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PASSIVE STORAGE TECHNOLOGIES

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Passive storage of cryogens in space has been used for sometime in scientific instruments. This paper will describe some recent advances in storage technology and how passive techniques could be applied to the storage of propellants at the Space Station. The devices considered here are passive orbital disconnect struts, cooled shield optimization, liftweight shields and catalytic converters.

Cooled Shields

Cooled shields can greatly reduce the tank heat load in cryogen storage systems. This is the case for both passive and refrigerated systems. In passive systems, the enthalpy of the effluent gas is used to cool shields embedded in the insulation and thus intercept the heat before it gets to the tank. In refrigerated systems, the shields and tank are cooled by an external refrigerator. The use of cooled shields can reduce the overall refrigeration power.

The analysis presented in this section are given in a parametric form. This has the advantage of being insensitive to the exact model used for the thermal conductivity of the insulation. The analysis also applies to insulations that are penetrated by struts, plumbing, and wires. All of these penetrating devices are assumed to be attached to the cooled shields and the thermal conductivity function adjusted accordingly. We have also assumed that the insulation is multilayered insulation (such as double aluminized Mylar with silk net spacers). The analysis uses a tank temperature of 20 K and an outer shell temperature of 256 K (460°R). It should be noted that the heat loads are extremely sensitive to the outer shell temperature.

The use of cooled shields has the further advantage of allowing thicker insulation. Practical limitations limit MLI to blankets of 5 cm (2 inches) thick. Thus, an unshielded system can only have 5 cm of insulation. In a shielded system, each shield can support a blanket allowing more insulation.

The optimization analyses presented here are not the only ones that can be done, and they may not be the appropriate ones in all cases. In doing the optimization, I have not considered the mass or volume constraints on the system.

PASSIVE ORBITAL DISCONNECT STRUTS (PODS-III)

- 0 VARIABLE CONDUCTANCE-VARIABLE STRENGTH TANK SUPPORTS
  - 0 HIGH STRENGTH DURING LAUNCH
  - 0 LOW CONDUCTANCE ON ORBIT
- 0 LAUNCH AND ORBITAL CHARACTERISTICS INDEPENDENTLY SPECIFIED
  - 0 EXAMPLE
    - 0 FOR LAUNCH RESONANCE FREQUENCY 35 Hz
    - 0 FOR ORBITAL RESONANCE FREQUENCY 20 Hz
      - 0 RESULTS IN x10 LOWER ORBITAL HEAT LEAK
    - 0 WOULD IMPROVE IF ORBITAL REQUIREMENT WERE LOWER
- 0 STATUS
  - 0 A SYSTEM OF 6 STRUTS TO SUPPORT A 430 KG TANK IS BEING LAB TESTED
  - 0 THESE STRUTS ARE LIMITED BY A MINIMUM GAUGE REQUIREMENT
  - 0 STRUTS FOR A BIGGER SYSTEM WILL PERFORM BETTER

Figure 1

COOLED SHIELDS

- 0 THE USE OF COOLED SHIELDS CAN GREATLY REDUCE THE TANK HEAT LOAD
  - 0 SHIELDS ARE USEFUL IN BOTH PASSIVE AND REFRIGERATED SYSTEMS
  - 0 ANALYSIS APPLIES TO COOLED STRUTS, PLUMBING, WIRES, ETC., ALSO
- 0 THE PERFORMANCE OF BOTH SYSTEMS IS SENSITIVE TO THE OUTER SHELL TEMPERATURE

Figure 2

The analysis of the passive system is based on the method of minimum mass flow.[5] This type of analysis is appropriate for storage systems where there is no internally generated (from a scientific instrument, for example) heat load. The heat load on the tank is converted to a mass flow which then cools the shields. The mass flow is minimized with respect to position and temperature of the shields within the constraints of thermodynamics. This can be reduced to a system of 2N+1 simultaneous equations, where N is the number of shields. These equations can be easily solved by an iterative procedure. This gives the optimum locations, temperatures, and heat loads of the shields. These are given in the attached Table. The shield locations have been normalized by dividing by the total insulation thickness (not including the shield thicknesses). The temperatures are given in Kelvin. The resulting heat loads have been normalized by dividing by the heat load of an unshielded tank with 2 inches (5 cm) of insulation, this being the thickest blanket that can be easily manufactured. There are two columns of heat loads shown. The first is for a total insulation thickness of 2 inches. The second column assumes that the thickest insulation blanket (the outermost one) is 2 inches and the others are increased proportionally.

An interesting result of this analysis is that the shields are not evenly spaced in position or temperature. Rather, they are closer to the tank. This is a result of the thermal conductivity decreasing with temperature. One can also see that the heat load decreases as the number of shields is increased. The first shield results in a large reduction. There is still a significant gain by using 2 shields, but not for more than 2 shields. The relative heat load for the case of an infinite number of cooled shields is given for reference.

The analysis for refrigerated systems uses the same assumptions as above but takes a different approach. It uses the method of minimum entropy production. This method is appropriate for active refrigerated systems and for passive systems with sufficient internal heat generation. The analysis presented here is for the later case where the efficiency of the refrigeration can be ignored (Carnot efficiency is assumed). For an active refrigerated system the entropy produced by the refrigerator inefficiency would have to be included. The entropy produced at the shields and at the tank is $S_i = Q_i/T_i$ where Q_i is the heat flux absorbed at the ith shield (or at the tank) and T_i

COOLED SHIELDS
PASSIVE SYSTEMS

- 0 BASED ON METHOD OF MINIMUM BOIL-OFF
  - 0 OPTIMIZED LOCATION AND TEMPERATURE OF SHIELDS
- 0 ASSUMPTIONS: TANK AT 20 K, OUTER SHELL AT 256 K
  DOUBLE ALUMINIZED MYLAR/SILK NET INSULATION

NUMBER OF SHIELDS	NORMALIZED[1] LOCATION	SHIELD TEMP., K	RELATIVE HEAT LOAD 2" TOTAL[2]	RELATIVE HEAT LOAD 2" THICKEST[3]
0			1.0	1.0
1	.35	114	.34	.22
2	.19, .50	77, 162	.26	.13
3	.13, .32, .60	61, 120, 188	.24	.097
∞			.18	

1) TANK = 0, OUTER SHELL = 1, THICKNESS OF SHIELDS NOT INCLUDED
2) TOTAL THICKNESS OF INSULATION (NOT COUNTING SHIELD THICKNESS) IS TWO INCHES
3) THICKEST INSULATION BLANKET IS TWO INCHES THICK

Figure 3

is the respective temperature. The total entropy produced is $S = - Q_0/T_0 + Q_i/T_i$ where T_0 is the outer shell temperature and Q_0 is the sum of the Q_i's. The most thermodynamically efficient system occurs when S is minimized with respect to the location and temperature of the shields. (This will give the system that requires the least refrigeration.) This method involves a simple iterative process similar to the one used in the passive case.

The parametric results are shown in the attached Table. The values in the last columns refer to the heat load refrigeration required on the tank due to the insulation. The heat that must be removed from each of the shields is also simple to calculate but has been left off the chart for clarity. More heat must be extracted at the shields than in the storage case. In a passive system this additional cooling must be supplied by an increased mass flux caused by the instrument dissipation. The principal result of this analysis is that the optimum location of the shields is slightly farther out in the insulation and their optimum temperatures are slightly colder.

COOLED SHIELDS
PASSIVE INSTRUMENT OR REFRIGERATED SYSTEMS

- 0 BASED ON METHOD OF MINIMUM ENTROPY PRODUCTION
  - 0 OPTIMIZED LOCATION AND TEMPERATURE OF SHIELDS
- 0 ASSUMPTIONS: TANK AT 20 K, OUTER SHELL AT 256 K
  DOUBLE ALUMINIZED MYLAR/SILK NET INSULATION

NUMBER OF SHIELDS	NORMALIZED[1] LOCATION	SHIELD TEMP., K	RELATIVE HEAT LOAD 2" TOTAL[2]	RELATIVE HEAT LOAD 2" THICKEST[3]
0			1.0	1.0
1	.42	95	.19	.11
2	.26, .59	60, 141	.11	.044
3	.19, .41, .68	46, 95, 168	.082	.026

1) TANK = 0, OUTER SHELL = 1, THICKNESS OF SHIELDS NOT INCLUDED
2) TOTAL THICKNESS OF INSULATION (NOT COUNTING SHIELD THICKNESS) IS 2 INCHES
3) THICKEST INSULATION BLANKET IS 2 INCHES THICK

Figure 4

Both the passive and the refrigerated systems require good heat exchangers. These are particularly difficult to model in the passive case because the effluent gas passes through several flow regimes. It starts as laminar flow at the tank and ends as sonic flow at the exhaust nozzle. The flow also spans a large temperature range with the concomitant variation in the properties of the gas. Fortunately, there are models available to handle this problem.[8]

If shields are to be used, it is important that their mass be kept small to keep the system mass down. However, the shields must be stiff enough to meet the resonance requirements. A possible choice is to use thin aluminum sheet bonded to aluminum honeycomb.[1] We have done an analysis for a 2 m^3 (70 ft^3) instrument system. This analysis showed that a 0.13 mm (0.005") Al sheet bonded to 0.64 cm (0.25") thick Al honeycomb (1.3 cells/cm of 0.033 mm gauge) should have sufficient thermal conductivity and strength. The density of such a structure is 0.57 Kg/m^2.

COOLED SHIELDS

- 0 HEAT EXCHANGERS FOR PASSIVE SYSTEMS
  - 0 DIFFICULT TO MODEL
    - 0 LARGE TEMPERATURE DEPENDENCE OF GAS PARAMETERS
    - 0 SPANS DIFFERENT FLOW REGIMES - SONIC AT EXIT
- 0 MASS OF SHIELDS ARE IMPORTANT IN EITHER PASSIVE OR ACTIVE SYSTEMS
  - 0 USE LIGHTWEIGHT CONSTRUCTION
    - 0 THIN AL SHEET BONDED TO AL HONEYCOMB
    - 0 $\sim$0.6 KG/M^2 POSSIBLE

Figure 5

Para-Ortho Conversion

The heat load in passive hydrogen systems can be reduced by using a catalyst on the shields.[9] The converter speeds up the conversion of para-hydrogen to ortho-hydrogen. The equilibrium mixture of para (anti-parallel proton spins) and ortho (parallel proton spins) is temperature dependent. At 20 K it is >99% para and at 300 K it is 25% para. The para-ortho conversion is an endothermic reaction that is usually too slow to be of use. However, it can be speeded up by using appropriate catalysts. The heat of conversion has a maximum of 400 J/g at 100 K. This compares with the enthalpy of the gas of 900 J/g for a change in temperature from 20 K to 100 K. Thus, the reaction can be used to supply additional refrigeration to the shields. Conversion efficiencies of $\sim$100% are possible with commercially available catalysts. For example, the use of APACHI-1 would require about 100 g of catalyst for each g/s of hydrogen flow.

One way of using a catalyst would be to increase the performance of a passive system. This is most effective if the catalyst can be attached to a shield running near the 100 K peak in the heat of conversion. From the previous Tables it is seen that the 1 shield and 3 shield cases are ideal for this. An analysis of the one shield case is given in the attached Table. A 15% reduction in heat load can be achieved.

PARA-ORTHO CONVERSION

- 0 TWO FORMS OF H_2
  - 0 PARA-ANTI-PARALLEL PROTON SPINS
  - 0 ORTHO-PARALLEL PROTON SPINS
  - 0 AT 300 K - 25% PARA
  - 0 AT 20 K - >99% PARA
  - 0 PARA TO ORTHO CONVERSION IS ENDOTHERMIC
  - 0 PEAK HEAT OF CONVERSION $\sim$400 J/G, @ $\sim$100 K
    (ENTHALPY 20 K - 100 K $\sim$900 J/G)
  - 0 CONVERSION REQUIRES CATALYST
    - 0 100% CONVERSION POSSIBLE
    - 0 $\sim$100 G CAT/GH_2S^{-1} FOR APACHI-1
- 0 IMPROVED PASSIVE SYSTEM

SHIELDS	LOCATION	TEMPERATURE	RELATIVE HEAT LOAD 2" TOTAL	2" THICKEST
1	.35	106	.29	.19

Figure 6

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