A BLIND TEST ON THE PULSE TUBE REFRIGERATOR MODEL (PTRM)

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ABSTRACT

The Stirling Refrigerator Performance Model (SRPM) has been validated extensively against the Lockheed built Stirling Coolers and various units in the literature. This model has been modified to predict the performance of the Pulse Tube Coolers (PTCs). It was successfully validated against a Lockheed in-house-built PTC. The results are to be published elsewhere[1]. In this paper, the validation of PTRM against a NIST (National Institute of Standards and Technology) orifice pulse tube cooler is reported. Dimensions and operating condition of the PTC were obtained from NIST without prior knowledge of the performance. In other words, this is a 'blind test' on the PTRM with the help of the National Institute of Standards and Technology. Good correlation was found between the test data and the prediction. PTRM is a generic model that gives accurate performance prediction of the pulse tube coolers.

INTRODUCTION

The Pulse Tube Performance Model (PTRM) was modified from the Stirling Refrigerator Performance Model (SRPM) which is a third order model that has been validated extensively against various Stirling coolers in the literature. They include the Lucas-Lockheed 60K unit[2], the NASA/Philips Magnetic Bearing unit[3], the Oxford refrigerators[4], and the Astronomic Infrared Sounders (AIRS) units A, B, and C. A detailed description of the model can be found in Reference 5. In this report, the PTRM model is validated against a large laboratory PTC built by the National Institute of Standards and Technology.

THEORY

The equations and assumptions used in the PTRM model were discussed elsewhere[5]. The model breaks up the pulse tube cooler into a number of nodes. The number of nodes in each section depends on the value of the state variables.

For examples, more nodes are required in the regenerator because of the large temperature difference and large pressure drop in the axial direction. Conservation of energy, momentum and mass are solved until the solutions converge. Equation of states and empirical equations for pressure drop and heat transfer are also used. No fudge factors are used in the program.

Figure 1 is a schematic diagram of the nodal network used in the Pulse Tube Refrigerator Model (PTRM).

This model was modified from the Stirling Refrigerator Performance Model. The expansion space of the SRPM was replaced by the pulse tube with an orifice and the surge volume. The volumetric variation and the flow passage (to the displacer motor) at the hot end of the regenerator (of the Stirling model) were also eliminated. The gas transport in the pulse tube is modeled as unidirectional laminar or turbulent flow, depending on the Reynolds number. Heat transfer in the axial direction is modeled as enthalpy flow whereas the radial heat transport is predicted by the forced flow heat transfer coefficient (for both laminar and turbulent regimes). The transport across the orifice is modeled by the discharge flow coefficient.

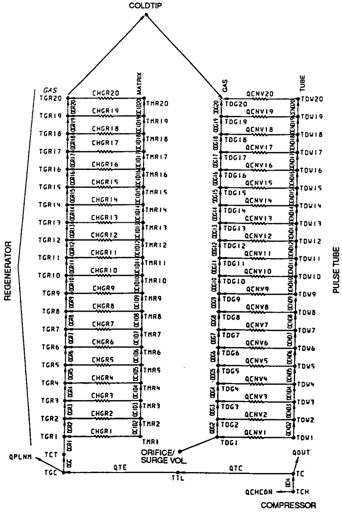

Figure 1. Schematic of the nodal network diagram.

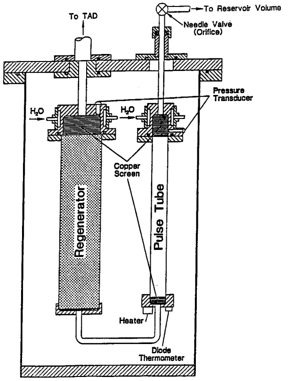

Figure 2. Cross-section NIST large pulse tube cooler.

EXPERIMENTAL SETUP

The pulse tube refrigerator was originally designed to be used with a thermoacoustic driver (TAD) that operated at a frequency of 30 Hz, a pressure ratio of 1.1, and an average pressure of 3.0 MPa. After some measurements in 1990, NIST changed the size of only the pulse tube and operated the pulse tube refrigerator with a mechanical compressor. The system achieved an efficiency of about 15% of Carnot when referred to input PV power and is one of the most efficient pulse tube refrigerator ever measured, in spite of the fact that it was not optimized for the conditions that it experienced with the mechanical driver. The measurements on this pulse tube refrigerator were made on January 1991.

Geometry

Figure 2 shows a cross-section of the pulse tube refrigerator. Not all of the dimensions are to scale on this drawing. The compressor and the connecting line are not shown in this figure. They would be connected in place of the TAD. A commercial dry-lubricated compressor was used. The top head with inlet and outlet valves was removed and replaced with a steel plate with a threaded hole for the output connection. Only one of the two cylinders was used to generate the oscillating pressure. This single piston/cylinder has a swept volume of 347.5 cm^3. At the outlet of the compressor was an aftercooler that consisted of seven copper tubes in parallel which were surrounded by water. Each of these copper tubes had an inside diameter of about 0.125 inch and a length of about 22 inches. At the exit of the aftercooler the working helium gas passed through 18 inches of 0.25 in. ID copper tubing before entering the 2 inch length of 0.285 inch ID stainless steel tube just inside the vacuum can. This stainless steel tube then connects on to the second aftercooler as shown in Fig. 2.

Table 1. Dimensions of three heat exchangers

	Aftercooler	Cold HX	Warm HX
Diameter (mm)	39.5	25.4	19.1
Length (mm)	19.0	8.4	19.1
Mesh (copper)	#80	#80	#80
Wire Diameter (mm)	0.140	0.140	0.140
Porosity	0.65	0.65	0.65
Gas volume (cc)	15.1	2.77	3.56

The dimensions of this aftercooler, the cold heat exchanger, and the pulse tube warm heat exchanger are given in Table 1. The dimensions of the regenerator and the pulse tube are given in Table 2.

The connecting tube between the regenerator and the pulse tube is stainless steel with a length of about 6 inches and an inside diameter of 0.230 in. The connecting tube between the pulse tube warm end heat exchanger and the needle valve is 4 in long and 0.23 in ID. The 11-turn needle valve has an orifice diameter of 3.18 mm. It is a Nupro needle valve, model #SS-4LA, with a full open C_v of 0.15. A few measurements were made with the secondary valve partially open, but the improvement in performance was barely noticeable. The reservoir volume is 2700 cc.

Operating Parameters

Table 3 lists the operating parameters and the performance of the pulse tube refrigerator when the second orifice was closed. The orifice settings used for these data are approximately equal to the optimum values.

RESULTS AND DISCUSSION

The prediction on the NIST PTC was performed without prior knowledge of the performance of the pulse tube. Inputs to the model were supplied by NIST on May 10, 1994. The prediction was completed on July 14, 1994 and the experimental performance was obtained from NIST and compared with the analysis on July 19, 1994.

Table 2. Dimensions of regenerator and pulse tube

	Regenerator	Pulse Tube
Outside diameter (mm)	41.28	25.4
Wall thickness (mm)	0.89	0.51
Length (mm)	208	208
Mesh	#200 stainless	NA
Wire Diameter (mm)	0.0533	NA
Porosity	0.69	NA
Gas volume	175.8 cc	97.1 cc

Table 3. Operating parameters and performance of pulse tube refrigerator

T_c (K)	37.5	43.0	58.9	78.4
T_h (K)	298.9	304.9	310.2	315.8
P Average (MPa)	1.739	2.551	2.504	2.488
P Ratio (pulse tube)	1.255	1.301	1.334	1.384
P Ratio (regenerator)	1.337	1.370	1.394	1.453
P Ratio (compressor)	1.404	1.424	1.445	1.491
Frequency (Hz)	4.6	4.5	4.5	4.5
Orifice turns	7.0	7.0	7.0	7.0
PV Work (W)	454	653	642	619
Net Cooling (W)	0	0	14.5	29.9

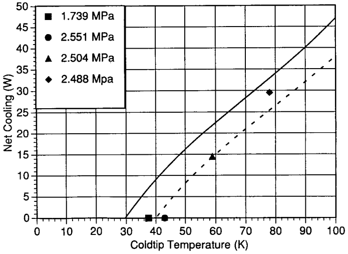

Figure 3. Net cooling versus coldtip temperature.

Figure 3 is a plot of the net cooling versus the coldtip temperature at two different fill pressures. (In both Figures 3 and 4, the solid line represents prediction at 1.739 Mpa average pressure; the dashed line represents prediction at 2.551 MPa). Excellent agreement is found between the prediction and the experimental data. Figure 4 shows the specific power (net refrigeration / PV power) of the unit versus the coldtip temperature. The agreement at low temperature is remarkable. The experimental data at 80K tends to give better performance than the prediction. The dis repancy is mainly due to the difference in the compressor PV work between the experimental data and the prediction. The predicted PV work is about 50% higher. This might have been caused by the uncertainty in knowing where the PV work was measured experimentally, and the unaccounted dead volumes in modeling the pulse tube system.

It should also be mentioned that the experimental data being compared here were taken with a fixed orifice setting (of 7 turns). As for the PTRM prediction, better net cooling was found with a large valve opening at high

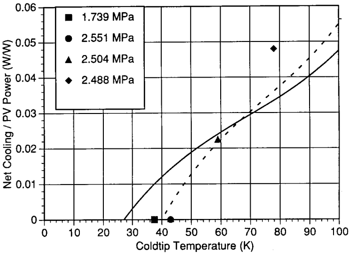

Figure 4. Specific power versus coldtip temperature.

temperatures, and a small valve opening at low temperatures. Unfortunately, at large orifice openings, the PV work is also much larger. This might also have contributed to the discrepancy between the experimental data and the prediction. Since the predicted net cooling at 80K is higher than the experimental data, one might be able to run the model at a smaller orifice opening to reduce the PV work and get even better correlation. In other words, running PTRM at a small orifice opening (within the limit) results in better specific power performance, but the net cooling is worse accordingly.

CONCLUSIONS

The Pulse Tube Refrigerator Model (PTRM) was validated against a NIST pulse tube cooler without prior knowledge of the performance. Good agreement was found between the prediction and the experimental data. The PTRM was proven to be a generic model that is invaluable for the analysis and design of pulse tube coolers.

REFERENCES

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