MRT Control

The power weight penalty decreases with decreas-
ing MRT. However, selection of an MRT must con-
sider, in addition to weight implications, the effect
of higher metabolic rates, control concept, and con-
densation. Figure 7 shows the allowable range of
temperatures to prevent condensation on the vehicle
walls and equipment. A minimum MRT control of
10°F above the maximum dew-point temperature was
selected. This temperature difference allows a 2°F
control tolerance, 3°F for scatter below the MRT,
and 5°F safety margin. For example, as indicated
in Figure 7, the minimum MRT control for 75°F air temperature and 60 percent relative humidity would be 70°F. Lower air temperatures will allow lower MRT’s.

A knowledge of the compartment MRT is, therefore, essential to sizing the crew comfort system. In industrial or residential air-conditioning work, it is customary to assume that the MRT and the air temperature are equal. The presence of natural convection coefficients ensures that this assumption will not be substantially in error.

The absence of natural convection in a space vehicle poses an entirely different question. In general, the forced convection coefficients on the electronic equipment cabinets, walls, and storage cabinets are small and the temperature within the spacecraft is largely determined by the radiation exchange. The temperature distribution in a typical two-man space station was calculated. The results showed that some local temperatures were below the condensation temperature during significant portions of the orbit.

The study ground rules forbid any temperature below the dew point to prevent the presence of free water in the compartment. Free water in the compartment of an operational vehicle causes such problems as bacteria growth, corrosion, and shorting of electrical and electronic components.

A trade study was conducted to determine the best method of preventing local temperatures from falling below the dew-point temperature. The study results showed that the least vehicle penalty resulted when the cold spots were heated by radiation from storage and electronic equipment cabinets. The heat was supplied to the cabinets by tubing containing the heat-transport fluid. The weight penalties for heating in this manner are small. Almost all of the penalty is in the weight of the tubing and the fluid contained in the tubing. Water was used as the heat-transport fluid in the occupied compartments and the pressure drop in the panels is negligible.

If the same technique of controlling the cold location temperature was extended to controlling the temperatures on the warm faces of electronic equipment cabinets, the compartment MRT would be subject to control. The weight penalty for controlling the MRT of the whole compartment was calculated and found to be approximately 20 pounds, the bulk of which is required to prevent some surfaces from falling below the dew-point temperature.

The pressure drop in the MRT control loop is small and the loop is integrated with the cabin heat-transport loop. Therefore, the weight penalty for MRT control is constant and does not influence the optimum design point.

Design Point Selection (0.50 Clo)

For the highest thermal resistance considered (0.50 Clo), a minimum system weight results when the air temperature is 73°F and the MRT is 70°F as shown in Figure 6. Examination of the system at off-design condition is shown in Figure 8. As noted, during maintenance activities (780 Btu per hour), approximately 32 percent of the body is covered with perspiration. Evaporation capacities between 25 and 70 percent are considered tolerable. While exercising (1400 Btu per hour), approximately 90 percent of the body will perspire (heat storage = 0). Although 90 percent is above the tolerable range, the exercise period is expected to be of short duration. Air Force physiologists at Wright-Patterson AFB have concurred that it is permissible for men to sweat for short periods every day.

The system was not examined at lower than average metabolic rates (for example, during sleeping) because the vehicle studied had a separate sleeping compartment. In addition, the crew has the option of raising the temperature or donning additional clothing.

The foregoing shows that the crew comfort system optimized for average metabolic rates can provide comfort during a wide range of activities. Therefore, the system may be designed at the optimum weight point.

Design Point Selection (0.10 Clo)

For the lowest clothing thermal resistance considered (0.10 Clo), a minimum system weight results when the air temperature is 75°F and the MRT is 70°F as shown in Figure 6. However, before an MRT can be chosen, the system must be examined at off-design conditions. System A in Figure 9 shows how the comfort parameter (fraction of wetted skin) varies with metabolic heat production.

As noted, during maintenance activities, approximately 62 percent of the body is covered with perspiration. While exercising, the body is required to store heat. Lowering the air temperature 5°F will allow maintenance activities to be performed with 40 percent of the body covered with perspiration but the body is still required to store heat during exercise. This condition is not acceptable.

Examination of off-design conditions at a new design point is shown as system B, also on Figure 9. System B design point is 75°F air and 75°F MRT. As noted, during maintenance activities approximately 45 percent of the body is covered with perspiration. While exercising the body is required to store heat. Lowering the air temperature and MRT to 70°F will allow maintenance activities to be performed with approximately 20 percent of the body covered with perspiration. During exercise, approximately 100 percent of
the body is covered with perspiration, but the body is not required to store heat. As in the case of 0.50 Clo, sweating should be permissible during exercise.

The foregoing demonstrates that a crew comfort system optimized for average metabolic rates must be examined at off-design conditions.

The effect of lowering the MRT during increased activity but maintaining a constant air temperature is shown in Figure 10. Lowering the MRT from 75°F to 70°F increases the radiative losses from 220 to 295 Btu per hour, thereby allowing the crew to work at an increased rate without increased evaporative losses. Beginning at a work activity corresponding to a heat production of approximately 605 Btu per hour, the evaporative losses increase to maintain a heat balance. At 605 Btu per hour, the man's body is effectively 10 percent wet. Further increase in activity results in increased wetness and eventually heavy sweating will occur.

If the 70°F MRT design (System A) were selected, obviously no lower MRT temperature capability would be allowed to avoid condensation as shown in Figure 6. In this instance, control of higher metabolic rates would require a variable air velocity. This design approach requires a multiple-speed blower or multiple blowers with the associated problems of hardware availability, development cost, reliability, and system complexity.

Reference to Figure 3 shows that, under design conditions, the 70°F MRT system will require approximately 25 fpm air velocity and the 75°F MRT system (System B) requires approximately 47 fpm air velocity. A review of Figure 6 shows a 50-pound weight penalty for the 75°F MRT design as compared to the 70°F MRT design. The Δ penalty is largely the power penalty associated with the higher air velocity.

The factors influencing the selection of MRT are summarized below for the case of the lowest clothing resistance.

<table>
<thead>
<tr>
<th>System A</th>
<th>System B</th>
</tr>
</thead>
<tbody>
<tr>
<td>( T_a = 75°F, \ MRT = 70°F )</td>
<td>( T_a = 75°F, \ MRT = 75°F )</td>
</tr>
<tr>
<td>1. Δ Weight</td>
<td>0 pounds</td>
</tr>
<tr>
<td>2. ( f_s ) at 780 Btu (maximum sustained level)</td>
<td>45%</td>
</tr>
<tr>
<td>3. Control concept for peak metabolic rates</td>
<td>air temperature and additional equipment</td>
</tr>
<tr>
<td>4. Air velocity</td>
<td>25 fpm</td>
</tr>
</tbody>
</table>

A review of the above data indicates that the lowest clothing thermal resistance system allowed use of lower air velocity across man and less air flow through the humidity-control system. Approximately 75 pounds of weight is saved in the power system by this approach. An additional weight saving is obtained because the lower circulation rates will require smaller system components such as heat exchangers, water separators, and fans. The primary advantage of the highest clothing thermal resistance is system flexibility. This system can operate at higher metabolic rates without changing either MRT or air temperature.
Conclusions

The following conclusions were reached as a result of this study:

1. The environmental parameters of air velocity, air temperature, clothing thermal resistance, and MRT can be optimized to obtain the least vehicle weight penalty for providing crew comfort during a wide range of activities.

2. The lowest vehicle weight penalty is obtained when the crew wears clothing with the lowest thermal resistance.

3. MRT control reduces vehicle weight penalty.

4. MRT control is required for low-air-velocity systems.

5. Refinements in heat and mass-transfer coefficients will improve the value of the optimization technique.

References


3. Correlation of data obtained from the following:


FIGURE 1: CLOTHING THERMAL RESISTANCE
**Latent Heat**

- Respiration
- Perspiration

**Sensible Heat**

- $Q_{\text{MET}}$ = Metabolic Heat
- $T_S$ = Skin Temperature
- $T_{\text{CL}}$ = Clothing Temperature
- $T_A$ = Air Temperature
- $\text{MRT}$ = Mean Radiant Temperature
- $R_{\text{CL}}$ = Clothing Resistance
- $R_C$ = Convective Resistance
- $R_R$ = Radiation Resistance

**Figure 2: Heat Balance**
METABOLIC RATE = 520 BTU/HOUR
RELATIVE HUMIDITY = 50%
PRESSURE = 7.0 PSIA (O₂-N₂)
FRACTION OF WETTED SKIN = 10%
CLOTHING THERMAL RESISTANCE

- 0.5 CLO
- 0.1 CLO

FIGURE 3: AIR VELOCITY REQUIREMENTS
METABOLIC RATE = 520 BTU/HOUR
RELATIVE HUMIDITY = 50%
PRESSURE = 7.0 PSIA (O₂-N₂)
FRACTION OF WETTED SKIN = 10%
CONDENSING TEMP. = 45°F
CLOTHING THERMAL RESISTANCE

0.5 CLO

0.1 CLO

FIGURE 4: HUMIDITY CONTROL REQUIREMENTS
FIGURE 5: HUMIDITY & AIR CIRCULATION POWER REQUIREMENTS
FIGURE 6: COMBINED HUMIDITY & AIR CIRCULATION POWER REQUIREMENTS
FIGURE 7: PSYCHROMETRIC CHART — 7.0 PSIA O₂-N₂
FIGURE 8: 0.50 CLO OFF DESIGN CONDITIONS
FIGURE 10: 0.10 CLO HEAT REJECTION DISTRIBUTION
FIGURE 9: 0.10 CLO OFF DESIGN CONDITIONS