Dissipating the heat inside mine refuge chambers

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Heat Dissipation Inside Mine Refuge Chambers

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1. Abstract

Following the SAGO Mine disaster in January 2006, Congress enacted the MINER Act that requires mine operators to install and maintain mine refuge chambers in all underground coal mines.

Refuge chambers serve as a temporary shelter for escaping miners or as a refuge in case escape from the mine is cut off.

The occupants of a refuge chamber create metabolic heat that must be dissipated so that the temperature inside the chamber will not exceed an apparent temperature or heat index of 95 HI.

This paper deducts a heat dissipation model from thermodynamic fundamentals and demonstrates that the calculated results from the mathematical model compare quite well to the results of a series of refuge chamber function tests conducted at NIOSH's Lake Lynn Experimental Mine in 2007.

The theoretical approach from the model can be used by chamber designers and mine operators to estimate the suitability of certain chambers for their purposes. Based on thermodynamics, the paper makes recommendations for a "cooler" chamber design and offers sensitivity analyses studying the temperature inside the chamber as a function of the mine temperature as well as the effects of reducing chamber occupancy in order to lower the temperature inside.

2. Introduction

The 2006 MINER Act as well as state mining laws in certain states require mine operators to provide refuge chambers on each working section and additionally, at certain strategic locations along the escapeways from the mine. Prior to 2006, refuge chambers had been used in coal mines only on an exception basis and general design and performance requirements did not exist.

Miners have always been trained that, during an emergency such as a fire, explosion or roof collapse, their first and best option is to escape from the mine. Barricading or awaiting rescue in a refuge chamber must be considered as a measure of last resort. If escape is cut off due to a toxic atmosphere, heat, water or debris, refuge chambers may offer a safe haven to miners until a rescue team can open up the passage and safely lead the trapped miners to the surface. Refuge chambers may also serve to provide a safe atmosphere in which escaping miners can rest and switch self-contained self rescuers without the danger of inhaling toxic gases.

Federal law requires that refuge chambers can provide at least 96 hours of breathing support, food and water to each occupant.

The temperature and humidity inside the chamber must be such that the apparent temperature or the so-called Steadman Heat Index (Steadman, 1979, see also MSHA 2011) remains below 95 HI (NIOSH 2007 and 30 CFR § 7.504 (b) (1)).

This means that the chamber needs to be designed such that it can dissipate the heat generated inside by human metabolism and other heat sources such as CO_2 scrubbing.

In this paper, the thermodynamics of heat transfer from the chamber to the mine air are examined and conditions analyzed that are required to maintain the desired maximum heat index.

3. Heat Production from Human Metabolism and Other Processes Inside the Chamber

3.1. Metabolic Heat Output

The human body produces metabolic heat from slowly combusting food in the stomach, intestines and muscles. The heat generated depends on the oxygen consumption and the lean body mass; body fat does not contribute to metabolic heat generation. The metabolic heat output generated by occupants of mine refuge chambers is significant and must be considered when designing such chambers.

Several standard values can be found for the heat produced by human metabolism: According to Williams (2009), a person weighing 75 kg consumes 0.24 liters of oxygen per minute at rest and generates 4.18 kJ/kg*hr of heat. At this rate of, this person would put out 87 W of heat. This heat output is confirmed in the NIOSH (1986) criteria document that lists 1.16 W/kg for a person at rest (equivalent to 87 W for a 75-kg man) as well as in Berne et al. (2004).

A series of tests with several refuge chambers were conducted at the NIOSH Lake Lynn Experimental Mine (LLEM) using a metabolic heat output of 117 W (400 BTU/hr) per person (Bauer and Kohler, 2009). This value goes back to a NASA standard and is also used as a design parameter in the State of West Virginia (Harris 2006).

The thermodynamic model used for the comparisons in this paper is using 117 W per person as the input rate since the chamber testing at the LLEM was conducted at this heat rate.

Additionally, the following should be noted: The metabolic heat output is considered for a person "at rest." If the occupants enter the chamber in an excited and perhaps exhausted state, heavy breathing and exercise would temporarily cause a significantly higher metabolic rate. Still, Williams (2009) expects that the "at rest" condition will set in after 20 to 30 minutes so the "at rest" metabolic heat rate is justified for the model calculations. According to NIOSH measurements, it took between 4 and 15 hours for the temperature inside the chamber to rise to a static thermal equilibrium. Therefore, the at-rest metabolic rate is used when evaluating a chamber at thermal equilibrium.

3.2. Other Heat Sources

In addition to human metabolic heat, chemical and electric devices (CO and CO_2 scrubbers, chemical toilets, self-heating meals, electric fans, lights etc.) may produce additional heat that needs to be transferred to the outside as well. With the exception of CO_2 scrubbing, additional heat producing components have not been considered but could be easily incorporated into the model.

At LLEM, CO_2 was injected into the test chambers at a rate of 0.51 l/minute and person.

In three of the chambers tested by NIOSH at the LLEM, CO_2 was scrubbed using anhydrous soda lime, $Ca(OH)_2$. The chemical reaction

$$Ca(OH)_2 + CO_2 \rightarrow CaCO_3 + H_2O + heat$$

generates 113 kJ per mol of CO_2 (using heats of reaction from Weast 1990) resulting in a heat flow of 42.8 W/person.

One chamber used lithium hydroxide (LiOH) curtains to scrub CO_2. This reaction,

$$2\ LiOH + CO_2 \rightarrow Li_2CO_3 + H_2O + heat$$

generates 138 kJ/mol of CO_2 resulting in 52.2 W/person heat flow.

The chemical heat of reaction is dependent on the state of the water produced in the reaction. If the water is in vapor form, some of the heat from the reaction will be consumed to evaporate the water, thereby reducing the net heat of reaction. This leads to some confusion when comparing literature values.

The lithium reaction is happening at a temperature where the water formed by the reaction is largely vaporized and therefore, in most cases, the heat-of-reaction calculation is considered assuming that all water is vaporized. This would reduce the heat generated to 93 kJ/mol in the LiOH reaction. This value corresponds with Westinghouse (1971) where the heat production is listed as 875 Btu equivalent to 90 kJ/mol.

White and Walker (1982) experimentally determined the heat production from a breathing apparatus using a soda lime scrubber from direct calorimetry and found it to be about 105 kJ/mol compared to the theoretical 113 kJ/mol for 100% liquid water. For comparison, if all water was evaporated, the heat of reaction would only be 68 kJ/mol.

In the refuge chamber application, a steady state is considered where water condenses along the chamber walls. In the condensation process the heat of vaporization is gained back. Therefore, considering the overall chamber, the heat produced by the scrubber reaction must consider water in its liquid state. It does not matter if some water is initially in vapor form because it will all condense out on the walls.

Similar to CO_2, scrubbing processes for CO must also be considered when evaluating refuge chambers. However, in the testing at LLEM no CO was used so this heat source has not been considered in the model used in this comparison.

3.3. Water Evaporation and Condensation

To simulate the metabolic release of water vapor from perspiration and exhalation, piezoelectric humidifiers were operated in the test chambers at the LLEM. It was assumed that each chamber occupant would evaporate 1.5 kg of water per day. The humidifiers generate a fine mist of water droplets dispersed in the air where they evaporate, taking away heat from the chamber air. This heat is significant: The vaporization enthalpy for water is 40.7 kJ/mol. Evaporating 1.5 kg of water per day thus requires 39.2 W of heat. However, as the water condenses on the chamber walls (which was observed with all chambers during the steady-state phase of the tests), this same amount of latent heat is gained back, creating a net effect of zero.

The condensed water will pool on the chamber floor. Assuming a floor space of 15 sq. ft per person (MSHA required minimum), 1.5 liters (per person and day) will amount to just over 1 mm per day or about 3/16th of an inch over 4 days unless drainage is provided.

In the steady state, there is no fundamental thermodynamic difference between the test setup using light bulbs and humidifiers and actual chamber use with humans inside:

Human metabolic heat (117 W) is composed of sensible and latent heat. Sensible heat (about 2/3 or 78 W) is transferred to the chamber air resulting in a temperature increase while latent heat (39 W) is consumed to evaporate water (exhalation or perspiration).

In comparison, the light bulbs generate 117 W of sensible heat, of which 39 W cause the water mist to evaporate, leaving 78 W of sensible heat that leads to a temperature increase.

In both cases, the latent heat (39 W) is converted back to sensible heat as the water condenses on the walls, leaving the total net human heat input to the chamber at 117 W.

3.4. Heat Loss from adding Oxygen and Make-up Air

Oxygen is piped into the chambers to make up for the human metabolic oxygen consumption. Also, make-up air is supplied to maintain a slightly positive pressure inside the chamber since a slight amount of leakage is unavoidable. It should be assumed that both oxygen and air are supplied from pressurized cylinders at mine temperature. Leakage air (including water content) is leaving the chamber at chamber temperature.

In the LLEM tests, 0.62 l/min O_2 was piped into the chamber. The 0.51 l/min of CO_2 that was piped in would have been consumed in the scrubber reaction so it would not displace chamber air.

0.62 l/min and person O_2 displace the equivalent volume of air. At 25 C, the air density is 1.16 kg/m^3 thus $1.20 * 10^{-5}$ kg/s air are lost. The enthalpy of the air at 25 °C and 100% relative humidity is 75 kJ/kg. Therefore the heat lost from adding oxygen is about 0.9 W per person, which is considered insignificant.

Heating the oxygen from mine temperature to chamber temperature consumes heat as well especially since expanding the compressed oxygen from the tank will further cool the gas before it enters the chamber according to the Joule-Thomson (JT) effect. This heat loss was determined at 0.6 W per person and considered insignificant as well.

If make-up air is provided, additional heat loss occurs since warm air is displaced from the chamber. Adding dry make-up air also contributes slightly to lowering the humidity inside the chamber. For this comparison, the impact of make-up air was neglected as no information was available about leakage and the amount of make-up air used in the LLEM tests.

4. Fundamentals of Heat Transfer

Heat flow from the chamber to the ambient air in the mine can be described in three separate thermodynamic processes,

1. Heat transfer from radiation
2. Heat transfer from convection
    a. Convective heat transfer from the air inside the chamber to the chamber walls, roof and floor
    b. Convective heat transfer from the outside walls and roof of the chamber to the mine air
3. Heat transfer from conduction
    a. Conductive heat transfer through the roof and walls of the chamber
    b. Conductive heat transfer into the mine floor

The following sections present detailed calculations for these heat transfer mechanisms.

4.1. Heat Transfer from Radiation

Radiation heat transfer follows the Stefan-Boltzmann law,

$$q_{rad} = \sigma\, \varepsilon\, A\, (T_1^4 - T_2^4) \tag{1}$$

where

q_{rad} is the radiation heat flow (heat energy per unit time in W),

σ is the Stefan-Boltzmann constant, $\sigma = 5.67 * 10^{-8}$ W/(m^2*K^4)

ε is the radiation emissivity. For a black body, ε=1, and for polished silver, ε=0.02.

A is the area of the radiating surface (m^2),

T_1 is the absolute temperature of the radiating body (K) and

T_2 is the absolute temperature of the environment, in this case, the mine.

The value for ε depends on the texture of the radiating surface and on its geometric arrangement with respect to the surface receiving (and possibly, partially reflecting back) the radiated heat. The geometric effect is quite complex when considering the rounded surfaces of the human body in relation to the chamber walls and roof. Therefore, for the purposes of this application, it will be assumed that heat is radiated from the chamber occupants' bodies to the chamber walls and the same heat flow will be radiated from the outside chamber surface to heat the mine air and walls (symmetric flow). It is further assumed that none of the radiated heat is reflected back from the mine walls, i.e., the mine is considered a black body radiator.

Literature values show that ε is about 0.95 for painted surfaces (independent of their color) and about the same for plasticized fabrics (Holman 1972, Kneer 2009).

It should be noted that radiation heat transfer will occur from the chamber floor to the mine floor in case of rigid chambers that have an air space under the floor. In practice, however, the heat transferred is negligible since the temperature difference between the underside of the chamber and the mine floor calculated in the model is quite small. For inflatable chambers with a tent floor there is no radiation into the mine floor.

4.2. Convective Heat Transfer Inside and Outside the Chamber

Heat dissipates from the air inside the chamber to its walls in a process called convective heat transfer. Convective heat transfer happens in two separate instances: First between the occupants and the inside of the chamber walls, roof and floor, and second, between the outside of the chamber walls and roof (not floor) and the mine air.

If the occupants are at rest and not moving about the chamber (conservative assumption), there is no forced air movement inside the chamber and the convective heat transfer occurs through natural or "free" convection. This means the heat transfer process itself will create the heat that causes the air to move around inside the chamber.

Outside the chamber, the same convective heat transfer happens along the exposed surfaces of the chamber. Since refuge chambers are generally placed in cross cuts in order to keep them out of the path of explosive airblasts, it should be conservatively assumed that there is no ventilation or forced air movement outside the chamber. Mine ventilation is often compromised following a fire, explosion or fan outage which might also result in still air around the chamber. Therefore, it must be assumed that the convective heat transfer outside the chamber is also happening from natural convection only and not from forced convection. It is, however assumed that there will be sufficient space separating the chamber walls and roof from the mine ribs and roof to allow some air to circulate from natural convection in order to facilitate the heat transfer. A transfer of heat from the mine air to the mine walls is not considered – it will be assumed that the mine wall area is sufficiently large around the chamber so that this transfer happens without creating a significant temperature differential. The mine is considered a quasi-infinite heat sink.

Although the physical process of natural convection is quite complex, it can be evaluated in a simplified manner where the heat transfer depends on the chamber surface area A, and the differential ΔT_1 in temperatures between the air inside the chamber and the chamber wall.

The fundamental equation for convection ("Newton's law of cooling") can be expressed as

$$q_{conv} = h\ A\ \Delta T_1 \tag{2}$$

where

q_{conv} is the convection heat flow (energy per unit time in W),

h is the convective heat transfer coefficient (W/m^2*K)

ΔT_1 is the temperature differential between the inside of the chamber and its wall (K)

For air in free convective flow (no forced air movement), h ranges between 3 and 20 W/ m^2*K (Kneer, 2009). Since this parameter has a significant impact on the result, a more detailed determination of h is required. The textbook calculations differ for vertical and horizontal surfaces. For the horizontal surfaces, it also makes a difference if the heat transfer happens to the bottom of the roof, from the top of the roof or to the top of the floor.

The first step of the calculation is to determine Rayleigh's number Ra (sometimes also referred to as the product of Grashof's number, Gr and Prandtl's number, Pr). All three numbers are dimensionless.

$$Ra = Gr * Pr = g * \Delta T * L^3 / (T_{film} * \alpha * v) \quad (3)$$

where

g is the gravity acceleration constant (9.81 m/s^2)

L is the "characteristic dimension" of the surface in m (for vertical surfaces, the height, for horizontal, rectangular surfaces, their mean dimension, Holman 1972)

α is the thermal diffusivity of air; $\alpha = 2.25 * 10^{-5}$ m^2/s

v is the kinematic viscosity of air; $v = 1.59 * 10^{-5}$ m^2/s

T_{film} is the average absolute temperature of the convective flow near the surface (K), defined as

$$T_{film} = 0.5 * (T_s + T_{inf}) \quad (4)$$

T_s is the surface temperature (K), and

T_{inf} is the temperature at "infinity", i.e. the temperature at a distance away from the surface where it is no longer affected by near-surface phenomena.

If Rayleigh's number is greater than 10^9, the convective air flow is considered turbulent. This was the case for in all refuge chamber calculations.

From Rayleigh's number, under turbulent conditions, Nusselt's number Nu can be calculated as follows:

$$Nu = C * Ra^m \quad (5)$$

where C and m are constants determined from the Table 1 below.

Finally, the convective heat transfer coefficient is determined from Nusselt's number:

$$h = Nu * \kappa_{air} / L \quad (6)$$

where κ_{air} is the thermal conductivity coefficient for air, $\kappa_{air} = 0.0263$ W/(m*K)

The chamber surface area *A* available for heat transfer (roof, floor and walls) is dependent on the chamber dimensions. It should be noted that for m=1/3 the value of h is independent of L since this dimension cancels out of the equation.

Table 1: Coefficients to calculate Nusselt's number

Application	C	m	Source
Horizontal plate, heated, face up (roof of chamber,	0.13	1/3	Kneer 2009

outside)			
Horizontal plate, cooled, face down (roof of chamber, inside)	0.15	1/3	Incropera 2006
Horizontal plate, cooled, face up (floor of chamber, inside)	0.27	1/4	Incropera 2006

Other factors, for example the presence of cooling fins on the chamber walls, are not being considered at this time. Also, for simplicity it is assumed that the heat is equally distributed inside the chamber, i.e., the air temperature is the same throughout the inside of the chamber.

Convective heat transfer happens from the inside of the chamber to the walls, roof and floor. Outside the chamber, convection happens at the walls and on the roof. No convection is considered at the floor since for inflatable chambers, the tent surface is in direct contact with the mine floor. Rigid chambers may have an air space underneath the color but since the chamber floor is warmer than the mine floor, convection does not happen. Rather, the air space under the chamber acts as an insulation layer.

Calculated values for h for walls and roof in the numeric models ranged between 2.6 and 3.1 W/(m^2*K), i.e., at the lower end of the range of literature values cited above. It should be stressed here that increased turbulence caused by forced airflow both inside and outside the chamber could significantly improve the convective heat transfer, leading to cooler temperatures inside the chamber. Possible sources for forced air movement would be mine ventilation outside and leakage make-up air inside the chamber.

4.3. Conductive Heat Transfer through the Chamber Walls and Floor

Heat must be moved through the chamber wall material by a process called conductive heat transfer. This transfer depends on the following linear relationship.

$$q_{cond} = \kappa * A * \Delta T_2 / s \qquad (7)$$

where

q_{cond} is the conductive heat transferred per unit time (W),

κ is the thermal conductivity coefficient (W/(m*K)), for sample values see Table 2.

Table 2: Constants used for κ (Beardmore 2009):

Material	κ [W / (m*K)]	Comment
Steel	50	Values vary but variations are insignificant for results
PVC Plastic	0.19	Other plastics have similar values, uncritical
Air	0.0263	

ΔT_2 is the temperature difference between the inside and the outside of the chamber wall or the mine floor, respectively, and

s is the wall thickness (m).

Heat transfer through the chamber floor deserves special consideration. In order to permit a steady-state calculation, the model makes the assumption that the mine floor provides an “infinite” heat sink able to dissipate the heat induced from the chamber without warming significantly. This assumption is incorrect but simplifies the calculation considerably. In reality, the floor will heat up slightly. However, the impact on the chamber temperature is considered insignificant: The numeric model shows that the heat transferred into the floor is less than 10% of the total convective heat flow. A sensitivity analysis shows that cutting the heat transfer through the floor in half (i.e. simulating that the temperature of the floor has increased by half of the difference between inside temperature and mine temperature) would only increase the inside temperature by 0.2 °C. Eliminating heat transfer through the floor altogether would result in an increase of 0.3 °C over the base case. Therefore this simplifying assumption appears justifiable.

Inflatable chambers generally have a tent floor that is in direct contact with the mine floor. Therefore, heat flows in conductive transfer through the chamber floor and into the mine floor. Since the conductive heat flow capacity is much larger compared to the convective heat flow, the temperature difference between the chamber floor (inside) and the mine floor is quite small (less than 0.1 °C).

It should be noted that, at the LLEM test site, the chambers were erected on a level concrete floor. In mines, chambers may sit on uneven rock or gravel surfaces where the chamber floor may not be in good contact with the mine floor and air may be trapped under the chamber, acting as a heat insulator.

For rigid chambers, there are two effects to be considered. Rigid chambers have an air space between the support rails under the floor that acts as an insulation layer. However, if the chambers are constructed of steel, aluminum or another good heat conducting material, a few support rails in direct contact with the mine floor are sufficient to conduct significant amounts of heat through the floor. Model calculations show that the difference between treating the floor as solid steel or as a PVC layer does not make a significant difference for the temperature inside the chamber since the floor temperature difference even for the inflatable chambers is so small.

For simplification and in consideration of the slight impact that heat transfer through the floor has on the temperature inside the chamber (see sensitivity analysis above), it was assumed that the heat conducting capability into the mine floor is the same for rigid steel and inflatable chambers.

5. Heat Transfer Calculation Model

As indicated by equations (2) and (7) above, conductive and convective heat transfers are linear dependent on the temperature while radiated heat (eq. 1) is proportional to the fourth power of the absolute temperature. Therefore a combined, iterative mathematical model was developed to determine the temperature inside the chamber for a given mine temperature.

The following summarizes the simplifying assumptions discussed above:

- The temperature of the mine air is uniform and equal to that of the mine rock. This assumption may not hold true in all cases, especially in geothermally warmer mine environments. However, if the air is still around the chamber, it will eventually assume mine rock temperature. The tests at the LLEM showed that the mine air remained almost constant during the 96 hours of testing time.
- The mine floor is treated as an infinite heat sink, i.e., it does not heat up from the chamber heat. The proportion of heat going into the floor is relatively small compared to that transferred from walls and roof so this assumption is not considered critical to the outcome of the calculation.
- The thermal conditions are evaluated at steady-state, equilibrium temperatures, not during transient stages.
- The temperature inside the chamber may have a vertical gradient, i.e., the air could be warmer closer to the roof. The model uses a simple linear gradient. It turns out that the gradient does not have a significant impact on the average temperature inside so more detailed vertical gradient modeling is not considered necessary.
- The temperatures are assumed to be constant across the surface areas of the chamber walls.
- The calculation only considers the metabolic heat produced by the chamber occupants and the heat generated by the CO_2 scrubber. Additional heat producers such as chemical beds to scrub CO and all other heat generating components (lights, fans etc.) should be taken into consideration in realistic applications.
- Thermal effects of adding oxygen and make-up air to compensate for slight leakage are considered insignificant and ignored in the model.
- The chambers were treated as smooth, rectangular boxes without cooling fins, external or internal frames or gussets etc.
- Some rigid chambers have structures inside including benches, waste water tanks built into the floor, etc. which may constitute thermal barriers and should be considered for a more detailed analysis. However, this model does not take such structures into account.

To model the convective and conductive heat transfer, a system of 17 linear equations (eq. 8 through 24) with 17 unknowns (temperatures T_i and heat flows q_i) can be established:

The indices are used as follows: w=wall, r=roof, rinf=roof at “infinity”, f=floor, finf=floor at “infinity”, i=inside (average), o=outside and c=conductive.

The sketch in Figure 1 illustrates the various temperatures and heat flows. The left half of the figure shows a rigid chamber, the right side, an inflatable type. The thickness of the walls has been exaggerated to better show the heat flow vectors. It was assumed to be 2 mm for all chambers tested – the exact value does not have a significant impact on the results.

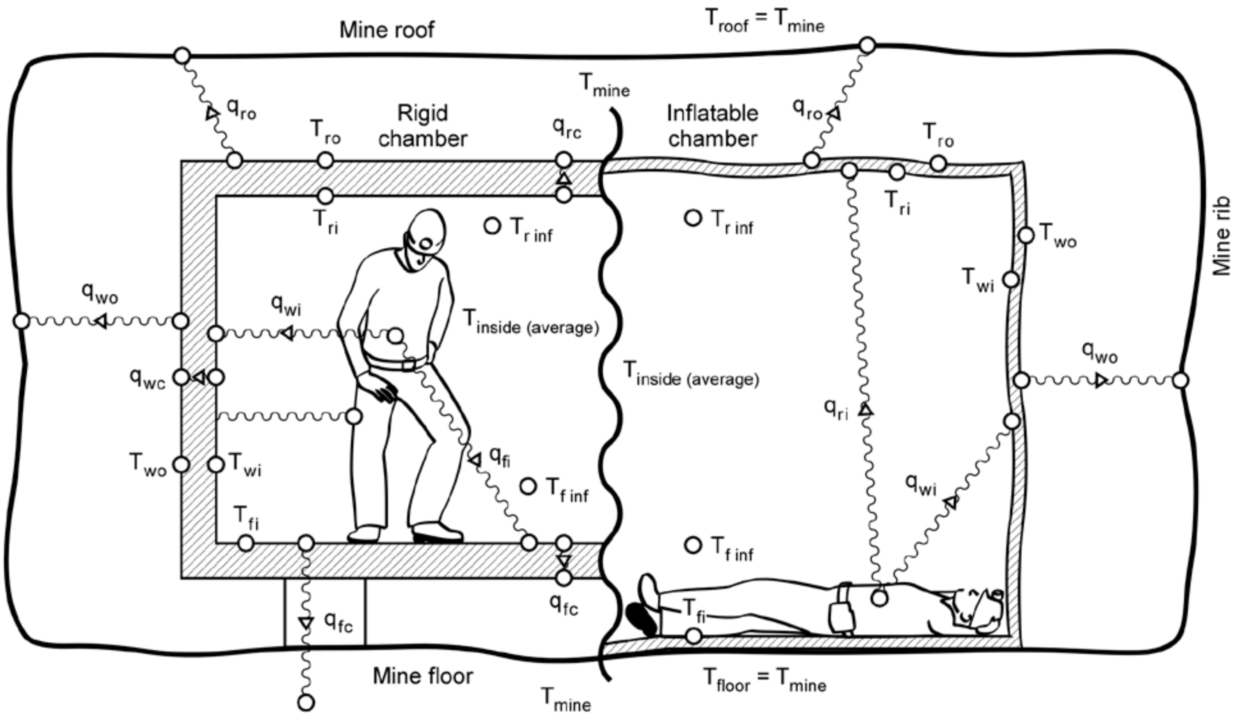

Figure 1: Cross section of a refuge chamber showing nomenclature (not to scale). The left half depicts a rigid chamber, the right, an inflatable chamber.

The first law of thermodynamics requires that

$$q_{total} = q_w + q_r + q_f = constant \tag{8}$$

Eight linear equations are set up for the convective and conductive heat transfer components, as follows:

$$q_{wi} = C_{wi}\ (T_{inside}\text{-}T_{wi}) \tag{9}$$

$$q_{wc} = C_{wc}\ (T_{wi}\text{-}T_{wo}) \tag{10}$$

$$q_{wo} = C_{wo}\ (T_{wo}\text{-}T_{mine}) \tag{11}$$

$$q_{ri} = C_{ri}\ (T_{inside}\text{-}T_{ri}) \tag{12}$$

$$q_{rc} = C_{rc}\ (T_{ri}\text{-}T_{ro}) \tag{13}$$

$$q_{ro} = C_{ro}\ (T_{ro}\text{-}T_{mine}) \tag{14}$$

$$q_{fi} = C_{fi}\ (T_{inside}\text{-}T_{fi}) \tag{15}$$

$$q_{fc} = C_{fc}\ (T_{fi}\text{-}T_{mine}) \tag{16}$$

Where $C_{mn} = k_n * A_n$ for all m = {w, r, f} and n = {i, o},

and $C_{mc} = \kappa * A_n /s$ for all m = {w, r, f}; and

s is the wall thickness (m) or the thickness of the air space underneath the steel chamber, respectively.

Two additional temperature equations are introduced to account for a vertical temperature gradient inside the chamber (i.e., near the roof, temperatures may be hotter than near the floor)

$$T_{rinf} = T_{inside}\ (1 + C_{sp}) – C_{sp} * T_{ri} \tag{17}$$

and

$$T_{finf} = T_{inside}\ (1 – C_{sp}) + C_{sp} * T_{ri} \tag{18}$$

where C_{sp} is a “spread” factor between 0 and 1 that describes the gradient. 0.5 was chosen as a starting value. Sensitivity calculations show that C_{sp} does not have a significant influence on the average inside temperature.

The first law of thermodynamics yields five additional equations

$$q_{wi} = q_{wc} \text{ and } q_{wc} = q_{wo} \tag{19-20}$$

$$q_{ri} = q_{rc} \text{ and } q_{rc} = q_{ro} \tag{21-22}$$

$$q_{fi} = q_{fc} \tag{23}$$

and, lastly, the mine temperature is a known constant:

$$T_{mine} = constant \tag{24}$$

All constants C_{ik} can be determined from calculated and material parameters. The input value for q_{conv} is the total heat transferred by convection and conduction, i.e., the total heat produced by the occupants of the chamber minus the portion that is transferred by radiation.

An iterative correction must be applied to correct the value for ΔT in eq. (3) when calculating Rayleigh’s number. Technically, this calculation could be incorporated directly in the matrix but it would unnecessarily complicate the mathematical relationships since the exponent m in eq. (5) is non-linear. Since the convection calculations are symmetric, one can assume that ΔT inside the chamber is equal to that outside the chamber. Since the temperature differential across the walls, roof and floor is quite small (less than 0.2 K), it can be approximated as

$$\Delta T = 0.5 *(T_{mine} – T_{inside}) \tag{23}$$

The matrix is solved using an approximate guess for ΔT and then the value is then iteratively adjusted so that eq. (23) is satisfied.

Since the radiated heat q_{rad} depends on the 4^{th} power of the outside wall temperature, the iteratively determined ΔT is also used to calculate the wall temperatures required to calculate the radiated heat (cf. eq. (1)).

6. Comparison of the Model to In-mine Tests

6.1. Modeling Results

The model calculations yield reasonably close results when compared to test data from four chambers tested at the LLEM in 2007. Table 3 summarizes these results:

Table 3: Chamber Test Data

Chamber Model	Hours to	Average	Average	Model

	steady temp.	T_{mine}, °C	T_{inside}, °C	T_{inside}, °C
Model A:	5	16.7	25.3	30.0
Model B:	4	15.8	22.6	24.9
Model C:	6	17.4	30.7	31.0
Model D:	15	16.6	32.1	35.3

The comparisons show that, in three of the four cases, the model estimates a slightly higher temperature than what was measured. Unfortunately, the tests at LLEM were primarily focused on the technical function of the chamber and not on the heat issues. Therefore, temperatures were only recorded at one single point inside the chamber and detailed readings at the walls, roof, floor etc. are not available. Also, no measurements of leakage air were recorded. If there had been significant leakage, it would explain a lower inside temperature both from the cooling (loss of warm air through leakage and replacement with air at mine temperature) and from increased turbulence inside the chamber which would improve the conductive heat transfer.

It is therefore suggested to conduct an additional chamber test to specifically investigate the heating phenomenon and to validate the model with more detailed temperature and leakage air measurements.

6.2. Apparent Temperature or Heat Index

The apparent temperature, also referred to as the heat index (HI), takes into account the effects of heat and humidity on the human body and is a measure of relative discomfort that a human experiences. It is based on research by Steadman (1979) who conducted physiological studies of evaporative skin cooling. A quadratic curve-fit function allows the calculation of the heat index HI

$$HI = -42.379 + 2.04901523T + 10.14333127R - 0.22475541TR - 6.83783\times10^{-3}T^2$$
$$- 5.481717x10\text{-}2R2 + 1.22874x10\text{-}3T2R + 8.5282x10\text{-}4TR2 - 1.99x10\text{-}6T2R2 \qquad (24)$$

where

T is the ambient dry bulb temperature in degrees Fahrenheit (inside the chamber) and

R is the relative humidity (%). Note that the excessive number of significant digits in eq. (24) were those provided in the source document.

According to Steadman's equation, the apparent temperature of 95 HI will be reached with a dry bulb temperature of 82 °F at 100% relative humidity and with a dry bulb temperature of 83 °F at 90% relative humidity. During the tests at LLEM, all chambers showed condensation on the inside walls, indicating that the humidity was near or at saturation (close to 100%).

6.3. Sensitivity Analysis

The graphs in Figure 2 show the temperatures inside the chambers as a function of mine temperatures and the calculated heat index. Since the heat index is defined based on temperature measurements in Fahrenheit, all temperatures were converted to Fahrenheit.

Figure 2 shows that the temperature inside is an almost linear function of mine temperature. The Heat Index is a fourth-order polynomial that is only defined over a certain temperature range. The heat index increases much steeper than the inside temperature. All chambers tested at the LLEM reached a heat index of more than 95 at mine temperatures below 65 °F. If the chamber cannot be cooled by other means, these chambers would need to be de-rated in their capacity, allowing fewer persons inside than the full capacity.

Figure 3 shows the required de-rating according to the model, calculated for a relative humidity of 90%. The de-rating function is also quasi-linear – the waviness of the graphs results from the fact that a heat index of exactly 95 cannot always be achieved with an integer number of persons occupying the chamber. The maximum occupancy rating goes to zero at 83 °F mine temperature since this will result in a 95 HI at 90% relative humidity.

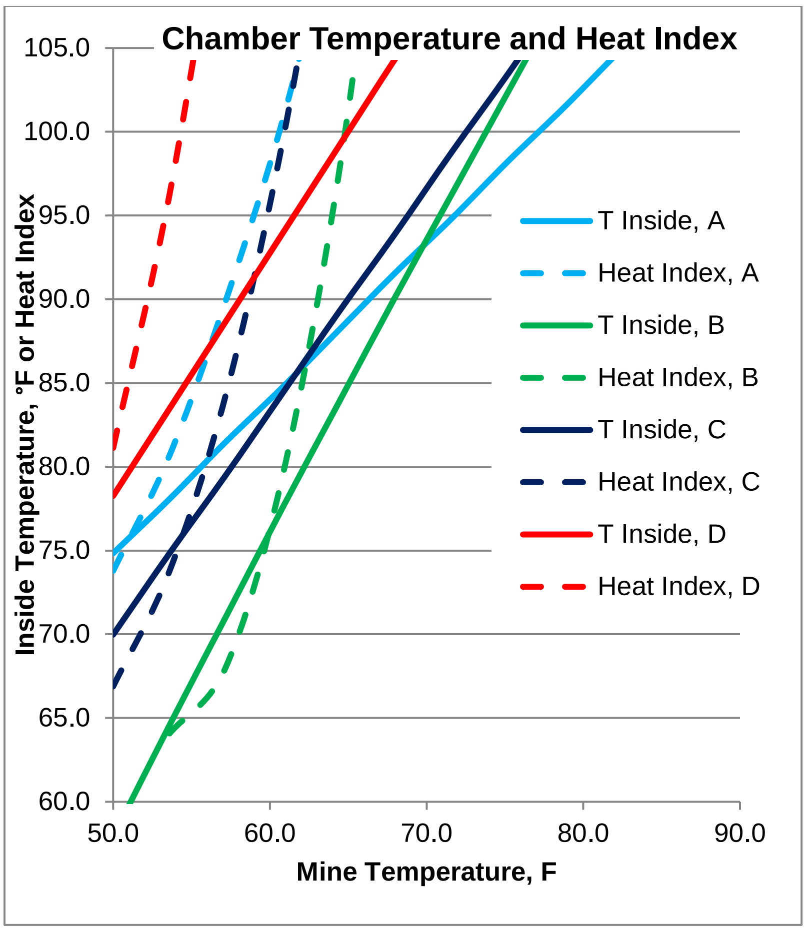

Figure 2: Chamber inside temperature and heat index as a function of mine temperature, for chamber models A, B, C and D.

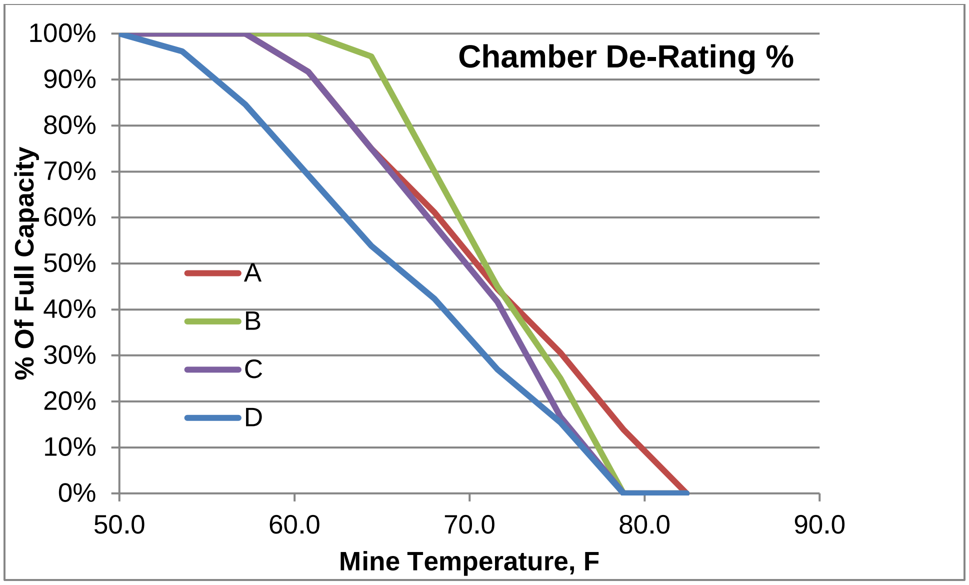

Figure 3: De-rating in % of full capacity for refuge chamber models A, B, C and D as a function of mine temperature

Depending on the chamber model, de-rating becomes necessary at mine temperatures of about 60 °F. Above 70 °F, the required de-rating is significant for all the chamber models tested.

7. Discussion and Conclusions

This modeling effort, in combination with the in-mine tests yielded the following conclusions:

- The complexity of the thermodynamic model requires several simplifying assumptions. A chamber designer should conduct temperature measurements using a full-scale prototype in a mine-like setting in order to determine the actual thermal balance.
- The models show that radiation cooling is a significant factor, contributing about 60% of the total heat transfer. In order to generate a radiation emissivity close to 1, the walls should not be made of polished, shiny surfaces but be painted in flat color or made from dark fabric material. If required, reflective markings should be applied in narrow strips in areas where heat transfer is already poor (structural ribs, corners etc.).
- Convective heat transfer from roof or walls can be improved by installing cooling fins on both inside and outside surfaces. The model shows that doubling the available surface area for convection along the wall and roof surfaces can reduce the inside temperature by 5 to 10 °C (9 to 18 °F).
- The numerical model can also be used to determine the requirements for mechanical cooling in case the mine temperatures are too high to permit sufficient passive heat transfer.

- A different thermodynamic modeling approach could be made using a computational fluid dynamics (CFD) model. However, CFD modeling would also require a number of simplifying assumptions.

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