

Start-Up Characteristics and Gravity Effects on a Medium/High-Lift Heat Pump using Advanced Hybrid Loop Technology

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ABSTRACT

Thermal characterization was performed on a vapor compression heat pump using a novel, hybrid two phase loop design. Previous work on this technology has demonstrated its ability to provide passive phase separation and flow control based on capillary action. This provides high quality vapor to the compressor without relying on gravity-based phase separation or other active devices. This paper describes the subsequent work done to characterize evaporator performance under various startup scenarios, tilt angles, and heat loads. The use of a thermal expansion valve as a method to regulate operation was investigated. The effect of past history of use on startup behavior was also studied.

Testing under various tilt angles showed evaporator performance to be affected by both adverse and favorable tilts for the given compressor. And depending on the distribution of liquid in the system upon startup, markedly different performance can result for the same system settings and heat loads. In this sense, the specific configuration and settings of the system are not mutually exclusive to a given performance. In general, four basic states of operation were identified which can result. It was also shown that active control of a thermal expansion valve may be used to recover from a non-optimal state.

Recommendations for future work include optimization of the evaporator port geometry and wick structure to better mitigate against detrimental effects of compressor suction. The compressor itself was identified as an area in need of technology development to better match compression ratios, suction pressure, and throughput to the specific mission requirements. It is recommended that future prototypes consider the use of an actively

controlled thermal expansion valve to mitigate against off-nominal states of operation.

INTRODUCTION

NASA initiatives for exploration of the moon and Mars will require advanced thermal control technologies to be fully realized. Accommodating the extreme environmental and power profile transients associated with a mission's cruise and surface phases pose serious challenges to the thermal design. Examples include extended sorties and science outposts near the lunar equator as well as robotic missions to the moon and Mars using advanced nuclear power sources. Such missions are associated with large diurnal environment swings and power dissipations – begging for a thermal bus design for robustness.

These future missions, such as those to the lunar equator where high sink temperatures exist, may benefit greatly or are enabled by the use of a medium/high-lift heat pump. To allow flexibility in radiator placement and orientation, it is necessary to elevate radiator temperatures by 50°C to 60°C over the equipment or payload temperature. And missions which require large radiator areas may save considerable mass, at the expense of input compressor power, by utilizing a heat pump.

The vapor compression two phase hybrid loop (VC-HTPL) technology is a heat pump system developed by Advanced Cooling Technologies, Inc., (ACT) under a NASA Small Business Innovative Research Phase I contract. ACT is currently on contract for Phase II. The work described in this paper was performed on a prototype built after the Phase I contract. The objective being to further characterize and understand the VC-HTPL system and to feedback this experience into the Phase II design and build.

The VC-HTPL system is a hybrid of a capillary and pumped system. The evaporator (also known as the heat exchanger or cold plate) which attaches to the equipment/payload to be cooled is an ACT proprietary design. It utilizes a wick structure to provide thin film evaporation off the wick surface. Menisci-based pumping is generated in this structure but is not necessarily the governing potential for flow due to the use of a downstream compressor. The pumped component of the loop is provided by a gear pump which is used to feed the evaporator with a constant flow of subcooled liquid. The other source of active pumping is from the compressor, which provides the larger source of flow potential.

The maximum heat flux is limited by the capillary limit of the wick structure. It provides the passive phase separation and is able to provide 100% quality vapor to the compressor. Passive phase separation in the wick structure permits a simpler design over active phase separators or interstage heaters associated with active systems. It provides a very low thermal resistance interface between the heat source and working fluid due to thin film, liquid-vapor interface boiling. And the active pumping component of the design, namely the compressor work, provides a way to mitigate start-up issues associated with fully passive capillary systems.

Figures 1 and 2 show a concept diagram and schematic of the VC-HTPL system, respectively. A gear pump was used to supply the evaporator in order to best accommodate off-nominal tests where insufficient subcooling exists and two-phase flow enters the pump. The compressor used was an off-the-shelf unit. It is a model G5Twin by Appion, Inc., and is shown in Figure 3. It contains an integrated, air-cooled condenser. Although it is generally used for refrigerant recovery operations, it was chosen for this prototype system because it does not require any lubricant to be added to the working fluid - a key requirement when working with a capillary based evaporator. Note that the need for compressor technology development was recognized. For this study, the compressor was treated as a “black

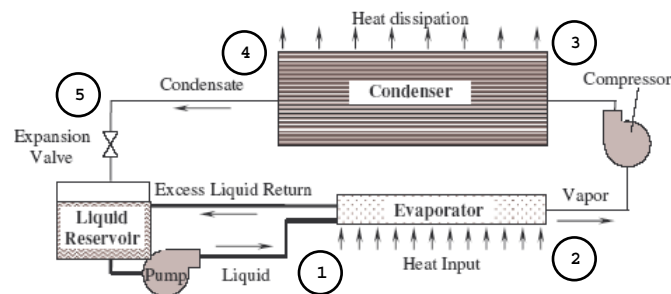


Figure 1. Concept of the VC-HTPL system. Note that the design is similar to a generic vapor compression heat pump, but the novel evaporator design involves an additional liquid loop to permit excess liquid to return to the reservoir.

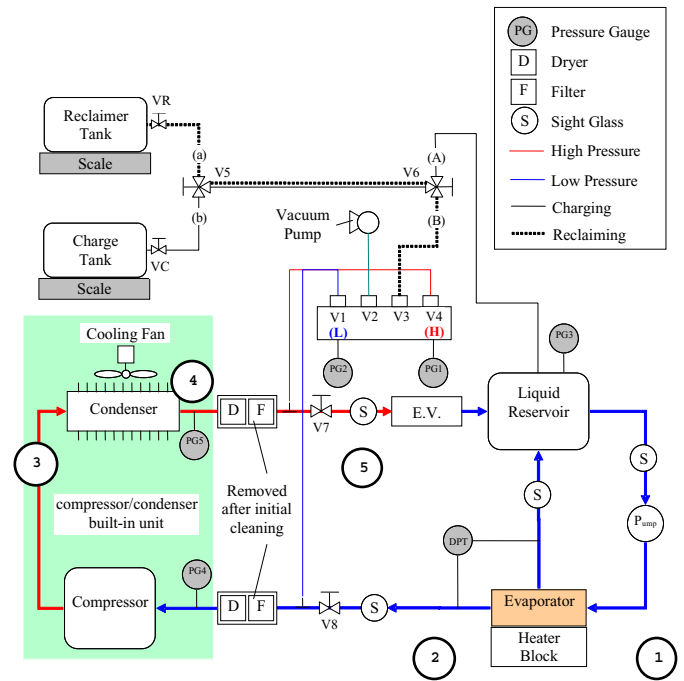


Figure 2. VC-HTPL schematic for the prototype tested.



Figure 3. The COTS compressor used during this study was an oil-free, reciprocating piston design. [Model G5Twin by Appion, Inc.].

box” component which provided a reasonably constant level of performance in which to study trends involving other parameters.

The working fluid used was HFC-134a. This fluid was chosen based on its compatibility with wetted materials, low toxicity and flammability, compatibility with the temperature range of interest, and its workable figures of merit for vapor transport, liquid transport, and compression factor. The configuration as tested is shown in Figure 4.

NOMINAL OPERATION

Prior to a discussion on transient behavior from start-up conditions and various tilt angles, a description of the steady, nominal loop operation is provided as a standard for comparison.

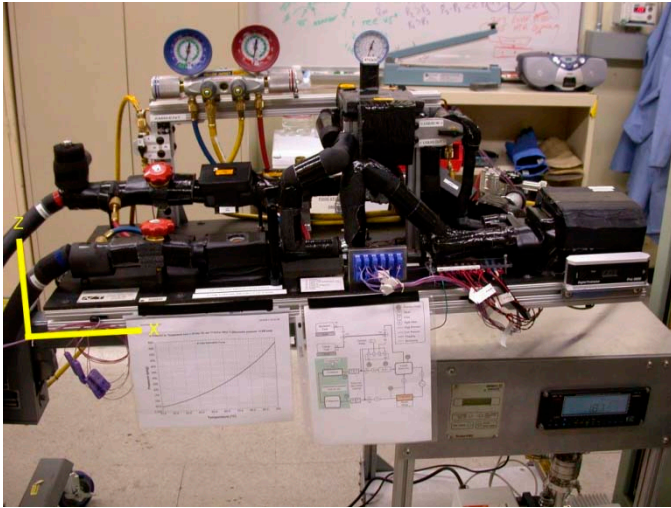


Figure 4. The VC-HTPL system as tested at JPL. Note the use of the test fixture to impose a slight adverse tilt.

BASIC HEAT-PUMP OPERATION – Generic heat pump operation may be considered to consist of the following:

- the working fluid lifts the heat load from the evaporator through sensible and latent heat removal;
- the mixture exiting the evaporator is either separated by phase or additional heating is applied to bring it to superheated vapor.
- 100% quality vapor enters the compressor, is compressed further into the superheated vapor region;
- the radiator condenses the high pressure vapor to saturated or subcooled liquid;
- the subcooled liquid is expanded into a low pressure, low quality two-phase mixture;
- the mixture is brought to saturated or subcooled liquid by either mixing in a reservoir or through additional cooling.

BASIC VC-HTPL OPERATION – The specific design condition for the VC-HTPL is similar to the heat pump described above. Figures 5 and 6 illustrate typical thermodynamic performance for the VC-HTPL during tests at JPL. Figure 5 is a simplified, somewhat idealized, temperature-entropy diagram with the saturation curve for the working fluid, HFC-134a, shown. The circled points on each graph represent a state for which either temperature and/or pressure data exists. The point labels correspond to the locations shown in Figures 1 and 2. Label 1 in the figure is the state of the liquid prior to entering the evaporator. A portion of the flow is then superheated to state 2, which is measured just after the evaporator, several inches upstream on the

vapor line. After the vapor is compressed (state 3) and subcooled (states 4 and 5), the high pressure liquid is then expanded to an unknown state of higher entropy, lower pressure, and lower temperature as shown by the dashed line. The low quality mixture then mixes in the reservoir with the liquid return flow from the evaporator. The liquid is then metered by the gear pump which feeds the evaporator. Figure 6 is the corresponding pressure-enthalpy diagram. Note that the number of pressure measurements were limited and were not generally co-located with the temperature measurements. The low pressure vapor line was one such area. The temperature measurement was made near the evaporator while the pressure measurement port was about 100 cm downstream toward the compressor. Despite the difference in the two measurement locations, the state of the fluid can be inferred and extended to also apply to the compressor inlet. The reasons for which are the low mass flow rate and near-ambient operating temperature of this line. The low mass flow rate results in negligible pressure loss, and the near ambient fluid temperature results in negligible heat transfer to the environment.

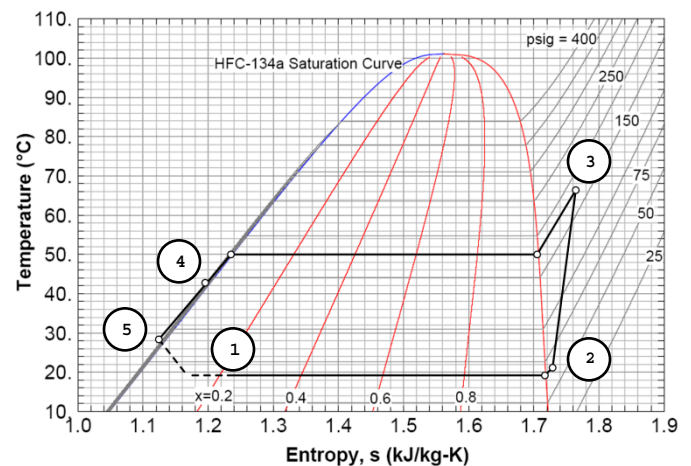


Figure 5. Temperature-entropy states for typical VC-HTPL performance (saturation curve for HFC-134a; steady-state; 240W load; 3° favorable tilt)

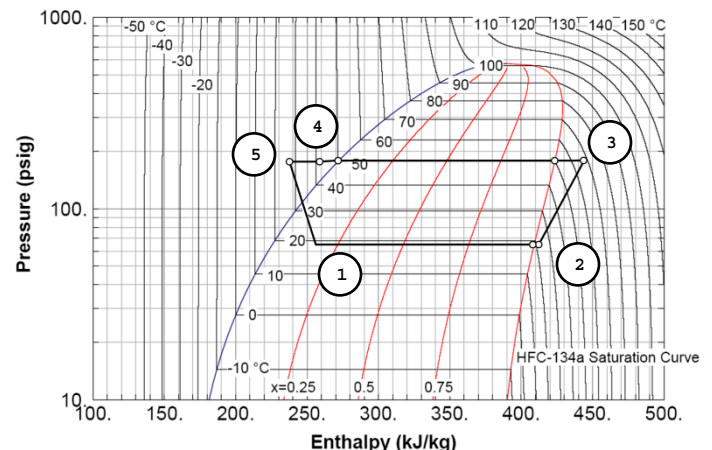


Figure 6. Pressure-enthalpy states for the same steady-state performance of Figure 5.

A typical data set was analyzed to quantify the magnitude of the flow rate and expected pressure loss in the vapor line. Nominal operation of the VC-HTPL relies on latent heat with a minimal amount of superheating to remove the imposed heat load. With a typical, applied load of 200W and evaporator temperature of 16.5 °C, the mass flow rate, $\dot{m}$, was

$$\dot{m} = 3.8 \frac{kg}{hr}$$

and mass flux, G :

$$G = \frac{\dot{m}}{\left(\frac{\pi ID^2}{4}\right)} = 6.1 \frac{kg}{m^2 s},$$

where ID is the low pressure vapor line inside diameter, 1.49 cm. The frictional and gravitational component of pressure loss from the location of the temperature measurement to the pressure port was determined from the following:

$$-\Delta P_{vap_fric} = \frac{2L f_{vap} G^2}{\rho_{vap} \cdot ID} = 2.9 \times 10^{-4} \text{ psid}$$

$$-\Delta P_{vap_grav} = \rho_{vap} \cdot g \cdot h = 4.21 \times 10^{-3} \text{ psid}$$

where the Blasius correlation for turbulent, single-phase flow in a round tube was employed to determine the Fanning friction coefficient, f_{vap} :

$$Re_{vap} = \frac{G \cdot ID}{\mu_{vap}} = 7974 \text{ (turbulent)}$$

$$f_{vap} = 0.079(Re_{vap})^{-0.25} = 0.00836$$

μ_{vap} is the dynamic viscosity of R-134a saturated vapor at 16.5 °C. The low mass flow rate of the vapor results in a negligible pressure drop from the evaporator to the compressor. Heat loss to the environment is also minimized because, under nominal operation, this line was run either at or below ambient temperature. For comparison, if single-phase liquid cooling were employed with a 5 °C temperature rise from evaporator inlet to outlet, the corresponding mass flow rate would be ~ 100 kg/hr.

DETAILED VC-HTPL OPERATION – A more in-depth look at the VC-HTPL reveals some application-specific advantages over the generic heat pump, along with imposed restrictions in order to realize them.

The evaporator promotes liquid-vapor interface boiling through an advanced, passive design. A proprietary wick structure provides passive phase separation with saturated or slightly superheated vapor exiting the evaporator. Excess liquid is returned to a liquid reservoir. Although menisci-based pumping pressure may be available from this wick structure, it does not

necessarily provide the governing potential for flow due to the presence of a compressor. The compression and condensing stage are directly analogous to the standard heat pump. But thermal expansion through a thermal expansion valve (TXV) can be used to tune the fluid temperature, pressure, and flow rate entering the evaporator. This hybrid system combines passive and active technology to provide distinct advantages over both. Passive phase separation in the wick structure permits a simpler design over active phase separators or interstage heaters associated with active systems. It provides a very low thermal resistance interface between the heat source and working fluid due to thin film evaporation off of the wick. Utilization of compressor work through active TXV control provides a way to mitigate start-up issues associated with fully passive capillary systems.

To achieve this intended, nominal design state of the VC-HTPL, certain conditions must be met which can be achieved by tuning the system. Figures 7 and 8 are plots of performance data for a case executed with a 200W load applied to the evaporator. This set of test data is presented to help exemplify system dynamics and is not necessarily typical of start-up behavior. (As will be shown, start-up behavior is dependent on the previous state of the system). The thermal resistance plotted in Figure 7 is defined as the following:

$$\text{Thermal Resistance} = \frac{\text{Evaporator Temp} - \text{Vapor Temp}}{\text{Applied Heat Load}},$$

where the Evaporator Temp is measured on the evaporator near the heater interface, and the Vapor Temp is measured on the vapor line downstream of the evaporator exit.

The ΔP of Figure 8 is the pressure differential measured between the vapor line downstream of the evaporator and the fluid inlet line prior to the evaporator.

Although a direct measure of mass flow rate was not available, a derived value was obtained using the temperature knowledge along the high pressure liquid line and an estimation of the heat loss along that line. The heat loss was estimated based on saturated evaporator conditions for which the flow rate was known. This mass flow rate, $\dot{m}_{hp,liq_line}$, was then normalized against the ideal mass flow rate based on the entire heat load being lifted by latent heat only, $\dot{m}_{2-\phi}$. The normalized mass flow rate shown in Figure 8 is defined as the following:

$$\dot{m}_{normalized} = \frac{\dot{m}_{hp,liq_line}}{\dot{m}_{2-\phi}}$$

where

$$\dot{m}_{hp,liq_line} = \frac{(\dot{q}_{sat} / h_{fg,sat}) \cdot Cp(\bar{T}) \cdot \Delta T_{hp,liq_line}}{(\rho(\bar{T})) \cdot \Delta T_{hp,liq_line}}$$

$$\dot{m}_{2-\phi} = \frac{\dot{q}_{evap}}{h_{fg}(T)}$$

During the first two minutes of operation shown in the data of Figures 7 and 8, the evaporator is being cooled by vapor-liquid interface boiling and likely also by saturated liquid or a low quality mixture flow: the temperature of the vapor line downstream of the evaporator is the same as that of the subcooled liquid entering the evaporator. The thermal expansion valve (TXV) was then tuned to obtain slightly superheated vapor conditions by decreasing the orifice opening and thus lowering its downstream pressure and temperature. The system then stabilizes to a steady-state condition given at the end of the plots. Figure 7 shows the minimum thermal resistance is obtained at this time (1:35pm) where stable liquid-vapor interface film boiling is occurring in the evaporator. And the normalized mass flow rate plotted in Figure 8 indicates near-optimum conditions with a value near unity. The ΔP between the vapor space and liquid side of the evaporator increases as the system progresses from an initial state of having liquid exiting the evaporator to the final state of slightly superheated vapor at the exit. The final ΔP value of about -0.02 psid thus corresponds to a state where the generated vapor is of pressure sufficient to offset the compressor suction pressure from exceeding the capillary limit of the wick. The data at the beginning of the run shows smaller ΔP values which were indicative of an unfavorable pressure differential which was unable to prevent liquid from being forced out of the wick structure into the vapor line. Note the effect of reducing the TXV orifice opening, done to promote the slightly superheated vapor condition, has the effect of lowering the evaporator pressure toward achieving a favorable ΔP .

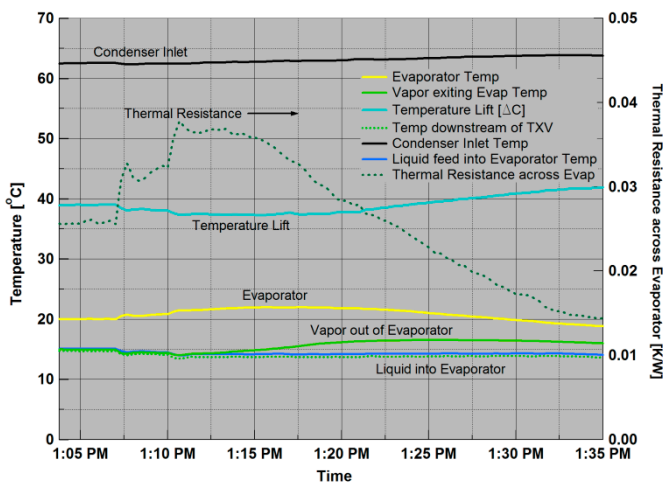


Figure 7. Component temperatures and evaporator thermal resistance when starting from a flooded condition and ending in the nominal operation mode. The transition was accomplished by adjusting the TXV. (No tilt, 200W applied to evaporator.)

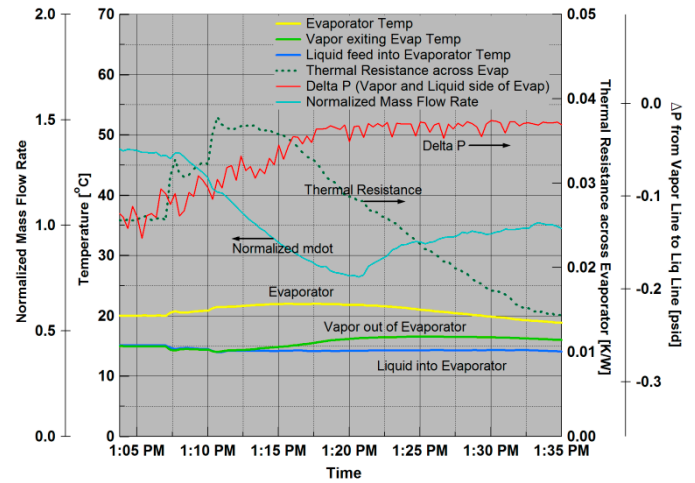


Figure 8. Pressure difference between the exit vapor pressure and inlet liquid pressure for the evaporator along with the normalized mass flow rate are shown for the same case in Figure 7.

The optimal performance characteristics of the VC-HTPL is considered to be the following:

1. minimal thermal resistance between the heat source and evaporator working fluid;
2. minimum flow rate;
3. 100% quality vapor at the compressor inlet for minimizing stresses;
4. high temperature lift for achieving radiator temperatures sufficiently greater than the environmental sink temperature or to minimize radiator size;
5. efficient expansion to near saturated liquid conditions to minimize the need for subcooling;
6. high coefficient of performance (COP).

The first three conditions have been shown to occur when we have achieved the state shown at the end of Figures 7 and 8. The high temperature lift for minimizing radiator or condenser size is largely dependent on the compressor – an area of technology development previously identified. For the system tested at JPL, temperature lifts of 35 to 45 °C were typical for the off-the-shelf single-stage compressor used. Lifts as high as 50 °C were achieved for some cases. Note that a similar system previously tested⁽³⁾ achieved lifts as high as 63 °C using a compressor of the same make and model. In general, the compressor used was not necessarily sized for the specific VC-HTPL system at JPL. The compressor provided a means to characterize the system dynamics associated with the advanced evaporator design. The compressor's capacity was about 4x larger than what was needed and the coefficient of performance reflects this. COP values on the order of 0.3 to 0.5 were measured, where

$$COP_{Heat Pump} = \frac{\text{Power Lifted}}{\text{Total Power Input}}$$

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