Micro Channel Recuperator for a Reverse Brayton Cycle Cryocooler

C. Becnel, J. Lagrone, and K. Kelly

Mezzo Technologies
Baton Rouge, LA USA 70806

ABSTRACT

The Missile Defense Agency has supported a research program to design, fabricate and test a recuperator that is compact, light weight, durable, and provides extremely high effectiveness. The recuperator is designed to be installed in the warm stage of a two-stage reverse Brayton cycle cryocooler that operates between 300 K and 60 K. The effectiveness of the recuperator must equal or exceed 99.6%. The specifications of the system are as follows: the working fluid is helium, the mass flow rate of the working fluid is 0.56 grams/sec, and the absolute pressures of the two flow streams are 2.3 and 1.6 atmospheres. The allowable fractional pressure drop of the recuperator was specified to be no more than 2%. Additional specifications included acceptable leakage rates, burst pressure, g-loading, etc.

Mezzo Technologies has spent the last year designing and fabricating a recuperator that meets required performance specifications while providing substantial weight savings (less than one third the weight) and volume savings (less than half the volume) over the current recuperator technology. The recuperator performance was quantified for the case where either air or argon was used as the working fluid, operating between two thermal reservoirs at 0 °C and 100 °C. The comparison between model prediction and actual performance is provided. A model-validated estimate of actual performance using helium at the actual operating conditions is also provided.

INTRODUCTION

The goal of this project was to manufacture a micro channel regenerator that would match the performance of the regenerator currently being used in a two stage reverse Brayton cycle cryocooler being developed by Creare while providing substantial savings in weight and size

It is well known that one of the advantages of micro scale flow passages is enhanced heat transfer. As the characteristic dimension of the heat exchanger channel size decreases, the heat transfer area/volume (A/V) and convective heat transfer coefficients (h_{conv}) at the fluid-solid boundaries both increase. Therefore, small scale provides a high thermal conductance/volume ratio (h_{conv}A/V), where thermal conductance is equal to (h_{conv}A). As a result of this scaling trend, the volume (and usually weight) of heat exchangers decreases as the channel size decreases (assuming a given allowable pressure drop and given overall geometry). This principle has been employed in this project.

Table 1. Recuperator Technical Requirements List.

Envelope	
Current HX Mass	12 kg
Volume	423 in3
Operating Conditions	
Heat Transfer	5W @ 60K
Flow Rate	0.56 g/s of 4- Helium
High Pressure Inlet	300K @ 2.3 atm
Low Pressure Inlet	60K @ 1.6 atm
Thermal Effectiveness	>99.6%
Fractional Pressure Drop	<0.020
Mechanical	
External Leakage	< 1E -7 std cc/s of helium
Cross-stream Leakage	< 1% of flow rate
Burst Pressure	> 10 atm
Proof Pressure	> 5 atm
Acceleration	GEVS-SE page 2.4-19
Interfaces	Fluid: 1/2 in OD x 0.035 in wall thickness Stainless Steel

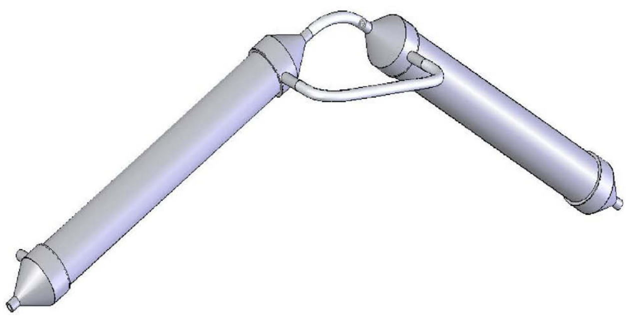

Figure 1. Two-module recuperator

DESCRIPTION OF RECUPERATOR

At the beginning of the project, Mezzo was supplied with the Technical Requirements List (TRL) shown in Table 1. These performance specifications match those of the existing recuperator. The goal of the project was to provide substantial improvement over these specifications, especially in terms of weight and volume savings. The term fractional pressure drop requirement is defined by the equation: $\Delta P_H/P_H + \Delta P_L/P_L < .02$. Mezzo's final recuperator design consists of two counter flow micro channel heat exchanger modules plumbed in series and shown in Figure 1.

A single module is shown in Figure 2. The core of the module is 22.5 inches in length, with an outer diameter of 3 inches. The unit is made of 304 stainless steel. The mass of a single module was originally predicted to be 2.3 kg. The mass of the two-module recuperator is approximately one third that of the existing recuperator and the volume is approximately one half the existing recuperator. The internal micro channel geometry of the recuperator is proprietary. Factors that were examined in detail included:

i) Axial conduction heat transfer is an important source of irreversibility in recuperators with very high effectiveness. Considerable design effort focused on reducing axial conduction heat transfer.[1,2]
ii) The length of the core must be sufficient to provide adequate heat transfer, yet not too long (since extra length implies extra mass and volume). Considerable effort was expended designing a core with the minimum length that would provide the desired effectiveness and satisfy the pressure drop specification.

Figure 2. Photo of single module

Table 2. Comparison of existing and Mezzo recuperators

	Specifications	Proposed Design
Module (Half of Heat Exchanger)		
Core Length	NA	17.8in (2" per inlet and outlet added)
Core Diameter	NA	2.43in
Recuperator		
Volume	423 in^3	200 in^3 (core)
Weight	12 kg	4.0kg (core)
Effectiveness	99.60%	99.60%
Fractional dP	0.02	0.0131
High Pressure Side	NA	0.0066
Low Pressure Side	NA	0.0065
Material	NA	304 S.S.

A comparison of the performance of the existing recuperator and the *predicted* performance of the *two* module unit shown in Figure 1 is summarized in Table 2. This comparison shows that the new micro channel recuperator provides significant volume and mass savings (1/2 the volume and 1/3 the weight). Furthermore, the predicted fractional pressure drop is also less than the requirement.

EXPERIMENTAL FACILITY

A. Description of facility

A model was developed by Mezzo to predict the performance of the recuperator using the actual working fluid (helium) at design conditions. To verify the accuracy of the model prediction, an experimental facility was designed and built to quantify the performance of the Mezzo recuperator. Specifically, the facility was designed to test the performance of the unit operating between 0°C and 100°C, using air or argon as the working fluid. Experiments using air and argon were performed over a range of Reynolds numbers that encompassed the actual operating conditions. A close correlation between test results and model prediction over the appropriate range of Reynolds numbers was assumed to validate the accuracy of the model.

Differences between the experiments performed and the actual operating conditions include i) the working fluid was argon or air instead of helium, and ii) the pressures of the counter flowing streams were nominally both equal to one atmosphere absolute instead of the 2.6 atmospheres and 1.6 atmospheres (absolute) that would exist in the real system, and iii) the temperature difference between the two fluid streams was 100 K per module instead of 120 K.

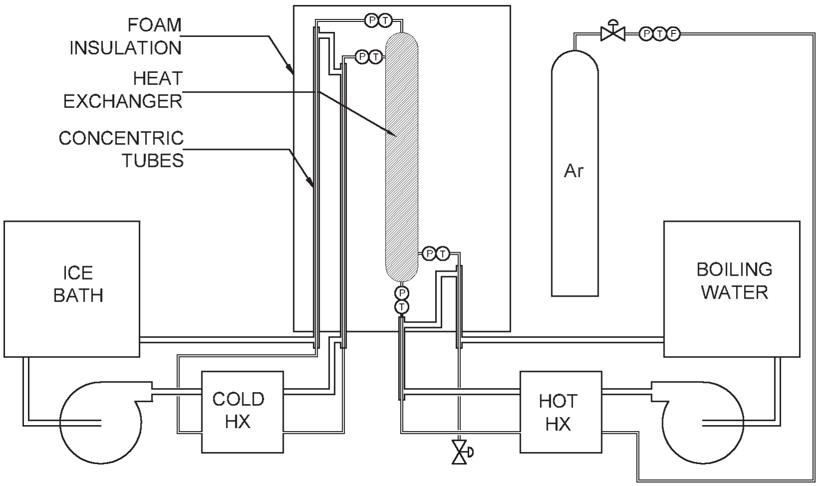

Figure 3. Test diagram

The general test setup is shown in Figure 3. Constant inlet temperatures for both flow streams are maintained by the two phase-change baths (an ice bath and a boiling water bath). The working fluid (argon or air) is fed to a heat exchanger (coiled copper tubing) located in the high temperature boiling water bath. The mass flow rate of the working fluid is controlled by a mass flow control valve. After exiting the high temperature heat exchanger, the working fluid enters the high temperature end of the recuperator. Between the boiling water bath and the recuperator, the working fluid is jacketed with hot water pumped from the hot water bath to prevent heat loss from the working fluid and to ensure that the working fluid enters the high temperature end of the recuperator at a temperature extremely close to 100°C.

After exiting the low temperature end of the recuperator, the working fluid (now at a temperature almost equal to the low temperature reservoir) passes through a long coil of copper tubing in the ice bath (the low temperature reservoir) before reentering the low temperature end of the recuperator (at 0°C). Both piping legs between the recuperator and ice bath are jacketed with ice water to prevent heat transfer to the working fluid from the ambient. The working fluid then passes back through the recuperator, exits the high temperature end of the recuperator, passes through a hot-water-jacketed section of tubing, then exits to the atmosphere.

The recuperator and tubing were insulated with at least six inches of fiber glass insulation. The environmental heat losses with the foam insulation are estimated to be ~1W. This insulation increased the axial conduction parameter, λ, by approximately 25% (estimated using a FEA model). A photograph of the test facility is shown in Figure 4.

B. Measurements

For each test condition, steady state values of temperature, pressure, and flow rate were measured and recorded. Both pressure and temperature were measured on the inlet and outlet on both flow streams passing through the recuperator. Temperature measurements were made by Hart Scientific PRTs. The combined error of the probe and recorder is ~25 mK. This error yields an effectiveness error of 0.025% for a single module or 0.013% for the two module unit. Pressure was measured by pressure transducers, and flow was measured and controlled by a Gas Mass Flow Controller.

EXPERIMENTAL RESULTS

Sets of experiments were performed using both argon and air as the working fluid. The range of flow rates for both sets of tests was selected to provide Reynolds numbers ranging from 0.25 to 3.0 times the operational Reynolds number. Table 3 shows data taken from one set of argon tests. Even

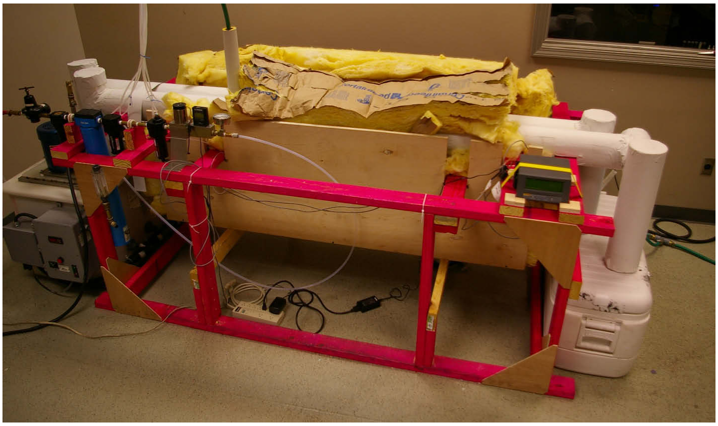

Figure 4. Experimental facility

Table 3. Experimental data for Argon

Mass	High pressure side						Low pressure side				
Flow Rate	Mdot Cp		Tin	Tout	Model ΔP	Exp. ΔP		Tin	Tout	Model ΔP	Exp. ΔP
L/min	W/K	Re	C	C	in H2O	in H2O	Re	C	C	in H2O	in H2O
14.9	0.2118	16.28991	99.813	3.161	1.3637	1.4	6.7219	0.144	96.048	0.6719	0.65
29.8	0.4236	32.55733	100.137	2.171	2.7711	2.75	13.4449	0.115	96.881	1.4249	1.75
59.5	0.8458	65.02249	100.013	1.585	5.5703	5.8	26.8448	0.114	97.434	3.116	5.45
89.3	1.2694	97.59893	99.963	1.498	8.251	8.75	40.2924	0.093	97.628	5.0157	11
119	1.6916	130.0776	99.896	1.552	10.6882	11.4	53.6941	0.087	97.593	7.0184	17.85
178.4	2.536	195.0308	99.839	1.869	15.0299	15.9	80.499	0.075	97.321	11.3284	35.6

Flow Rate	Mdot Cp		Model	Meas. HP	Meas. LP	Exp.
L/min	W/K	Re	ε	ε	ε	ε
14.9	0.2118	16.28991	92.79%	96.97%	96.22%	96.60%
29.8	0.4236	32.55733	95.90%	97.94%	96.74%	97.34%
59.5	0.8458	65.02249	97.41%	98.53%	97.42%	97.97%
89.3	1.2694	97.59893	97.74%	98.59%	97.66%	98.13%
119	1.6916	130.0776	97.75%	98.53%	97.69%	98.11%
178.4	2.536	195.0308	97.47%	98.20%	97.48%	97.84%

though the pressures of the two streams were effectively equal in the tests that were performed, the two flow streams are called "high pressure side" and "low pressure side". Inlet and exit temperatures of both flow streams are used to calculate the effectiveness values of the high pressure and low pressure flow streams, respectively. For example, using the data from Table 3 for the case where the flow rate of argon is 14.9 L/min, the effectiveness on the high pressure side is calculated as the ratio of the actual temperature difference experienced by the high pressure side working fluid divided by the difference in temperature of the two streams entering the recuperator:

$$\varepsilon_{high-pressure} = \frac{T_{in-high-pressure} - T_{out-high-pressure}}{T_{in-high-pressure} - T_{in-low-pressure}} = \frac{99.813 - 3.161}{99.813 - .144} = 0.9697 \quad (1)$$

Similarly, the effectiveness on the low pressure side at the same flow rate is calculated as follows:

$$\varepsilon_{low-pressure} = \frac{T_{in-low-pressure} - T_{out-low-pressure}}{T_{in-low-pressure} - T_{in-high-pressure}} = \frac{.144 - 96.048}{.144 - 99.813} = 0.9622 \quad (2)$$

It should be noted if there were no heat transfer between the environment and the recuperator, the measured high pressure side and low pressure side effectiveness values would be identical. However, although the recuperator is insulated, there is a small heat loss through the insulation to ambient. This heat loss results in the low pressure side effectiveness being lower than it would be if an adiabatic thermal boundary condition existed on the outside of the recuperator. Similarly, heat

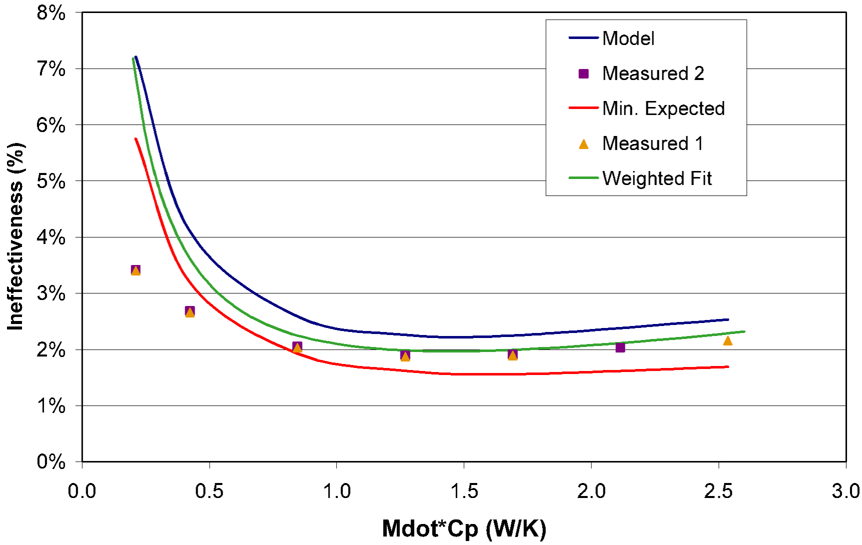

Figure 5. Results using Argon as the working fluid

loss causes an increase in the high pressure side effectiveness value. Interestingly and fortunately, the *average* value of high and low pressure side effectiveness values is insensitive to heat transfer from the outside of the recuperator shell, so the average of the high and low pressure side effectiveness values accurately characterizes the effectiveness of the recuperator.

Figure 5 plots the ineffectiveness (1-ε) as a function of flow rate (or Reynolds number) of a single module. The curve labeled "Model" provides the original model prediction prior to testing. The curve labeled "Min. Expected" provides the "most optimistic" prediction based on making certain modeling assumptions that would increase performance. An example of a factor that differentiates the original model from the optimistic "Min Expected" model is the assumed length of the flow development region. In the original model, the thermal conductance (UA) of the core was calculated by only considering the recuperator volume where fully developed flow was assumed to be established. In the "Min. Expected" case, the entire core length was used in calculating the thermal conductance of the core. Obviously, the extra length of assumed active core gave improved predicted performance. The two curves, "Model" and "Min. Expected" provide a range of model-predicted ineffectiveness values.

Two sets of experiments using argon were completed. They are labeled "Measured 1" and "Measured 2". Both sets of data are identical, indicating that the experimental results were repeatable. Above a specific heat capacity rate of around 0.75 W/K the experimental results indicate that the recuperator performance is better than the original model. Only at low heat capacity rates do the experimental results fall outside the range of model predictions. While the cause of the discrepancy between model prediction and experimental result at low flow rates is still being examined, the actual system will operate at specific heat capacity rates that are well within the range where the model accurately (and conservatively) predicts performance.

At the top of Figure 5 are two numbers that can be used to characterize the performance of the recuperator. With argon as the working fluid, the thermal conductance, UA, of the recuperator is 147 W/K. This value varies only slightly with flow rate, since the flow is laminar on both the high and low pressure side, and the Nusselt numbers of both flow streams are effectively constant. The other number, R, characterizes the axial thermal resistance which is required to calculate axial conduction. The R value of the recuperator is 71 K/W. The values of UA and R were calculated by a weighted curve fit of the data, with heavier weighting being applied to data points taken at higher flow rates (Reynolds numbers).

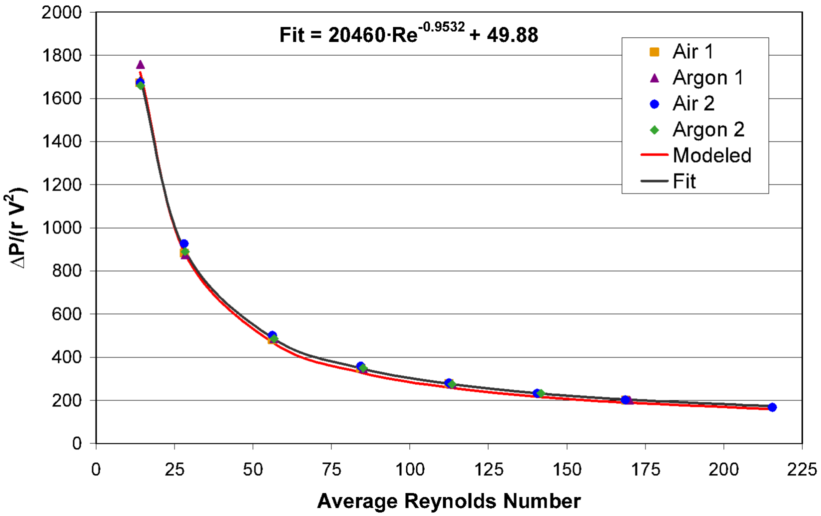

Figure 6. High side dimensionless pressure drop

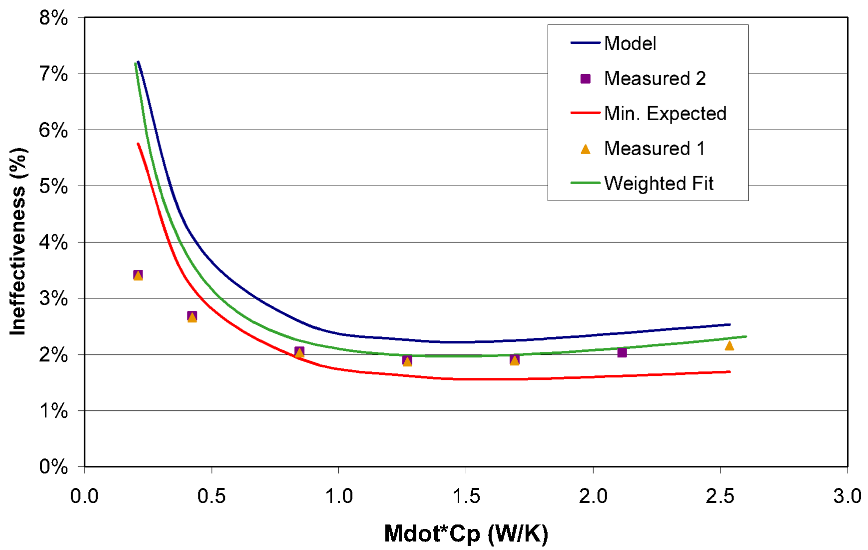

Figure 7. Dimensionless pressure drop on low pressure side

In addition to calculating heat transfer, it is important to predict pressure drop of the two fluids. Figure 6 plots the non-dimensional pressure drop of the high pressure side as a function of high pressure side Reynolds number. It can be seen that experimental results for air and argon match the model prediction very well. The pressure loss term on the high pressure side is understood very well.

Figure 7 plots the non-dimensional pressure drop on the low pressure side as a function of Reynolds number. The pressure drop on the low pressure side was very poorly estimated. The model prediction (red line) crosses the experimental data from the argon and air tests at a Reynolds number of around 70. Fortunately, the Reynolds number at which the actual system is expected to operate is near 70, so the actual error in predicting pressure drop on that side may be low. Nevertheless, model improvements are needed to more accurately predict pressure drop on the low pressure side.

Table 4. Predicted results for helium using updated model results

	Effectiveness		DP Used		NTU		Lamda (x10^3)		UA (W/K)		Resistance (K/W)		Mass(kg)	
Total	99.65%	99.60%	79%	80%	567.5	520.7	1.784	2.078	1651	1514	192.6	165.5	4	4.6
Hot	99.30%	99.22%	56%	59%	352.1	321.5	4.152	4.833	1024	935.1	82.83	71.15	2	2.3
Cold	99.25%	99.16%	22%	20%	218	199.2	2.986	3.478	634.4	579.4	115.1	98.83	2	2.3

Experimental (highlighted, first value of each pair) Modeled (unshaded, second value of each pair)

The modeling and experimental results were considered to be in very good agreement. This implies that the model predictions for the actual system should be valid. The primary source of error at this point is probably associated with inaccuracies in predicting the pressure drop on the low pressure side. The experimentally-derived values of UA and R, obtained using argon as the working fluid, can be used to predict the performance of the recuperator in actual operating conditions. Eqs. (3) and (4) are used to predict, respectively, the UA and R values when helium is used as the working fluid. With these values, the ineffectiveness, I, as well as the effectiveness, ε, can be calculated using Eq. (5).

$$UA_{helium} = UA_{argon} \frac{k(T)_{helium}}{k(T)_{argon}} \tag{3}$$

$$R_{axial} \propto k(T)_{solid} \tag{4}$$

$$I = 1 - \varepsilon = \frac{1}{NTU+1} + \lambda = \frac{1}{\frac{UA}{\dot{m}C_p}+1} + \frac{1}{R_{axial}\dot{m}C_p} \tag{5}$$

A comparison between the original model prediction and new model prediction is given in Table 4. The predicted effectiveness has increased to 99.65%, which represents a significant improvement over the original requirement of 99.6%. This is due to the fact that the heat transfer (captured by the NTU term) is greater than originally modeled, and the axial conduction (captured by the λ term) is less than originally modeled. The total fractional pressure drop is 79% of the allowable and almost identical to the original model prediction. The weight is slightly less than predicted. Overall, the actual performance will exceed the original predicted performance.

CONCLUSIONS

A recuperator was designed, fabricated and tested. The test results indicate that the recuperator will meet target heat transfer and pressure drop performance specifications. The recuperator offers significant volume and weight savings over the existing recuperator. In addition, the cost for fabrication is expected to be greatly reduced. The recuperator will soon be subjected to vibration testing to determine if it passes structural qualification testing.

ACKNOWLEDGMENTS

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