DESIGN AND MODELING OF DRONE POD THERMAL MANAGEMENT SYSTEM

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Abstracts should be shorter, just to entice the reader. I would recommend shortening it.

ABSTRACT

A solicitation put forward by the Department of Defense requested proposals for a pod enclosure, with a thermal management system that could be mounted on a drone to house electronic equipment. The team modeled the shell as a rectangular structure with the given dimensions, evaluating several different designs, each utilizing a unique material and shell thickness. A thin shell and a thick shell design, consisting of varying thicknesses of a Fiberglass Reinforced Plastic (FRP) outer layer, internally lined with rigid foam insulation board, were selected for further analysis based on superior demonstrated thermal properties. Using the shell designs, expected heat loads for the thermal management system were calculated, based on an expected outdoor operating temperature range of -20°C to 50°C, and a given heat load due to the electronic equipment inside the pod. Modeling the thermal management system as an ideal refrigeration system, three different refrigerants were analyzed, yielding ideal compressor work inputs, refrigerant mass flow rates, and coefficients of performance for each shell design. Based on the ideal analysis, a working fluid of ammonia was selected, and heat exchangers were designed to fulfill the heat transfer requirements of the system. A condenser and evaporator both made out of aluminum were selected, and further analyzed to estimate pressure drop in the working fluid through each heat exchanger. The refrigeration cycle was then re-evaluated, including the losses in the heat exchangers, as well as a compressor with 80 percent efficiency. The resulting actual cycle for the worst-case heat load corresponding to each shell design was modeled, allowing the team to make a recommendation for overall design. The final design recommendation presented by the team includes a 15 mm thick shell, with 5 mm of FRP, and 10 mm of rigid foam insulation. The heat exchangers recommended are coiled ⅜" aluminum tubing with forced convection across the tubes. Using ammonia as the refrigerant, this pod design is cable of protecting and maintaining the equipment necessary while satisfying the specifications of the solicitation.

What Dimensions

Extra space

Take out. Should be in final design section

INTRODUCTION

The Department of Defense issued a solicitation, identifying design requirements for a drone mountable pod that is capable of housing electronics equipment, while also maintaining its internal temperature. The drone pod shell was given explicit dimensions of 7 feet long by 16 inches tall and 15 inches wide. The internal pod temperature was required to be maintained at 65°F +/- 5°F with an estimated heat load of 0.5 - 2 kW generated by the internal electronics. The solicitation also specified the pod maximum weight, including all components of the thermal management system, ~~could not exceed~~ of 125 lbs.

With these specifications in mind, the team designed two pods with different shell designs and evaluated them as options for satisfying the solicitation. Three different working fluids were analyzed, generating ideal requirements for each shell design. Heat exchangers were designed to fulfill these requirements, while taking care to balance weight with heat transfer capability. Finally, after identifying a compressor, the actual refrigeration cycle of the designed thermal management system was modeled, including heat transfer in and out of the shell for an outside temperature range of -20°C to 50°C. Based on the modeling of the actual cycle, a final design was identified.

Mention Δ Pressure drops

NOMENCLATURE

COP - Coefficient of performance
D - diameter
k - thermal conductivity
h_c - convection coefficient
A - surface area
L - characteristic length
Cp - Specific heat
μ - viscosity
ρ - density
h - specific enthalpy
$\dot{Q}$ - heat transfer
$\dot{W}$ - work

Good

Already told me

POD SHELL DESIGN

The pod shell dimensions given by the solicitation specify an 84" L x 16" H x 15" W enclosure for the pod. Initially the team evaluated for a shell entirely made of aluminum or steel, but the high thermal conductivities, as well as the densities of these materials, made them unsuitable for constructing the drone. As a lightweight alternative, shell designs consisting of Fiberglass Reinforced Plastic (FRP) and rigid foam insulation board were compiled and analyzed.

The provided dimensions allowed the team to calculate allow for surface area and volume calculations

Check sentence.

for each thickness. By using the volume and the density of each material, an estimation of shell mass for each variation was calculated, allowing the team to balance thermal properties against the principal concern of mass.

Thermal performance estimations for each shell variation were formulated for outside temperature ranges between -20°C and 50°C. The internal pod temperature was assumed to be uniform at 65°F. Using Equation 1 the thermal resistance of the shell was calculated.

Already told me.

Why? Any consequences?

$$\frac{1}{R_{tot}} = \frac{L}{kA} + \frac{1}{h_c A} \qquad (1)$$

The thickness of the shell was varied, and one third of the thickness was designated as FRP, while the remaining two-thirds were designated as rigid foam insulation. The FRP thermal conductivity was assumed to be 0.057 W/mK [SOURCE] and the thermal conductivity of rigid foam was assumed to be 0.026 W/mK [SOURCE]. The convection coefficient of outside air was calculated to be 133 W/m²K using an estimated operating velocity of 250 m/s, and the heat transfer relationships between calculated Reynolds (Eq. 2), Prandtl (Eq. 3), and Nusselt numbers (Eq. 4).

Any error in these values?

$$R_{eD} = \frac{\rho V D}{\mu} \qquad (2)$$

$$\Pr = \frac{\mu * Cp}{k} \qquad (3)$$

$$Nu = \frac{h_c * D}{k} \qquad (4)$$

Resulting thermal resistance, in conjunction with the temperature difference between the internal space and outside environment (Eq. 5) was used to calculate the heat transfer through the shell for each thickness in the exterior temperature range.

$$\dot{Q} = \frac{\Delta T}{R_{tot}} \qquad (5)$$

Based on mass considerations, in conjunction with the thermal performance, a thin shelled model with a total thickness of 10 mm and a thick shelled model of 15 mm were identified for further analysis. At the coldest temperature of -20°C, the thin shelled model rejected 338.8 W of heat while the thick shelled model rejected 227.8 W of heat. At the highest temperature of 50°C, the thin shelled model absorbed 280.5 W of heat while the thick shelled model absorbed 188.5 W of heat. A plot of heat transfer for thickness vs. temperature can be seen in Fig. 1 below. The mass of the thin shelled pod was found to be 16.1 kg while the mass of the thick shelled pod was calculated to be 24.1 kg.

Graph? Seems very light for Pod mass with a thickness of 1 and 1.5 cm

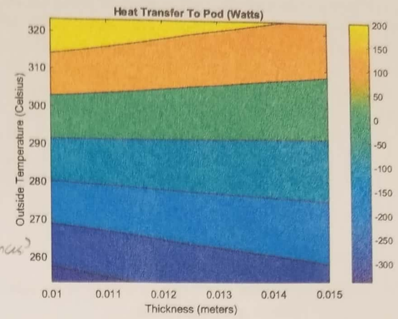


FIGURE 1 – HEAT TRANSFER FROM OUTSIDE TO POD FOR FIBERGLASS REINFORCED PLASTIC SHELL DESIGN

Use FRP

REFRIGERANT

Three different refrigerants, R-410a, R-134a, and Ammonia, were initially selected as viable options as a working fluid in the thermal management system. The total heat load for each shell case was calculated by adding the specified heat load of the internal electronics to the heat transferred in or out of the shell. Using an iterative analysis of the ideal refrigeration cycle, the enthalpy of each cycle state point was identified for each refrigerant. By relating the enthalpies at different state points (Eq. 6), the coefficient of performance for each refrigerant was found for the cycle process occurring at every temperature in the estimated operating range. This process was repeated for both the thin and thick shelled pod design.

Why?

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{\dot{Q}_l}{W_{comp}} \qquad (6)$$

A plot of the coefficient of performance vs. outside environment temperature (Fig. 2) below shows that for any given temperature the refrigerant Ammonia has a higher calculated COP when compared to R-410a and R-134a. This held true for both the thin and thick pod shell designs.

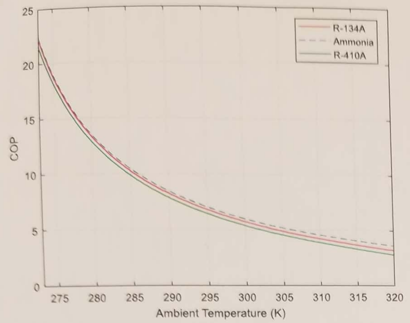


FIGURE 2 – COP VERSUS AMBIENT TEMPERATURE

The ideal cycle analysis was also used to evaluate the mass flow rates of the refrigerant required at any given temperature by relating the heat transferred into the system to the change in enthalpies across the evaporator (Eq. 7).

$$\dot{m} = \frac{\dot{Q}_l}{h_1 - h_4} \tag{7}$$

The mass flow rate data also showed that, when compared to R-410a and R-134a, Ammonia exhibited the most desirable mass flow rate for any given temperature, in either shell design.

The team weighed environmental and safety concerns of the refrigerants as well. All three refrigerants have a zero rating for Ozone Depletion Potential, but Ammonia also holds a zero rating for Global Warming Potential, while R-410a and R-134a both have a GWP above 1400 [LINDE GAS SOURCE]. While ammonia can be more harmful than either R-410a or R-134a, the concern was considered low, as the drone will be unmanned, and any leaks are likely to disperse without significant human contact.

Ammonia kills when not handled perfectly. Prove this

Considering the above information, the team selected Ammonia as the proposed refrigerant for the thermal management system. The lower mass flow rates for each given temperature resulted in a lower compressor work requirement, which ultimately led to higher COP. The higher COP then gave the team more flexibility in other areas of design such as compressor size and weight, and heat exchanger design.

HEAT EXCHANGER DESIGN

Two heat exchangers were designed to function as the condenser and evaporator in the ideal refrigeration cycle. Using the maximum heat load for each shell thickness, the maximum required mass flow rate of ammonia was calculated using Eq. 7. With the mass flow rates known, the relationship between the Prandtl number (Eq. 3), Reynolds number (Eq. 7), and Nusselt number (Eq. 8) could be used to find the convection coefficient of the refrigerant. This same relationship was used to calculate the convection coefficient of the flow over the heat exchanger pipes for both the condenser and evaporator. The flow of air over the condenser was estimated to be equivalent to the speed of the drone, 250 m/s. The flow of nitrogen over the evaporator is set at 10 m/s, facilitated by a small fan to increase the velocity where the nitrogen flows through the heat exchanger.

$$R_{eD} = \frac{4\dot{m}}{\pi D \mu} \tag{8}$$

$$Nu = 0.27 R_{eD}^{0.8} * Pr^{n} \tag{9}$$

where n is 0.4 for a cold fluid being heated and 0.3 for a hot fluid being cooled

For ease of calculation, the thermal properties of the refrigerant (density, viscosity, and thermal conductivity) were averaged between the two state points at the inlet and outlet of the heat exchangers. In practice this assumption will not hold, and experimental data would be needed to confirm the properties at any given point.

You can get better numbers for this online

The team analyzed copper and aluminum piping as possible materials for heat exchanger piping. An equivalent thermal resistance was found using the calculated convection coefficients, thermal conductivity of the piping, and a ⅜" piping size. Specifying a dT value of 20C for the evaporator, and 10°C for the condenser, the thermal resistance was used in Eq. 9 to find the necessary length of piping in each heat exchanger for both shell cases.

$$L = \frac{\dot{Q} * R_{eq}}{\Delta T} \tag{10}$$

Accounting for the density of the pipe material, the mass of the heat exchangers was also calculated for each case, allowing the team to select a heat exchanger design while remaining sensitive to mass restrictions.

For both pod shell cases, the team selected aluminum tubing, based not only on mass considerations, but also its compatibility with ammonia as a working fluid, and its excellent heat transfer capabilities. Copper was revealed to have slightly better heat transfer capabilities; however, after inquiring about the chemical properties of ammonia, it became clear that ammonia would corrode the copper piping and therefore shifted the decision towards ammonia [SOURCE]. Table 1 below shows the relevant values for each heat exchanger in both shell cases.

Graph? I want numbers and stuff

Aluminum?

(TABLE WITH VALUES FOR Mdot, Qdot, HX mass, Pipe Length, etc…)

IDEAL AND NONIDEAL CYCLES

The thermal management system requires the incorporation of a compressor, either a traditional or a variable frequency compressor. A traditional compressor has one constant power input and outlet pressure. Consequently, the cycle must operate at the worst-case scenario at all times to ensure that heat can be rejected to any ambient temperature. The required power for the traditional compressor is the same as the required power for the worst-case scenario for the variable frequency compressor. Our simulations estimated the required input to be approximately 673 Watts. In ambient temperatures lower than the worst case, the traditional compressor would operate with the same power input while switching on and off, operating just enough to maintain the required pod temperature. Unlike the traditional compressor, the variable frequency compressor adjusts the power input to maintain a more efficient cycle. The power requirement decreases from the worst-case scenario for lower outside temperatures, and, at the same time, the COP increases. Our team made use of a variable frequency compressor to guarantee an efficient cycle in all scenarios. → Good.

To estimate the general relationships between temperature, specific entropy, pressure, and specific enthalpy, ideal cycle diagrams for both the thick shell and thin shell scenarios were plotted for the best and worst case outside temperature conditions. Figures 3 and 4 depict these relationships, which include isentropic compressor operation, isobaric heat transfer across both the condenser and evaporator, and isenthalpic pressure drop in the expansion valve.

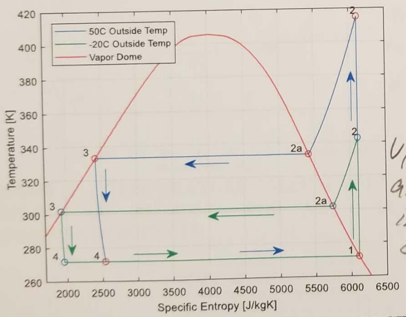


FIGURE 3 – TEMPERATURE VERSUS SPECIFIC ENTROPY OF AMMONIA FOR AN IDEAL REFRIGERATION CYCLE

Nice graph. Up the axis label in font a bit

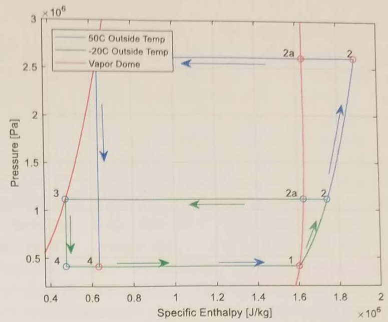


FIGURE 4 – PRESSURE VERSUS SPECIFIC ENTHALPY OF AMMONIA FOR AN IDEAL REFRIGERATION CYCLE

To create a more accurate model of the cycle, the team investigated several assumptions made in the creation of the ideal cycle. To start, when modelling the ideal cycle, we assumed the heat exchanging processes, over the condenser and evaporator, to be isobaric. In practice, this assumption does not hold and thus the head loss over the heat exchangers was approximated. From a manipulation of Bernoulli's equation, ignoring changes in both kinetic and potential energy, head loss (H_L) can be defined by the equation,

$$H_L = \frac{\Delta P}{\rho g} = f \frac{L}{D} \frac{V_{avg}^2}{2g} + K_L \frac{V_{avg}^2}{2g} \quad (11)$$

Good

The rightmost equation solves for head loss accounting for major and minor losses, the first and second term in the sum respectively. For this analysis, minor losses were ignored as they ~~requires~~ require more assumptions and does not further enhance the model. The friction factor (f), used to account for major losses, is determined differently for varying Reynold's numbers. For our heat exchangers, our team worked with smooth turbulent flow, or Reynolds numbers above 2300 and below 10^5. In this case, the friction factor can be calculated using Blasins equation,

Nomenclature

$$f = \frac{0.316}{R_{eD}^{0.25}} \quad (12)$$

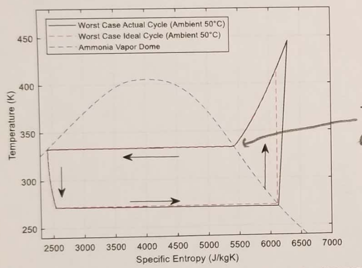
Good

Higher Reynold's numbers are not encountered in the heat exchangers for this pod so equations for friction factor of higher Reynolds number flows are not necessary. Upon calculation of the characteristic friction factor, head loss and pressure drop were calculated using the head loss equation (Eqn. *), the characteristic friction factor and a rough estimate of average density. Utilizing an average density is not necessarily leading to perfectly accurate results, but in an effort to provide a well-rounded approximation of the cycle, multiple assumptions were addressed in the ideal process and in turn to simplify the methods of accounting for inefficiencies.

The next inaccurate assumption investigated from the ideal cycle was the isentropic compression process. Ideally, a compressor results in no change in entropy, but in practice there is entropy generation due to friction and heat from operation. To address this issue, we researched experimentally calculated isentropic efficiencies and according to {insert source info}, a typical compressor efficiency ranges from approximately 0.75 to 0.85 so an averaged value of 0.80 was used for the analysis (SOURCE NUMBER). The isentropic efficiency used calculate the actual state encountered at the end of the non-ideal compressor process, is defined as;

$$\eta_{comp} = \frac{w_{ideal}}{w_{actual}} = \frac{h_i - h_{fs}}{h_i - h_{fac}} \qquad (13)$$

With the head loss/pressure drop and compressor efficiency calculations, more accurate plots of temperature versus specific entropy and pressure versus specific enthalpy were compiled as seen in Fig. 5 and 6. These plots illustrate the range of ammonia properties throughout the vapor compression refrigeration cycles.



These should be greater pressure drop from HL.... maybe 10-15%

FIGURE 5 – TEMPERATURE VERSUS SPECIFIC ENTROPY OF AMMONIA FOR AN ACTUAL REFRIGERATION CYCLE

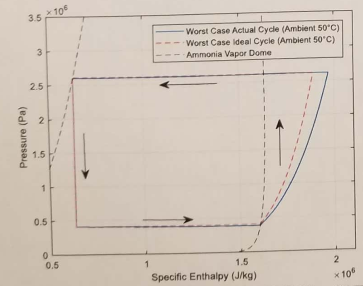


FIGURE 6 – PRESSURE VERSUS SPECIFIC ENTHALPY OF AMMONIA FOR AN ACTUAL REFRIGERATION CYCLE

RECOMMENDATION

After the iterative calculation procedure and analysis regarding refrigerant types, shell materials, pipe materials, and their effects on system weight and efficiency, final design parameters were identified. A pod shell consisting of 3.3 mm Fiberglass Reinforced Plastic (FRP) and 6.7 mm polyurethane foam board. This material and thickness combination produce the most lightweight option of 16.09 kg for the shell while still limiting heat transfer from the surrounding environment to between -340 and 280 Watts depending on the surrounding temperature. For the condenser and evaporator heat exchangers, aluminum ⅜" piping provided the best results regarding length and mass. The two heat exchangers total to 60.6 meters of piping and a total mass of 3.67 kg. With both the shell and heat exchangers, the total mass of the pod comes to 19.78 kg or 44 lbs. This is well within the necessary 125 lb or 56.7 kg weight criteria and allows ample room to factor in the fan, compressor mounting system, and transfer tubing masses. This system will operate with a minimum COP of 3.39 and since ammonia is an excellent refrigerant for thermal heat transfer as previously discussed, the maximum mass flow rate of 0.0024 kg/s translates to a maximum required power input of 673 Watts to the compressor.

how Many?

This weight seems very small. I want Piping these are accurate