Validation of an Integrated Modeling Tool for Study of Pulse Tube Coolers

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ABSTRACT

Fluid models of a miniature pulse tube cooler are used to assess the feasibility of using a commercial fluid simulation tool for design and optimization purposes. Validation of initial coupled fluid models is provided via comparison with a lab prototype. The analysis incorporates coupled solutions of conductive and convective heat transfer in a realistic 3-D in-line pulse tube configuration, with consideration for compressibility and turbulence of the helium gas. Initial studies focus on establishing capability to determine phase relationships of fluid properties and spatial variations in enthalpy flow rate. Flow parameters from testing of a prototype pulse tube cryocooler are compared with analytical results obtained using the computational fluid dynamics software. Interpretation of the results demonstrates the usefulness of the model in diagnosis of key performance issues with the prototype.

INTRODUCTION

Closed-cycle cryocoolers have proven to be an enabling technology for Earth observation and space science applications, having accumulated heritage over the last 20 years. However, a limited number of qualified products are available for any particular mission that should arise. This is particularly apparent in the case of long-life pulse tube coolers on small/micro-satellite payloads where availability of low mass coolers may enable science payloads. A trend in selection of pulse tubes has continued for space-borne scientific instruments, which tend to prescribe to the philosophy of fewer moving parts. Efficiency of these coolers has been improved with the use of inertance tubes[1], which are used to achieve a controlled phase shift between mass flow and pressure, with lower losses compared with an orifice-type phase shifter. Since this aspect has been embraced, pulse tube cryocoolers have shown much improved efficiency. However, trends observed in small-sat buses, which are limited in power and mass resources, are not being matched by the rate of development of small cryocoolers, indicating that there is a gap in terms of cost and availability of proven long-life cooling technology for use on such platforms. Contributing to this slow evolution

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is a lack of standard tools to facilitate rapid development. Most coolers are developed using proprietary in-house software, with the exception of codes like REGEN 3.2.

1-D and 2-D simulations of complete pulse tube systems have been presented in the literature with some comparison between models, but less often versus test data. Sensitivity to geometry has led to an adherence to cylindrical geometries throughout designs. Many features contribute to 3-D disturbances, including transition fittings, transducer connections, coils and reservoir connections, as well as porous regenerator materials to a lesser extent. Losses studied in dual-opposed compressor configurations require the flow to turn dramatically in exiting the compressor. Sensitivity to gravity is a common challenge, which has shown varying effects believed to vary with scale of the cooler. Where these effects are generally seen as nuisances, other features with distinct 3-D flow aspects, including flat-plate type heat exchangers, can be exploited more effectively in the aftercooler, hot and cold heat exchanger sections. Clearly, these features can act positively as flow-straighteners or produce detrimental 'jet-effects' that cannot be represented in a 2-D model.

It is surmised that development can be hastened with the use of high-fidelity simulation techniques, reducing the amount of prototyping required and providing opportunities to generate more compact and higher efficiency devices through more flexible geometry. Analysis of individual components can usually be shown to correlate; yet models of complete systems are very sensitive to cumulative effects/errors resulting from simplified components. Prediction of the phase angles often performed using electrical equivalent models, which may differ significantly from measurement depending on the circumstances. Compressibility has been shown by analysis to be a factor over a wide range of test cases to reduce the phase shift between pressure and mass flow at the inertance inlet. Results of past modelling have demonstrated some success, though recent studies are reporting discrepancies in phase shift predictions with geometry variation and with application of different turbulence models.[2]

A decision to model the entire system in 3-D necessarily involves exploring computational limits. Such a model would allow a fuller view of heat transfer within the system and to the boundary conditions. Some aspects of accuracy are expected to improve over conventional 1-D and 2-D models, as loss effects related to transition elements and overall boundary conditions can be better represented. Heat transfer can be modelled throughout without simplifying assumptions for isothermal or adiabatic processes. Notably, ESC-TMG also possesses the capability to accurately model radiative parasitic heat loads through coupled model solutions with TMG™, with algorithms proven for modelling of complex spacecraft systems. This feature of the integrated software will be explored at a later time.

STUDY APPROACH

The cryocooler research program plan at CSA is exploring techniques to more rapidly optimize scaled versions of pulse tube coolers, though the software is likely to be useful for a range of refrigeration applications. A cost-effective program has been adopted to address the above objectives by establishing a test facility at the Canadian Space Agency (CSA) with the help of undergraduate students, while partnering with MAYA Heat Transfer Technologies. MAYA provides technical support for the computational fluid dynamics (CFD) analysis package called Electro-Systems Cooling (ESC™). CSA is performing the experimental validation of the models, while assisting MAYA to learn about the technology. The software applied for the study makes use of the IDEAS-TMG™ interface, a widely used thermal analysis package in the spacecraft industry that is capable of managing very large models. Based on existing skills with the TMG tool, it is reasonable to assume that someone with some thermal modelling experience could perform more advanced computations with the goal of reducing the number of prototyping iterations in development.

The key objectives of the project were established as follows:

- Demonstrate and explore the capability for realistic coupled fluid simulations in 3-D on a standard PC workstation with commercial software
- Use basic hardware to validate principles and provide a learning platform with room to evolve

- Determine areas of design that may benefit most from 3-D capability

The approach applied is summarized below:

- Model complete pulse tube (accepting simplification of compressor and regenerator)
- Use a known 1-D model (NASA ARCOPTR) as a baseline design
- Build a prototype based on the baseline parameters, with modularity of components
- Demonstrate phase shift phenomena in linear arrangement of a 3-D model
- Develop more realistic model including 3-D features, compressibility, buoyancy, fluid property nonlinearities, heat transfer throughout, turbulences models
- Incorporate regenerator pressure drop and surface area compensation into model
- Develop post-processing routines to calculate phase relationships between key fluid properties, characterize enthalpy flow at various cross-sections
- Perform correlation study of variation of geometry, frequency & pressure
- Correlate with measured pressures, apply model to improve prototype performance
- Final steps to incorporate radiation transfer for parasitic calculations, and to explore improved modelling of compressor and regenerator elements

COMPUTATIONAL FLUID MODEL OF CRYOCOOLER

The cryocooler model (Fig. 1) has been configured in ESC to account for effects of compressibility, heat transfer for all internal walls, turbulence, viscous-heating, buoyancy and nonlinear thermal properties based on the NIST12 pure fluids database. Initial studies have made use of a relatively simple mixing length turbulence model[3], though a more recent model confirms that a k-epsilon model offers significant improvements. Nonetheless, the mixing length model is described as follows:

$$\mu_t = \rho\, l^2\, S \tag{1}$$

where l is the mixing length and S is the mean strain rate, as defined by:

$$l = \min(f_l\, \kappa\, y_n,\ 0.09 y_{\max}),\ \text{where}\quad f_l = 1 - \exp\left(\frac{-y_n^+}{26}\right) \tag{2}$$

$$S = \sqrt{2 S_{ij} S_{ij}} \tag{3}$$

$$S_{ij} = \frac{1}{2}\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right)\ \text{within and}\quad S = \frac{dU}{dy_n} = \frac{u_s}{\kappa\, y_n}\ \text{at walls} \tag{4}$$

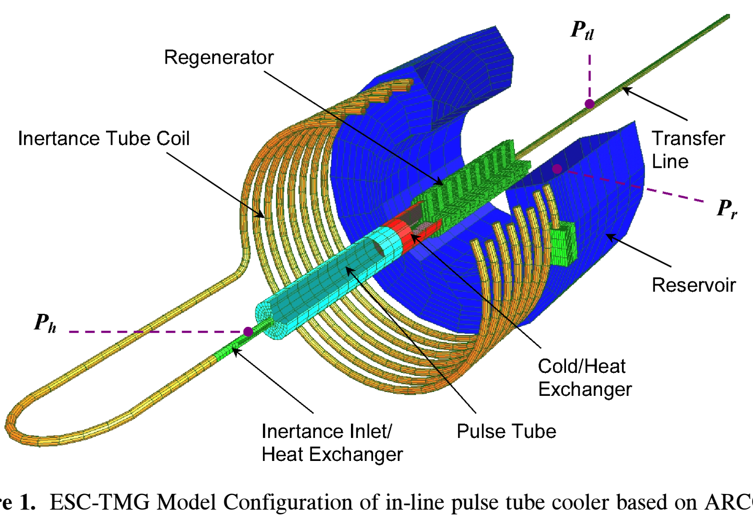


Figure 1. ESC-TMG Model Configuration of in-line pulse tube cooler based on ARCOPTR.

where κ=0.41 is the Von Karman constant; y_n is the normal distance from node to wall; y_{max} is a characteristic length scale for the model, and u_s is the shear velocity. As this model is not likely sufficient for regenerator modelling, a k-epsilon model is now being applied. Addition of a k-ω of turbulence is planned for inclusion in the software in the near future and will be considered when available.

Porous blockages are modelled using a resistance factor, R_{matrix}, related to the square of the velocity, which is consistent with a friction factor model as described in regenerator literature[4] in that it is related to the square of the velocity. The resistance force per unit volume is given by:

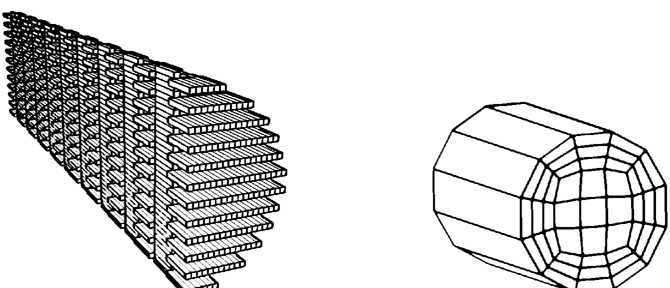
$$F = -\frac{1}{2}\rho\, R_{matrix} \left|\vec{V}\right|^2 \qquad (5)$$

where ρ is the gas density. ESC also allows for models that combine velocity and velocity-squared relations.

Conduction is easily modelled in all housing components, including the pulse tube walls. Nonlinear properties of heat capacity and thermal conductivity are considered in metals as well as the pure helium gas. Metals are modelled using properties obtained from the NIST on-line cryogenic database. By modelling heat-transfer at all internal surfaces, a better understanding of the distribution of rejected heat may be gained. A simplified regenerator has been attempted; with a coarse set of ducts assuming enhanced heat transfer. This was deemed acceptable for the start of the program, since this was not the focus of the study. However, by implementing a representation of the regenerator, the benefit is to have a view of the overall system, even if for the purpose of identifying related limitations.

The various components have been configured as follows using a total of 36900 elements in the flow model and 25600 nodes in the flow model, with all gas-affected elements configured to experience heat transfer effects:

1) 'Transient Vent' feature simulates the compressor. Such vents may be defined by data or a function for transient mass flow or pressure, also allowing for a transient temperature profile to be imposed as required. The aftercooler is not modelled.
2) The regenerator, shown in Fig. 2, is modelled coarsely with porosity ≈ 0.65 (close to the prototype value) using a manually meshed solid to achieve a distributed flow with heat exchange throughout the volume and some diffusion at the pulse tube exit. The goal is to represent the porosity of the regenerator matrix accurately enough to preserve time transient properties, while compensating for the lack of surface area using enhanced heat transfer. An enhanced heat transfer relationship is applied to account for reduced surface area of the model. The matrix is modelled with the heat capacity of stainless steel, but with reduced conductivity, intended for tuning to give representative stack conduction.
3) Cold-tip heat exchanger is modelled as a variation of a flat-plat heat exchanger with heat transfer to oxygen-free copper walls.



(a) (b)

Figure 2. Finite Element Model of Cryocooler: (a) simplified regenerator mesh model and (b) cross-section of inertance tube allowing for higher gradients toward the walls.

4) Pulse tube modelled with a 0.1 mm wall of titanium 6Al-4V, with heat transfer included.
5) The inertance tube is modelled with a wall of 0.75 mm. The cross-section of the model in Fig. 2 uses a coarse division with 9 elements across, but could be readily increased for future studies without difficulty. No temperature boundary was applied.
6) Reservoir volume modelled more coarsely (except at the exit of the inertance tube) with a solid shell mesh equivalent to 5 mm of copper surrounding the volume. The base of the reservoir is fixed at ambient temperature.

PROTOTYPE COOLER AND TEST STATION

The prototype cooler, which was never intended to represent the state-of-the-art, is shown in Fig. 3. It provides a baseline for the study, with no proprietary restrictions, and is intended as the subject of a sample investigation. It may be improved incrementally over time with the aid of the fluid model and test results. Thus, it is treated more in this case as a 'sick patient' requiring diagnosis and treatment. The laboratory cooler was designed based on the NASA ARCOPTR, which was formerly available to the public over the NASA internet site. This tool was used to select a baseline design. The baseline was intentionally selected to be a relatively small cooler, but perceived to be in the general operational ranges of existing miniature pulse tube coolers. For this reason, operating parameters in the range of 20-40 bar and 45-70 Hz were targeted. Some key parameters of the baseline prototype cooler are summarized below:

- Aftercooler: copper 80-count mesh (wires/inch)
- Regenerator: 10 mm diameter x 37 mm length, stainless steel 400-count, packing verified by mass measurement to an approximate porosity $\alpha = 0.63$
- Cold Heat Exchanger: copper 100-count mesh
- Pulse Tube: 8 mm diameter x 36 mm length
- Inertance Tube: 1.6 mm ID (varying lengths, first test at 1.4 m) terminated at a reservoir volume of 52 cm

To reduce costs, 400-count mesh was substituted in place of 500-count mesh, without any changes in the other cooler parameters. This would result in a lower pressure drop and impact performance from the outset, since the regenerator dimensions had been chosen based on the higher count mesh.

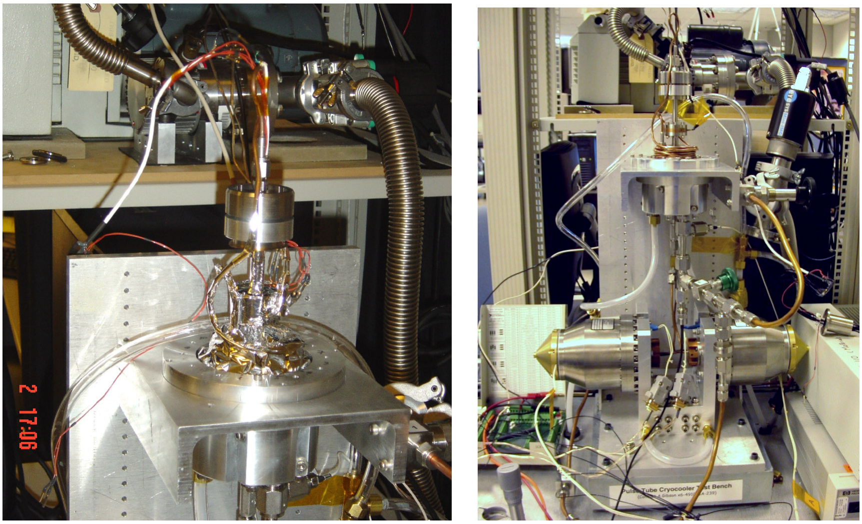


Figure 3. Prototype in-line pulse tube cryocooler based on ARCOPTR type design *(prototype shown at left, right is test station with back to back 80K BAe compressors).*

RESULTS OF PRESSURE TRANSDUCER MONITORING

Instrumentation of the test station includes three high frequency (<10 kHz) pressure sensors located at the transfer line (p_{tl}), hot end heat exchanger (p_h) and reservoir (p_r). Platinum resistance thermistors are used to monitor the temperature of the pulse tube cold-tip, reservoir, and hot-end heat exchanger. The two back-to-back BAe 80K compressors shown in Fig. 3 were used to provide the oscillating pressure input (2 x 1.8 cm^3 available) and were driven by a linear amplifier using open-loop control with the input signal generated by Labview™ routines. Compressor position pick-offs were sampled and available for phase studies. FFT routines were used to monitor amplitude of compressor piston position and pressure signals and to calculate relative phase angles between any pair of signals.

An initial set of tests was performed to characterize pressure drops in the setup for comparison with the model as well as to understand the basic functioning of the prototype. The initial inertance tube configuration was implemented with a length of 1.4 m of tubing, though this is perceived to be too short to achieve optimal performance of the unit. Nonetheless, it allowed for correlation with the CFD model and validation of the instrumentation. The ARCOPTR computer runs could not be updated to be more representative of the final configuration due to unavailability of the website. Simulations were performed under similar, not equivalent conditions, and have been included in Fig. 4 for reference.

Pressure drops between the sensors were measured, each with respect to the pressure at the hot-end heat exchanger section at the inlet to the inertance tube (essentially same pressure as inside pulse tube). The measure of (p_{tl}-p_h) includes the losses associated with the aftercooler, regenerator, cold heat exchanger and fittings, though dominated by the regenerator, whereas (p_r-p_h) is only associated with the drop across the inertance tube and associated fittings.

Phase shifts up to the 3rd harmonic were measured across the two main sections of the non-optimized cooler (short inertance tube, reduce mesh density and poor hot heat exchanger), with the results of the fundamental provided in Fig. 5. The test data show uniform trends with pressure and frequency variations. The phase shift at the reservoir was demonstrated to increase negatively with

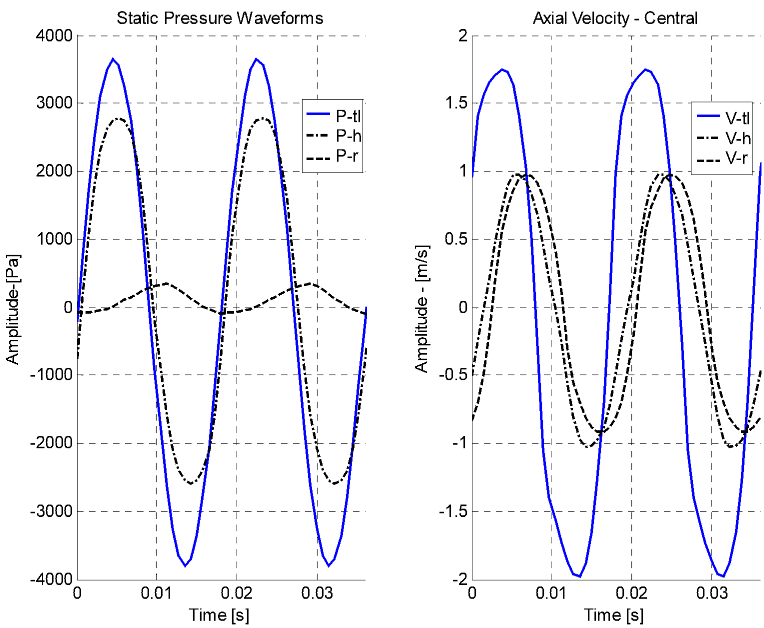


Figure 4. Predicted waveforms for (a) pressure and (b) axial velocity at three pressure monitoring points, demonstrating phase shifting due to the regenerator and inertance.

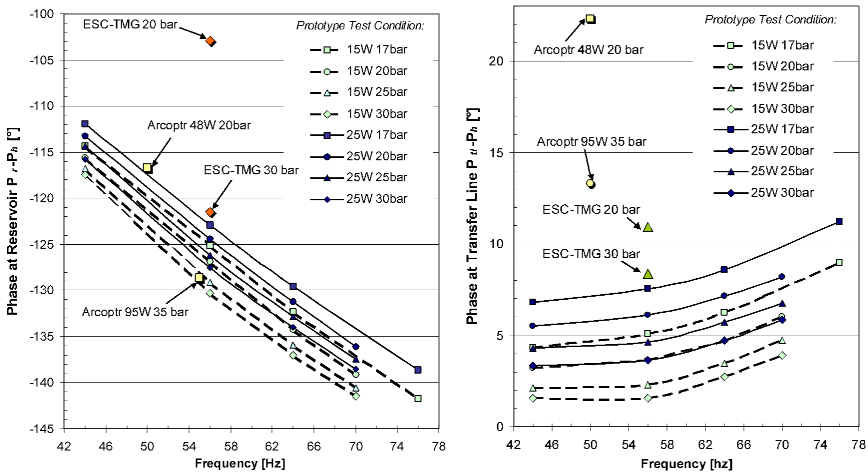


Figure 5. Phase differences between fundamental harmonics of pressure across regenerator and inertance tube sections *(Measured points indicated by series with lines, modelled points are indicated by individual points. Power numbers refer to measured or calculated PV power values.)*

increasing pressure and frequency. Phase shifts in the harmonics were also characterized, using the FFT-based algorithms, for comparison with harmonics obtained from the modelling. These results are not included herein. Multiple tests were performed in each case, with some careful averaging performed to compensate for reduced precision in frequency resolution at higher frequencies.

MODELLING RESULTS AND OBSERVATIONS

A simple 3-D model, with a linear inertance tube, typically took one day to complete 2-3 thermodynamic cycles of simulation running on a PC workstation. Initial runs of the more realistic CFD model geometry were troublesome and lengthy due to the presence of some distorted elements, most of which were representing the metal surfaces. Less effort had been made in meshing these areas as the focus had been placed on fluid elements. This improved dramatically with implementation of a squared element configuration within the regenerator. A total of 67 days of runtime (~20 cases) was dedicated to studying the convergence behavior of the model with varying relaxation parameters, residual requirements, regenerator resistance and heat transfer activation. The study led to somewhat predictable conclusions that small, steady gains could be made with use of smaller timesteps and tighter residuals.

Some convergence issues surfaced periodically, with the highest imbalance numbers in mass always present in the regenerator section. After testing the k-epsilon model, which immediately improved stability of the calculation, it is believed that the coarseness of the simplified mesh, particularly in the direction of flow does not lend itself to modelling flow that must turn rapidly around the torturous path within the regenerator. Imbalance numbers for energy of less than 10% were shown to be attainable (2-3 cycles in 3-8 days for a 3.2 GHz processor), which is reasonable considering the scale of the problem and the relative simplicity of the meshes used for both the regenerator and tubing elements (shown in Fig. 6 and previously in Fig. 2). Further improvements with residual, step size and grid refinement can be anticipated.

For the analysis, the first cycle was discarded to allow for pseudo-stabilization of the cycle. The remaining cycles were used to perform and FFT to study frequency content and relative phase. Two of the successful mixing length runs are discussed here as a preliminary assessment of feasibility of using the technique. The first cases were for 56 Hz operation at 20 and 30 bar fill pressures as illustrated in Fig. 7. The reliability and speed is expected to improve significantly with some refinement of the model in these areas, as convergence of the model is improved.

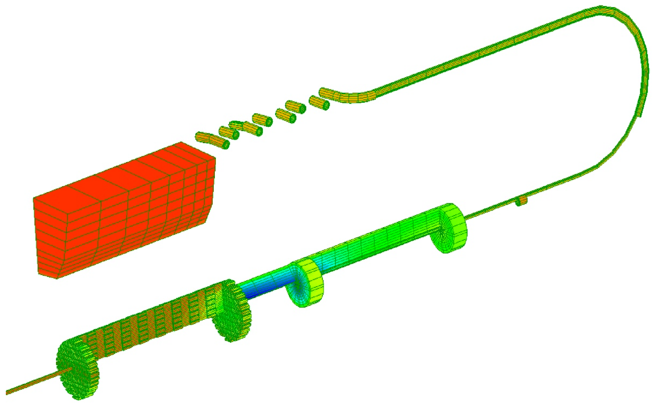


Figure 6. Cutaway view showing key pulse tube components, with a pronounced temperature distribution developing in the fluid toward the end of the first thermodynamic cycle.

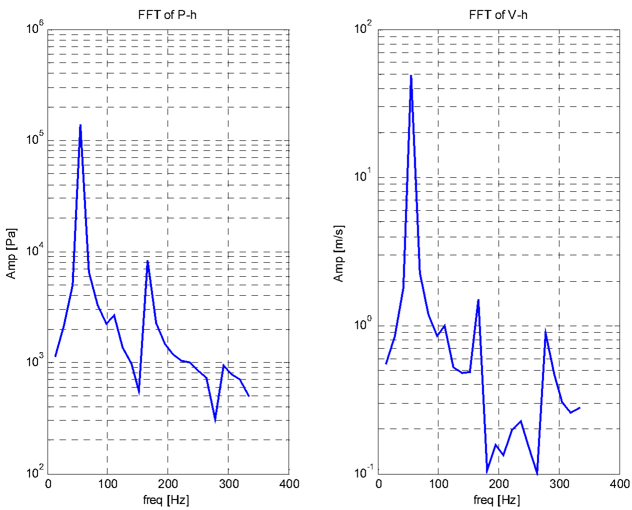


Figure 7. Typical calculated FFT response for the modelled pressure waveform at the hot end of the pulse tube (case shown is for 30 bar, 55.6 Hz).

A Matlab™ routine was written to calculate pressure drop and phase shift in both mass flow/velocity and pressure. The routine was used to monitor shifts of the fundamental, 1st, 2nd and 3rd harmonics, with respect to the pressure p_h. A second routine was composed to map and integrate enthalphy flow at any point in the model, or in the transient run. This made use of the density, velocity and temperature solutions from the ESC-TMG run.

$$\frac{1}{\tau}\int_0^\tau \oiint C_p \rho \, V_z T \, dAdt \tag{6}$$

The enthalpy flow calculation, based on Kuriyama[5], was used to successfully demonstrate a primary deficiency in the prototype cooler & model. As suspected, the jet disturbance at the hot end of the pulse tube results in circulation and effectively cancels much of the strong flow characteristic at the centre of the tube. In this configuration, the ESC-TMG model predicted much lower net enthalpy flow compared with the ARCOPTR runs. In fairness, the ARCOPTR design assumes an ideal heat exchanger, and it is clear that it is easy to use meshes to improve upon the baseline model applied for this study. Results for the 2 ESC-TMG runs mentioned above are

Table 1. Summary of 1-D, 3-D and lab test results with similar parameters.

Parameter	ARCOPTR-1	ESC-TMG1	Ref Lab Test-1	ARCOPTR-2	ESC-TMG3	Ref Lab Test-2
Fill Pressure [bar]	20	20	20	30	30	30
Frequency [Hz]	50	55.6	56	50	56.6	64
$P_{ratio}=P_{tl0}/P_a$	1.95/20	1.90/20	1.73/20	2.17/30	1.90/30	1.85/20
Regenerator Screen	500 mesh	R=0.8/ cm	400 mesh	500 mesh	R=0.8/ cm	400 mesh
Inertance Length [m]	1.7	1.4	1.4	1.7	1.4	1.4
Inertance OD [mm]	1.3	1.6	1.6	1.6	1.6	1.6
Phase (vs.P_h)						
θ_{Ptl}	22.6	10.9	6.1	15.3	8.36	3.9
θ_{Pr}	-116.8	-102.9	-124.5	-117.7	-121.5	-135.2
$\theta_{mflow\text{-}tl}$ or θ_{Vtl}	46.5	32.2	-	43.8	25.0	-
$\theta_{mflow\text{-}h}$ or θ_{Vh}	-10.2	-15.6	-	-11.9	-18.9	-
$\theta_{mflow\text{-}r}$ or θ_{Vh}	-28.6	-37.2		-29.4	-36.7	
H_{ptc} [W] Enthalpy Flow	4.1	2.1	-	6.4	0.86	-
Turbulence Model	Mixing Length		N/A		Mixing Length	N/A

included in Table 1, with a graphical representation of the time-averaged profile across the pulse tube toward the cold heat exchanger shown in Fig. 8. The negative enthalpy flow contribution from the outer annulus is directly apparent and nearly cancels the positive effect of the central region. Notably, the performance reduces with pressure according to the ESC-TMG result. This was observed in the poor performance of the lab prototype, where the pressure had to be reduced to 17

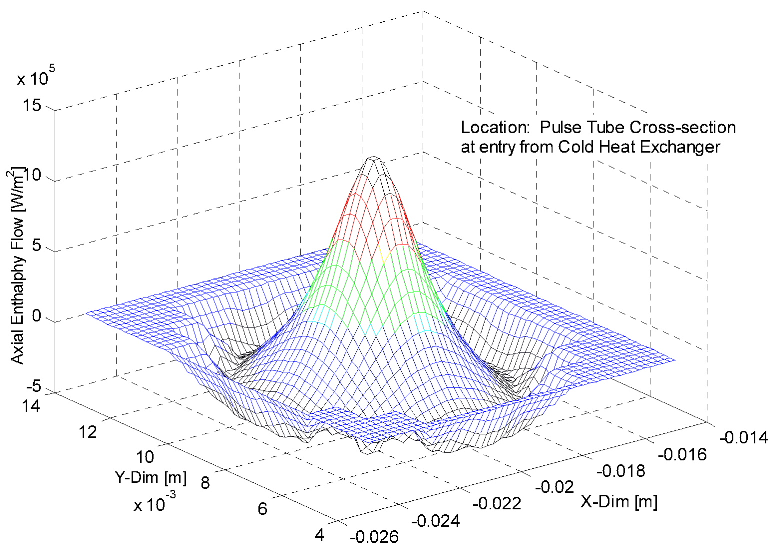


Figure 8. Enthalphy Flow per unit area in pulse tube (time-averaged for 20 bar, 56Hz condition).

bar from the intended 30-40 bar range in order to obtain 80 degrees of cooling. Results appear a bit rough at the edges due to larger spacing between nodes and the simple interpolation algorithm applied.

Where the fluid model realism can be fully validated, there exists an opportunity to develop designs using more radical geometries, materials, and processes that are better suited for scaling of microcoolers.

CONCLUSIONS AND RECOMMENDATIONS

A capability to conduct correlation studies has been established, and some encouraging results have been obtained from a three-dimensional fluid model of a pulse tube cryocooler. Predictions of pressure phase shift along the inertance line were consistent with measurements at fundamental frequencies, following similar trends. The regenerator model was demonstrated to be inadequate in its current form as a design tool, but is considered to be workable for system-level studies, given some improvements and further correlation studies are performed with tests and/or other regenerator design software. Enthalpy calculations performed on the results files provided insight to better understand the performance of the prototype.

It is recommended that a finer mesh be applied to the regenerator model to resolve turbulence better, in combination with more sophisticated turbulence models, with the aim to improve imbalance figures of the model further. The lab prototype should be refitted with better heat exchangers at both cold and hot ends of the pulse tube and a higher density mesh (500-count) to better represent the ARCOPTR baseline, allowing for higher phase shifts due to this resistance.

Subsequently, the unit should be characterized for a few different inertance geometries, to test the model over a wider range of conditions. Higher power runs should be tested to directly correlate the effect of compressibility versus models, with and without this feature activated. Finally, performance analysis routines are to be adapted to use the industry standard method to communicate operating conditions via phasor diagrams, for both the model and prototype results.

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