On Demand Cooling System for Mine Rebreathers
NIH Phase I SBIR Final Report

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List of Terms & Abbreviations

ABMS = Automated Breathing Metabolic Simulator
CFD = Computational Fluid Dynamics
FEM = Finite Element Method
GWP = Global Warming Potential
MFC = Mass Flow Controller
NIOSH = National Institute of Occupational Safety and Health
NPPTL = National Personal Protective Technology Laboratory
PCV = Pressure Controlled Valve
PVA = Polyvinyl Alcohol
RH = Relative Humidity
TCV = Temperature Controlled Valve
SCBA = Self Contained Breathing Apparatus
SLM = Standard Liters per Minute

Abstract

In this SBIR project, TDA Research is developing a new, on-demand cooling system for the long duration rebreathers often used for mine rescue operations (i.e. mine rescue reentry rebreathers). Long duration rebreathers are self-contained, closed circuit, pure oxygen backpack sized units used by rescue personnel to reenter an incident area after an accident to rescue stranded personnel and perform other tasks necessary to stabilize the area. Rebreathers conserve air by recirculating exhaled air through a system that chemically scrubs out CO_2 and meters in O_2 (from a small compressed gas cylinder) to maintain an O_2 content of at least 19.5% in the inhaled air.

TDA Research set out to develop a long duration rebreather cooling system that can cool 40ºC, 100% RH inlet air to a constant 35ºC, even with changing air flow rates and heat loads. To achieve this goal, we accomplished these four tasks in the Phase I project:

1. Analyze heat exchangers
2. Build test apparatus
3. Test proof-of-concept cooling system
4. Develop preliminary compact design for Phase II

We also fabricated and tested the preliminary Phase II prototype.

The proof-of-concept system worked extremely well. It cooled the air in ~6 minutes, held a constant outlet air temperature for the full four hour test, and maintained pressure in the 30 psig heat exchanger, which prevented moisture in the air from freezing on the surface. Our test rig included a system for conditioning the inlet air to 40ºC, 100% RH, a system for cooling the air using the boiling of R134a and evaporation of water into dry R134a gas, and a system for capturing and recycling the used R134a. We also ran some variable air flow rate experiments, including one with seven different air flow rates (20 L/min to 62 L/min). The proof-of-concept prototype handled all of these test conditions excellently to accommodate a large range of air flow rates, and varied its R134a consumption rate to handle the changing heat loads, while maintaining constant outlet air temperature. While working with the proof-of-concept system, we were able to simplify the TCV, moving from proportional control to binary open/close control. We also simplified the PCV by moving from a pneumatic valve to an electronic solenoid valve. We also increased the water capacity (by >3.4x) of the low pressure heat exchanger by using a robust high capacity PVA chamois.

After testing our proof-of-concept system, we designed a preliminary prototype which combined the 30 psig and low pressure heat exchangers into a single unit that was 50% smaller and 76% lighter than our first generation cooling prototype. It also used a smaller and lighter TCV and a smaller, lighter, and fully passive PCV. We used additive manufacturing to print the prototype design out of aluminum and replaced the proof-of-concept cooler with the new prototype in our test rig. We tested the new compact heat exchanger with the same variable air flow rate protocol used for the proof-of-concept prototype system and it performed even better. Both units were able to quickly adjust the R134a flow rate as the air flow rate changed, so as to maintain constant outlet air temperature, and both consumed an almost identical amount of R134a (only 0.14% difference). Additionally, the compact system was able to cool the air even faster than the original proof-of-concept prototype (going from ~6 min cooling time to <1 min). With additional design work, we can likely shrink the temperature swings while further reducing the size and weight of the cooling system.

1. Executive Summary

In this SBIR project, TDA Research is developing a new, on-demand cooling system for the long duration rebreathers often used for mine rescue operations (i.e. mine rescue reentry rebreathers). Long duration rebreathers are self-contained, closed circuit, pure oxygen backpack sized units used by rescue personnel to reenter an incident area after an accident to rescue stranded personnel and perform other tasks necessary to stabilize the area. Long duration rebreathers conserve air by recirculating exhaled air through a system that chemically scrubs out CO_2 and meters in O_2 (from a small compressed gas cylinder) to maintain an O_2 content of at least 19.5% in the inhaled air. This allows the rebreather to be considerably lighter than an equivalent self contained breathing apparatus (SCBA) and lets the rescue workers stay in the mine for 2-4 hours instead of 30 minutes. Mine rescue rebreathers are designed so that the rescue worker can have four hours of breathable air. A diagram of a Drager's BG4 mine rescue rebreather is shown in Figure 1.

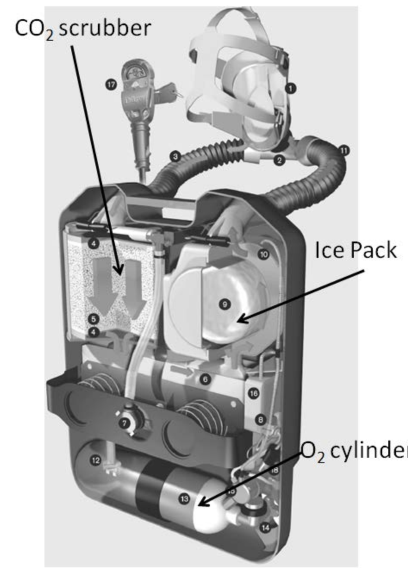

Figure 1. Dräger's BG4 rebreather [Dräger 2004]

Considerable heat is generated when the CO_2 is removed in the scrubber bed and moisture is condensed, and this heat must be removed or the air rapidly becomes too hot to breathe. Rebreathers currently employ ice to cool the air so that it stays within safe temperature limits. However, the ice tends to cool the air more than necessary early in the mission, but loses its cooling capability as the ice melts near the end of the mission. Additionally, it cannot adjust the amount of cooling based on the user's exertion level. Extra heat loads such as greater than normal exertion can result in premature consumption of the ice and in fact, the lack of ice has been implicated in at least one fatality during a training session using rebreathers in Nevada [Burden 2015]. The objective of our project was to develop a small, lightweight cooling system that keeps the rescuer's air from becoming dangerously hot for the full four hours that a modern rebreather can provide air. In Phase I, TDA designed, built, and tested a proof-of-concept prototype to prove that our approach to air cooling would work.

TDA's cooling technology replaces the ice pack with the combination of a heat exchanger and R134a, a common refrigerant. Instead of relying on the latent heat required to melt ice for cooling, we used the latent heat required to boil the R134a. We produced extra cooling by evaporating a small amount of water into the dry R134a gas before it is expelled. Unlike ice based cooling, TDA's technology controlled the temperature of the recirculation air (neither under nor over cooling it) by metering the flow rate of R134a using a thermostatic valve. This way, we used our cooling capacity more efficiently by keeping the air at a constant safe temperature (35°C) instead of wasting some of our cooling capacity early and not having enough later. Due to the thermostatic control of the R134a flow, our system automatically adjusted to changing exertion levels and ambient conditions so that the breathing air was always at a safe temperature.

In Phase I of this project, TDA designed and built a proof-of-concept prototype of our cooling system, along with an experimental rig for testing it. Our proof of concept prototype worked extremely well. Starting the apparatus at steady state with 40°C air, the system cooled the breathing air to the proper temperature within six minutes, and kept it there for the entire four hour duration of the experiment. It properly maintained a constant air temperature over a wide range of air flow rates (20 L/min - 62 L/min) which simulated a wide variety of exertion levels. We also subjected our cooling system to a variable flow rate test (which is equivalent to a variable heat load test), which included five step changes in flow rate of ~10 L/min and one step change of ~20 L/min. The R134a flow rate automatically adjusted to these changing flow rates, while keeping the breathing air at a constant temperature.

This proof-of-concept prototype was designed to be easy to use and fabricated quickly, it was not optimized to be small or light. It was fairly bulky and weighed >9.5 lbs. After using this prototype to prove that the air could be thermostatically controlled by boiling R134a and using the R134a to evaporate water, TDA designed a smaller, compact prototype of its cooling system. This compact prototype was 50% smaller and 76% lighter than the proof-of-concept prototype; it weighed 2.3 lbs. We fabricated it by 3D printing aluminum via laser powder bed fusion and tested it using the same experimental rig. We subjected it to the same variable flow rate experiment as the proof-of-concept prototype. The compact prototype also adjusted its R134a consumption rate to maintain constant breathing air temperature with results that were remarkably similar to the proof-of-concept design. In fact, it cooled the air down from our starting temperature of 40°C to the appropriate temperature in less than one minute.

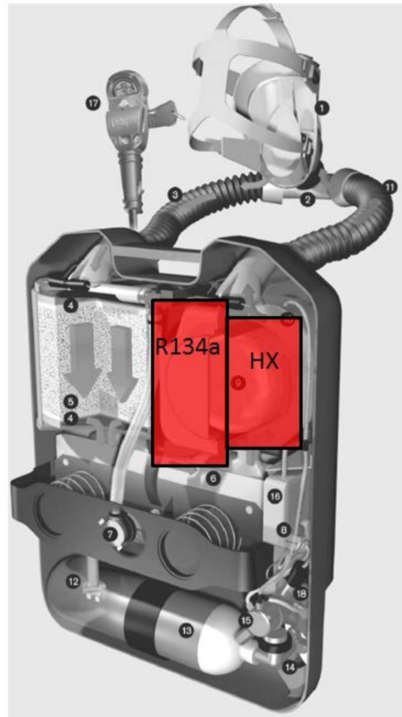

Figure 2. Dräger's BG4 rebreather [Dräger 2004] with TDA's heat exchanger and R134a canister replacing the ice pack

In a Phase II project, we will work to further optimize the system. R134a was chosen as the refrigerant because it is cheap, widely available in the US, non toxic, and non-ozone depleting. However, there are refrigerants with a higher latent heat of vaporization and refrigerants with a lower global warming potential (GWP). Using these could reduce the weight of our consumable. We will investigate using alternative refrigerants to improve system efficiency, reduce weight, and minimize global warming potential. We will also work to improve our heat exchanger design. Moving from the proof-of-concept to the printed design reduced the size by 50% and the weight by 76%. This printed version had a specific surface area of ~60 m^2/m^3 while the most compact heat exchangers can have a ratio of closer to 400 m^2/m^3. Since the R134a has to be kept at 30 psig during boiling (to prevent ice formation on the surface of the heat exchanger) and we need to maintain a very low pressure drop, we may not be able to make it 400m^2/m^3, but we should still be able to significantly increase the surface area to volume ratio, which will considerably reduce the size and weight. By improving the design for manufacturability, we are hoping to reduce the cost of our system down to ~$260 per system when produced in large quantities

Dräger lists their rebreather as weighing 34.17 lbs and having dimensions of 23.43" x 17.72" x 7.28". Based on this, we can use their picture in Figure 1 to estimate the size of the ice pack and the surrounding air channel, which is approximately 8.3" x 8.3" x 7.3". TDA's printed heat exchanger is only 3.8" x 7.4" x 2.4". Additionally, a 12 oz. can of R134a from A/C Pro has a diameter of ~2.5" and a height of ~6". If we were to scale this up to 3 lbs of R134a, we would need a can with a diameter of 4" and a height of 9". This means that with some of the open space in the rebreather, we should be able to fit both TDA's current heat exchanger and a canister of R134a in the space currently occupied by the ice pack. This is shown schematically in Figure 2.

We will also work to test our cooling technology under more realistic conditions. We will partner with the National Personal Protective Technology Laboratory (NPPTL) and test our cooling technology using their rebreathers and automated breathing metabolic simulator (ABMS). This way we can see how our cooling system works under real world conditions, instead of in the simulated conditions of our experimental rig.

TDA is working to set up the partnerships that it will need to commercialize this technology. We have had multiple conversations with Avon Protection Systems, who are developing their own rebreather that will require air cooling. They've purchased a proof-of-concept prototype of our cooler to test with their rebreather, and we are in talks with them about purchasing our compact prototype. We are hoping to leverage this relationship into a commercial partnership to ease the commercialization path for our important cooling technology.

2. Scientific Report

2.1. Background

2.1.1. The use and purpose of mine rescue rebreathers

In the event of a fire or a mine collapse, miners can be trapped, requiring rescue workers to go in and find them. While mines are designed to have adequate ventilation, depending on the exact scenario, often times the air is unsafe to breath within the mine during fires, cave-ins, and other accidents [Hartman et al 1992, DOL 1997]. Hypothetically, someone could use a self contained breathing apparatus (SCBA), but carrying all of the air needed for a rescue operation is prohibitively heavy. Alternatively, someone could use compressed air from the outside fed to them via a hose, but this considerably hinders flexibility and maneuverability. Additionally, if anything happens to the hose, then the rescuer is in peril, creating an unnecessary hazard to the rescuer. Instead of carrying all of the necessary air or having it pumped from the outside, mine rescue workers rely on rebreathers. Air is primarily nitrogen (~78%), with some oxygen (~21%) and small amounts of other gases. During breathing, some of the oxygen is turned into carbon dioxide. In a rebreather, the nitrogen and leftover oxygen are recycled, the carbon dioxide is chemically scrubbed out (as described in section C.2), and a small amount of additional oxygen is added to make up for the amount converted to CO_2 during breathing. Since the user is only carrying around a small amount of oxygen, instead of all of the air that would be needed, a rebreather can be far lighter than a full SCBA and doesn't bring any of the hazards or limits on mobility of a compressed air hose.

CO_2 absorption

$$Ca(OH)_2 + CO_{2(g)} + H_2O_{(g)} \rightarrow CaCO_3 + 2H_2O_{(g)} \quad \Delta H = -5.8\,kcal/mole$$

Water condensation

$$2H_2O_{(g)} \rightarrow 2H_2O_{(l)} \quad \Delta H = -21.5\ kcal/mole\ CO_2\ removed$$

Figure 3. Reactions for CO_2 removal using soda-lime (mostly calcium hydroxide). Note that a mole of water is produced for each mole of CO_2 removed which adds to the heat load.

The use of rebreathers where CO_2 is removed from exhaled air and oxygen is added back to the recycled air for breathing has enabled miners to conduct rescue operations that might otherwise be impossible if they had to carry an equivalent amount of compressed air in tanks (SCBA) or if they had to rely on air supplied from hoses attached to compressors outside the mine. While 4-hour mine rescue rebreathers weigh about 30-35 lb, this is considerably lighter than carrying the equivalent amount of air in an open circuit self-contained breathing apparatus (SCBA), which would weight ~88 lbs. Early rebreathers used potassium superoxide (KO_2) to generate O_2 by its reaction with CO_2 and water vapor in the breath. While this is suitable for some short term applications, all of the commercially available 4-hour rescue rebreathers use a combination of a CO_2 sorbent such as LiOH or soda-lime, along with a small cylinder of compressed O_2. A constant flow leak valve is used to maintain the O_2 concentration at 19.5 vol% or greater [eCFR 2015].

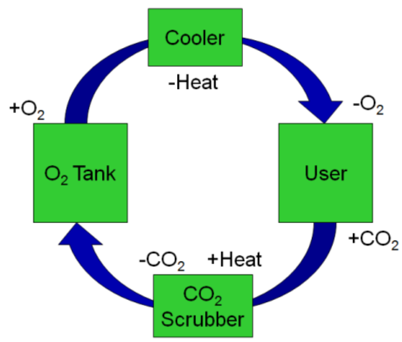

Figure 4. Diagram showing steps in a rebreather.

2.1.2. Current cooling technology for rebreathers

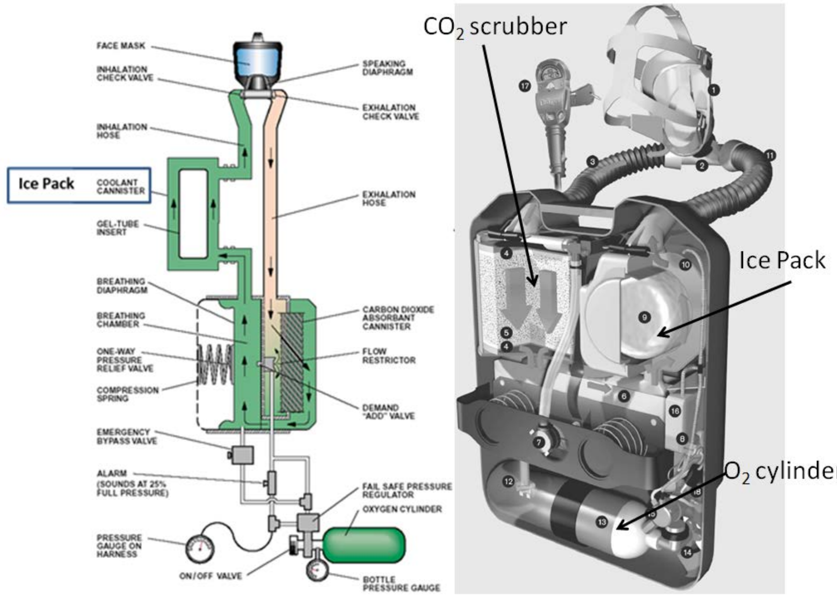

Figure 5. Schematic of Dräger's BG4 rebreather [Dräger's 2004]

The reaction between CO_2 and LiOH (or alternatively soda-lime, $Ca(OH)_2/NaOH$) is exothermic (Figure 3) so the return air must be cooled to a safe temperature before being reoxygenated for breathing. The basic process of this rebreather cycle is shown schematically in Figure 4. This cooling is now done using ice packs (see Figure 5 for a schematic of a typical rebreather). Unfortunately, it is not possible to control the rate of heat transfer from the hot air (from the CO_2 scrubber) to the ice, and therefore the return air is over-cooled early in the mission, and then under-cooled later as the ice melts. Also, because neither the heat transfer area nor the heat transfer coefficient for melting the ice in the ice pack are affected by the heat load, an increase in the heat load due to an increase in the miner's level of exertion will cause the return air temperature to increase. In a worst case, this could lead to premature consumption of the ice, and a loss of cooling which would result in a premature, dangerous and sometimes fatal end to the rescue mission (at least one fatality has been attributed to be at least in part due to loss of ice cooling during a training exercise in Nevada [Burden 2015]).

According to 42CFR 84.103, the maximum inhaled air temperature cannot exceed 35°C (95°F) where the service life of the rebreather is 4 hours when the inhaled air is between 50 and 100% relative humidity (RH). The testing is rather rigorous, and there are a series of 5 tests where the wearer, walks, runs carries and drags different weights, climbs a vertical treadmill, crawls on hands and knees, etc. for various time intervals [eCFR 2015]. As for the test subject, the regulations simply state that the apparatus be "worn by institute personnel trained in the use of self-contained breathing apparatus." While the testing specified in 42CFR 84.103 is thorough, the ice pack used in current 4-hour rebreathers cannot be guaranteed to provide sufficient cooling to keep the return air temperature below 35°C for all possible differences in individual users metabolic rates of CO_2 generation, different levels of exertion, different ambient temperature conditions and other uncontrollable factors. Thus, there

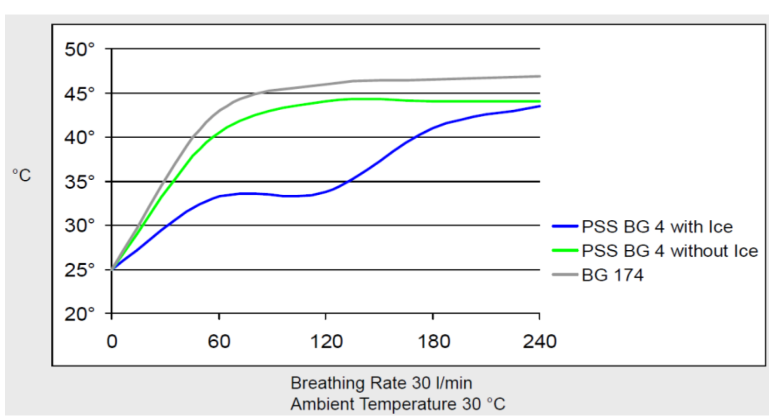

Figure 6. Temperature vs. time curve showing the effect of adding ice to Dräger's BG4 rebreather [Eisenbarth 2005]

is a need for advanced heat management technology. In fact, Dräger's (a major manufacturer of mine rescue rebreathers) own data shows that under certain conditions, their BG4 rebreather with ice can only keep the return air temperature below 35°C for ~2.25 hrs and below 40°C for <3 hrs (see Figure 6).

The rate of cooling with ice packs cannot be controlled because the heated air from the CO_2 scrubber simply passes around a cylinder filled with ice without any feedback. Early on, the amount of cooling is determined by the rate of heat transferred from the hot air through the plastic walls of the container and into the ice. Later, as the ice melts, liquid water builds up between the ice and the plastic wall of the ice pack and acts as an additional thermal resistance for heat transfer (the convective heat transfer coefficient for the air flowing over the plastic wall of the ice pack remains unchanged). At the same time, the surface area of the ice itself is reduced. Both of these effects reduce the cooling rate (Figure 7) and the overall result is that with time, the ice pack becomes less efficient at cooling, and the temperature of the return breathing air increases.

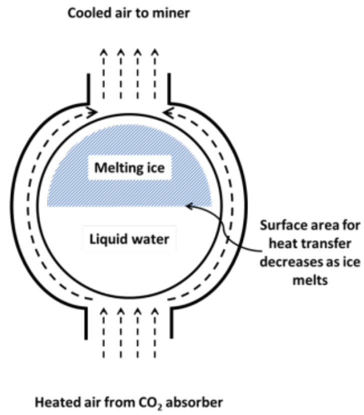

Figure 7. Schematic of ice pack and how the liquid water decreases the rate of heat transfer as the ice melts.

Because the rate of cooling cannot be controlled with ice, the size of an ice pack is a compromise between cooling capacity toward the end of the mission and weight. Current ice packs are sized so that the return air to the miner at the end of the mission does not exceed 35°C (eCFR 2015). If the ice pack were sized to have extra cooling capacity toward the end of the mission (to keep the return air temperature below 35°C) it would be considerably larger and heavier. Unfortunately, because of the need to reduce weight, any unexpected heat load on the system or unanticipated extra time in the rebreather can result in the air temperature exceeding 35°C, and possibly going much higher.

2.2. Approach

2.2.1. TDA's on-demand cooling concept

TDA developed a proof-of-concept cooling system that maintained the breathing air at a constant 35ºC for four full hours, instead of overcooling at the beginning and undercooling at the end like ice does. The cooling mechanism in our process was the evaporation of R134a via its latent heat of vaporization (~198 kJ/kg).

The R134a was stored in a compressed cylinder. It was metered into a compact gas-to-gas heat exchanger where it boiled, absorbing heat. This cooled the return air on the other side of the heat exchanger. The flow rate of the R134a was metered via a thermostatic valve controlled by the exit temperature of the air (see Figure 8). This way, when the heat load increased (due to increased metabolic rate, increased exertion, increased ambient temperature, etc.), the R134a flow rate increased, increasing the R134a consumption rate and hence the heat rejection rate which kept the air temperature at 35ºC. Conversely, as the heat load dropped, the R134a flow rate decreased, decreasing the consumption rate and the heat rejection rate, which kept the air temperature at 35°C. This way, we used our cooling capacity as efficiently as possible.

Additionally, another advantage of our system over using ice is that it could be recharged almost instantaneously. Recharging our system required replacing the empty R134a canister with a full one (which only took a few seconds) while recharging the ice based system required someone to refreeze the ice, which generally took multiple hours. Finally, our system can be implemented using off-the-shelf R134a canisters (which have an extremely long shelf life), while ice based systems require the infrastructure needed to produce and store ice. This is a major advantage for mine rescue operations in poor countries or remote locations

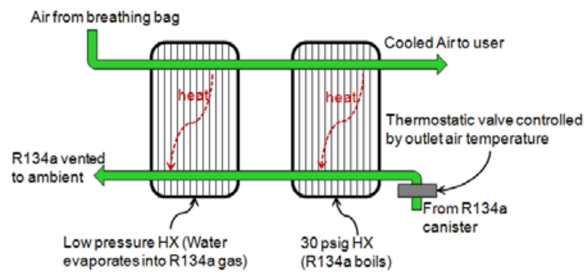

Figure 8. TDA's cooling concept where air is cooled via the boiling of R134a and the evaporation of water into R134a gas.

2.2.1.1. Using dry R134a to evaporate water

The heat of vaporization of R134a is considerably lower than that of the melting of ice (198 kJ/kg vs 334 kJ/kg). All things being equal, this puts TDA's on demand cooling technology at a significant disadvantage with regard to weight compared to the standard technology that is currently in place. However, the R134a can also be used to evaporate water (which is very endothermic).

The heat of vaporization of water is 2257 kJ/kg, dwarfing both R134a and ice. Unfortunately, it is impossible to rely on water evaporating into the ambient air because there can be huge variability with ambient air humidity. In humid air (and mine shafts can often have high humidity), it is difficult to get significant water evaporation. However, when the R134a in TDA's system boils it creates a dry gas which water can then evaporate into. By supplementing the cooling capacity of the R134a with the cooling capacity of water, we can increase the R134a's effective cooling capacity by 30%, closing a considerable amount of the gap between the cooling capacity of R134a and that of ice.

2.2.1.2. Determining total cooling capacity of ice

In an ice-based cooling system, the total amount of heat that can be removed is determined by the mass of the ice and the heat of fusion of water (ΔH_{fusion} = 144 Btu/lb). Existing rebreathers use ice packs that contain about 2.5 lb of ice, which corresponds to 360 Btu of cooling. This is actually far less than the total heat generated by the CO_2 scrubber. Most of the heat is actually dissipated by the mass of the rebreather (which acts as a heat sink) and by convective and radiative losses to the environment. For example, if the miner is breathing moderately heavily (60 breaths/min) and releasing 0.037 grams of CO_2 per breath [CDIAC 2011] the total heat generated by absorption of the CO_2 is 336 Btu/hr (1345 Btu for a 4 hour mission). Of this, ~1000 Btu is dissipated by heating the equipment and surroundings, with about 345 Btu going into melting the ice pack. Under these conditions, the rebreather equipment temperature will increase by less than 20°C over the course of the 4 hour rescue; this calculation is very approximate and assumes that the rebreather

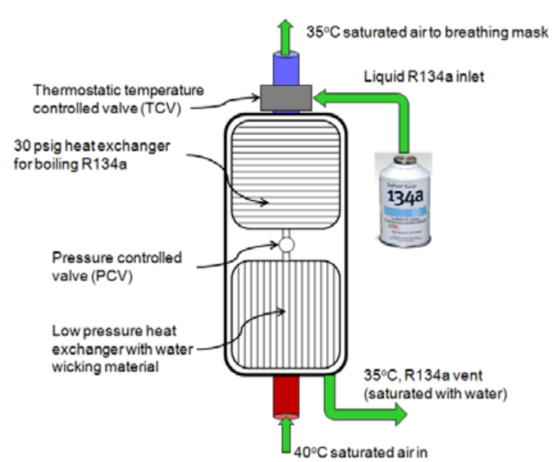

Figure 9. Schematic of on-demand cooling system.

absorbs half of the lost heat (500 Btu) as Q=mCpΔT, where Q is the heat load, the equipment mass is m = 35 lb and Cp (heat capacity) ≅ 0.5 Btu/lb/°F. Since 2.5 lb of ice melting corresponds to 360 Btu, we have designed our R134a based cooling system to remove the same 360 Btu of heat.

2.2.1.3. Implementation of TDA's on-demand cooling system

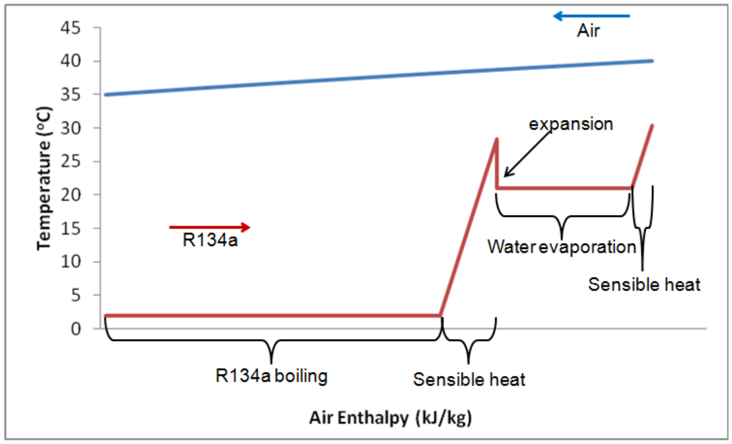

Figure 10. Temperature-Enthalpy diagram of TDA's cooling system.

TDA's on-demand cooling system replaced the ice pack with an R134a canister and a counterflow gas-to-gas compact heat exchanger. The compact heat exchanger had two parts, a 30 psig side and a low pressure side (shown in Figure 9). From the R134a's perspective, R134a passed through a thermostatic valve into a pressurized chamber (p ~ 30 psig). The pressure kept the R134a boiling temperature above 0°C, which prevented any water in the breathing air from freezing onto the heat exchanger surface and degrading its performance. The R134a boiled, absorbing heat from the air. It then exited the 30 psig heat exchanger (via a pressure controlled valve) into the low pressure side of the heat exchanger. The expansion cooled the gas a bit more (allowing it to absorb more heat from the air). It then flowed over a wicking material saturated with water. The warm dry gas caused the water to evaporate, removing even more heat from the air. The R134a was then exhausted to the surroundings. R134a is non-toxic, non-flammable, and non-ozone depleting [Elsheikh et al. 2006], so it poses no harm to the mine rescue workers. This process is made clear via the temperature-air enthalpy graph shown in Figure 10.

From the air's perspective: hot air entered the low pressure side of the heat exchanger where it was cooled via the evaporation of water into dry R134a and the expansion of R134a out of the 30 psig side. It then entered the 30 psig side where the air was cooled via the boiling of R134a. The air then exited the heat exchanger at 35°C and was inhaled by the user.

2.2.1.4. Criteria for Phase I success

The success criteria for Phase I was to build and test a proof-of-concept version of TDA's on-demand cooling system. This system had to be able to properly cool 30 L/min of 40°C air at 100% RH down to 35°C by controlling the flow rate of R134a. It also had to be able to accommodate changes in air flow rate while maintaining constant outlet air temperature.

2.3. Results

In this Phase I project, we designed a proof-of-concept prototype that included a thermostatic valve, a 30 psig heat exchanger for boiling R134a, a pressure control valve, and a low pressure heat exchanger with water wicking material for evaporating water into R134a gas. Using that proof-of-concept prototype, we showed that this approach could effectively maintain a constant outlet air temperature, even with extreme changes in air flow rate (and heat load).

We then improved upon that original design with the goal of making the components smaller and lighter while maintaining thermostatic operation. We developed a preliminary design for a compact heat exchanger which combined both the 30 psig and low pressure units into a single component. Using additive manufacturing, we printed this heat exchanger and incorporated it into our experimental rig. We combined this heat exchanger with miniaturized valves for metering R134a and controlling the pressure in the 30 psig side. This preliminary design drastically reduced the size (reduction in volume by 50%) and weight (reduction in weight by 76%) of our original prototype while still effectively maintaining constant outlet air temperature. Our second generation unit weighed 2.3 lbs and had a volume of 83 in^3.

2.3.1. Designing and building Phase I proof-of-concept rig

The goal of our initial proof-of-concept rig was to demonstrate that we could use the boiling of R134a to maintain constant outlet air temperature, even with changing inlet air conditions. This prototype was designed to be manufactured quickly, and to be easy to use so that we could perform a large number of experiments in quick succession. It was not designed to minimize size or weight. The rig allowed us to control the flow rate, temperature and humidity of the air entering our cooling system and then test our cooling unit.

2.3.1.1. General Schematic

As we described in the Phase I proposal, our proof-of-concept system needed to be able to cool saturated air from 40°C down to 35°C at a flow rate of 30 L/min. The air coming off of the CO_2 scrubber would be hotter than 40°°C, but as described in section D.3, a considerable amount of that heat would be absorbed/rejected by the rebreather equipment so that by the time the air is entering the cooling system, it would be approximately 40°C and 100% RH. The mine rebreather test rig that we built to meet our cooling goal had three main subsystems: (i) air conditioning, (ii) cooling system proof-of-concept, and (iii) R134a capture system.

The air conditioning system is what we used to control the air inlet properties. This system included a metering valve for controlling the flow rate of compressed air, heaters for maintaining the air temperature, and a humidifier that ensured that the air was saturated at ~100% RH. Altogether, it ensured that the correct amount of inlet air entered the cooling section and that the air was at 40°C and 100% RH.

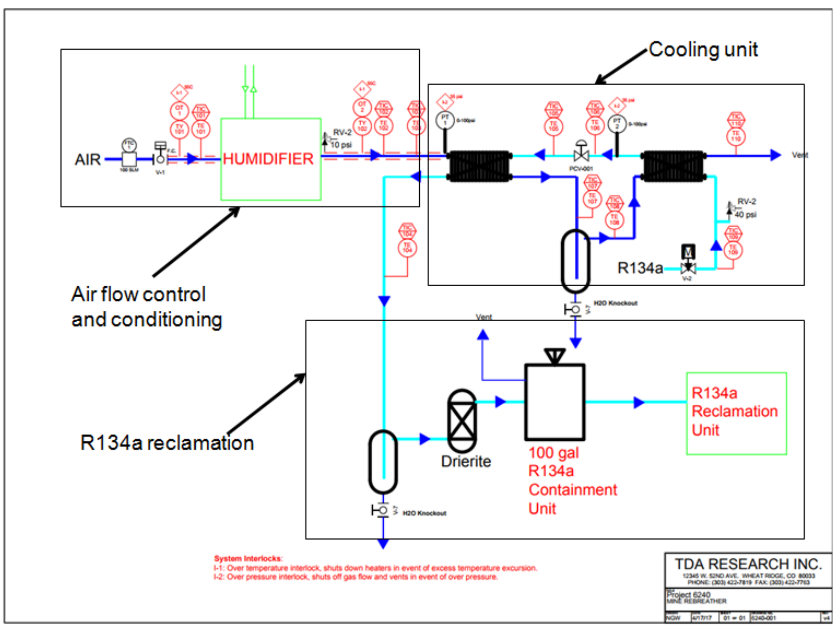

Figure 11. Schematic of experimental system used for proof-of-concept prototype. Box in top left is the air conditioning section, the box in the bottom right is the R134a reclamation section, and the box in the top right is the cooling proof-of-concept prototype.

The cooling section was our proof-of-concept prototype. It included the R134a cylinder, the solenoid valve that metered in the R134a, the 30 psig heat exchanger, the pressure relief valve for the 30 psig heat exchanger, and the low pressure heat exchanger. This is the subsystem where heat from the air was transferred to the R134a.

The last subsystem was the R134a reclamation unit. In real-world use, the spent R134a will be exhausted to the environment, where it is non-toxic and non-flammable. However, in the laboratory setting, we setup an R134a capture system to recycle the R134a. While the experiment was ongoing, the used R134a was dried using a Drierite column and then exhausted into containment bags within barrels. We used containment bags to maintain constant ambient pressure as they were filled, while the barrels were there to protect the bags and hold the R134a in case of a leak. After the experiment was done, an R134a compressor was used to pump the R134a from the containment bags into a recycle cylinder. When our initial R134a cylinder was spent, we replaced it with the recycle cylinder.

The whole system is shown schematically in Figure 11. The components in the square at the top left were used for the air conditioning subsystem and the components in the rectangle at the bottom right were used for the R134a recycling subsystem with the rest being the cooling subsystem. When this idea is implemented into a real rebreather, only the cooling subsystem will be used. The other two subsystems were there for ease of experimentation, not for cooling purposes

2.3.1.2. Details of air and R134a loops.

It can be difficult to understand the cooling process just by looking at the schematic in Figure 11, so here we'll go into more detail about the air and R134a loops. The air loop is fairly simple. Air left the compressed air cylinder, where it was conditioned to the proper humidity and temperature (100% RH and 40°C). Next, it entered the low pressure heat exchanger, where it was cooled via the evaporation of water into dry R134a gas. Then it entered the 30 psig heat exchanger where it was cooled via the boiling of R134a down to 35°C. Then it was exhausted into the room (in the real system, it would be inhaled by the user).

The R134a flowed counter to the air stream. It is shown in Figure 12. R134a started in an R134a cylinder; it was metered through a valve controlled by the outlet temperature of the air, through an orifice into the 30 psig heat exchanger. Here it boiled at ~0°C and ~30 psig. It absorbed some of the heat from the air, increasing its temperature. It then went through a pressure control valve where it expanded, which reduced its temperature and allowed it to absorb more heat from the air as it entered the low pressure heat exchanger (at 0 psig). It then flowed over a water wicking material, which evaporated water and cooled the air further. It exited the low pressure heat exchanger and entered the R134a capture system where it was recycled into a R134a cylinder (in the real system, it would skip the capture system and be exhausted to the environment).

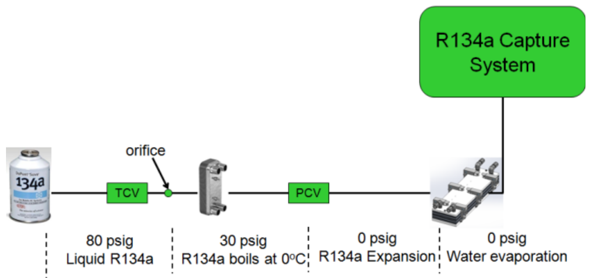

Figure 12. Diagram of R134a stream (TCV = temperature controlled valve, PCV = pressure controlled valve)

2.3.1.3. Brief description of main components

Almost all of the components in this experimental system were bought off-the-shelf. The main components are listed below, organized by subsystem.

Air Conditioning Subsystem:
- Compressed Air
- Air MFC--603aaafsvvaa from Parker
- Humidifier--Perma Pure Humidifier Tube from Permapure
- Heater--AWH-051-020DM-MP heat tape from HTS/Amptek
- Temperature controlled water bath for saturating air at 40°C--F25 from Julabo

Cooling Subsystem:
- Temperature controlled solenoid valve--AVP-31C1-24D from Nitra
- 30 psig Heat Exchanger--35115K61 from McMaster-Carr
- Pressure controlled valve--1001gct36svcpp12sx with the "N" trimset from Badger Meter, and PSV-1 from Aalborg
- Low Pressure Heat Exchanger--custom built at TDA

R134a Reclamation Subsystem:
- R134a drums--4142T4 from McMaster-Carr
- R134a reclamation pump--RG3 from Robinair

2.3.1.4. Details on the heat exchangers

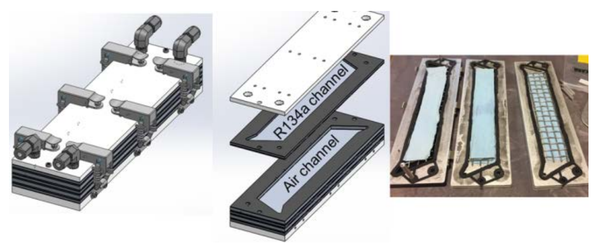

Figure 13. Custom built flat plate heat exchanger. On the left is the heat exchanger fully assembled, in the middle is an exploded view, and on the right is a photograph of the heat exchanger disassembled.

The two heat exchangers that we used served slightly different purposes. The 30 psig heat exchanger held the boiling R134a. It needed to be able to withstand at least 30 psig to keep the R134a boiling above 0°C. It was important that the R134a boiled above 0°C to prevent water in the air from freezing on the heat exchanger surface, which would have acted as an additional thermal barrier and degraded the heat exchanger's efficiency. We found an affordable off-the-shelf brazed flat plate heat exchanger that met these requirements from McMaster-Carr. It weighed ~4 lbs, had dimensions of 7.81" x 3.31" x 2.63" and was rated up to 450 psi. This heat exchanger was thoroughly overbuilt for our purposes, but it was acceptable for a proof-of-concept prototype.

The low pressure heat exchanger was considerably more complicated. It needed to be able to be opened up so that wetted wicking material could be inserted into the R134a channels. We were unable to identify any off the shelf heat exchangers that met our needs. We decided to custom build our own heat exchanger as shown in Figure 13 (left). It was a flat plate heat exchanger held together with corner draw latches. There were three R134a channels and two air channels in between them with the two streams in counter flow (Figure 13 middle). By using latches, it was fairly easy to disassemble (Figure 13 right) so that the wicking material could be wetted before each run. The low pressure heat exchanger weighed 3.8 lbs and had dimensions of 12" x 3.5" x 1.78".

We used the CFD package in SolidWorks to analyze our custom low pressure heat exchanger. The simulation was used to make sure that we had fairly even flow across the wetted surfaces (to ensure effective evaporation and heat transfer), that we had an acceptable amount of heat transfer, and that the pressure drop would not be excessive. SolidWorks cannot do coupled heat and mass transfer, and cannot simulate phase change. So, we could not directly simulate the heat transfer mechanism in this heat exchanger.

Instead, we relied on the relationship between heat and mass transfer coefficients. SolidWorks can determine the thermal convective coefficient, h_{th}. Then, using the Chilton-Colburn analogy, we related the mass convection coefficient, h_m, to the thermal convection coefficient [Incropera 2011].

$$h_m = \frac{h_{th}}{\rho c_p}\left(\frac{D_{AB}}{\alpha}\right)^{2/3}$$

In this equation, D_{AB}=mass diffusivity, α=thermal diffusivity, ρ=density, and c_p=specific heat. This way, we used SolidWorks CFD to determine an effective thermal convection coefficient. Then we used the Chilton-Colburn analogy to determine an effective mass convection coefficient. With that, we could determine an average water evaporation rate and use that to determine the amount of cooling. These calculations were used to develop our final low pressure heat exchanger design shown in Figure 13.

2.3.1.5. Water wicking material

A water wicking material was needed in the R134a paths of the low pressure heat exchanger that has a high capacity for holding onto water but also allowed water to evaporate into the dry R134a gas. We initially used paper towels as the wicking material. The paper towel was acceptable, but we found that it tended to dry out before the full four hour experiment was completed, leading us to search for alternative wicking materials. Additionally, the paper towel would probably not be robust enough to survive a large number of uses in a real world system.

The ideal wicking material needed to be robust and able to hold onto a lot of water while also able to release the water so that it could evaporate. Some materials, like absorbent gel materials in diapers are great at holding onto water, but terrible at releasing it. While searching for possible alternatives, we found a product, a PVA chamois that looked promising. Its two main applications are as towels for absorbing lots of water, and as a cooler where it is wetted and put over the neck to cool someone on a hot day. Thus, it has high water capacity while allowing water to evaporate. The PVA chamois material is also extremely cheap (we bought a full towel from Amazon for <$10).

We did experiments to see how much water per unit area the PVA chamois holds compared to the paper towel. We cut both the paper towel and the PVA chamois into 10 cm × 10 cm squares (i.e. 100 cm^2) and weighed them dry, then saturated them, and weighed them wet. The paper towel held 4.54 g of water while the PVA chamois held 15.63 g of water (that is >3.4x more water) for the same area. Additionally, the PVA chamois was much more robust than the paper towel and survived a high number of uses.

We should note that in this application, the water capacity per unit area was most important, since we had a limited amount of surface area that the wicking material could make contact with. On a per gram basis, the values of water capacity for the paper towel and the PVA chamois were almost identical. We did not do measurements for volumetric water capacity.

2.3.1.6. Control System

Along with specifying components, and building the low pressure heat exchanger, we also had to develop the control scheme for controlling the temperature controlled valve (TCV) metering in the R134a and the pressure controlled valve (PCV) maintaining a constant 30 psig in the 30 psig heat exchanger.

We had initially intended to use the temperature controlled solenoid valve which metered in the R134a as a proportional valve, where the size of the opening would depend on the current outlet air temperature. However, after a few preliminary experiments, this control mechanism proved too unwieldy for this application. Instead, we inserted an orifice behind the temperature controlled valve and used the valve as a binary open/close valve, with it being opened when the outlet air temperature was above the set temperature, and closed when it was below the set temperature. The open/close control mechanism drastically simplified the controls, and still worked extremely well.

Our original PCV used a large pneumatic valve with the "N" trim set from Badger Meter. This valve was large and heavy, but worked well to control the pressure in the 30 psig heat exchanger. We later replaced that valve with a much smaller, lighter, and cheaper PSV-1 valve from Aalborg. This significantly reduced the size and weight, and still properly controlled the pressure in the 30 psig heat exchanger.

2.3.1.7. Thermodynamic accounting

The goal of our proof of concept prototype was to be able to cool the saturated air from 40°C down to 35°C. In real world conditions, the cooling unit will be inside the rebreather container, insulating it from the ambient conditions. In the lab, the ambient air temperature was only 20°C, and while we insulated all of the lines, there was still some heat leakage. While running 100% RH, 40°C air through our system, without any cooling from R134a, the exit air was ~35°C. Clearly we could not claim credit for cooling the air down to 35°C with R134a if the outlet air was already 35°C without any active cooling.

To solve this issue, we decided to evaluate our cooling system on an energy basis instead of a temperature basis. Specifically, we tracked the enthalpy change of the air. The change in enthalpy of the gas told us the amount of energy (both sensible heat due to change in temperature and latent heat due to condensation of water) that was removed from the air during cooling. To cool 100% RH air from 40°C to 35°C required 45.04 kJ/kg of air (i.e. we went from 195.52 kJ/kg to 150.48 kJ/kg). The sensible heat and latent heat were calculated using the following equations:

$$q_{sensible} = c_p \Delta T$$
$$q_{latent} = \lambda \Delta W$$

Where q = heat per unit mass (J/kg), c_p = specific heat of air, T = temperature, λ = latent heat of vaporization of water, and W=absolute humidity (kg water/kg dry air). The change in enthalpy could also be calculated using a psychrometric chart or a psychrometric calculator as done in [Singh 2002] using the equations from [ASHRAE 2005].

To determine the exit temperature that our air should have been at with cooling, we subtracted 45.04 kJ/kg from the outlet air enthalpy without any cooling (150.48 kJ/kg) to get 105.44 kJ/kg. This equated to 28.34°C and 100% RH. _This means that cooling our air down to 28.3°C with the heat losses inherent in our system was equivalent to cooling the air down to 35°C without any heat losses._ Of course, in the real system, we would be looking to cool to 35°C regardless of the amount of heat loss, we were only doing this enthalpy accounting so that we could be sure that our cooling system was capable of cooling to 35°C under real-world conditions.

Along with accurately accounting for the amount of energy rejected due to our cooling system, we also had to be very careful in accounting for the air flow rate. Our mass flow controller was designed to control the air flow rate based on "standard liters per minute" (SLM). The "standard" in "standard liters per minute" refers to dry air at 20°C, while our air was saturated at 40°C. In order to convert SLM into actual L/min we did our calculations on a mass basis.

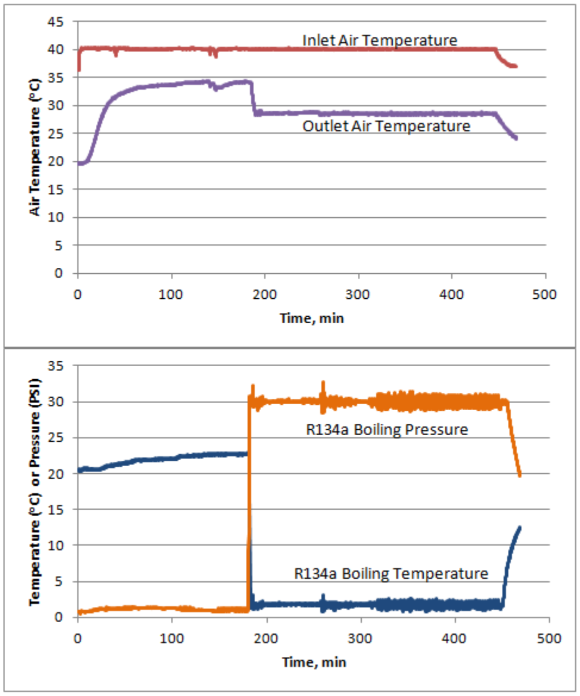

Figure 14. Data from 25 SLM experiment. Top: Inlet and outlet air temperature, Bottom: R134a pressure and temperature in 30 psig heat exchanger

First, we converted the SLM to a mass flow rate. We multiplied the SLM by the specific volume of dry air at 20°C (which is listed in the thermodynamic tables of air in [ASHRAE 1989]) and took into account unit conversion from L to m^3. This gave us the mass flow rate in kg/min. To determine the volumetric flow rate at 40°C and saturated, we multiplied the mass flow rate by the saturated specific volume and took into account unit conversion from m^3 to L. Overall, converting from SLM to actual L/min for saturated air at 40°C meant multiplying by 1.172. _Throughout the rest of this report, SLM will refer to "standard liters per minute" as was measured by our MFC, while L/min will refer to "actual liters per minute" when the air was saturated and heated to 40°C._

2.3.2. Testing proof-of-concept device

The average air flow rate that Drager uses to test its rebreather is 30 L/min, so we started testing our proof-of-concept test rig using a constant flow rate of 25 SLM (i.e. 29.3 L/min). We monitored the temperature of the air and R134a throughout the system, as well as the pressure in the 30 psig heat exchanger. We also monitored the consumption of R134a by keeping our R134a cylinder on top of a scale and measuring the weight change of the cylinder.

Figure 14 shows typical data for a 25 SLM (29.3 L/min) experiment. In the beginning, we just ran hot air, to bring the system to steady state without any cooling. At minute 185, the

Table 1. Summary of single flow rate experiments

Date	Air Temperature	Air flow rate (SLM)	Air flow rate L/min	whicking material	R134a consumption rate
8/21/2017	40	25	29.3	paper towel	3.75
8/21/2017	40	30	35.2	paper towel	4.74
8/22/2017	40	25	29.3	paper towel	3.83
8/24/2017	40	33	38.7	paper towel	4.56
8/30/2017	40	53	62.1	paper towel	5.22
9/18/2017	40	25	29.3	paper towel	3.43
9/20/2017	40	25	29.3	PVA	3.34

R134a cooling system was activated. The outlet air (in purple on the top graph of Figure 14) dropped to ~28.3°C in ~6 minutes and then held steady throughout the experiment (which ended at the 445 minute mark). In the bottom graph of Figure 14, we can see that the pressure control system worked effectively, with the pressure holding around 30 psig and the R134a inlet temperature holding just above 0°C. The cooling system was run for ~4.3 hrs before it was turned off at the 445 minute mark. In this experiment, four hours of cooling required 3.43 lbs of R134a (slightly higher than the 3 lbs of R134a we had calculated using thermodynamic equilibrium and assuming 100% efficiency).

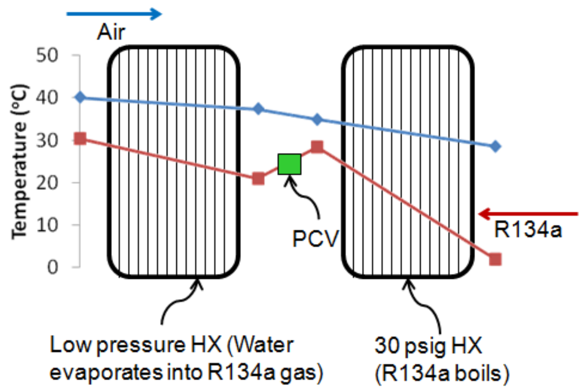

Figure 15. Typical steady state temperature profiles for air and R134a streams.

We performed four repetitions of the 25 SLM (29.3 L/min) flow rate experiments, as well as additional experiments at flow rates of 30, 33, and 53 SLM (35.2, 38.7, and 62.1 L/min). The heat that had to be removed from the air is: $\dot{q} = \dot{m}c_p\Delta T$. where $\dot{q}$ is the cooling power (W), $\dot{m}$ is the mass flow rate (kg/s), c_p is the specific heat (J/kg-°C) and ΔT is the temperature change of the air (°C). Based on that equation, it was clear that we could vary the heat load by either changing the temperature of the incoming air or by changing the flow rate of the incoming air. To simplify running our rig, we chose to change the heat load by changing the air flow rate.

Table 1 summarizes the data for these constant flow rate experiments. In each of these experiments, the outlet air cooled to the appropriate temperature quickly, and held steady throughout the entire 4 hr experiment. Additionally, the pressure control system worked well, maintaining R134a pressure within the 30 psig heat exchanger at ~30 psig and maintaining the inlet temperature into the high temperature heat exchanger at just above 0°C (which prevented moisture in the humid air stream from freezing on the heat exchanger surface, degrading its efficiency). Our system was able to accommodate a wide variety of flow rates because the solenoid valve which metered in the R134a was controlled via the outlet air temperature. When the air flow rate was lower, less R134a was consumed and when the air flow rate was higher, more R134a was consumed while a constant outlet air temperature was always maintained.

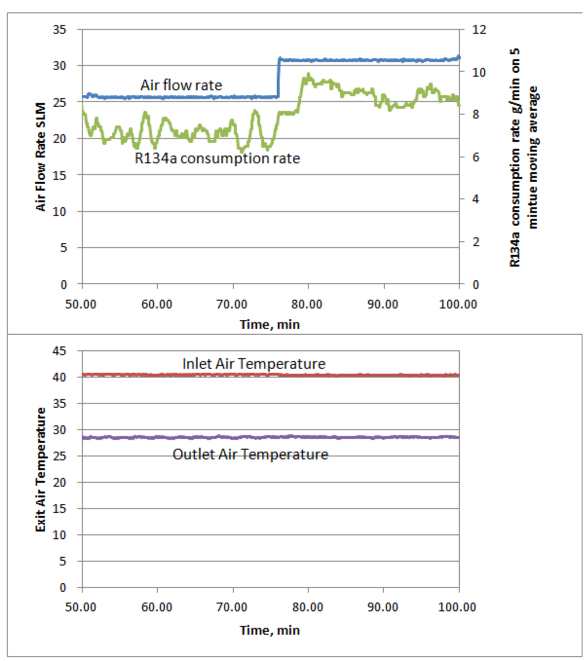

Figure 16. Data from variable flow rate experiment. Top: air flow rate (left axis) and R134a consumption rate (right axis), Bottom: Air inlet and outlet temperatures.

2.3.2.1. Temperature Profile

As described above, the goal of this proof-of-concept prototype was to remove 45 kJ/kg of energy from the air. While accounting for the heat loss to ambient, this translated to cooling the air to ~28.3°C. The steady state temperature profiles of the air and R134a streams in a typical experiment are shown in Figure 15, along with the position within the cooling system. The R134a began on the right, boiled in the 30 psig heat exchanger (at ~30 psig and 2°C), then absorbed heat from the air in the heat exchanger and warmed up to 28.3°C. It then went through the PCV where it expanded (with pressure at 0 psig) and cooled to 21°C. Then it entered the low pressure heat exchanger where it evaporated water and absorbed heat from the air. It was then vented to ambient at 30.3°C.

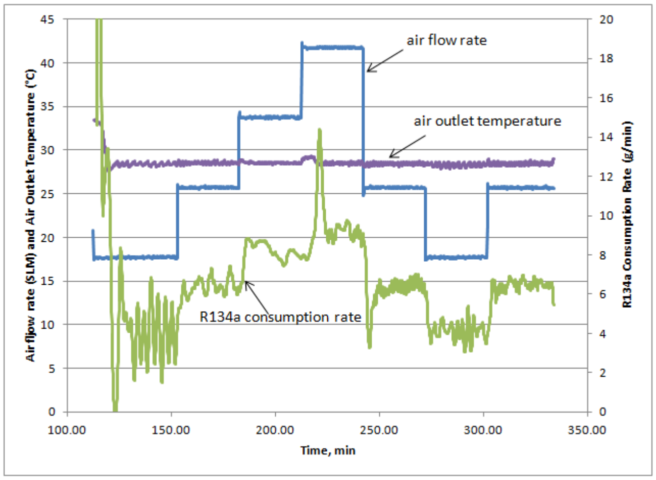

Figure 17. Results from variable air flow rate experiment. The left axis gives values for air flow rate (blue) and outlet air temperature (purple) while the right axis gives values for the R134a consumption rate moving average (green).

The air began on the left at 40°C, 100% RH as it entered the low pressure heat exchanger. It gave up heat to the R134a stream, cooled to 37.3°C as it went through the heat exchanger. It then moved through the piping between the low pressure and 30 psig heat exchangers, leaking heat to ambient, and cooling down to 34.9°C. Lastly, it entered the 30 psig heat exchanger where it gave up heat to the boiling R134a, cooling it down to 28.3°C.

2.3.2.2. Variable flow rate experiments

Our constant flow rate experiments helped us develop the control systems for the TCV and PCV and show that they work over a wide range of flow rates. It also helped us determine the R134a consumption rates over a wide range of flow rates. However, one of the most attractive attributes of TDA's on-demand cooling technology is that it could handle variable heat loads without overcooling or undercooling. In our first test to investigate variable heat loads, we started the experiment at 25 SLM, then part way through increased the air flow rate to 30 SLM. We maintained the inlet air at constant saturated 40°C. As shown in Figure 16, the TCV quickly and accurately adjusted the R134a consumption rate to maintain a constant outlet air temperature.

Figure 18. Inside of Drager's BG4 rebreather [Drager 2017]. (Left) shows the size and location of the ice pack while (Right) shows the size of TDA's compact heat exchanger superposed over the icepack.

Our cooling system's ability to adjust to changing heat loads was even more apparent in our second variable load experiment. Here we went through seven different air flow rates ranging from 17 SLM - 42 SLM (20 L/min - 49 L/min). We also used the PVA chamois as the water wicking material instead of paper towels for this experiment. The results from this experiment are shown in Figure 17. Even as we had huge variations in air flow rate (and hence heat load), *<u>the R134a adjusted quickly and accurately enough so that the air outlet temperature stayed constant</u>*.

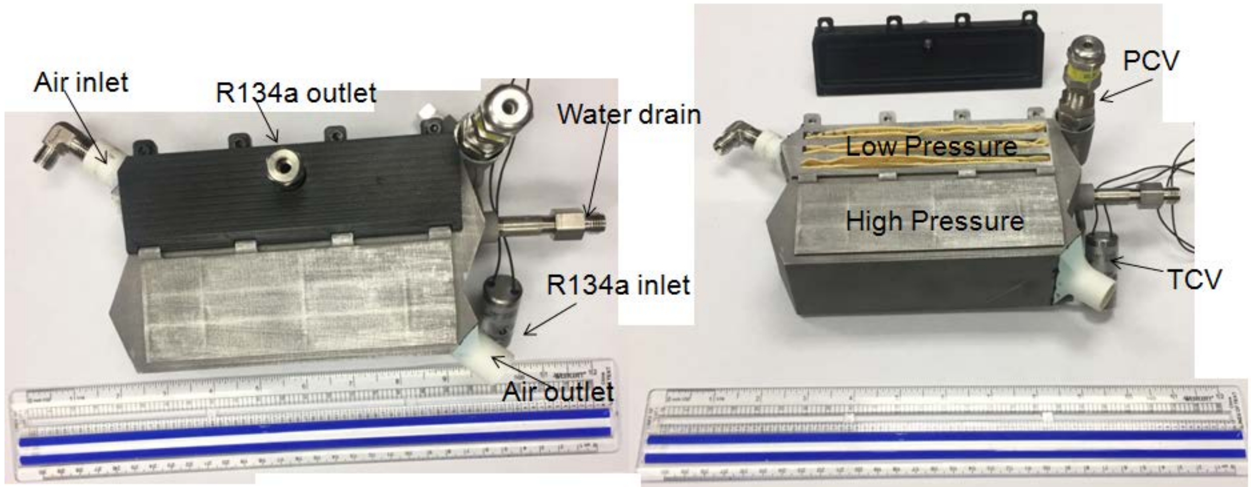

Figure 19. Printed heat exchanger combining 30 psig and low pressure components. (Left) heat exchanger fully assembled, (Right) heat exchanger partially disassembled to show low pressure wetted region.

2.3.3. Preliminary compact integrated heat exchanger

The last task that we proposed in Phase I was to use what we learned in designing the proof-of-concept prototype to develop a preliminary design for a compact heat exchanger for Phase II. TDA used our CAD program, SolidWorks to develop a custom heat exchanger that combined both the 30 psig and low pressure heat exchangers into a single unit.

We then used the additive manufacturing capabilities at Qualified Rapid Products to print our design out of aluminum. <u>*This heat exchanger was considerably smaller and lighter than the combined size and weight of the two heat exchangers from the proof of concept rig (54% smaller and 78% lighter) at only 7.4" x 3.8" x 2.35" and 1.8 lbs.*</u> In Drager's brochure for its BG4 rebreather [Drager 2017], it lists the rebreather's dimensions as 23.43" x 17.72" x 7.28". It also includes a picture of the rebreather, with the ice pack clearly visible (see left side of Figure 18). Using the picture and the dimensions listed, we estimated the size of the ice pack as roughly 8.33" x 8.33" x 7.28". Our heat compact exchanger was considerably smaller than this ice pack, as demonstrated on the right side of Figure 18.

Furthermore, this design replaced the Nitra solenoid valve that we used as a TCV in the proof of concept prototype with a considerably smaller/lighter 411L1112HVS solenoid valve from Asco. It also replaced the PCV valve from the proof of concept prototype with a smaller, lighter, and totally passive modified RL3 spring valve from Swagelok. The printed heat exchanger, along with the TCV, PCV, and additional connectors is shown in Figure 19 along with a 12" ruler for scale. On the left, we see the heat exchanger fully closed and ready for operation. On the right, we see the door removed which provides access to the low pressure section so that we could quickly and easily re-wet the PVA chamois.

The heat exchanger was locally cross flow and globally counterflow, i.e. at any particular location, the air and R134a streams were moving perpendicular to one another, but over their total path lengths, they were moving in opposite directions. This idea is somewhat clearer when the two paths are highlighted, as shown in Figure 20.

2.3.3.1. Designing compact heat exchanger

Developing this heat exchanger required an intense iterative design approach where we had to balance multiple competing interests, including minimizing size and weight, minimizing air pressure drop, while also designing in enough support so that the 30 psig section did not burst due to pressure. To balance these goals, we used various simulations packages built into SolidWorks.

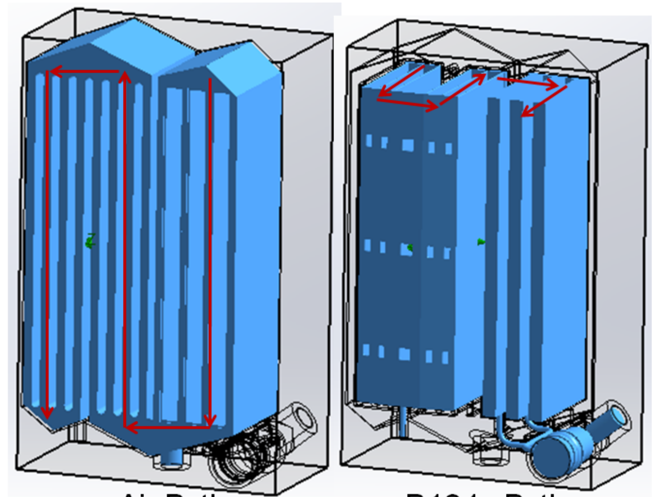

Figure 20. Air (left) and R134a (right) paths within the compact heat exchanger.

We used the finite element method (FEM) in SolidWorks to calculate stress on a part. To speed up the simulation, we only focused on the 30 psig section. We used 107,244 linear tetrahedral elements. We added enough constraints to prevent rigid body motion, and placed a 100 psig pressure load on the surfaces exposed to pressurized R134a (we used 100 psig instead of 30 psig so that we would have a built-in factor of safety). Our initial design led to excessive stresses which would have caused rupture at 100 psig (see Figure 21). We went through eight additional iterations, making changes to the design like thickening the walls, adding ribs, and filleting corners while working to minimize stress and weight. After the eighth iteration, we had a version that kept the stress to within acceptable limits and still only weighed 1.8 lbs (78% less than the combined weights of the proof-of-concept heat exchangers).

Once we had a design that could survive the pressure, we also had to make sure that our heat exchanger had reasonable flow paths (improving heat exchanger efficiency) and had minimal pressure drop on the air side. We used computational fluid dynamics (CFD) in SolidWorks Flow Simulation to investigate air pressure drop and the flow paths. This required 2.6 million cells and ~eight hours for the solution to converge. Some of the results are shown in Figure 22. The calculated pressure drop was extremely low, less than 10 Pa (<0.05 inches of water). Additionally, the air flow paths were fairly uniform, which lead to an efficient heat exchanger.

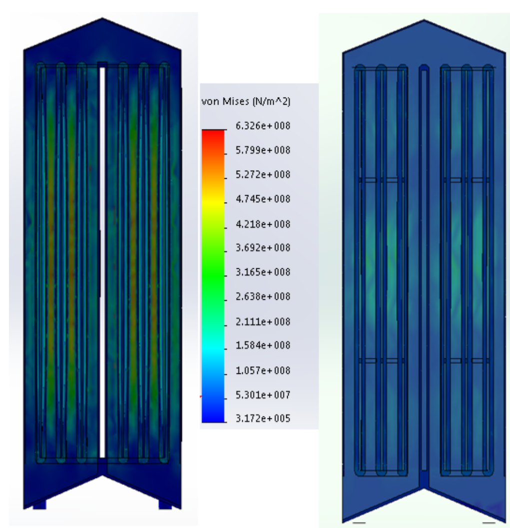

Figure 21. Von Mises Stress of original design (Left) and modified design with thicker walls, ribs, and filleted corners (Right).

Our second generation cooling system shown in Figure 19 includes the PCV and the TCV and was significantly smaller and lighter than our first generation proof-of-concept prototype that we had tested earlier. Table 2 shows the size and weight of the components in our first generation original proof of concept prototype and our newer second generation compact prototype. The second generation prototype was 50% smaller and 76% lighter than the first generation prototype.

2.3.3.2. Testing the compact heat exchanger

We decided to test our new compact heat exchanger by inserting it into our proof-of-concept test rig. This replaced the 30 psig heat exchanger, low pressure heat exchanger, TCV, and PCV with the combined assembly shown in Figure 19.

Table 2. Summary of size and weight of the two generations of cooling prototypes

Gen 1 Prototype				
Part	Part Number	Vendor	weight (lbs)	size (in^3)
PCV	PSV-1	Aalborg	0.88	11.21
TCV	AVP-31C1-24D	Nitra	0.94	12.25
High Pressure Heat Exchanger	35115K61	McMaster-Carr	4	67.99
Low Pressure Heat Exchanger	N/A	custom	3.8	74.76
		Totals:	**9.62**	**166.21**

Gen 2 Prototype				
Part	Part Number	Vendor	weight (lbs)	size (in^3)
PCV	RL3	Swagelok	0.33	1.44
TCV	411L112HVS	Asco	0.17	15.92
Combined Heat Exchanger	N/A	custom	1.8	66.08
		Totals:	**2.30**	**83.44**

Using the new compact cooling system, we performed the same variable air flow rate experiment as we performed on the original proof-of-concept prototype (data from the original proof-of-concept prototype is shown in Figure 17). The data for the new prototype is shown in Figure 23. The results from the new heat exchanger were remarkably similar to the results from the old one, which can be seen by comparing Figure 17 and Figure 23.

Both did a phenomenal job of holding a constant outlet air temperature. Also, both accommodated the changing air flow rate (and hence the heat load) by changing the R134a consumption rate. In fact, over the 4 hour experiment, the proof-of-concept system used 3.518 lbs of R134a while the new compact heat exchanger used 3.523 lbs of R134a. This is a difference of only 0.14%. There were also some important differences between the new system and the proof-of-concept one that we need to highlight.

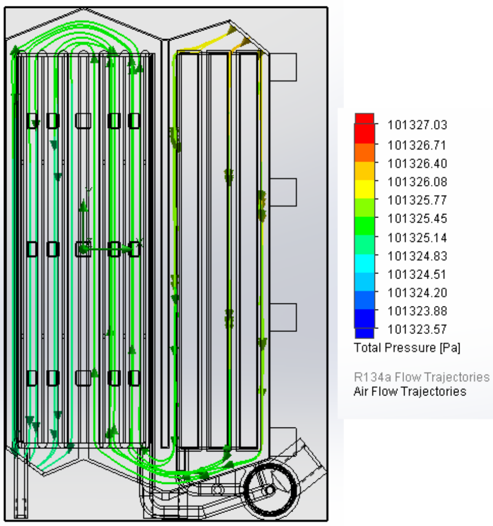

Figure 22. Air pressure results from the CFD simulation

First, since the new system was much smaller, there was less heat leak. As discussed earlier, the proof-of-concept system had enough heat leak that without any R134a cooling, the outlet air was 35ºC. So, to do as much cooling as would be needed to cool 40ºC, 100% RH air to 35ºC, we cooled the air down to 28.3ºC. In the new heat exchanger, the outlet air temperature without any R134a cooling was 36.3ºC. This means that we only needed to cool the outlet air down to 30.2ºC in order to provide the appropriate amount of cooling.

Also, the smaller/lighter heat exchanger had considerably less thermal mass. The proof-of-concept system took ~6 minutes to cool the air down to an acceptable level. Our new heat exchanger required <1 minute. The only drawback of the new heat exchanger compared to the proof-of-concept system was that the outlet air temperature was slightly less stable, with temperature swings that were somewhat larger than for the proof-of-concept system. However, considering that the temperature swings were generally less than 2ºC, and were done conservatively (i.e. always holding the temperature below the maximum), we were very happy with the results from the miniaturized cooling system. With more time, we are confident that we could improve on the system, further reducing its size, weight, and the size of the temperature swings.

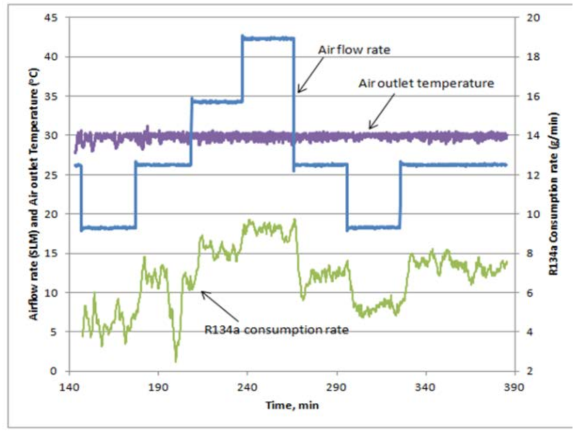

Figure 23. Results from a variable air flow rate experiment with the new compact heat exchanger

2.4. Discussion

2.4.1. Goals vs Accomplishments

TDA Research set out to develop a long duration rebreather cooling system that can cool 40°C, 100% RH inlet air to a constant 35°C, even with changing air flow rates and heat loads. To move toward this goal, we set ourselves four specific tasks for the Phase I project:

5. Analyze heat exchangers
6. Build test apparatus
7. Test proof-of-concept cooling system
8. Develop preliminary compact design for Phase II

We accomplished all four of these tasks, and also fabricated and tested the preliminary Phase II design.

Our test rig included a system for conditioning the inlet air to 40°C, 100% RH, a system for cooling the air using the boiling of R134a and evaporation of water into dry R134a gas, and a system for capturing and recycling the used R134a. Our proof-of-concept prototype cooled the air in ~6 minutes, held a constant outlet air temperature for the full four hour test, and maintained pressure in the 30 psig heat exchanger, which prevented moisture in the air from freezing on the surface (see Figure 14). We performed a variety of experiments, including constant air flow rate and variable flow rate and showed that the R134a consumption rate could accommodate a large range of air flow rates (we tested as low as 20 L/min and as high as 62 L/min as shown in Figure 17 and Table 1) and could accommodate changing air flow rates, all while maintaining constant outlet air temperature (see Figure 16 and Figure 17).

Prior to starting the project, our thermodynamic calculations (which assumed 100% efficiency) estimated that we would need 3 lbs of R134a consumable to cool 30 L/min of 40°C, 100% RH air down to 35°C. During our experiments, we ended up using 3.34 - 3.83 lbs of R134a for four hours of cooling. Most of this efficiency drop was likely due to imperfect insulation in the R134a lines and around the heat exchanger and imperfect insulation around the heat exchanger itself. From our steady-state test without any cooling, we knew that heat leaked out of our system, this indicated that during active cooling, heat leaked into our system (especially into the cold 30 psig heat exchanger). This will be less of an issue when the cooling system is contained within the entire rebreather system (as opposed to exposed to ambient) and improving the insulation will be straightforward to fix in future work.

While working with the proof-of-concept system, we were able to simplify the TCV, moving from proportional control to binary open/close control. We also simplified the PCV by moving from a pneumatic badger valve to an electronic solenoid valve, which was considerably smaller, lighter, and cheaper without sacrificing performance. We also improved the water capacity of the low pressure heat exchanger by changing the wicking material from paper towels to high capacity PVA chamois. This increased water capacity by >3.4x. The PVA chamois was also considerably more robust than the paper towel, meaning that it should last through many uses.

The proof of concept system worked extremely well. It handled flow rates as low as 20 L/min and as high as 62 L/min while maintaining constant outlet air temperature. We also ran some variable air flow rate experiments, including one with seven different air flow rates. The proof-of-concept prototype handled all of these excellently, and varied its R134a consumption rate to handle the changing heat loads of the changing air flow rates, while maintaining constant outlet air temperature (shown in Figure 17).

After testing our proof-of-concept system, we designed a preliminary prototype which combined the 30 psig and low pressure heat exchangers into a single unit that was 54% smaller and 78% lighter than the two heat exchangers (combined unit shown in Figure 19). It also used a smaller and lighter TCV and a smaller, lighter, and fully passive PCV. We used additive manufacturing to print the prototype design out of aluminum and replaced the proof-of-concept cooler with the new prototype in our test rig. Overall, the cooling unit was 50% smaller and 76% lighter than our first generation cooling prototype.

We tested the new compact heat exchanger with the same variable air flow rate protocol used for the proof-of-concept prototype system (see Figure 23). The results from that experiment showed that the compact heat exchanger was even better than the original proof-of-concept one. They both were able to quickly adjust the R134a flow rate as the air flow rate changed, so as to maintain constant outlet air temperature. And both consumed an almost identical amount of R134a (only 0.14% difference). Additionally, the compact system was able to cool the air even faster than the original proof-of-concept prototype (going from ~6 min cooling time to <1 min). The compact heat exchanger did exhibit somewhat larger temperature swings in the outlet air, however since the swings were conservative (i.e. always stayed below the target temperature), this was not too big of an issue. With additional design work, we can likely shrink the temperature swings while further reducing the size and weight of this cooling system.

2.4.2. Cost Estimate

TDA's proof of concept cooling system (using the TCV and PCV from the compact version) cost approximately $1,421.33 to build as shown in Table 3. The most expensive component was the custom low pressure heat exchanger (~$600), then the PCV ($285), followed by the TCV ($265) and finally the 30 psig heat exchanger ($217),. Everything else combined including fittings, R134a, PVA chamois, etc., only cost $54.33. The compact heat exchanger system was considerably more expensive. The heat exchanger was 3D printed from aluminum (a fairly new and expensive process), and there were three other custom parts 3D printed from ABS. The total system cost was $2,562.28 with the heat exchanger costing $2,085.67, and the three plastic 3D printed parts costing an additional $165.

As this system moves toward commercialization, we will work to minimize these costs. The 3D

Table 3. Summary of cost estimates for the cooling system

Gen 1 Cooling System Cost				Gen 2 Cooling System Cost				Optimized Cost Estimate			
description	number	price	total price	description	number	price	total price	description	number	price	total price
combined heat exchanger	1	$817.00	$ 817.00	combined heat exchanger	1	$2,085.67	$2,085.67	combined heat exchanger	1	$150.00	$ 150.00
Nitra TCV	1	$285.00	$ 285.00	Asco TCV	1	$ 76.28	$ 76.28	Cheaper TCV	1	$ 20.00	$ 20.00
Aalborg PCV	1	$265.00	$ 265.00	Swagelok PCV	1	$ 181.00	$ 181.00	Cheaper PCV	1	$ 20.00	$ 20.00
PVA	0.17	$ 9.84	$ 1.64	PVA	0.17	$ 9.84	$ 1.64	PVA	0.17	$ 9.84	$ 1.64
orifice	1	$ 7.80	$ 7.80	3D printed door	1	$ 85.00	$ 85.00	injection molded door	1	$ 3.00	$ 3.00
12 oz R134a canister	4	$ 4.49	$ 17.96	3D printed nozzles	2	$ 40.00	$ 80.00	injection molded nozzles	2	$ 3.00	$ 6.00
mesh screen	0.33	$ 20.80	$ 6.93	12 oz R134a canister	4	$ 4.49	$ 17.96	12 oz R134a canister	4	$ 4.49	$ 17.96
miscellaneous parts	1	$ 20.00	$ 20.00	orifice	1	$ 7.80	$ 7.80	orifice	1	$ 7.80	$ 7.80
		total =	**$1,421.33**	mesh screen	0.33	$ 20.80	$ 6.93	mesh screen	0.33	$ 20.80	$ 6.93
				miscellaneous parts	1	$ 20.00	$ 20.00	miscellaneous parts	1	$ 20.00	$ 20.00
						total =	**$2,562.28**			**total =**	**$ 253.33**

printed metal heat exchanger can be broken down into multiple components, where each one can be cast. Making the molds for the cast tends to be very expensive (tens of thousands of dollars), but the individual cast parts can be very cheap (tens of dollars). If we produce enough of them that the cost of the molds is spread over a large number of parts, then we can significantly reduce the cost of the system. Similarly, we can reduce the cost of the 3D printed plastic parts by injection molding them. Lastly, we chose a PCV and TCV that were readily available and easy to install. There are much cheaper PCVs and TCVs available that may work with our system (can be closer to $20). In Phase II, we will investigate using these alternative valves. If we assume cast parts for the heat exchanger costs $150, injection molded plastic parts cost $9, and the PCV and TCV cost $20 each, then the entire on-demand cooling system would only cost $253.33 (with $190 from the heat exchanger, PCV and TCV). At this cost, the benefits of our on-demand cooling system clearly outweigh the added cost. In Phase II, we will work to reduce the cost of our cooling system design to meet these goals.

2.4.3. Commercialization plans

Mine rescue rebreathers are essential pieces of safety equipment that allow rescue workers into mines after a disaster such as a collapse or a fire to rescue mine workers. The rebreathers supply four hours of breathable air, but the ice based cooling currently employed often doesn't last a full four hours. In fact, according to Dräger's own data, when someone is breathing at 30 L/min in an ambient environment of 30°C, the ice is only able to keep the air below 35°C (the maximum allowable temperature for humid air over a four hour period) for less than 2.5 hours and it can't keep the air below 40°C for three full hours (see Figure 7).

Most of the reason that ice struggles to work effectively is that cooling system doesn't have any feedback control. In the beginning of the four hours, the ice is cooling the air below 35°C, which is unnecessarily using up some of its cooling capacity. So, by the end of the four hours, it is no longer capable of keeping the air at 35°C. With the exact same cooling capacity as the block of ice, by better controlling the cooling so that the air is kept at a constant 35°C, the air can be kept at a safe temperature for longer because cooling capacity won't be wasted.

This is TDA's approach for our on-demand cooling system. We replace the ice, whose cooling rate can't be controlled, with R134a, which we can meter in using a thermostatic valve. This lets us keep the temperature of the breathing air at a constant 35°C, so that with the same cooling capacity, we can maintain a safe temperature for a longer period of time.

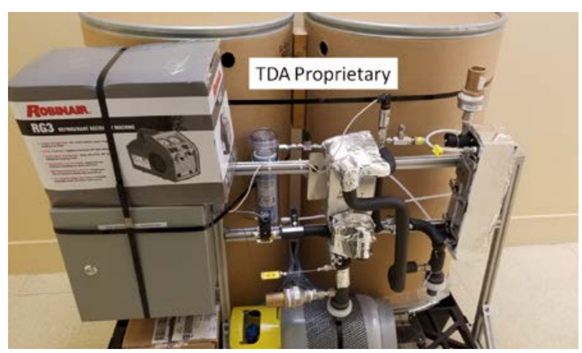

Figure 24. Photograph of test rig that TDA sent to Avon Protection Systems. Rig includes proof-of-concept cooling prototype and R134a reclamation unit (with most of the mass and volume taken up by the R134a reclamation unit)

TDA is in talks with Avon protection about its on-demand cooling technology. Avon purchased a copy of our proof-of-concept test rig (see Figure 24) so that they could test it in conjunction with a rebreather that they are designing. This test rig was very similar to the one TDA built (see Figure 11) except that the air conditioning section was removed, since Avon will be supplying air from its rebreather prototype. We are now in talks with Avon to supply them with the new miniaturized heat exchanger prototype.

Avon has considerable experience and market expertise in the personal protective equipment (PPE) space. They sell PAPRs (such as the Avon EZAir+), escape respirators to protect emergency personnel against chemical-biological-radiological-nuclear (CBRN) scenarios (such as their NH15), and SCBA equipment (such as their ST50). They are currently working on a rebreather, which will need cooling technology that can improve upon the ice cooling that is used in most other rebreathers. They are looking to TDA's on-demand cooling technology so that their rebreather can supply 35°C air over a full four hour mission.

2.4.4. Phase II Plans

In a Phase II project, we will work to further optimize the system and to test it under more realistic conditions. System optimization will be focused around reducing size, weight, and cost. More realistic test conditions will better take into account the effect of the rebreather equipment on the air temperature and will better simulate human breathing and metabolism.

2.4.4.1. Optimizing Cooling System

This will include two main thrusts: optimizing the refrigerant and optimizing the equipment. R134a was chosen as the refrigerant because it is cheap, widely available in the US, non toxic, and non-ozone depleting. However, there are refrigerants with a higher latent heat of vaporization and refrigerants with a lower global warming potential (GWP). For instance, R152a has a heat of vaporization that is 58% higher than that of R134a and a GWP that is 91% lower. The drawback of R152a is that it is slightly flammable (which is why we chose R134a in Phase I). However, with the proper flame suppressant (such as 3M's Novec 1230), we should be able to eliminate the flammability while still improving upon R134a's latent heat of vaporization and GWP. Since the consumable constitutes the majority of the weight of our system, finding a more effective refrigerant will significantly improve our cooling technology while also minimizing global warming potential.

We will also work to improve our heat exchanger design. Moving from the proof-of-concept to the printed design reduced the size by 50% and the weight by 76%. In fact, the heat exchanger without the refrigerant already takes up less room than the ice pack that it is replacing. However, considering that we only had a very limited amount of time to design, fabricate, and test our second generation cooling unit, we believe that we can reduce the size and weight further, while also reducing cost with the much greater amount of time and resources that will be afforded in a Phase II project. Our heat exchanger has a specific area of ~60 m^2/m^3 while advanced compact heat exchangers can approach 400 m^2/m^3. During Phase II, we will work to improve our design so that it has a higher specific area and lower weight. The Phase II heat exchanger will be designed to be built out of multiple parts instead of printing it as one single part to improve manufacturability and reduce costs. Finally, we will also investigate alternative PCVs and TCVs focusing on reducing cost and weight.

2.4.4.2. Testing Cooling System In Realistic Conditions

Importantly, we will work to test our cooling technology under more realistic conditions. We are in talks with Rohan Fernando at the National Personal Protective Technology Laboratory (NPPTL) about partnering to test our cooling technology using their rebreathers and automated breathing metabolic simulator (ABMS). TDA will modify the rebreather and/or our cooling system so that they can work together. Instead of using heated and humidified air to simulate the air exiting the CO_2 scrubber as done previously, the ABMS will exhale air at the proper temperature and humidity into the rebreather. This air will also have the correct CO_2

concentration to simulate typical exhaled air. The excess CO_2 will then be scrubbed out in the rebreather (the exothermic process that generates all of the heat). Some of the excess heat and moisture from this process are rejected to the rebreather because of its thermal mass. The remaining heated air that enters TDA's cooling unit will have the same thermodynamic properties that the cooling unit will experience in the field. This way, the test of our cooling system will be conducted under more realistic conditions that better simulate the performance of a rebreather during actual use.

2.5. Disclosure

This report was prepared by Dr. Girish Srinivas, Dr. David Eisenberg, Nathan Weinstein and Jeffrey Martin of TDA, who contributed the primary calculation, design, fabrication and testing efforts during the Phase I project. Additional minor effort was contributed by other TDA support personnel.

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