Numerical Modeling and Experimental Testing of a Mixed Gas Joule-Thomson Cryocooler

by

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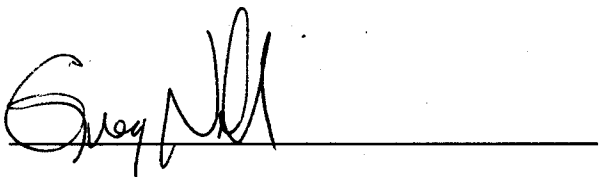
Approved by



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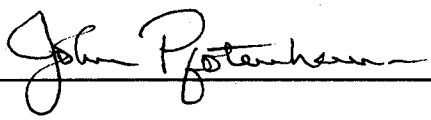

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Abstract

Mixed gas Joule-Thomson (MGJT) systems have been shown to provide order of magnitude improvements in efficiency relative to JT systems that use pure working fluids. This thesis presents theoretical and experimental work related to using a single-stage, low power (< 1 W) MGJT system for cooling the current leads required by high-temperature superconducting electronics. By thermally integrating the current leads with the recuperative heat exchanger of a MGJT cycle, it is possible to intercept the electrical dissipation and conductive heat leak of the wires at a relatively high temperature which provides a thermodynamic advantage. Also, directly cooling the leads rather than indirectly cooling the chips may provide some advantages relative to thermal integration.

To design the recuperative heat exchanger for the MGJT cycle, the composition of the gas mixture was optimized using a robust genetic optimization technique. Following mixture selection, the optimization model was modified so that it included the effect of frictional pressure drop, axial conduction through the heat exchanger, and the overall conductance available from the heat exchanger on the performance of the MGJT cycle. The individual influences of these loss factors on the refrigeration power of the MGJT cycle were investigated parametrically and conceptually in order to determine the target values for a low power system and develop some insight into the relative importance of each effect. A detailed model of the specific Hampson-style heat exchanger geometry was developed and used to obtain a design for an initial demonstration device.

The demonstration device was fabricated and integrated with a thermal vacuum test facility, gas handling equipment, and the appropriate instrumentation. Several tests were carried out. First, the heat exchanger alone was tested (outside of a JT cycle) using helium as the working fluid. These data provided some experimental verification of the detailed model. Next, the test facility was modified through the installation of a fixed orifice expansion valve to allow open cycle testing of the device using high pressure (9.745 MPa) pure Argon. These measurements provided further insight into the performance of the device.

The test facility was subsequently integrated with a compressor in order to allow measurements of the Device's performance using gas mixtures in a closed loop configuration. These test results ultimately revealed issues relative to contamination, which were addressed through the installation of a liquid nitrogen trap, as well as liquid management. The liquid management issue is thought to be related to inadequate vapor kinetic energy which does not provide sufficient momentum transfer to the liquid to move it through the system. The liquid management issue constrains the performance of the MGJT cycle at low mass flow rates and was explored over a very limited range of conditions. Further testing is suggested which will allow the liquid management constraint to be explored more completely in order to guide future designs.

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A	Area (m^2)
A_b	Surface area of tube base (m^2)
A_f	Area of a fin (m^2)
A_{ff}	Free-flow area (m^2)
cp	specific heat (kJ/kg-K)
D_{helix}	Diameter of helix (m)
err	Percent Error
Fin_{OD}	Fin diameter (m)
$Fins_{inch}$	Fins per inch (1/in)
G	Mass flux (kg/s-m^2)
h	Enthalpy (J/g)
htc	Heat transfer coefficient (W/m^2-K)
ID	Inner Diameter (in or m)
k	Thermal conductivity (W/m-K)
L	Length (m)
L_{fins}	Length or height of fins from tube base (m)
Lp	Longitudinal pitch (m)
$\dot{m}$	Mass flow rate (g/s or kg/s)
M	Number of heat exchanger segments in detailed model (equal lengths) (-)
N	Number of recuperative heat transfer segments in optimization model (-)
N_{fins}	Number of fins (-)
Nu	Nusselt number (-)
N_T	Number of Tubes (-)
OD	Outer Diameter (in or m)
P	Pressure (kpa or psig)
Pr	Prandtl number
$\dot{Q}_{load,cold}$	Heat load applied to cold end of heat exchanger (W)
$\dot{Q}_{load,dist}$	Distributed heat load along the heat exchanger length (W)
$\dot{Q}_{rec}$	Recuperated heat (kJ/kg)
$\dot{Q}_{rej}$	Heat Rejected from cycle to surroundings (J/g)
R	Thermal resistance (K/W)
r	Radius (m)
Re	Reynolds number (-)
rf_{ft}	Fin radius on finned tube (m)
Sbc	Spacing between coils (m)
Sbc_{adj}	Spacing between adjacent coils (m)
S_{ft}	Spacing between fins (m)
t	Thickness (m)
T	Temperature (K)
tf_{ft}	thickness of fin on the finned tube (m)

Tp	Transverse pitch (m)
UA	Heat exchanger stream-to-stream conductance (W/K)
$\dot{V}$	Volume flow rate (L/min or Ft^3/hr)
$\dot{W}_{comp}$	Isentropic compressor work (J/g)
X_T	Non-dimensional Transverse pitch (-)
X_L	Non-dimensional Longitudinal pitch (-)
y_i	Mole fraction of component *i* of mixture

Greek

ΔT	Hot end temperature difference (K)
η	Fin efficiency (-)
η_o	Overall fin efficiency (-)
μ	Dynamic viscosity (kg/m-s)

Subscripts

h/high/hot/HP	Associated with the high pressure or hot stream
c/low/cold/LP	Associated with the low pressure or cold stream
inlet	Entrance to cryostat/experiment
outlet	Exit of cryostat/experiment
freeze	Freezing temperature
tp	Triple point
sat	Saturation state
spec	Specified by geometry
axial	Straight line between the hot end and cold end of the heat exchanger
cond	Conduction heat transfer in the axial direction
ft	Associated with the finned tube
ot	Outer tube
it	Inner tube
tot	Total or sum

Chapter 1 Introduction

1.1 Actively Cooled Current Leads

High temperature superconducting (HTS) electronics are being developed in order to operate in the temperature range of 80 K to 120 K. These electronic components must be mounted on a cold stage that is actively cooled via a thermal connection with any one of several types of commercially available cryocoolers, as shown in Figure 1.1. A large part of the heat load on the system is related to the current leads that carry the bias current and signals from room temperature electronics. The heat leak from these wires onto the high temperature superconducting chips is related to conduction along the leads as well as ohmic dissipation (because these leads are not superconducting).

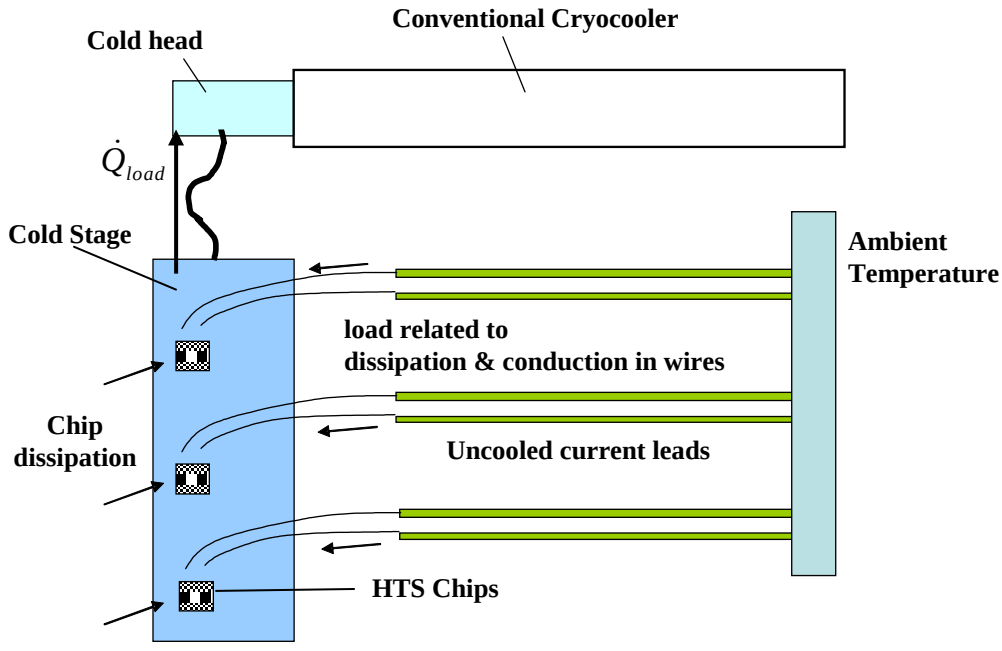

Figure 1.1: The conventional cooling configuration used for high temperature superconducting electronics.

There are two disadvantages associated with the cooling configuration shown in Figure 1.1. Thermal integration with the cryocooler is difficult. The current leads deposit thermal loads throughout the electronics package at the various electrical connections however the cooling load is removed from a single thermal interface with the cold stage. Therefore, chips mounted closer to the cold stage will be colder than those mounted further away. The performance of superconducting electronics is sensitive to temperature and it is important that interfacing chips be at the same temperature.

Also, the heat is allowed to reach the lowest temperature part of the cycle before it is removed. Entropy is the ratio of heat transfer to temperature and therefore, the heat load represents the largest entropy flow at the coldest point in the cycle. The power and size requirements of the cryocooler generally scale with the entropy flow that it must remove and therefore there is a thermodynamic advantage associated with removing the heat load at a higher temperature.

Ideally, the entire electronics package would be kept at a uniform cryogenic temperature and the heat generated by ohmic dissipation would be removed at a temperature very near where it was generated. Several researchers, including Williams (1963), have shown that cooling current leads carrying high currents for large superconducting magnets with vapor cryogen allow for much of the heat leak to be intercepted before it reaches the cold end. Vapor cooling current leads for high temperature superconducting electronics has not previously been studied. The direct implementation of vapor cooling via the boil-off

of a cryogen is not likely to be feasible for applications that require a gravity independent system due to the presence of the liquid.

The objective of this thesis is to study a system in which the current leads are thermally integrated with the recuperative heat exchanger within a mixed gas Joule-Thomson (MGJT) system, as shown in Figure 1.2. The recuperative heat exchanger removes the conductive and ohmic heat leak of the current leads at the temperature where it is generated and also provides cooling to the HTS computer chips at several locations in the electronics package. The cooling is provided via the leads rather than indirectly (as shown in Figure 1.2). This actively cooled current lead concept has thermal integration and thermodynamic advantages; the configuration may permit a more uniform temperature distribution from chip-to-chip and also allow heat to be removed from the system in a more reversible manner.

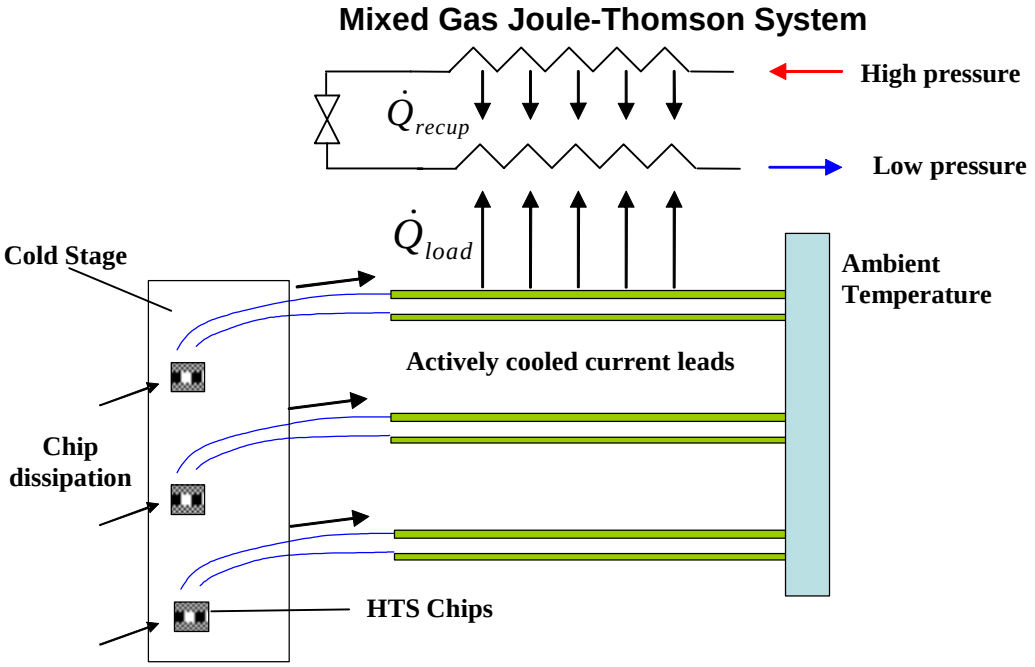

Figure 1.2: The proposed method of cooling HTS electronics.

This project has focused primarily on the details of the MGJT refrigeration cycle; while there has been substantial research in this area, there has been relatively little detailed theoretical work reported that supports the more numerous publications regarding specific experimental results. Also, the use of an MGJT system for high efficiency, low power, low temperature applications involving a distributed load has not been thoroughly studied. In the following sections, JT and MGJT systems are introduced and discussed.

1.2 Mixed Gas Joule-Thomson Cycle

The simplest, closed cycle configuration for a Joule-Thomson system is shown in Figure 1.3. The cycle consists of a compressor, an aftercooler, a recuperative heat exchanger, a valve, and an evaporator or load heat exchanger. In this cycle, the working fluid is compressed to a high pressure (state 2) and cooled to room temperature (state 3) in the aftercooler. The high pressure gas enters the recuperative heat exchanger and is cooled (by the returning low pressure gas) until it exits the heat exchanger at a cryogenic temperature (state 4). The high pressure, cold working fluid is expanded through a valve; this isenthalpic expansion causes the temperature of the fluid to drop (state 5). The cold, low pressure fluid accepts heat as it passes through the load heat exchanger or evaporator. The working fluid enters the cold end of the heat exchanger (state 6) and warms as it pre-cools the high pressure gas until finally it exits the heat exchanger and enters the suction side of the compressor (state 1).