

ESTCP Cost and Performance Report

(PP-9706)



Non-Ozone Depleting Mobile Heat Pump

January 2000



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TECHNOLOGY CERTIFICATION PROGRAM

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LIST OF ACRONYMS

AEF	Air Expeditionary Force
AFRL	Air Force Research Laboratory
ASC	American Superconductor Corporation
COP	Coefficient of Performance
ECU	Environmental Control Unit
ESTCP	Environmental Security Technology Certification Program
GE-CRD	General Electric Corporate Research and Development Center
GWC	Global-Warming Chemical
HVAC	heating, ventilating and air conditioning
ODC	Ozone-Depleting Chemical
OEM	Original Equipment Manufacturer
USAF	United States Air Force

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- General Electric Corporate Research and Development Center, and
- American Superconductor Company.

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1.0 EXECUTIVE SUMMARY

To address the growing environmental and logistical burden posed by continued use of Class I ozone depleting chemicals (ODCs), this co-funded Environmental Security Technology Certification Program (ESTCP) and United States Air Force (USAF) Science and Technology project sought to develop a heating/cooling system using a pulse tube refrigerator with helium (a non-ODC gas) as the refrigerant. Currently, the USAF relies on fluorocarbons, such as R-22, to act as the refrigerant in the environmental control units (ECUs) used to heat and cool fixed, low-volume enclosures on military installations and mobile enclosures for military deployments. Expanded provisions and expedited implementation of the Montreal Protocol as well as ISO 14000 will compromise the future use of Class I ODCs and the subsequent mission readiness of U.S. military forces if the military continues to rely on conventional heat pumps.

In addition to reducing ODC use, project goals included a reduction in the weight of mobile ECUs and an increase in their efficiency. Air Force Air Expeditionary Force (AEF) deployments are accomplished by moving a large quantity of equipment and material to various remote locations. A typical AEF kit that sustains an 1100-man force requires shipment of 4.23 million pounds of material with 91 C141 sorties. Reductions in the size and weight of the mobile ECUs transported during AEF deployments would promote military mobility. Increased efficiency of heating, ventilating and air conditioning (HVAC) equipment can reduce deployment costs through decreases in electrical power requirements and fuel consumption.

The pulse tube refrigerator is the simplest mechanical gas-cycle refrigerator suitable for application in a commercial heat pump. The essential components of the pulse tube refrigerator are a compressor to pressurize the helium working fluid, a condenser to remove heat of compression from the working fluid, a thermal regenerator to pre-cool the working fluid to the refrigeration temperature, a cooler to absorb heat from the surroundings, and a pulse tube to provide the refrigeration effect required to cool the gas in the cooler. An orifice and ballast on the warm side of the pulse tube provide synchronization of the pressure wave and the volumetric flow rate necessary to produce the refrigeration effect, while the heat exchanger between the orifice and the pulse tube maintains the top of the pulse tube at ambient temperature. Essentially, the pulse tube system absorbs heat through the cold end heat exchanger and rejects this heat through the warm end heat exchanger. In the heating mode of operation, heat rejected in the condenser and from the warm end of the pulse tube is used to heat the indoor space. During the cooling mode of operation, refrigeration from the pulse tube inner heat exchanger is used to cool the indoor space.

This particular advanced cycle pulse tube heat pump was originally developed by General Electric Corporate Research and Development Center (GE-CRD). GE-CRD designed and fabricated a breadboard pulse tube to evaluate its potential for use in an advanced cycle heat pump. The breadboard system demonstrated that the pulse tube concept could achieve higher heating efficiencies than conventional heat pumps, resistance (strip) heating and natural gas for external temperatures less than 30°F. Cooling mode capabilities were low by comparison and posed a possible limitation to successful development of the technology. American Superconductor Corporation (ASC) was contracted by the USAF to design, fabricate and test a prototype of the advanced cycle pulse tube heat pump.

This ESTCP project consisted of two phases: an optimization study to investigate and optimize the pulse tube design parameters; and a pilot scale study to design, fabricate and test a prototype pulse tube heat

pump. The first phase studied the effects of operating pressure, pressure ratio, orifice opening, double inlet opening, valve timing, and warm end and cold end heat exchanger configuration on system performance. Based on the results, prototype components were designed. To improve system heating performance, ASC developed a rotary valve, and optimized the air-cooled heat exchangers and start-up, de-frost, and temperature control logic. A deliverable prototype heat pump was assembled with circulation ducts that allowed air to be removed from a deployable room, heated and returned to the room. Tests were conducted on the prototype to determine its heating and cooling capacities.

The test results showed the system to be an exceptional adiabatic heater, as the heat from both the work of the compressor and the ambient air are utilized by the process. The pulse tube system did not meet the cooling efficiency requirements. Therefore, no further development of this technology will occur. The project will not be taken to the stage of field testing and site demonstration. Because the prototype contained custom-made components and because full-scale implementation would require the manufacture of parts not currently commercially available, the technology is at a stage too premature to support a reliable cost assessment. No cost evaluation is included in this report.

2.0 TECHNOLOGY DESCRIPTION

Gifford and Longworth developed the original pulse tube refrigerator in 1963. Mikulin (1984) and Zhu (1990) proposed new concepts known respectively as the orifice pulse tube and double inlet pulse tube. Extensive work has been performed on the pulse tube concept in several laboratories and companies. The pulse tube refrigerator can be compared to a Stirling cooler. Performances and efficiencies comparable to Stirling or Gifford-McMahon coolers have been demonstrated in the cryogenic cooling area. This technology is especially applicable to the cryo-pump, infrared detector, and magnet cooling areas. The technology is reliable and easy to integrate.

A pulse tube heat pump is a pulse tube refrigerator with cold end heat exchanger equilibrating with ambient temperature and warm end heat exchanger operating as a heater (Figure 1). The essential components of the pulse tube refrigerator are a compressor to pressurize the helium working fluid, a condenser to remove heat of compression from the working fluid, a thermal regenerator to pre-cool the working fluid to the refrigeration temperature, a cooler to absorb heat from the surroundings, and a pulse tube to provide the refrigeration effect required to cool the gas in the cooler. An orifice and ballast on the warm side of the pulse tube provide synchronization of the pressure wave and the volumetric flow rate necessary to produce the refrigeration effect, while the heat exchanger between the orifice and the pulse tube maintains the top of the pulse tube at ambient temperature (Figure 2).

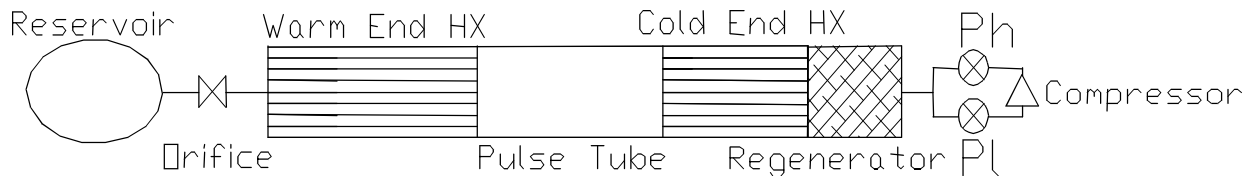


Figure 1. System Configuration of the Pulse Tube Assembly

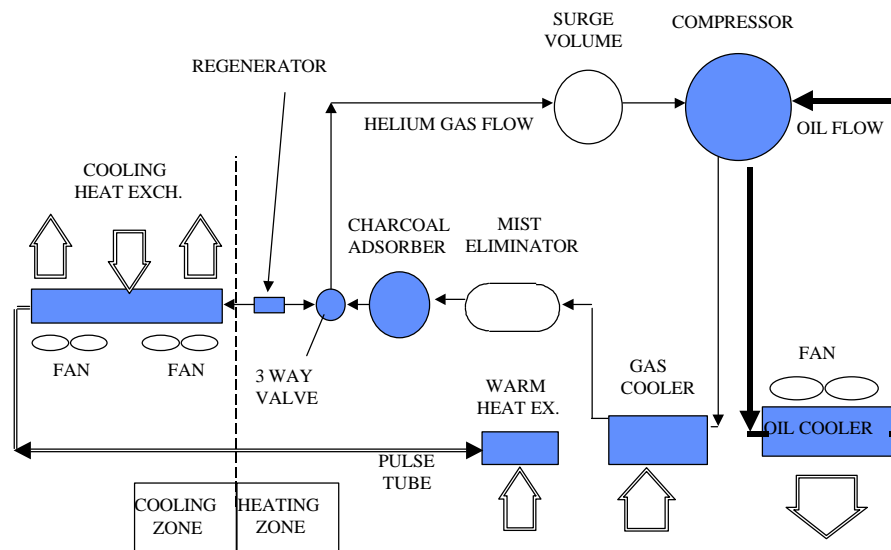


Figure 2. General Schematic of the Pulse Tube Heat Pump

Heat is continuously pumped from a low temperature (outside temperature in the heating mode, inside temperature in the cooling mode) to a high temperature. The heating and cooling effects result from periodic pressure fluctuations caused by either the movement of a compressor piston or the opening and closing of a three-way valve, pressurizing and de-pressurizing the pulse tube assembly. During the compression process heat is rejected from the warm end heat exchanger, cooling the compressed gas. During the expansion process, heat is added to the gas in the cold end heat exchanger. The expansion and compression processes within the pulse tube are realized by careful control of the operation of the valve system. Ideally, these two processes occur isothermally. The ideal performance of a pulse tube heat pump would attain the level represented by the Carnot cycle.

The working process of the pulse tube heat pump can be separated into four different steps. To better illustrate the thermodynamic cycle of the pulse tube heat pump, the time dependent pressure waves within the pulse tube and buffer obtained by the theoretical simulation are presented in Figure 3. The corresponding state points are also shown in the same graph.

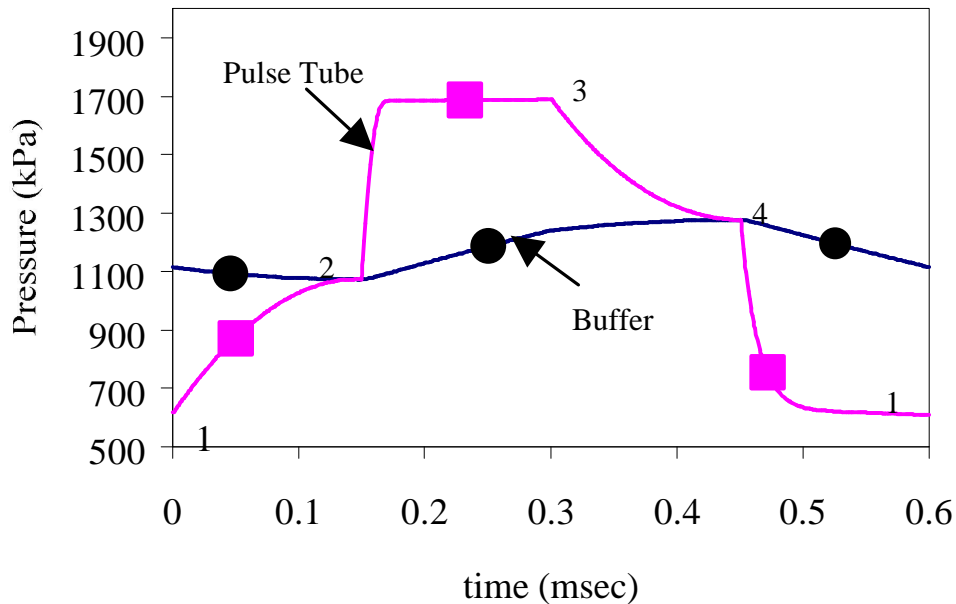


Figure 3. Pressure Wave Obtained by Theoretical Simulation

Step 1. Warm End Compression (1-2). During this process, both high and low valves are closed. Gas flows from the buffer to the pulse tube through the warm end of the tube due to the pressure difference between the buffer and the pulse tube. The pressure within the pulse tube increases while the pressure inside the buffer decreases until both pressures reach equilibrium. As expected, the pressure change within the pulse tube is more dramatic than that within the buffer due to volume difference.

Step 2. Cold End Compression and High Pressure Shuttle (2-3). The compressor high-pressure valve is opened, allowing the gas to flow into the pulse tube cold end through the regenerator. Initially part of the gas flows through the orifice into the buffer and part of the gas stays in the pulse tube until the pressure within the pulse tube reaches P_h . After the system pressure increases to P_h , gas flows through the pulse tube into the buffer.

Step 3. Warm End Expansion (3-4). The high-pressure valve is closed while the low-pressure valve is kept closed. Gas flows from the pulse tube into the buffer through the orifice due to the pressure difference between the buffer and the pulse tube. The pressure within the pulse tube decreases while the pressure inside buffer increases until both pressures reach equilibrium.

Step 4. Cold End Expansion and Low Pressure Shuttle (4-1). The compressor low-pressure valve is opened. The gas flows from and into the pulse tube through the regenerator and the orifice, respectively. Since the low pressure valve has larger flow coefficient than that of the orifice, more gas flows out of than into the pulse tube, the pressure within the pulse tube decreases to P_1 . After the system pressure reaches P_1 , gas flows through the pulse tube from buffer to the compressor suction end.

The thermodynamic cycle of the pulse tube heat pump can also be described by considering different gas segments within the pulse tube. Each gas segment goes through its own thermodynamic cycle. Figure 4 illustrates the different states of each gas column during one operating cycle. Segment 'C' represents the cold gas column, segment 'P' represents the gas piston, and segment 'B' represents the gas from the buffer. Segment P never flows out of the pulse tube. The pressure of the buffer and the intermediate pressures of the pulse tube are determined by the valve timing, valve flow coefficients and the buffer volume. Since the theoretical heat pumping rate equals the rate of heat absorption at the cold end of the pulse tube, the calculation of heat pumping rate can be based on thermodynamic analysis of the control volume at the cold end of the pulse tube (Figure 4). The sidewalls, the moveable boundary between the cold end volume and the gas piston, and the fixed boundary at the lower end of the cold end heat exchanger define the boundaries of this control volume. It is assumed that the control volume does work on the gas piston but exchanges no mass or heat with the gas piston, and that no heat is exchanged with the sidewalls.

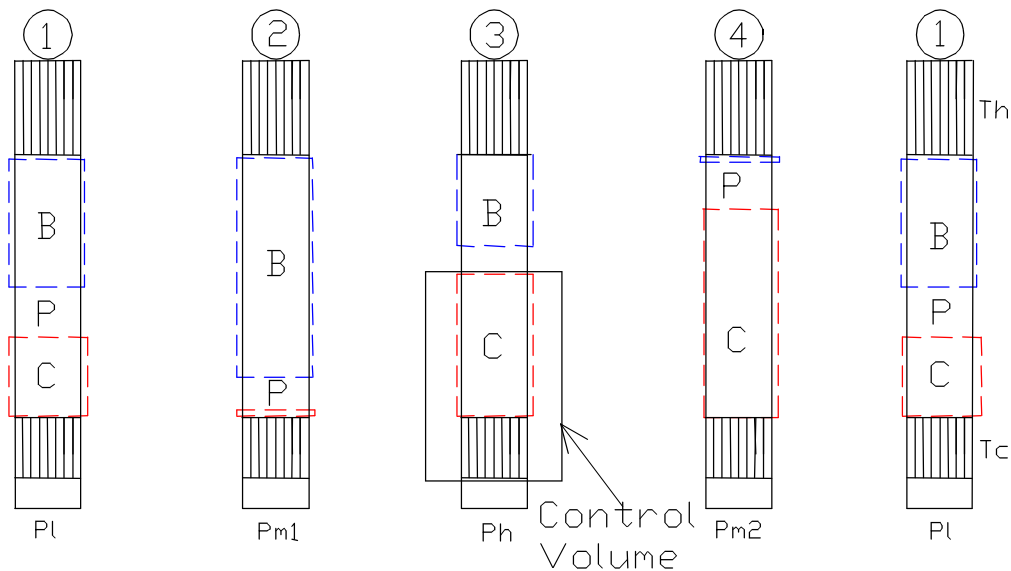


Figure 4. Gas Column States During One Operating Cycle

The energy balance for the control volume 'C' over one cycle is described by

$$\langle \dot{Q}_h \rangle = \langle \dot{Q}_c \rangle = \langle \dot{W}_c \rangle \quad (1)$$

where \dot{Q}_h is the heat pumping rate, \dot{Q}_c is the heat absorbed at the cold end, and \dot{W}_c is the net work which the cold space 'C' does on the gas piston. Theoretical analysis [1] shows that the work term is a function of the void volume of the regenerator and the cold end heat exchanger, the total volume of the pulse tube, the operating conditions of system including high /low pressure and high/low temperature, and the first and second intermediate pressures. The maximum heat-pumping rate can be reached by optimizing the relevant parameters.

The coefficient of performance (COP) is determined by

$$g = \frac{\langle \dot{Q}_c \rangle f + \dot{W}_{comp}}{\dot{W}_{comp}} \quad (2)$$

where f is the operating frequency and \dot{W}_{comp} is the compressor input power.

This pulse tube technology is applicable to heating and cooling systems for mobile enclosures and for fixed low to medium volume enclosures. The project was intended to replace the conventional heat pumps that rely on fluorocarbons such as freon. The strengths of pulse tube technology over current conventional heat pumps are reduction in the weight and volume of the ECU, increased efficiency, elimination of ODC/global-warming chemical (GWC) refrigerants, and decreased energy consumption. The technology operates without the use of restricted ODCs or GWCs, an emerging concern for future ECU systems. The efficiency of the prototype exceeds that of conventional heat pumps, resistance (strip) heat or natural gas at temperatures below 30EF. The weakness of the pulse tube technology is that it does not operate efficiently in its cooling mode.

The project requirements for the prototype system were to provide a greater heating efficiency than existing technology and a cooling efficiency equivalent to current technology. In addition, the system should be turnkey, as light as possible, contained within a stand-alone unit, and should not require an operator. The specific design requirements of the prototype were to provide a heating requirement of 5 kW with a COP of at least 1.1 and to heat an enclosure to 25°C with an external temperature of 0°C or below. For the prototype study, a compressor that supplied 30 standard ft³/minute with a discharge pressure of 300 psi, suction pressure of 80 psi at an input power of 5 kW was used.

3.0 DEMONSTRATION DESIGN

3.1 PERFORMANCE OBJECTIVES

The overall project goal was to develop a heat pump that would have greater heating efficiency (COP greater than 1.1) and equivalent cooling efficiency as compared to existing ODC-based technology, but that would use a non-ODC gas, helium, as the working fluid. The measure of performance was the COP calculated from prototype testing. Specific technical objectives were to optimize the design parameters for maximum heating and cooling efficiency, minimize the footprint of the unit, and minimize the weight of the unit. To achieve these goals, the project was divided into two phases: the optimization study to investigate the effects of design parameters on system performance; and the prototype study to design, fabricate and test a prototype unit.

3.2 PHYSICAL SETUP AND OPERATION

A test station (Figure 5) was built to study the effect of different design parameters on performance and to determine the most efficient operational parameters for the pulse tube. The investigation considered: valve timing; operating pressure and frequency; orifice opening; size of the cold and warm end heat exchangers; and surge volume.

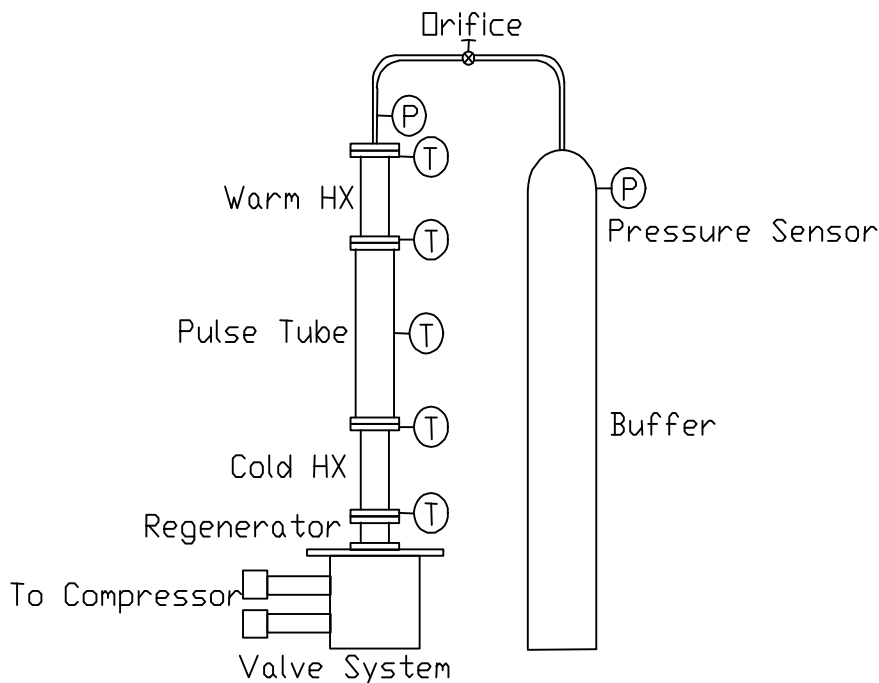


Figure 5. Schematic of Pulse Tube Heat Pump Test Station

The heat pump system consisted of a pulse tube, a cold end heat exchanger, a regenerator, a warm end heat exchanger, a buffer with an orifice, and a valve system. A pulse tube with total volume of 1 liter and a buffer with a volume of 7 liters was used for the entire experiment. Two pressure wave generators, one with an input power of 4 kW and the other with an input power of 6 kW, were used. Eight heat exchangers with different dimensions and/or cooling method were tested. Two different valve systems were investigated. Initially, a solenoid valve system controlled by a computer program was used to optimize the valve timing. When the optimized valve timing was obtained, a more reliable rotary valve driven by a motor was designed and tested. Two regenerators were used in the investigation.

Based on the optimization study results and theoretical equations, the prototype was designed and constructed. A schematic of the prototype is presented in Figure 6.

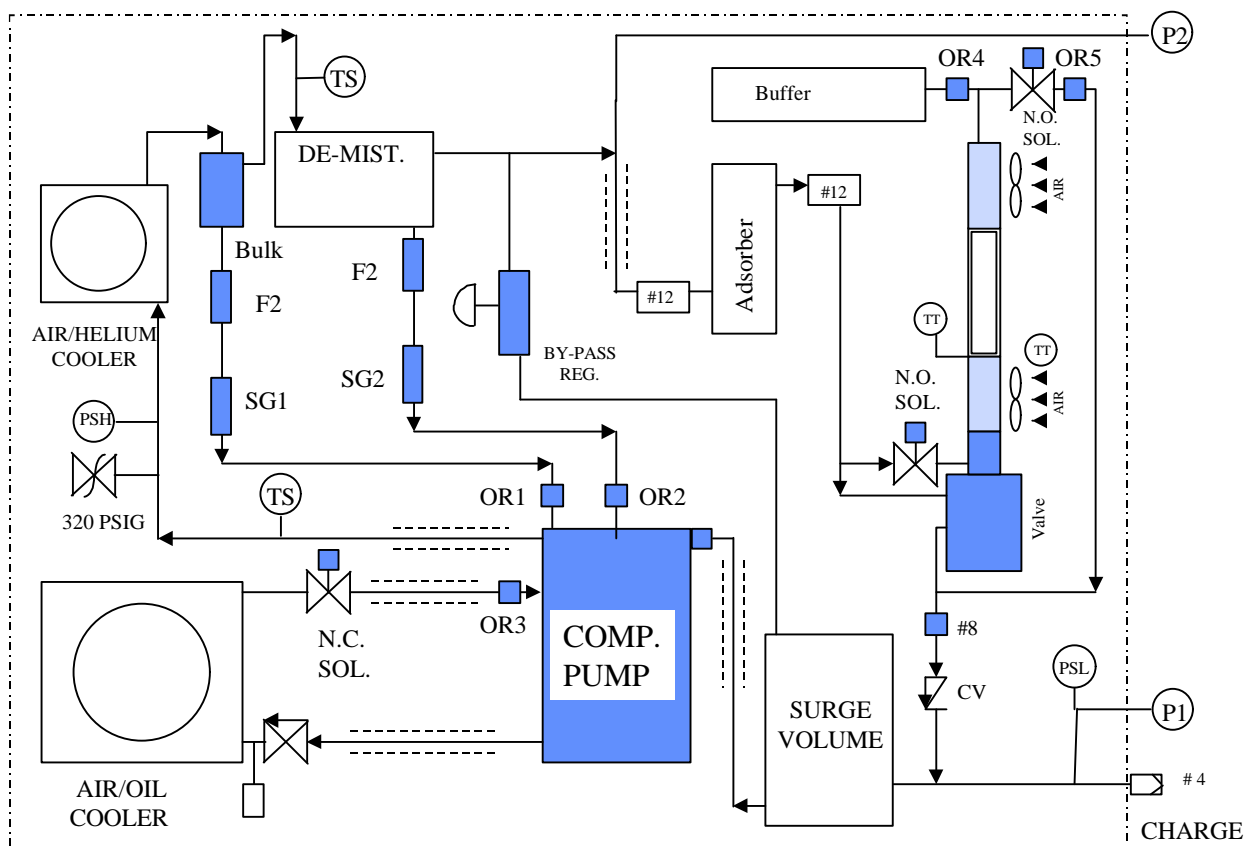


Figure 6. Prototype System Flow Diagram

3.3 MEASUREMENT OF PERFORMANCE

In the optimization study, the test station measured pressure, temperature, heat pump rate, system flow rate and power consumed by the compressor. The pressures in the buffer and pulse tube were measured with piezo-resistive pressure transducers manufactured by Omega. The temperatures at five locations were measured with temperature sensors manufactured by Omega. The heat pump rate was measured indirectly

by measuring the inlet and outlet water temperature and flow rate through the warm end heat exchanger. The compressor input power was measured with the power meter. System performance was assessed in terms of heat pumping rate.

Prototype performance was measured by monitoring the inlet and outlet temperatures of a water-cooled warm end heat exchanger. Earlier testing verified that the heating power of the warm end heat exchanger was equal to the cooling power of the cold end heat exchanger. From simultaneous measurement of the input power of the compressor, the overall prototype output and efficiency were determined.

3.4 DEMONSTRATION SITE/FACILITY BACKGROUND AND CHARACTERISTICS

This project encompassed only bench scale and pilot scale testing at the contractor's manufacturing facility. Due to the inability of the technology to meet the acceptance criteria, no site demonstration was performed. Therefore, no site or facility was involved with the project.

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4.0 PERFORMANCE ASSESSMENT

4.1 OPTIMIZATION STUDY

The pressure oscillations obtained during one set of experiments were compared with those obtained from theoretical simulation, presented in Figure 3. The experimental results were similar to the theoretical simulation.

Operating Pressure: Over the range of discharge pressures tested, it was observed that the heat pumping rate increased with increasing discharge pressure. In addition, the higher the ratio of discharge pressure to suction pressure, the better the performance, although the differential increase in the heat pumping rate decreased as the ratio increased.

Operating Frequency: For a given discharge pressure, there was an optimum frequency.

Orifice Opening: For a given pulse tube system and operating frequency, there was an optimum orifice opening. The higher the operating frequency, the larger the optimum orifice opening. The use of the double inlet opening decreased pulse tube performance; a simple orifice configuration was the most efficient.

Valve Timing: The valve timing strongly influenced system performance. The best performance was obtained when the ratio of the time during which the high pressure gas is admitted to the pulse tube, the time during which the gas flows from the buffer into the pulse tube through the orifice, the time during which the gas is exhausted from the pulse tube to the low pressure side of the compressor, and the time during which the gas flows from the pulse tube into the buffer have ratios of 1:1:1:1.

Heat Exchangers: It was found that performance was enhanced by use of a cold end heat exchanger with a small volume and large surface area. A water-cooled heat exchanger gave substantially better results than an air-cooled heat exchanger.

Surge Volume: A set of two surge volumes were added to the assembly, one on the high pressure side before the three-way valve and one on the low pressure side after the three-way valve. This was done to minimize pressure fluctuations on the compressor system and to more accurately measure the high and low side pressure of the PT system. Surge volume valve testing showed that the surge volume helped the efficiency of the pulse tube but had to be tuned to achieve the maximum performance.

4.2 PROTOTYPE TESTING

As expected, the prototype performed well in the heating mode and poorly in the cooling mode. The prototype pulse tube heat pump met the heating and unit size reduction goals of the project. In the heating mode, the pulse tube heat pump demonstrated the capability to meet the required portable shelter heating load down to 10EF without supplemental strip (resistance) heating, more than 30EF below the current A-39 ECUs. The prototype provided a total heating power of 8.0 kW (2.27 tons) with a coefficient of performance of 1.2. The prototype's cooling efficiency, however, did not meet the project acceptance requirements. In the cooling mode, the prototype had a cooling power of 1.3 kW (0.37 tons) with a coefficient of performance of 0.2. Based on the inability of the prototype pulse tube heat pump to provide sufficient cooling power, further technological development was discontinued.

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5.0 COST ASSESSMENT

The fabrication of the prototype required the use of custom built components. Full-scale use of this technology would entail further development and manufacture of these components. Because several components of the prototype heat pump are not readily available, it was not possible to perform a reliable cost estimate for full-scale implementation of the technology. Due to the premature status of this technology, the cost assessment section is not applicable.

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6.0 IMPLEMENTATION ISSUES

6.1 COST OBSERVATIONS

As noted above, the premature status of this technology precluded a cost assessment. Therefore, cost observations are not applicable.

6.2 PERFORMANCE OBSERVATIONS

The technology met the heating requirements for the project. In the cooling mode, the pulse tube heat pump failed to meet the acceptance criteria. Therefore, further development of the technology will not be pursued.

6.3 OTHER SIGNIFICANT OBSERVATIONS

The breadboard assembly showed that running the system with a periodic three-way valve was more efficient than running the system with a direct piston driven system. There were several reasons for this observation. First, the piston frequency was too high and caused flow problems in the regenerator. Second, the oil from the compressor infiltrated the pulse tube regenerator and heat exchangers, causing a loss in efficiency. Third, the heat rejection heat exchanger after the piston added dead volume to the system on the inlet side of the regenerator, causing system inefficiency. The three-way valve had the advantage of speed control for fine tuning the pulse tube, and allowing for an oil removal system after the compressor to protect the regenerator and heat exchangers. The dead volume before the regenerator could be minimized by mounting the valve on top of the regenerator.

Due to helium's higher specific heat ratio as compared to freon, it is necessary to include an additional gas cooling system in order to maintain the compressor temperature within its working limits. The compressor used in this project had an oil circulation loop. By adding a line into the high-pressure sump at the bottom of the unit and an oil injection port into the side of the unit, a loop was created that allowed the installation of an oil cooler onto the compressor skid. The oil cooler provided the majority of the cooling for the unit. Hot oil flowed out of the high pressure lower section, was cooled by the oil cooler, and was injected back into the unit at an intermediate pressure (150 psig) about halfway into the rotating scrolls. The cooled oil ran through the scrolls, cooling the helium gas as it was compressed, and helping to seal the scroll for more efficiency.

Because the oil was injected into the gas stream, it was necessary to separate the oil from the gas stream before the gas re-entered the pulse tube assembly. The majority of the oil was removed in the top bell area of the compressor. More oil was separated from the helium in the after-cooler. The third location for oil removal is the mist eliminator, where the oil content of the gas is reduced to the parts per million range. The final stage of oil removal is the charcoal adsorber. The unit removes any last mist or oil vapor that may still be in the gas. This adsorption unit is removable and requires recharge with new charcoal every 6000 hours of operation. The oil removed at the first three separation locations is recirculated. Thus, if used for heating, this system would require the purchase and disposal of activated carbon and, likely, periodic adjustment or monitoring of the oil volume and condition.

Because the flow in the bulk oil separator and mist eliminator lines is controlled via orifices, a small amount of gas is by-passed into the oil injection port along with the returning oil. It appears that system efficiency could be increased by eliminating this bypass gas flow. In future generations of these machines, the bulk oil separator could have a float trap to ensure that only oil would return to the injection port and that the flow from the mist eliminator would return to the low side gas inlet. The bypassing from the mist eliminator would have a slight impact on the overall output of the compressor. Testing of this possible modification was not attempted in the project.

Because the pulse tube assembly will not start if the oil temperature is below 60°F, heaters are mounted on the compressor unit and the oil cooler to maintain the oil temperature above 60°F. An indicator light on the starter/control box will illuminate if this occurs. When the oil is heated above the start point the light will shut off and the pulse tube assembly will then be allowed to re-start. In addition, an automatic warming control is built into the pulse tube assembly to ensure that it does not become too cold during long dwells or at initial start-up.

The de-frost cycle is required because the cold end heat exchanger temperature can drop below the point where it can freeze-out the moisture in the ambient air and accumulate a layer of frost. This frost will reduce the pulse tube efficiency. To prevent this from happening, a set of two temperature sensors are mounted in the ambient air loop to monitor the surface temperature of the cold end heat exchanger relative to the ambient temperature. As frost accumulates, the temperature difference between these two sensors increases. When the difference increases above the set point of the controller the pulse tube valve and the ambient air fan are de-activated. Two by-pass solenoid valves are opened and warm enclosure temperature helium gas is allowed to flow through the pulse tube. This warm gas melts the frost on the heat exchanger and clears the outer fan passages. When the temperature difference between the ambient air and the heat exchanger surface is low enough the two solenoid valves close, the pulse tube valve is re-activated, and the ambient air fan is turned back on. As the de-frost cycle is working the compressor and enclosure circulating fans are allowed to run as usual to maintain the enclosure temperature.

6.4 LESSONS LEARNED

At this stage of technological development, the advanced cycle helium pulse tube heat pump does not allow for as efficient cooling as is possible with the current fluorocarbon-based systems. AFRL has terminated work in this particular technology and is focusing on carbon dioxide cycle technology.

6.5 END-USER/OEM ISSUES

Due to the inability of the technology to meet the acceptance criteria, further development will not occur.

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Peter M. Winn, Jie Yuan ASC Westborough, MA
Submitted to WL/FIVC Tyndall AFB, FL
Contract No. F08637-98-C6016

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APPENDIX A

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