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Thermal Analysis of a Firefighter's Breathing Gas Cooler

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Abstract

A breathing gas cooler design is described to keep the life-support gas inside a firefighter's face mask at, or below, 120°F for a period of at least 30 minutes. Two basic heat exchanger designs are evaluated for their thermal performances. Both designs incorporate a modification to the existing firefighter's apparatus with MicroPCM™ wrapped around the breathing gas supply lines within a foamed metal matrix. The gas cooler designs, in the configuration of an air-to-liquid (or solid) heat exchanger, were designed to be worn comfortably by the firefighter and have quick connect and disconnect air supply fittings for ease of switch out. A 5 foot long, single pass heat exchanger, surrounded with a MicroPCM slurry in either a pig nose or bullseye configuration was found adequate to reduce a firefighter's breathing gas from a temperature of 200°F to 120°F with a respiratory minute volume of 62.5 LPM. Both design configurations have foamed aluminum matrix in contact with the outside of the breathing gas tube to enhance the heat transfer from the breathing gas to the MicroPCM slurry. Eicosane, C₂₀H₄₂, with a melting temperature of 98.1°F was shown to be the best candidate MicroPCM for maximizing the heat transfer efficiency in the breathing gas cooler. Less than 2 pounds of a slurry consisting of 60% Eicosane, and 40% water was found to provide sufficient heat storage capacity to operate the breathing gas cooler for a 30 minute duration. The entire heat exchanger assembly weighed less than 5 pounds.

Introduction

The existing equipment used by the U. S. Navy for firefighting and damage control activities utilizes a chemical process for producing a breathable gas for the user. The chemical reaction required to sustain the process is exothermic in nature, and transfers heat to the breathing gas. Due to the unique requirements for shipboard use, the majority of the candidates for replacement of existing equipment use similar technology. Additionally, this equipment is primarily used in localities of high ambient tempera-

ture. As a result, heat from the ambient environment is transferred through the apparatus to the breathing gas, further elevating the breathing gas temperature and potentially producing a debilitating situation for the user. Recent U. S. Navy tests have concluded that even open-circuit equipment, as shown in Figure 1, deliver a high temperature breathing gas to the user that may limit or prevent the completion of the mission when used in high ambient temperatures.



Figure 1. Photo of existing firefighter system.

The Navy has an ongoing R&D effort to reduce the breathing gas temperature of existing breathing apparatus used for firefighting and damage control activities. The applications of three separate technologies have been considered in this effort for lowering the breathing gas temperatures of existing breathing apparatus. The first technology considered was Composite Fabric Endothermic Material (CFEM). CFEM technology provides endothermic materials that can store large quantities of heat for extended periods of time (Hayes, 1984). These materials can be designed to absorb and release heat at desired temperatures by means of an embedded phase change component, and can be engineered into a variety of materials and configurations. Successful

uses of CFEM have been achieved in such varied applications as ballistic missile design and fast food delivery systems.

A second technology considered for breathing gas cooling was thermo-electric chips (TEC). TECs utilize a Peltier effect to produce a heat flux via voltage differentials, similar in operation to a thermocouple. They are small and can easily be configured into a breathing gas cooling application. TECs are currently used in commercially available portable refrigerators and are readily available for test and evaluation. The primary obstacle in the use of TEC technology is the need for an onboard power source, such as batteries, which must be carried by the

firefighter. Logistical drawbacks that batteries would present for shipboard firefighting applications eliminated this technology early in this R&D effort.

A third technology considered for the breathing gas cooler, and the major focus of this paper, was the use of a heat exchanger design employing a liquid slurry containing **paraffinic** Microencapsulated Phase Change Material (MicroPCM) as the cooling medium.¹ Similar to CFEMs in their heat storing abilities, MicroPCM technology encapsulates various

phase change materials (PCMs) from the paraffin family (see Table 1) in tiny spherical polymeric shells. By containing the phase change material during their transition from a solid to a liquid in a solid shell, MicroPCMs are finding applications where bulk phase change materials would not be appropriate due to the possibility of contamination. This technology has been applied to heating and cooling subsystems for spacecraft applications by NASA, and hot and cold protective clothing for the U.S. Air Force.

Technical Design Issues

The primary technical objective of this development effort was to maintain the breathing gas temperature for an **open-circuit** breathing apparatus below the threshold of pain for the duration that the user is in the hot environment. For environments having high relative humidities, such as those experienced with most closed-circuit breathing apparatus, extreme **discomfort** may occur at inhalation temperatures as low as 110°F, while an upper threshold has been shown to be approximately 122°F

during moderate work activities (Lind, 1977). The corresponding temperature for a dry breathing gas, as experienced with open-circuit units, is upwards to 200°F. Even though NIOSH requires closed-circuit inhalation temperatures to be no higher than 104°F, firefighting conditions aboard U.S. Navy ships often expose the firefighter to ambient temperatures in excess of 500°F for up to five minutes, and to temperatures in excess of 135°F for extended periods of time, potentially as long as one hour.

Table 1. Candidate phase change materials.

Name	Formula	Melting Point		Latent Heat of Fusion		Density	
		°C	°F	cal/gm	Btu/lb	gm/cm ³	lb/ft ³
Hexadecane	C ₁₆ H ₃₄	16.7	62.1	56.7	102.0	0.774	48.3
Octadecane	C ₁₈ H ₃₈	28.0	82.4		105.0	0.774	48.3
Eicosane	C ₂₀ H ₄₂	36.7	98.1	59	106.0	0.77	48.6
Heneicosane	C ₂₁ H ₄₄	40.2	104.4		86.0	0.75	47.3
Docosane	C ₂₂ H ₄₆	44.0	111.2	60	107.0	0.76	47.6
Tricosane	C ₂₃ H ₄₈	47.5	117.5	56	100.0	0.76	47.7
Tetracosane	C ₂₄ H ₅₀	49.4	120.9			0.77	48.0
Paraffin Wax		54.4	129.9			0.86	55.0
Hexadecane MicroPCM		16.7	62.1		84.7		60.9
Octadecane MicroPCM		28.3	83		85.5		60.9

¹ MicroPCM is a registered trademark of Triangle Research and Development Corporation, Raleigh, NC.

During these exposures, heat is transferred from the ambient environment into the breathing apparatus, dramatically increasing the breathing gas temperature. This high breathing gas temperature is one of the major restricting factors involved in **firefighting**. Successfully reducing these breathing gas temperatures will increase the efficiency of the firefighting crew and reduce the amount of equipment required for shipboard firefighting.

Another concern in the design is minimizing size and weight of a breathing gas cooler and maintaining a low pressure drop across the gas cooler. Minimizing the gas pressure drop will lessen the **effort** the user must exert in breathing on the apparatus, and thus, maximize his firefighting **capability**.

The objective of the analyses presented below was to quantify the required heat exchanger **surface** area to cool breathing gases under the following design parameters:

- a) The breathing gas entrance temperature into the heat exchanger would not exceed 200°F.
- b) The exit temperature from the heat exchanger should not exceed 120°F during 30 minutes of operation.

- c) The exterior surface temperature of the breathing gas apparatus will not exceed 200°F (this may vary depending on the type of insulation used in the backpack redesign.)
- d) The heat exchanger will see mid-stage pressure of 130 psig.
- e) The gas delivery flow rate to the user is 62.5 liters per minute (at 1 Ata and 98.6°F).
- f) The **MicroPCM™** will be capable of being recharged when submerged in 90°F water for 30 minutes.

In addition to determining the required heat exchanger surface area, the following analysis attempted to answer:

- 1) What is the appropriate **MicroPCM™** selection for this application?
- 2) Should **MicroPCM™** be mixed with another fluid in the heat exchanger capacitor side?
- 3) What percentage of **MicroPCM™** is needed if a **MicroPCM™/fluid** mix is used?
- 4) Should thermal conductivity enhancers (other than fluid) be mixed with the **MicroPCM™**?

Breathing Gas Cooler Designs

Two basic concepts for the breathing gas cooler were initially considered. The first concept was a "horse-collar" design that integrated a multi-path heat exchanger into a yoke worn around the firefighter's neck. This concept was quickly discarded due to the restrictions in motion that this design would cause to the wearer.

A second concept included a less radical modification to the existing firefighter's apparatus. This concept includes a heat exchanger design consisting of a long piece of stainless steel tubing which is surrounded with a foamed metal matrix and a **MicroPCM** slurry. This single-pass heat exchanger design was felt to provide the least amount of **emburiance** to the firefighter in accomplishing his mission. This heat exchanger is configured in the firefighter's backpack in a serpentine arrangement adjacent to the gas supply bottle, indicated in Figure 2, and the entire **backpack**, including the heat exchanger and the gas **bottle**, is encapsulated with rigid foam to insulate the backpack from the outside environment. A second length of tubing is included in this design to provide a supply of cooling water to the heat exchanger following an operational mission to facilitate regeneration of the **MicroPCM** slurry.

Two different versions of this water supply channel were considered. The first option included an identical water tube which was parallel to the breathing gas tube, both surrounded with the metal foam/**PCM** slurry matrix. This design, referred to as the "pig nose" option, shown in Figure 3, can additionally have a foamed metal matrix inside the air and water tubing to enhance the transfer of heat to and from the tube walls. An insulating outer jacket is used to contain the **PCM** slurry within the foam metal matrix and give an additional thermal barrier to resist heat entering from the outside environment.

A second version of this single pass option includes the water channel on the outside of the foam **metal/PCM** slurry matrix. This design, referred to as the "bullseye" option (Figure 4), maximizes the heat transfer area between the water channel and the foam metal/**PCM** slurry matrix during the regeneration phase. It also provides an additional thermal barrier to resist heat infiltration from the outside environment.

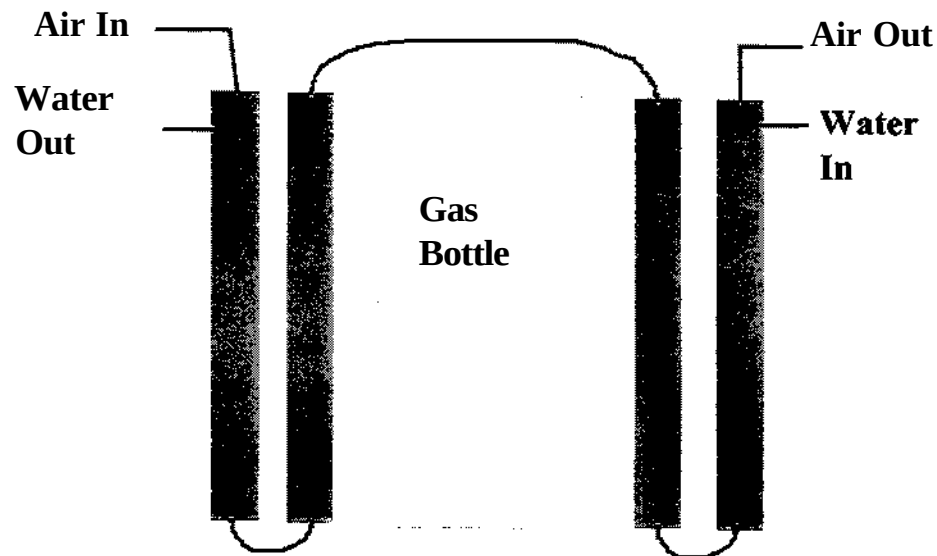


Figure 2. Conceptual single-pass gas cooler.

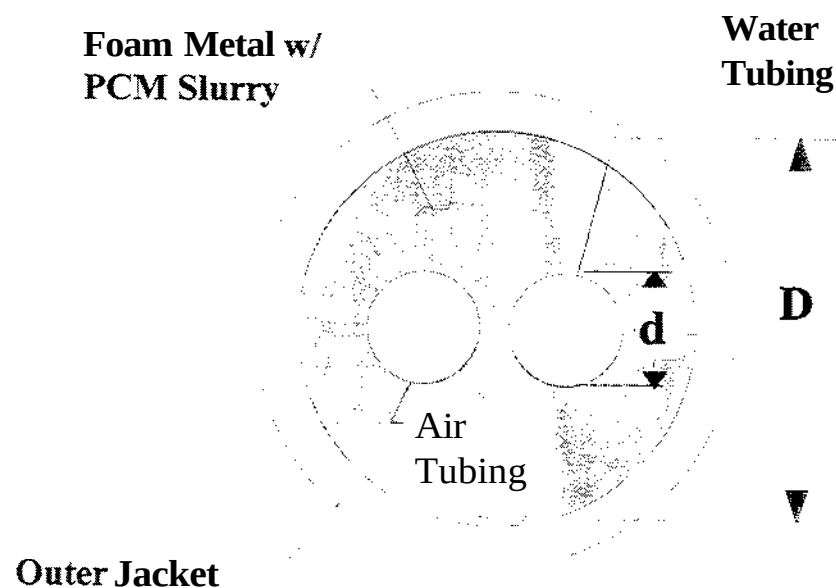


Figure 3. Pig nose design option.

Thermal Analysis of Design Concepts

In the following analysis, the breathing gas heat exchanger was assumed to consist of one or multiple, tubular air passages which are surrounded with a slurry of MicroPCM™ material as shown in Figure 5. The MicroPCM™ is contained within a matrix of "foamed metal" similar to Duocel Foam Metal manufactured by ERG, Inc to enhance the conduction of heat from the gas tubing into the

MicroPCM™ slurry. The melting temperature of the MicroPCM™ was selected such that it will be below the desired gas exit temperature from the cooler. Condensation of water vapor from the air as it cools in the heat exchanger was ignored. The tubing walls were assumed to have negligible heat storage capacity.

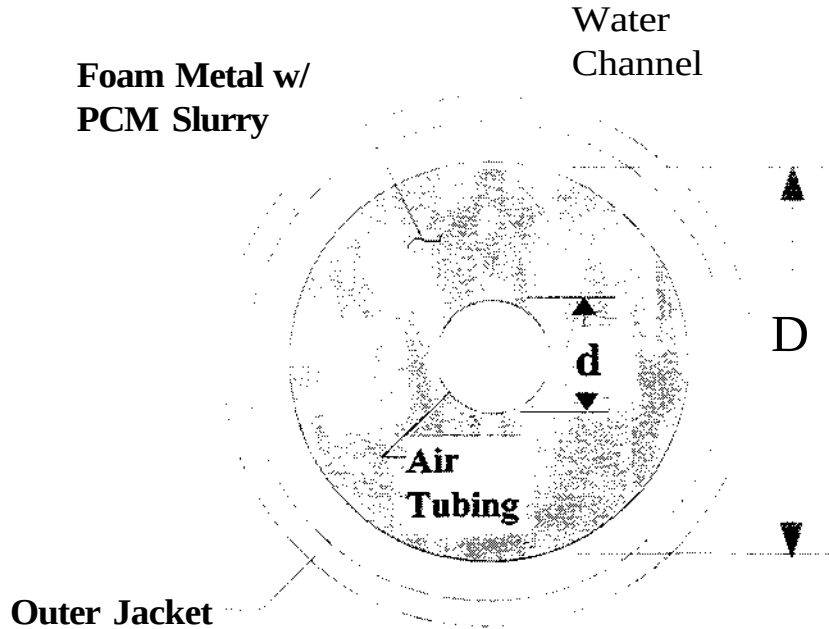


Figure 4. Bullseye design option.

An energy balance within the heat exchanger system can be written as

Energy Added to System = Change in Energy Level of Cooler

If we assume that the gas enters at T_{in} and exits at T_{out} at a mean flow rate of $\dot{V}$, and the MicroPCM™ slurry stays at the melt temperature T_M (the amount of MicroPCM™ is assumed to be adequate to absorb all lost heat from the air as the PCM goes from a solid to a liquid), we can write an expression for the amount of heat lost from the air as

$$\dot{q}_{in} - \dot{q}_{out} = \rho_{air} c_{p,air} \dot{V} (T_{in} - T_{out}) \quad (1)$$

where ρ_{air} and $c_{p,air}$ are the density and specific heat of air, respectively. The energy delivered to the MicroPCM™ slurry/foamed metal matrix through the tubing walls is

$$\dot{q}_{wall} = U_i A_i \Theta_M \quad (2)$$

where U_i is the overall heat transfer coefficient based on the inside surface area of the tubing; A_i is the inside surface area of the tubing; Θ_M is the log mean temperature difference between the air and the MicroPCM™ slurry, defined as

$$\Theta_M = \frac{T_{in} - T_{out}}{\ln \left(\frac{T_{in} - T_M}{T_{out} - T_M} \right)} \quad (3)$$

and

$$U_i = \frac{1}{\frac{1}{h_i} + \frac{r_i}{K_t} \ln \left(\frac{r_o}{r_i} \right)} \quad (4)$$

where K_t is the thermal conductivity of the tube wall material; and h_i is the convective heat transfer coefficient on the inside surface of the tube.

The change in the energy level of the MicroPCM™ can be written as

$$\dot{q}_{PCM} = h_{fg} \frac{dm_L}{dt} \quad (5)$$

where m_L is the mass of MicroPCM™ that goes from a solid to a liquid phase, and h_{fg} is the latent heat of fusion for the MicroPCM™.

Equating (1) and (5) gives

$$h_{fg} \frac{dm_L}{dt} = \rho_{air} c_{p,air} \dot{V} (T_{in} - T_{out})$$

or

$$dm_L = \frac{\rho_{air} c_{p,air} \dot{V}}{h_{fg}} (T_{in} - T_{out}) dt \quad (6)$$

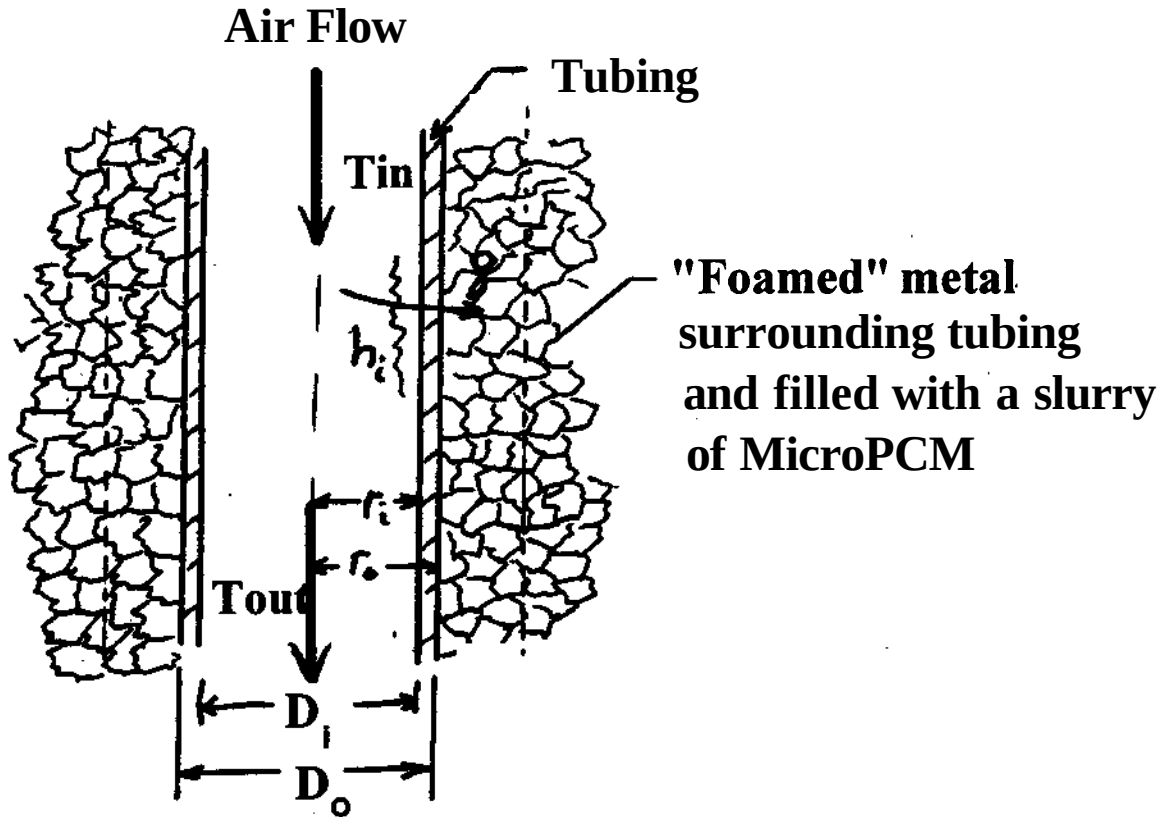


Figure 5. Heat exchange concept in breathing gas cooler.

Also, equating (2) and (5) gives

$$h_{fg} \frac{dm_L}{dt} = U_i A_i \Theta_M$$

or

$$dm_L = \frac{U_i A_i \Theta_M}{h_{fg}} dt \quad (7)$$

Now equating (6) and (7) gives

$$\frac{\rho_{air} c_{p_{air}} \dot{V} (T_{in} - T_{out})}{h_{fg}} dt = \frac{U_i A_i \Theta_M}{h_{fg}} dt$$

Simplifying, and restating the definition given in (3) for the log mean temperature gives

$$\ln \left(\frac{T_{in} - T_M}{T_{out} - T_M} \right) = \frac{U_i A_i}{\rho_{air} c_{p_{air}} \dot{V}}$$

or

$$\frac{T_{in} - T_M}{T_{out} - T_M} = e^{\left(\frac{U_i A_i}{\rho_{air} c_{p_{air}} \dot{V}} \right)}$$

This results in a air exit temperature from the breathing gas cooler given by

$$T_{out} = T_M + (T_{in} - T_M) \exp \left(- \left(\frac{U_i A_i}{\rho_{air} \dot{V}} \right) \right) \quad (8)$$

The overall heat transfer coefficient U_i shown in (8) can be determined from (4) after the convective heat transfer coefficient, h_i , has been found. This coefficient will be a function of the gas properties inside the tubing. We will assume that these properties can be determined based on a temperature that is the average of the inlet and exit gas temperature, T_A , where

$$T_A = \frac{T_{in} + T_{out}}{2}$$

The volumetric gas flow rate, $\dot{V}$, through a single path of a multi-path heat exchanger can be found from the design respiratory minute volume

(RMV given at 1 Ata and 98.6°F) as

$$\dot{V}_{path} = \frac{RMV}{\# paths} \left(\frac{14.7 psia}{P_{HX}} \right) \left(\frac{460 + T_A}{558.6^\circ R} \right) \quad (9)$$

where RMV is given in units of lit/min @ 1 Ata, 98.6°F; $\dot{V}_{th}$ is given in units of lit/min @ P_{HX} (pressure inside the heat exchanger) and T_A °F. After this volumetric flow rate has been found, the gas velocity through each path can then be calculated using the relationship

$$v_{path} = \frac{\dot{V}_{path}}{A_{cs}} = \frac{\dot{V}_{path}}{\pi r_i^2} \quad (10)$$

By assuming that the gas inside the heat exchanger behaves as an ideal gas, its density can be calculated as

$$\rho_{air} \cdot \frac{lbm}{ft^3} = \frac{144 \left(\frac{in^2}{ft^2} \right) P_{HX} (psia)}{53.3 \left(\frac{ft-lbf}{lbm-^\circ R} \right) (460 + T_A)^\circ R} \quad (11)$$

and the viscosity of the air inside the heat exchanger can be found as

$$\mu_{air} \cdot \frac{lbm}{ft-sec} = 1.212 \times 10^{-5} \left[\frac{460 + T_A}{530} \right] \quad (12)$$

The Reynolds number for air flow within a single path of the heat exchanger can then be obtained using

$$Re_{path} = \frac{\rho_{air} v_{path} (2 r_i)}{\mu_{air}} \quad (13)$$

For flow inside a tube, the dimensionless heat transfer coefficient, $\overline{Nu}_D$, is defined as

$$Nu_D = \frac{h_i (2 r_i)}{K_{air}} = 0.023 Re_{path}^{0.8} Pr^{0.3} \quad (14)$$

where Pr is the dimensionless Prandtl number for air (Pr for air is approximately 0.72).

This allows us to calculate the convective heat transfer coefficient on the inside surface of the tubing as

$$h_i = \frac{\overline{Nu}_D K_{air}}{2 r_i} \quad (14a)$$

Once the above variables have been determined, the gas exit temperature from the heat exchanger can be found using relationship previously derived in Equation (8).

Figures 6 through 8 show flow velocities, convective heat transfer coefficients, and exit air temperatures from various heat exchanger designs, each 24 inches in length, with up to 40 parallel flow paths when using phase change materials with melting temperatures between 104°F and 120°F. Figure 8 indicates that approximately 18 to 19 parallel flow paths, each 24 inches long, are required to reduce the breathing gas temperature in the heat exchanger from 200°F to 120°F with a MicroPCM™ melt temperature of 111°F (Docosane, C_{22}). This analysis indicates that the number of flow paths could be reduced significantly (5 to 6 parallel flow paths) to reach the desired exit temperature when the MicroPCM™ melt temperature is reduced to 104°F (Heneicosane, C_{21}). Likewise, a single pass would be adequate with a melt temperature of 83°F (Octadecane, C_{18}). Unfortunately, this low melt temperature would prevent regeneration of the PCM when ambient water temperatures are 90°F, as stated in the design requirements. Additionally, the published latent heats of fusion for heneicosane is only 86 Btu/lbm as compared to 107 Btu/lbm for docosane (see Table 1). This would require a larger mass of the heneicosane than docosane to store the same amount of lost energy from the breathing gases.

The required amounts of MicroPCM™ to cool the breathing gas from 200°F to 120°F can be determined by returning to Equation (6) and using the flow properties found above

$$dm_L = \frac{\rho_{air} c_{p,air} \dot{V}}{h_{fg} (wt\% PCM)} (T_{in} - T_{out}) dt \quad (15)$$

We will assume that the effective latent heat of fusion for the MicroPCM™ is 83% of the published

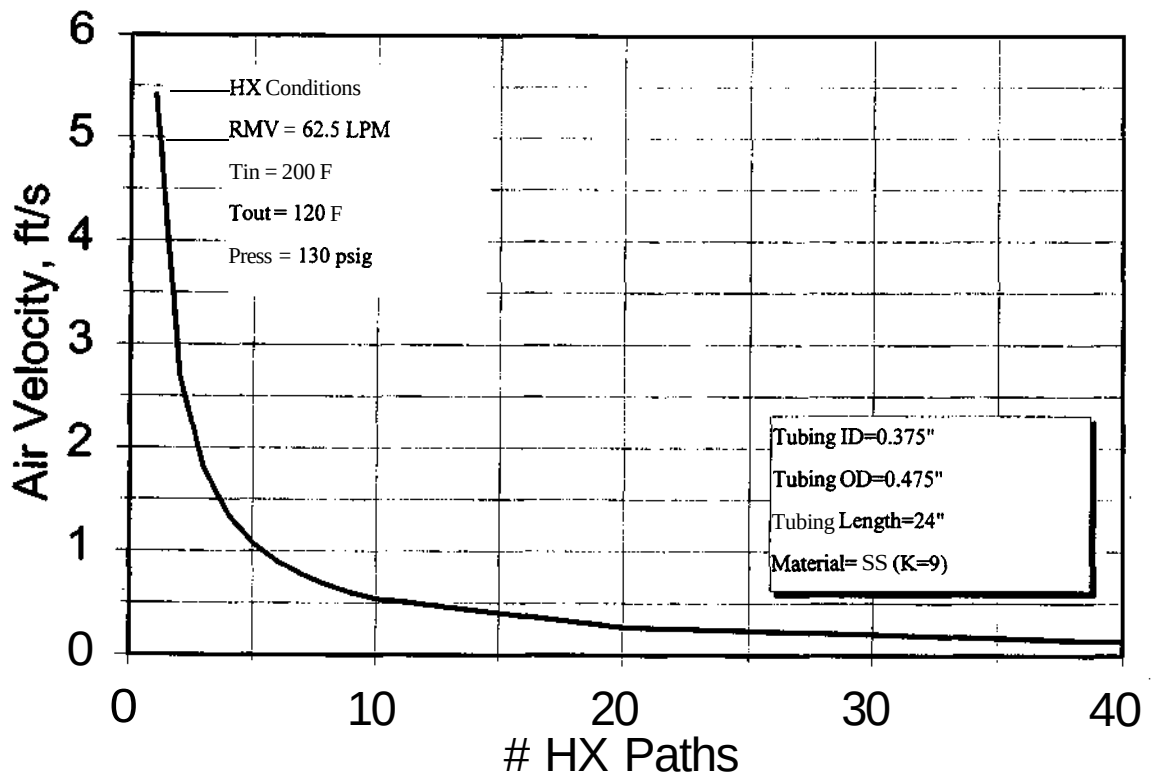


Figure 6. Air velocity in a multi-path heat exchanger.

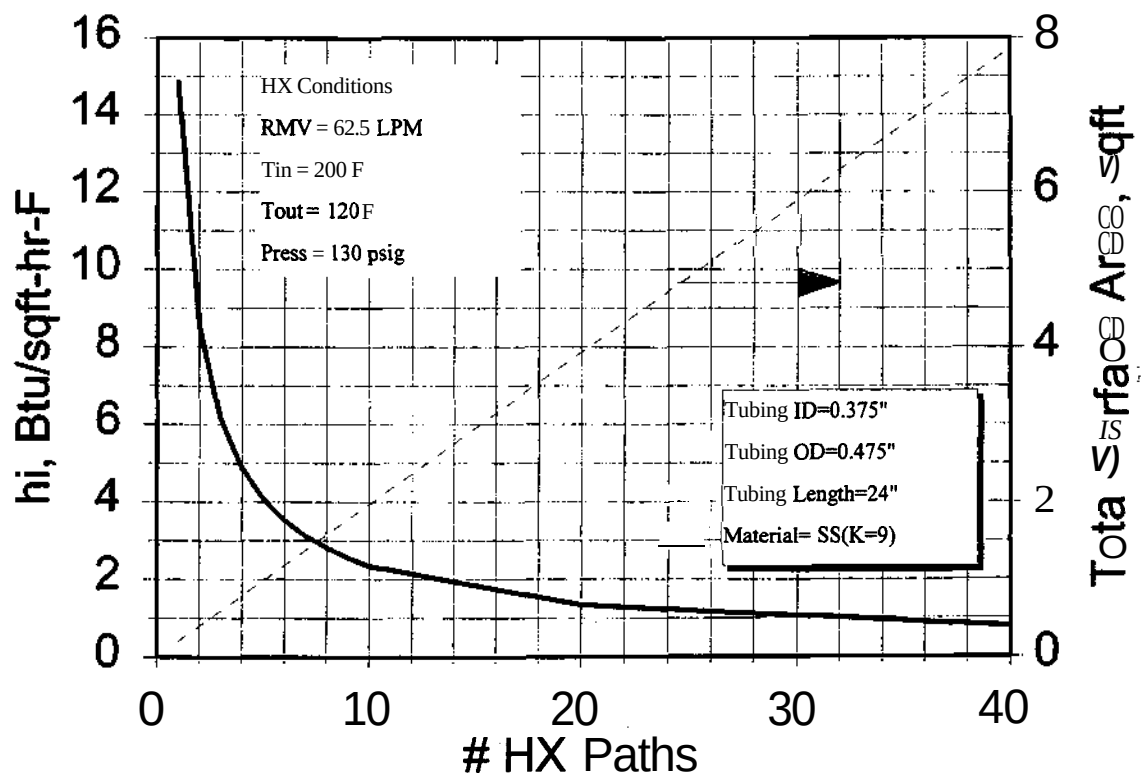


Figure 7. Convective heat transfer coefficient on inside of tubing walls for a multi-path heat exchanger, based on inside surface area.

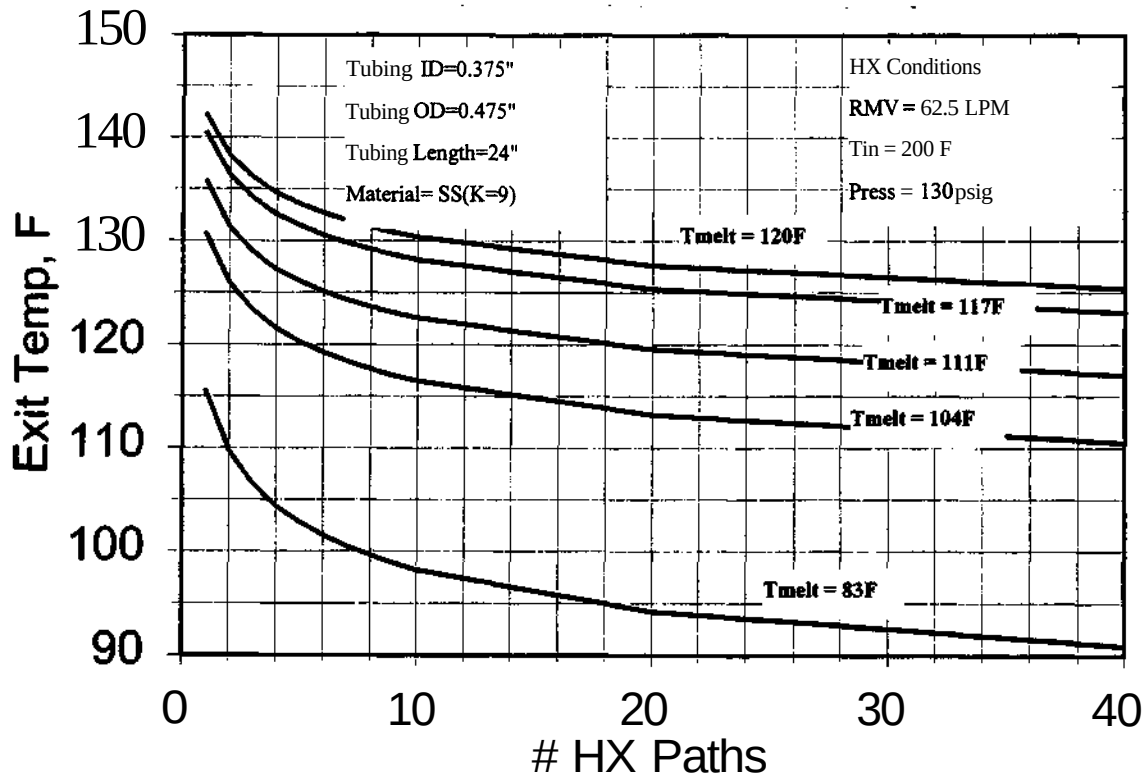


Figure 8. Air exit temperature from a multi-path heat exchanger with various PCM melting temperatures.

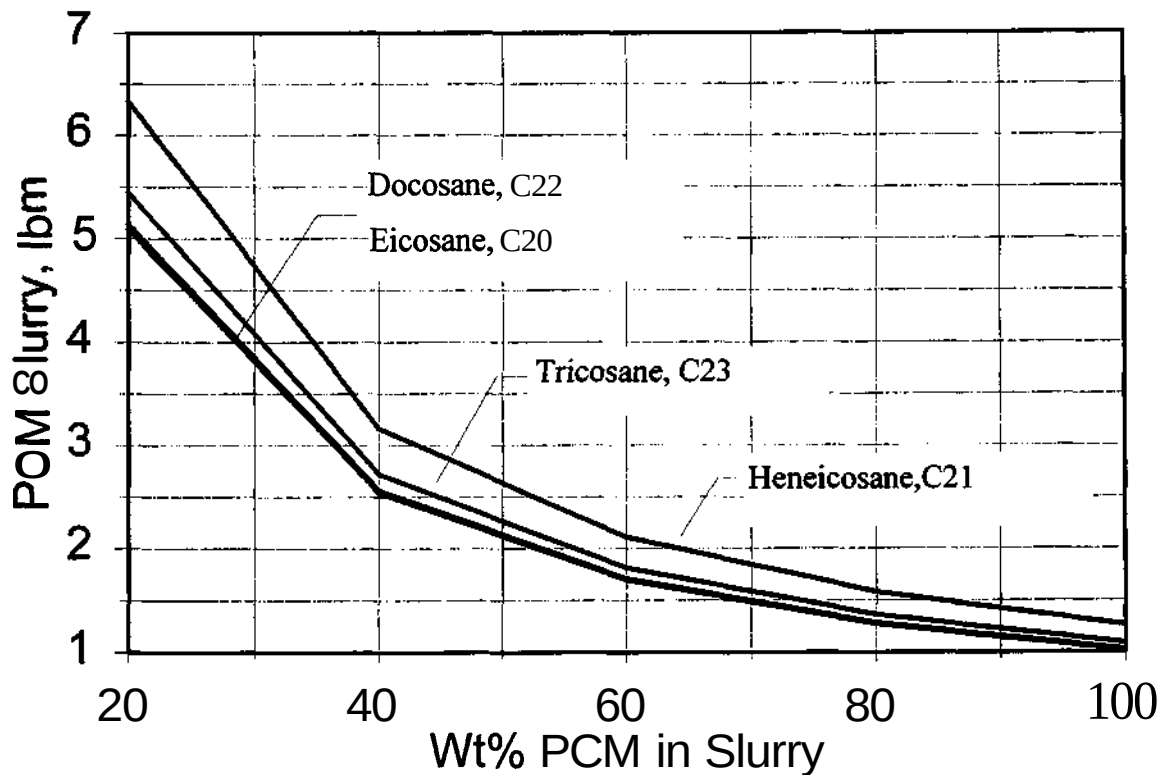


Figure 9. Required mass of a PCM slurry to absorb the lost heat from breathing gas during a 30 minute exposure.

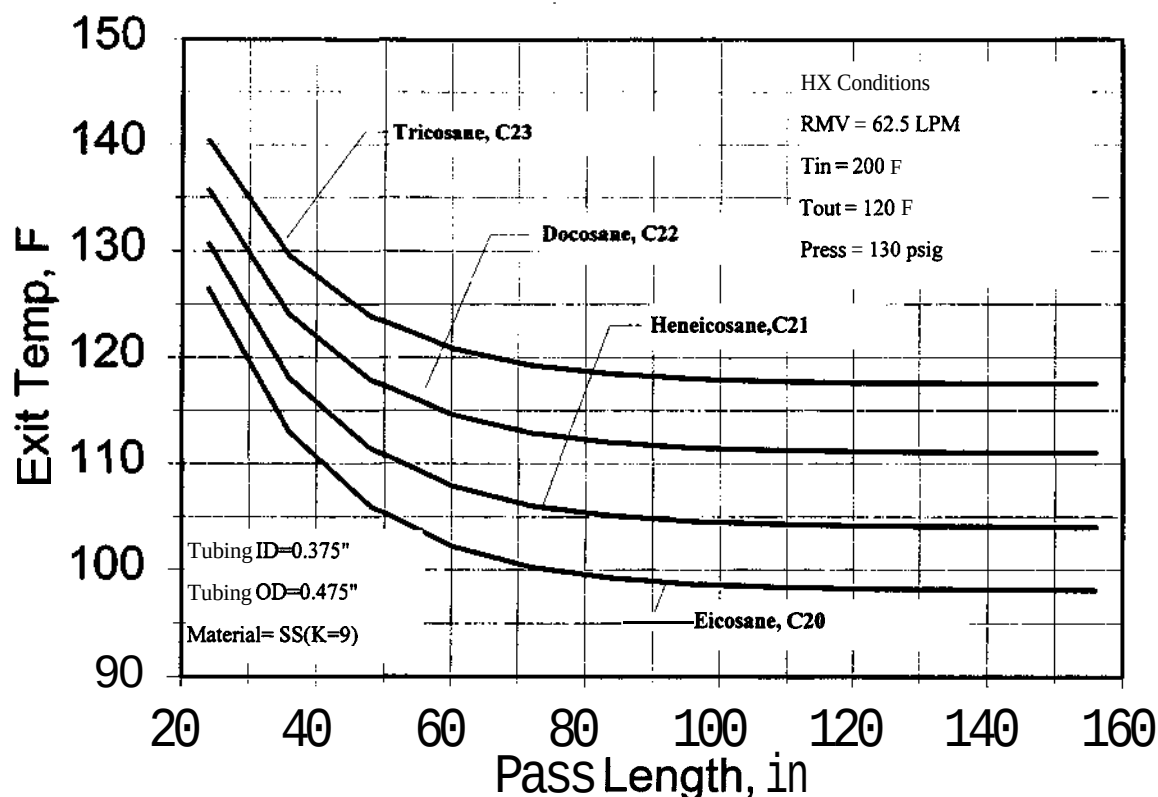


Figure 10. Exit temperature of breathing air from a single pass cooler with various pass lengths (based on heat transfer area).

values (this will account for the non-phase change material from which the encapsulating shell is constructed). In addition, as the weight percent of the MicroPCM™ in the slurry decreases the effective latent heat of fusion for the slurry will be reduced further. We will account for these reductions in latent heat storage capacity by multiplying h_{f} by the fraction of MicroPCM™ in the slurry ($\text{wt}\%$ PCM).

Using Equation (15), we can show that the required amount of a slurry consisting of 60% ($\text{wt}\%$ PCM = 0.6) docosane (h_{f} is 107 Btu/lbm) to cool the breathing gas for 30 minutes is approximately 1.7 pounds. Conversely, approximately 2.1 lbm of a slurry containing 60% heneicosane (h_{f} is 86 Btu/lbm) would be required. Figure 9 shows the required mass of different slurry mixtures of MicroPCM™.

Length Requirements for Single-Pass Designs

Two separate conditions must be satisfied to arrive at the required heat exchanger length for a single pass design. These conditions are:

- Sufficient surface area must exist between the inside of the breathing gas tube and the PCM slurry to insure an adequate heat transfer rate from the breathing gas, and
- Sufficient PCM material must be present in the breathing gas cooler to absorb all lost heat from the breathing gas.

The analysis presented above for multiple pass coolers can also be used to determine the required length of a single-pass heat exchanger (either the bullseye or pig nose designs) to give the desired exit temperature based on satisfying the first condition. Recall that this analysis assumed that there was sufficient PCM material present to absorb all lost heat from the air as the PCM goes from a solid to a liquid. Figure 10 summarizes the results of this analysis.

Observe from Figure 10 that, with the exception of Tricosane as a heat sink (melt temperature of

117.5°F), exit temperatures from the gas cooler below 120°F are feasible with tubing lengths less than 60 inches. When using *Eicosane* (melt temperature of 98.1°F) as the heat sink material, a single pass length of approximately 30 inches is indicated provided that a sufficient mass of *Eicosane* is present to absorb the lost heat from the breathing gas.

To satisfy the second condition stated above, we must find the amount of PCM that can be packaged in a single pass breathing gas cooler (either the bullseye or pig nose designs). This mass will depend on 1) the length of the gas cooler, 2) the outside diameter of the foam metal /PCM slurry matrix, 3) the density of the foam metal, and 4) the weight percent of the PCM material in the liquid slurry. The required mass of the PCM slurry to absorb all lost heat from the breathing gas was calculated previously using Equation (15) and shown in Figure 9 for a range of liquid slurries consisting of 20% to 100% PCM by weight. The required volume of PCM can be obtained by dividing this mass of PCM slurry by the density of the PCM slurry. A comparison of the required volume of PCM slurry with the available PCM volume packaged in the breathing gas cooler design allows us to determine the required design length.

Pig Nose Design

For the pig nose design, the available volume for the PCM slurry can be calculated as

$$V_{AVAIL} = (V_{OJ} - V_T) \cdot \left(1 - \frac{\%Foam}{100}\right) \quad (16)$$

where V_{OJ} is the volume of the outer jacket which contains the foam metal/PCM slurry, given as

$$V_{OJ} = \frac{\pi}{4} D^2 L_{PN}$$

V_T is the volume of the two tubes surrounded by the foam metal/PCM slurry, given as

$$V_T (2 \text{ tubes}) = \frac{\pi}{2} d^2 L_{PN}$$

and % Foam is the density of the foam metal, usually expressed as % of total volume (typically ranges from 2 to 20 %); where D in the above expressions is the inside diameter of the outer jacket, d is the outside diameter of the gas/water tubes, and L_{PN} is the length of the pig nose-designed breathing gas cooler. Equating the available volume of the PCM slurry

given in Equation (16) to the required PCM slurry volume V_{REQ} derived from Equation (15), and solving for the tubing length gives

$$L_{PN} = \frac{V_{REQ}}{\frac{\pi}{2} \left(\frac{D^2}{2} - d^2 \right) \left(1 - \frac{\% Foam}{100} \right)} \quad (17)$$

The resulting tube lengths for various MicroPCM materials, when the breathing gas cooler design is based on heat storage capacity, is shown in Figure 11. In each case these lengths exceed the pass lengths seen in Figure 10, lengths that were previously calculated based on the surface areas requirements to transfer sufficient heat to reduce the gas temperatures to 120°F. For instance, we see in Figure 10 that less than 45 inches of tubing is indicated to give the necessary heat transfer area when using *Docosane*. However, over 60 inches of tubing are called for to contain sufficient MicroPCM mass when packaging the 60% *Docosane* PCM slurry in an outer jacket with a 1.25-inch ID. Since both of these two design concerns must be met, we will have to select the longer tube length of 60.7 inches if the outer jacket diameter can not be increased.

An alternate approach would be to select an outer jacket diameter for the breathing gas cooler that would contain sufficient PCM mass in a length called for in Figure 10. Figure 12 indicates the required lengths for the breathing gas cooler to contain the necessary quantity of MicroPCM for a 30 minute exposure as a function of the outer jacket diameter. Note that by increasing the outer jacket ID from 1.25 inches to 1-7/16 inches the required 60% PCM slurry can be packaged around a tube length of approximately 45 inches, closely matching the length requirement called for in Figure 10.

Bullseye Design

A reduced length should be expected to package sufficient PCM for the bullseye design option since only a single tube is surrounded by the foam metal/PCM slurry matrix. A simple modification of Equation (17) to account for this single, centrally located tube gives the corresponding length for a bullseye-design breathing gas cooler, L_B , as

$$L_B = \frac{V_{REQ}}{\frac{\pi}{4} (D^2 - d^2) \left(1 - \frac{\% Foam}{100} \right)} \quad (18)$$

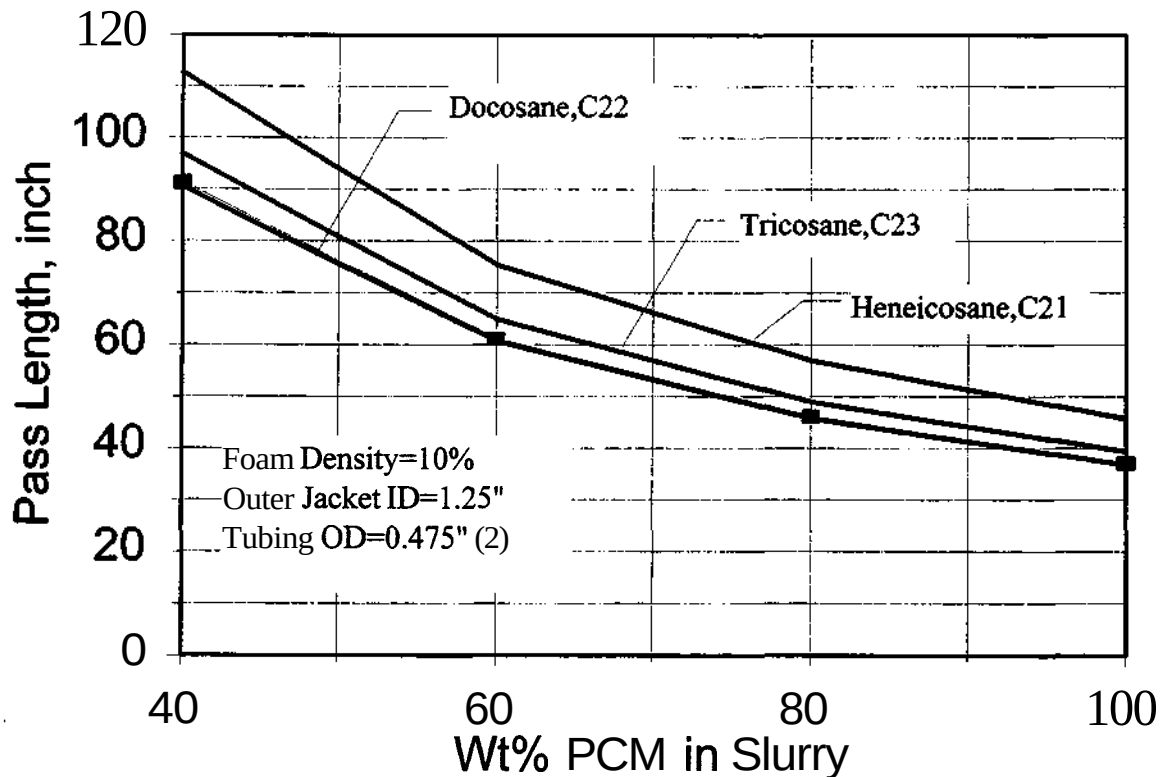


Figure 11. Required pass length for a breathing gas cooler based on PCM heat storage capacity for 30 minute exposure (Pig Nose Design).

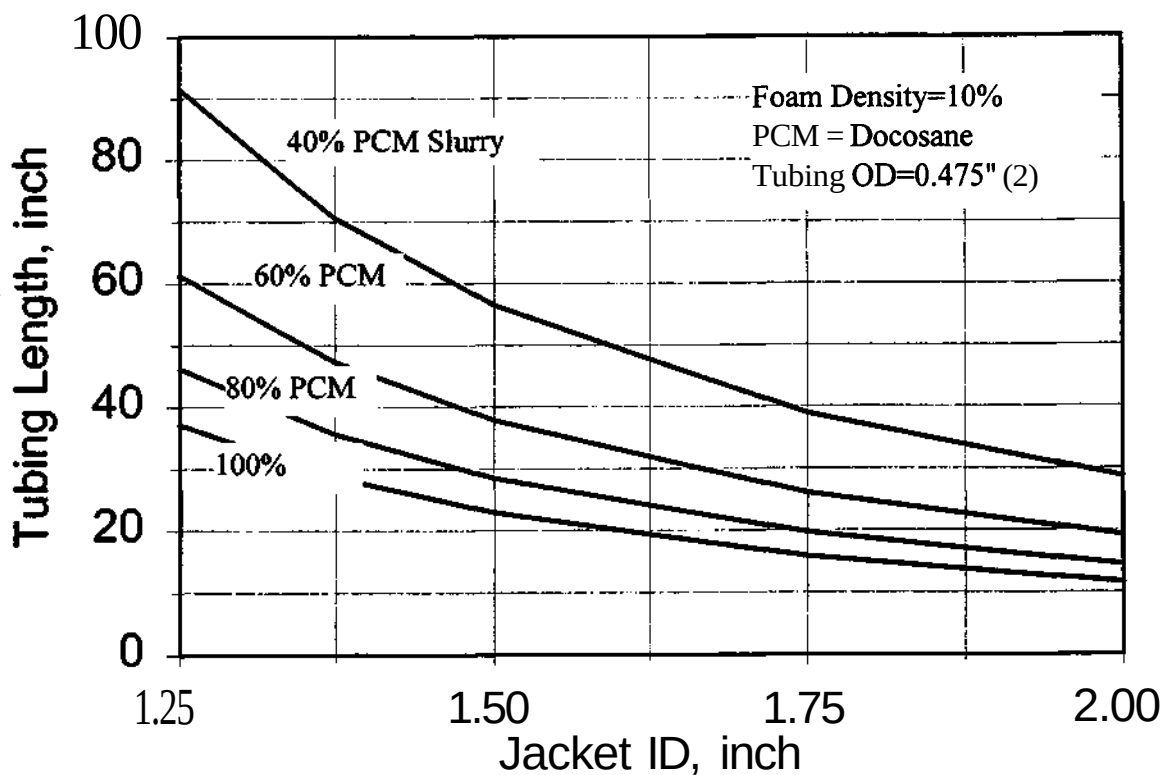


Figure 12. Effect of the outer jacket diameter on the required tube length for a single pass gas cooler based on heat storage capacity for a 30 minute exposure (Pig Nose Design).

Figure 13 shows the required lengths for the bullseye breathing gas cooler to contain the necessary quantity of PCM for a 30 minute exposure as a function of the outer jacket diameter.

$$\dot{q}_R = \dot{m}_w c_{p_w} \Delta T_w$$

$$\dot{q}_R = \frac{1 \frac{\text{gal}}{\text{min}}}{7.481 \frac{\text{gal}}{\text{ft}^3}} \times 62.4 \frac{\text{lb}}{\text{ft}^3} \times 1 \frac{\text{Btu}}{\text{lb} \cdot ^\circ\text{F}} \times 8^\circ\text{F} = 66.7 \frac{\text{Btu}}{\text{min}}$$

Whereas the pig nose design with a jacket ID of 1.25 inches requires in excess of 60 inches length to contain the 60% Docosane slurry for a 30 minute exposure, the bullseye design requires a length that is

approximately 10 inches less (50.5 inch) due to the single tube. By increasing the outer jacket ID from 1.25 inches to 1- 5/16 inches the required 60% PCM slurry can be packaged around a tube length of approximately 45 inches, again closely matching the length requirement called for in Figure 10. It should be noted however, that the additional annular ring contained in the bullseye design for the water channel will tend to offset any reductions in the assembly outside diameter when compared with the pig nose design. Thus, the primary advantage of the bullseye design would appear to be its increased contact area for maximizing heat transfer during the PCM regeneration phase. This improved regeneration capability would warrant the bullseye design to be selected over the pigs nose design provided that manufacturability of the bullseye design is not an issue.

PCM Regeneration

It is anticipated that the breathing gas cooler will be regenerated, i.e., the MicroPCM material will be re-solidified, using an available water source (i.e., ambient seawater or tapwater) with inlet temperatures to the cooler of up to 90°F. Exit water

temperatures from the breathing gas cooler can be approximated using a similar analysis that led up to Equation (8), with the exception that the properties of water will be substituted in place of the gas properties.

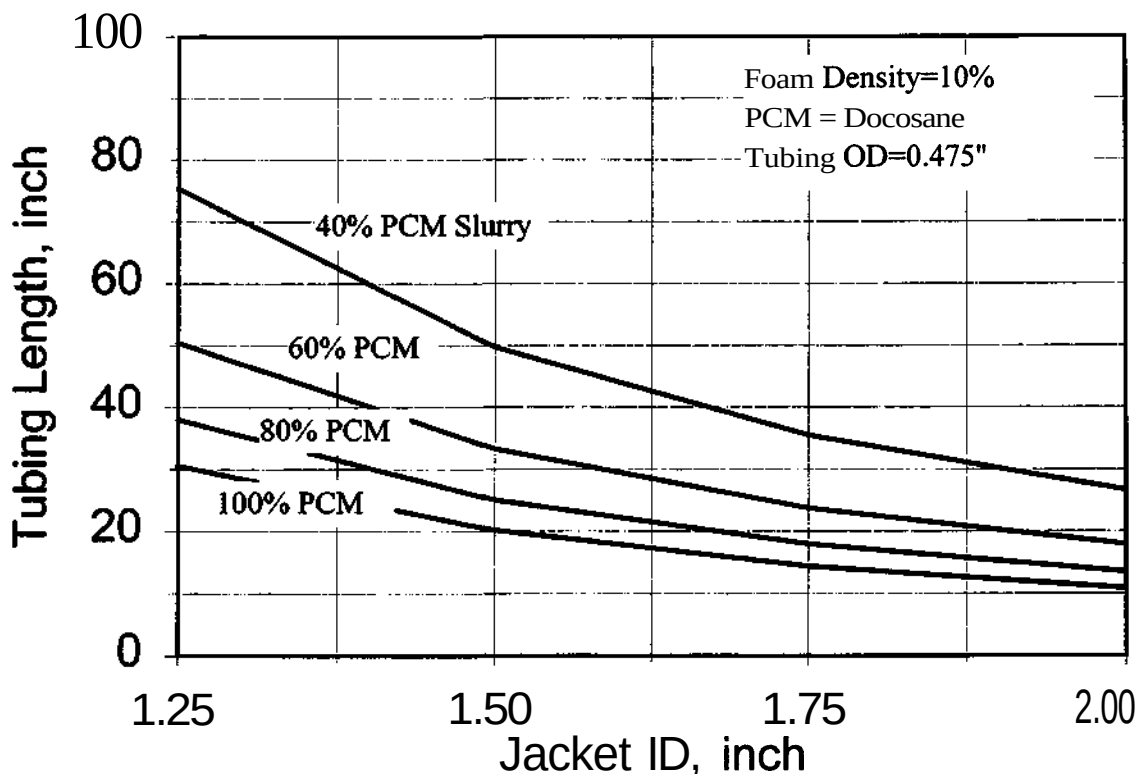


Figure 13. Effect of the outer jacket diameter on the required tube length for a single pass gas cooler based on heat storage capacity for a 30 minute exposure (Bullseye Design).

Figure 14 shows exit water temperatures from a pig nose cooler design with various metal foam/PCM slurry combinations surrounding the water tube.

Observe that exit temperatures will closely approach the PCM melting temperature within a tube length of 60 inches. Since the heat that can be absorbed by the water as it passes through the cooler will be proportional to the differential water temperature between inlet and exit conditions, the PCM's with the higher melting temperature will regenerate more rapidly.

However, if we assume that the PCM slurry consists of Eicosane, with the lowest melting temperature of 98.1°F, water entering at 90°F and exiting at 98°F, a flow rate of 1 GPM should be capable of removing heat from the breathing gas cooler at a rate of

$$\dot{q}_R = \dot{m}_w c_{p_w} \Delta T_w$$

$$\dot{q}_R = \frac{1 \frac{\text{gal}}{\text{min}}}{7.481 \frac{\text{gal}}{\text{ft}^3}} \times 62.4 \frac{\text{lb}}{\text{ft}^3} \times 1 \frac{\text{Btu}}{\text{lb} \cdot ^\circ\text{F}} \times 8^\circ\text{F} = 66.7 \frac{\text{Btu}}{\text{min}}$$

where $\dot{q}_R$ is the breathing gas cooler regeneration rate, $\dot{m}_w$ is the mass flow rate of water, c_{p_w} is the specific heat of water, and ΔT_w is the temperature rise as the water travels through the breathing gas cooler. In Figure 9, we saw that slightly over 1 pound (1.03 pounds) of 100% Eicosane, with a latent heat of fusion of 106 Btu per pound, would melt during the gas cooling phase. Based on the regeneration rate calculated above, we should be able to re-solidify this amount of PCM material in a matter of minutes with the 1 GPM flow of 90°F water through the pig nose cooler. Due to the presence of additional heat transfer area, regeneration with the bullseye design should progress even faster.

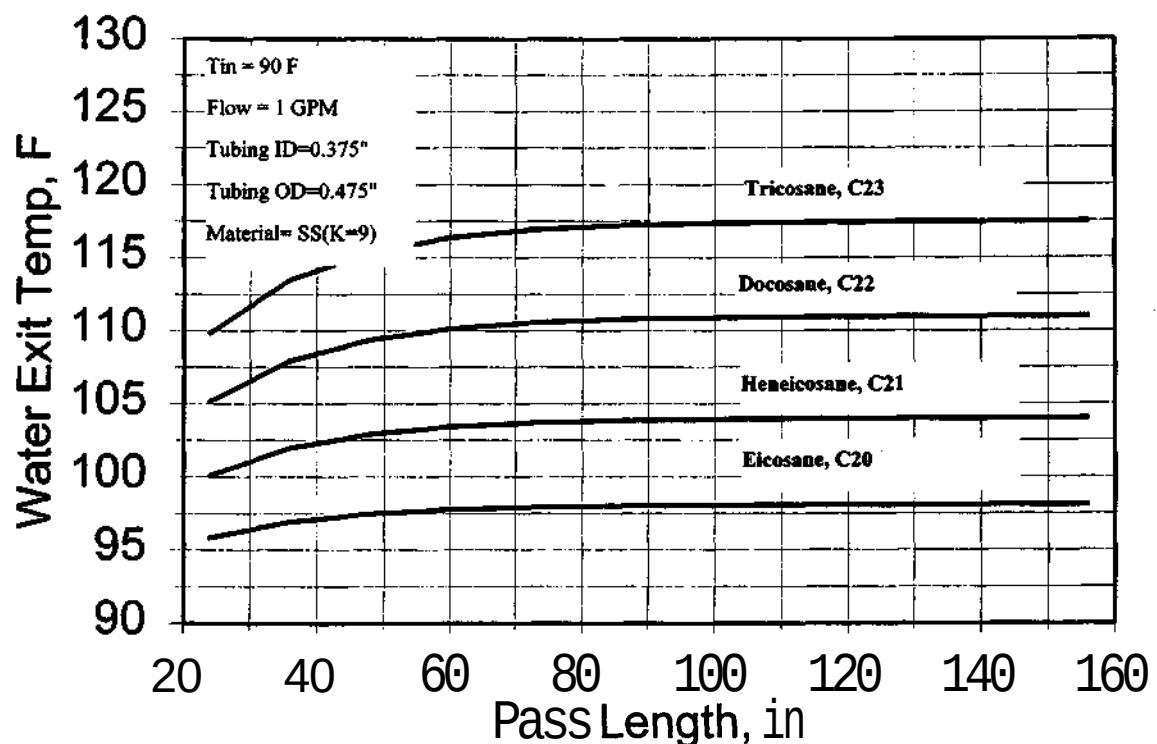


Figure 14. Water exit temperatures from a pig nose cooler design during PCM regeneration phase with a water flow of 1 GPM. (Assumes metal foam/PCM slurry temperature remains at the PCM melting temperature.)

Conclusions

The preceding analyses indicate that a 5 foot long, single pass heat exchanger, surrounded with a MicroPCM slurry in either a pig nose or bullseye configuration, is adequate to reduce a firefighter's breathing gas from a temperature of 200°F to 120°F with a respiratory minute volume of 62.5 LPM. Both design configurations have foamed aluminum, similar to Duocel manufactured by ERG, Inc in contact with the outside of the breathing gas tube to enhance the heat transfer from the breathing gas to the PCM slurry.

Eicosane, $C_{20}H_{42}$, with a melting temperature of 98.1°F appears to be the best candidate PCM for maximizing the heat transfer efficiency in the breathing gas cooler. However, docosane, $C_{22}H_{46}$, with a melting temperature of 111°F is also a viable candidate. Less than 2 pounds of a slurry consisting of either 60% Eicosane, or 60% Docosane, and 40% water was found to provide sufficient heat storage capacity to operate the breathing gas cooler for 30 minutes. Both of these PCM slurries can be packaged within a matrix of foamed metal in a heat exchanger length of approximately 60 inches with an outer diameter of the foamed metal/PCM slurry of

1.25 inches. (A 40% PCM/60% water mixture will require approximately 90 inches of heat exchanger length for the same mission.) By increasing the outer diameter from 1.25 inches to 1-5/16 inches, the required 60% PCM slurry can be packaged around a tube length of approximately 45 inches.

The presence of a foamed metal matrix surrounding the outside wall of the breathing gas tube was found to enhance the rate of heat transfer into the PCM slurry. The presence of foamed metal on the inside of the breathing tube could also maximize the transfer of heat from the breathing gas to the tube wall. However, since the lengths of both single pass cooler designs (pigs nose and bullseye) have been shown previously to be limited by PCM mass rather than heat transfer area, the application of foamed metal to the inside of the gas tube is not necessary.

Regeneration of the breathing gas cooler should not be a problem with either single pass configuration, even when using water temperatures as high as 90°F. Regeneration should be possible in a matter of minutes with a water flow rate of 1 GPM.

References

Hayes, C. Q., "Heat-Absorbing Heat Sink," U.S. Patent No. 4,446,916 (1984).

Lind, A. R., "Human Tolerance to Hot Climates," *Handbook of Physiology*, Oxford University Press, New York, pg 107 (1977).