Development of a Diaphragm Stirling Cryocooler

A. Caughley[1], M. Sellier[2], A. Tucker, M. Gschwendtner[3]

[1]Callaghan Innovation, Christchurch, New Zealand
[2]University of Canterbury, Christchurch, New Zealand
[3]TS-dot Engineering, Auckland, New Zealand

ABSTRACT

Callaghan Innovation, formerly Industrial Research Ltd, has developed a novel free-piston Stirling cryocooler concept using metal diaphragms. The concept uses a pair of metal diaphragms to seal and suspend the displacer of a free-piston Stirling cryocooler. The diaphragms allow the displacer to move without rubbing or moving seals, thus resulting in a long-life mechanism. When coupled to a metal diaphragm pressure wave generator, the system produces a complete Stirling cryocooler with no rubbing parts in the working gas space. Initial modeling of this concept using the Sage modelling tool indicates the potential for a useful cryocooler. A proof-of-concept prototype was constructed and achieved cryogenic temperatures. CFD modeling of the heat transfer in the radial flow fields created by the diaphragms shows the possibility of utilizing the flat geometry for heat transfer, reducing the need for, and the size of, expensive heat exchangers.

A second prototype has been designed and constructed using the experience gained from the first. Further CFD modeling has been used to understand the underlying fluid-dynamic and heat transfer mechanisms and refine the Sage[1] model. The prototype produces 29 W of cooling at 77 K and reaches a no-load temperature of 56 K. This paper presents details of the development, modeling and testing of the second iteration prototype.

INTRODUCTION

Callaghan Innovation has developed a novel metal diaphragm based pressure wave generator technology[2–4] that has been successfully used with pulse tube refrigerators [5–8]. To date, 15 metal diaphragm pressure wave generators have been made and operated with swept volumes from 20 cc to 1000 cc and with 500 W to 30 kW input power, respectively. The diaphragm concept uses a pair of opposed metal diaphragms to balance each other's average gas forces, transferring only the forces from the pressure oscillation to the driving mechanism.

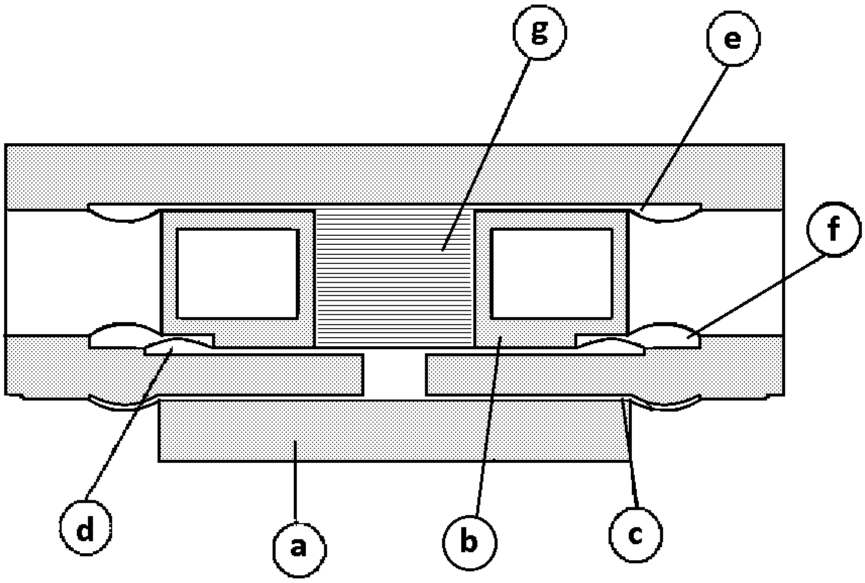

Figure 1. A cross-section of the diaphragm version of a free-piston system concept. a) Pressure wave generator driving piston, b) displacer, c) pressure wave generator compression space, d) warm side of displacer, e) cold expansion space, f) bounce space, g) regenerator.

The concept discussed in this paper is illustrated in Figure 1, it uses a pair of metal diaphragms to suspend the displacer of a free-piston Stirling cryocooler. The regenerator is housed within the displacer and an intermediate membrane separates the equivalent of the traditional bounce space[9] from the main pressure wave. Early Sage modelling was very promising and a proof-of-concept prototype was constructed[10]. The proof-of-concept prototype demonstrated cooling at cryogenic temperatures, but did not perform as well as initially expected. Refinement of the Sage model to accurately reflect compromises made in the manufacture of the prototype produced a good correlation with experiments.

The learnings from the proof-of-concept prototype initiated the construction of a second prototype with smaller displacer diaphragms and a central bounce space. Computational Fluid Dynamics (CFD) modeling was performed to better understand the heat transfer in the radial flows that exist in the gas spaces at the ends of the displacer. ANSYS® CFX[11] was the CFD software used and a series of validation experiments[12] confirmed its ability to accurately model the flows and heat transfers involved in oscillating compression typical of Stirling machines.

The following sections document the continuing development of the diaphragm Stirling cryocooler. The results of the testing of the second prototype are described. Characterization experiments at 200 K are compared with the Sage model of the cryocooler. Prototype operation at 77 K is compared to Sage and CFD models. Insights from the CFD modeling of the internal gas behavior at 77 K are discussed.

SECOND PROTOTYPE DEVELOPMENT

The second prototype, Figure 2, was designed in response to lessons learnt from the proof-of-concept prototype. The key improvements implemented on the second prototype were: removal of dead volume at the outer circumference of the expansion space; smaller diaphragms for the displacer to achieve a better swept volume to dead volume ratio; and a softer intermediate diaphragm to separate the bounce space. The second prototype was fitted to the same CHC200 pressure wave generator as the proof-of-concept prototype. The CHC200 pressure wave generator has a swept volume of 200 ml and is capable of efficient operation from 10 to 60 Hz. The displacer was suspended with diaphragms developed for the CHC60 pressure wave generator, which has a swept volume of 60 cc with a full stroke of 1.7 mm. Sage modelling indicated that the displacer should be as heavy as possible, so the displacer was made from solid stainless steel. The tube along the regenerator wall was used to take the compression from the gas forces on the displacer.

The intermediate diaphragm and annular bounce space of the proof-of-concept prototype was replaced with a central chamber sealed by a rubber diaphragm, thus minimizing the spring stiffness on the displacer. Rubber is suitable for the intermediate membrane seal as it only has to withstand the pressure wave, not to hermetically seal the whole gas pressure. The movement of the displacer was small and hence stresses in the rubber were very low which leads to a long life. The bounce space is not sealed from the compression space; a slow leakage between the bounce space and the rest of the machine ensures that the bounce space maintains the same average pressure as the cryocooler.

Sage was used to optimize the design. Figure 2 shows the second prototype design with reference to the components in the Sage model. The Sage model's predicted performance is shown in Figure 3. The refrigeration effect was predicted to be 75 W at 77 K when operating at 50 Hz frequency with 20 bar gas pressure. The Sage model accounted for 31 W of parasitic losses internal to the radiator matrix, but not for conduction losses down the displacer and tension walls. Conduction losses in the displacer were calculated to be 30 W and radiation within the displacer another 3 W. Accounting for losses, the net cooling power was then expected to be 42 W at 77 K.

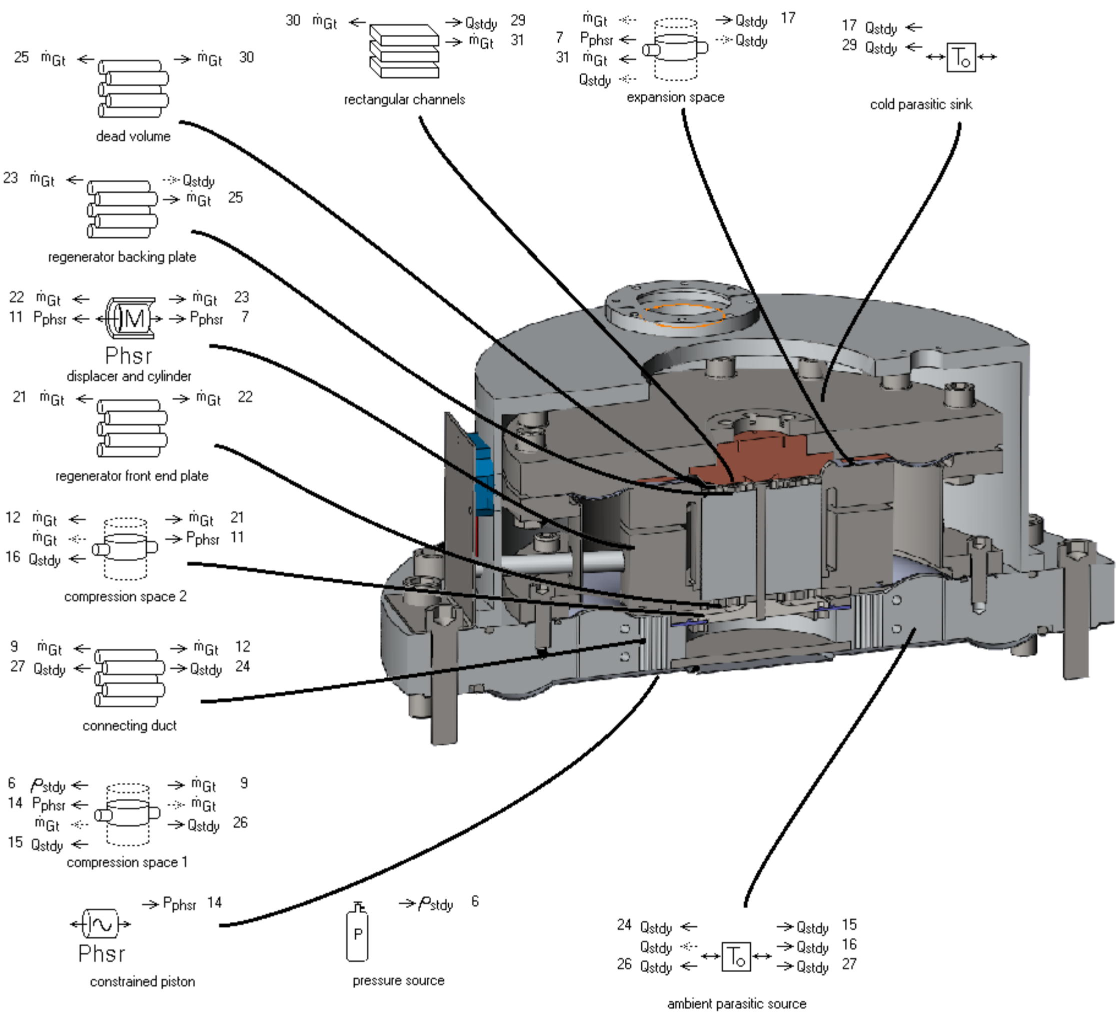

Figure 2 The second prototype and its associated Sage model components.

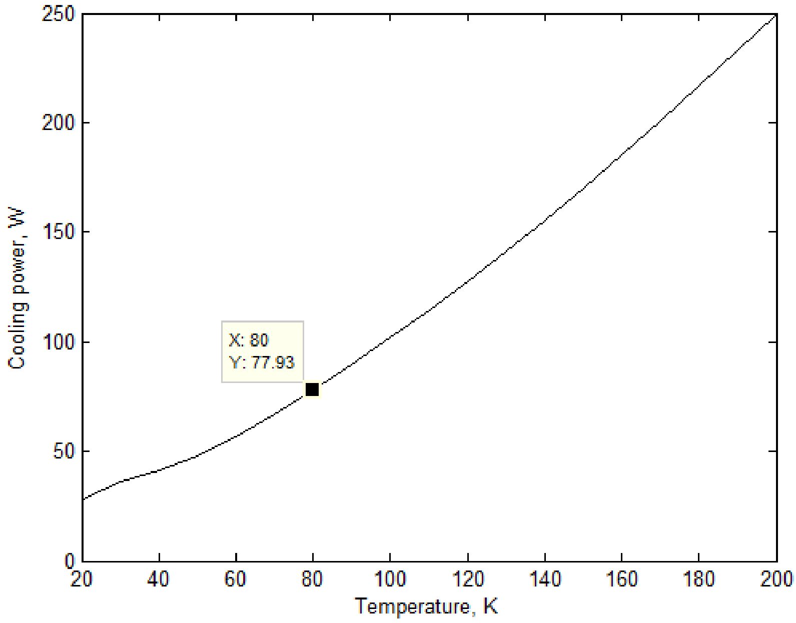

Figure 3. Sage model prediction of the second prototype's cooling power when operating at 50 Hz and 20 bar gas pressure.

VALIDATION EXPERIMENT AT 200 K

A series of characterization experiments were performed to validate the Sage model prediction of the displacer movement. The configuration of the cold head did not allow for the displacer's position sensor to be mounted within the cryostat, so experiments were performed in air with a cold plate temperature of approximately 200 K. Two cases were used for the validation. The first case was as-designed with the intermediate diaphragm and bounce space. The second case was with the intermediate diaphragm removed and the bounce space filled with a close-fitting aluminium plug. Figure 4 shows the predicted and measured displacer amplitude for both cases over a range of frequencies from 30 to 60 Hz. Figure 5 shows the predicted and measured displacer phase angles. The Sage model predicted the phase of the displacer movement well, with a clear distinction between the two cases. The experiment and the Sage model agreed well for the displacer amplitude in the case without the bounce space. However, when the bounce space was added, the experimentally measured amplitude was 0.1 mm less than predicted across the frequency range.

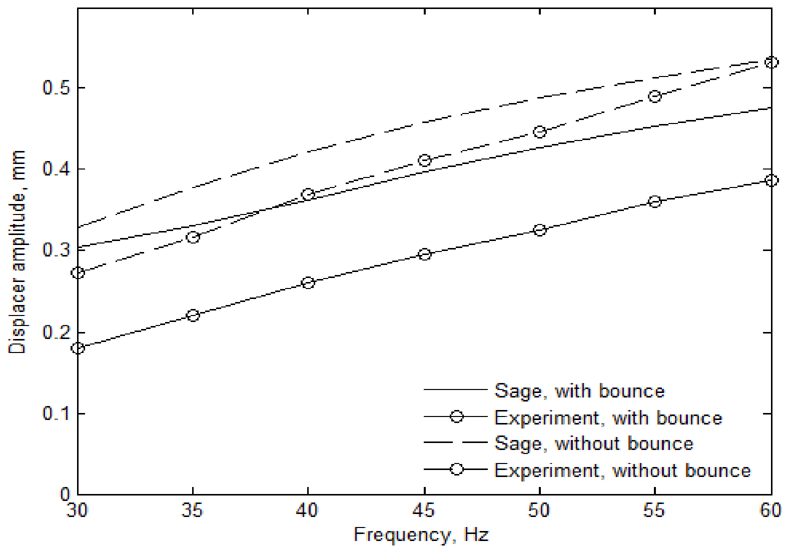

Figure 4. Experimental displacer amplitude at 200 K compared with Sage prediction.

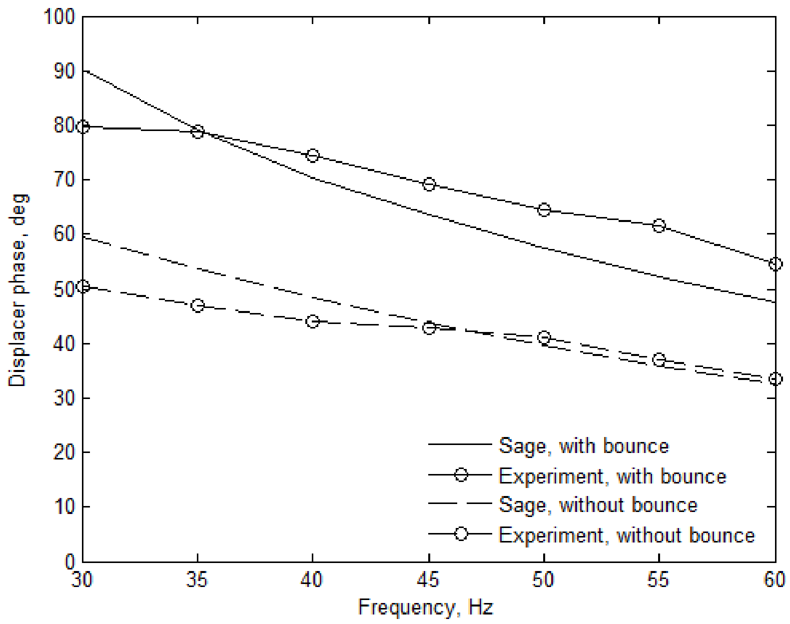

Figure 5. Experimental displacer phase at 200 K compared with Sage prediction.

COOL-DOWN AND PERFORMANCE AT 77 K

The cold head was fitted with the cryostat, multi-layer insulation, and a vacuum of 10^{-4} mbar was applied. Figure 6 shows the cool-down test. The cooler was initially run at 50 Hz and 20 bar gas pressure as per the Sage model. After 8 hours running it had reached 70 K. The speed was increased to 55 Hz and after 23 hours it had stabilized to 62 K. The speed was increased to 60 Hz and the gas pressure was increased to 22 bar, which resulted in a final temperature of 55.6 K on the cold face. The long cool-down time was due to the large heat capacity of the 23 kg stainless steel cold face. The cooling of the cold thermal mass can be considered as a dynamic load on the cooler, giving a quasi-static cooling power of 8 W at 77 K, for 50 Hz and 20 bar gas pressure.

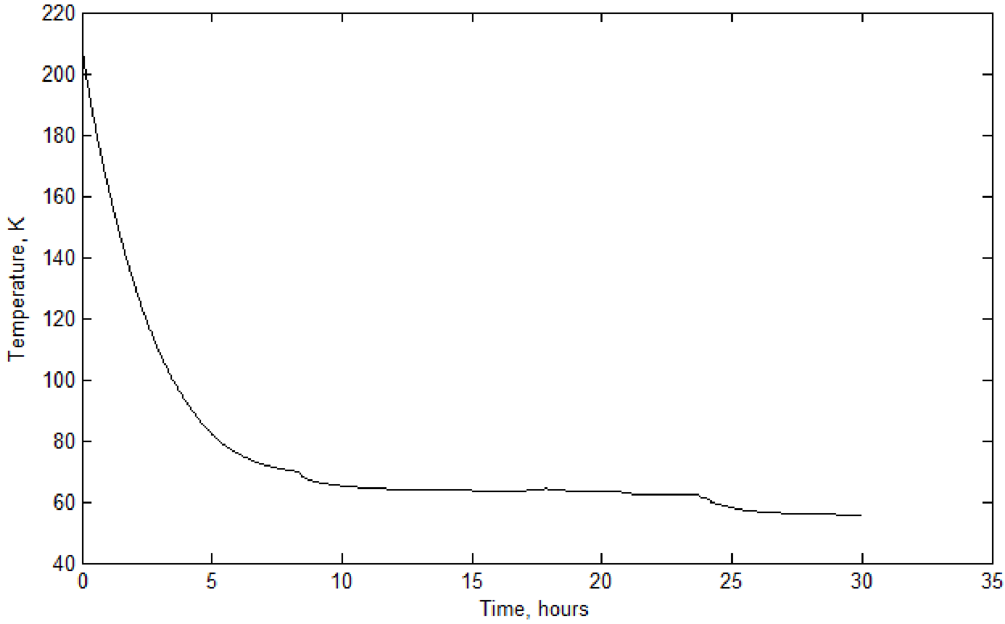

Figure 6. Cool-down test resulting in a final low temperature of 55.6 K.

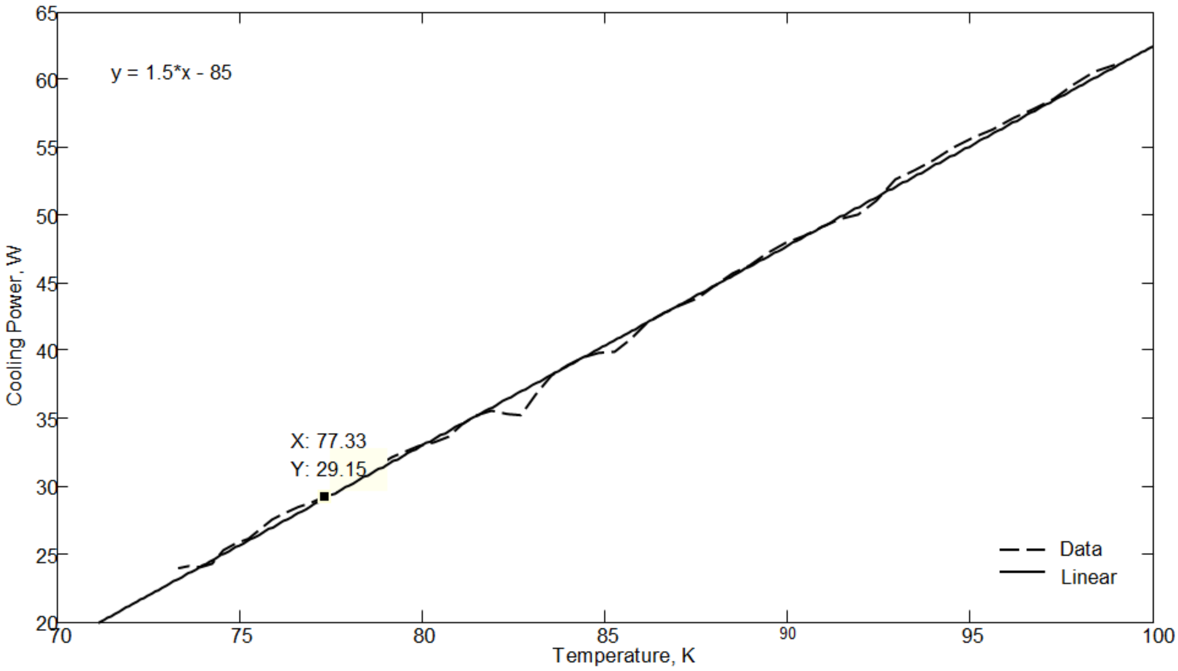

Figure 7. Net cooling power of the ST 200 cryocooler at 60 Hz.

A second cool down test was run at 60 Hz with a gas pressure of 22 bar. The cooling power was again calculated from the cool-down rate of the cold mass. Figure 7 shows the cooling power as a function of the cold temperature; 29 W of cooling power was achieved at 77 K. The reverse of the cool-down rate calculation was used during warm-up to estimate the heat load into the cold mass at a given temperature. The heat load was 60 W at 77 K which is very close to the expected parasitic losses of 63 W. Re-running the Sage model at 60 Hz, 22 bar gas pressure, and accounting for 33 W of conduction losses, produced a net cooling power of 65 W at 77 K.

CFD MODEL

Computational Fluid Dynamics (CFD) can model the fluid flow, compression and heat transfer in a three dimensional gas space and therefore has the potential to model the cryocooler more accurately than the one-dimensional modeler Sage. A CFD model of the cryocooler gas space was developed using ANSYS® CFX software[11]. CFX was validated against experimental work on gas spring dynamics by the author and detailed in a previous publication[12] . The displacer movement phase and amplitude for 77 K operation was taken from the Sage model.

The cryocooler gas space was modeled as five degree wedge with symmetry conditions on the wedge faces to produce a quasi-axisymmetric 2D model. The wedge contained 3D features such as the transfer holes and cold heat exchange slots. Figure 8 shows the computational domain.

The warm faces of the model were isothermal at 300 K and the cold faces held at 77 K. Initial conditions for gas temperatures were set to 300 K and 77 K for the warm and cold gas, respectively. A linear temperature gradient was set on the regenerator. The mesh was a combination of tetrahedral elements and extruded triangle or square elements. Sizing of the mesh was such that there were at least four elements across a heat exchange space and extra inflation layers were added in critical areas such as the cold surfaces. The analysis was run as a transient model for 50 cycles with 360 steps per cycle, the equivalent of 1 second elapsed time. It took approximately two weeks to complete the analysis on an i7 computer running seven cores. The regenerator matrix's heat capacity considerably slowed the analysis' ability to reach steady state. To speed up the process, after a run of 50 cycles, the final temperatures of the matrix and gas at the regenerator ends were set as initial conditions for the next run, with the regenerator being reset with a linear temperature gradient again. In this way, the movement of the central part of

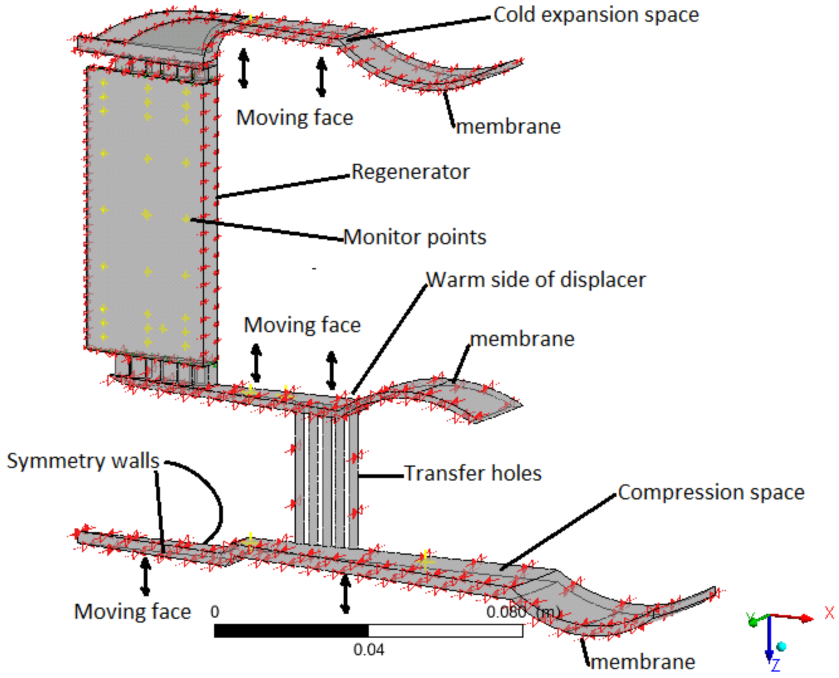

Figure 8. CFD Model of gas space in diaphragm Stirling cryocooler. The red arrows denote the symmetry faces of the wedge.

the regenerator's temperature profile was sped up. The process was repeated until the regenerator ends changed less than two degrees in a 50 cycle run.

The model's final prediction, for the cryocooler running at 60 Hz with 22 bar charge pressure and using Sage's displacer movement, was 131 W @ 77 K. Reducing the displacer's amplitude by 0.1mm, the difference between the Sage model and 200 K experiment, reduces the CFD's prediction of cooling power to **85** W at 77 K. Like the Sage model, the CFD modeled the conduction in the regenerator but not the 33 W of heat leak in the displacer. The resultant net cooling power of the prototype was then predicted to be **52** W at 77 K.

Figure 9 shows plots from the CFD model of temperatures, heat flux, and gas velocities at the point of maximum net heat flow into the expansion space. Maximum heat flow occurred 45 degrees before bottom dead center of the pressure wave generator piston. The plots show that cold heat transfer occurred predominantly in the center of the cold expansion space, where the gas was the coldest and flow velocities were the highest. The gas in the outer part of the expansion space did not take a large part in the refrigeration cycle. This dead volume of gas compressed and expanded as in a gas spring. Significant radial flows on the warm side of the displacer were observed, which contributed to the bulk of the machine's heat rejection.

DISCUSSION AND CONCLUSIONS

The second prototype performed considerably better than the proof-of-concept prototype; it achieved 29 W of cooling at 77 K and had a no-load temperature of 55.6 K. For the same 60 Hz and 22 bar gas pressure as the experiment, Sage predicted 65 W of cooling power at 77 K, while the CFD model predicted **52** W.

Sage's predicted cooling power is very good compared to the CFD model and experiment, especially considering that Sage assumes that the thin flat compression and expansion spaces are actually long tubes[13] of the same hydraulic diameter and volume. While the hydraulic diameter is maintained between the two geometries, the velocities and distances that the gas travels in a cycle are very different. The tube produces higher velocities due to the smaller flow area and higher pressure losses due to its longer length. Therefore heat transfer and flow losses would be

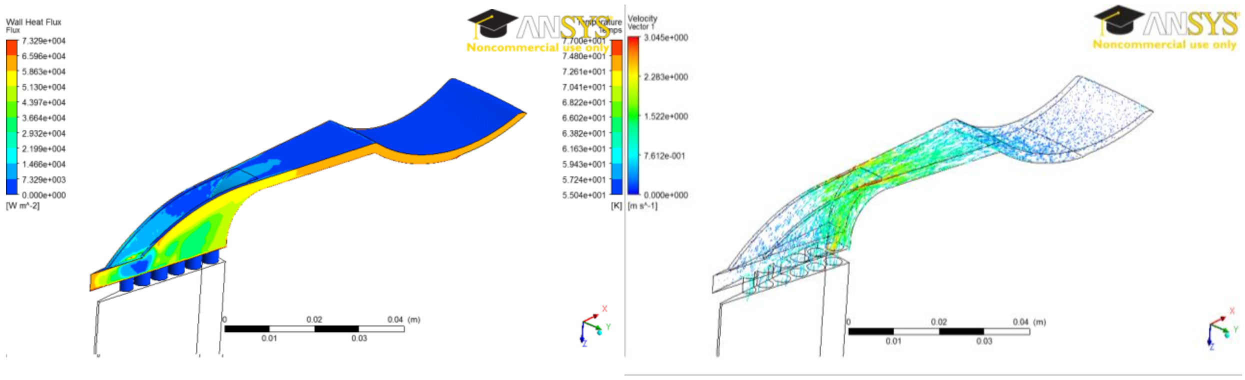

Figure 9. CFD plots of the cold gas at the point of maximum wall heat flux into the gas. Left, gas temperatures (on wedge sides) and wall heat flux of the cold section of the cryocooler. Right, gas velocity vectors.

over-estimated. The over-estimation of heat transfer would lead to an overestimation of the cooling power; which is then, in part, counter-acted by the extra entropy generated by the over-estimation of flow losses.

The Sage prediction of the displacer's movement amplitude at 200 K was larger by 0.1 mm than measured experimentally for the case with the bounce space. Without the bounce space, Sage accurately predicted the displacer amplitude. For the displacer phase, the correlation between Sage and the experiment held for both cases, with and without the bounce space, over a wide range of frequencies.

A CFD model of the second prototype was produced which, after some considerable running time, predicted a net 98 W of cooling at 77 K, using Sage displacer movement. Reducing the displacer amplitude by 0.1 mm to match the experimental value reduced the predicted net cooling to 52 W at 77 K which is much closer to the experimentally measured value. The 46 W reduction in cooling power with reduced displacer amplitude shows a high sensitivity of cooling power to displacer movement amplitude.

The CFD model confirmed the experimental observations that most of the cooling happened at the centre of the radial expansion space. The model showed that the centre of the expansion space is the area where the gas is coldest and has the highest velocities.

The first area for future improvement of the cryocooler is to increase the amplitude of the displacer movement while keeping its phase angle high. The Sage model is a good tool to investigate options for optimizing the displacer and the bounce space to get a better movement. A second area for improvement is to increase the heat transfer at the centre of the expansion space and to encourage more heat transfer on the warm side of the displacer. The CFD model is a good tool to optimize the gas/wall heat transfer in the warm and cold spaces. The third possible area of improvement is to minimize the dead volume in the cold gas.

ACKNOWLEDGMENT

The authors acknowledge Callaghan Innovation, New Zealand, for support of this work, New Zealand's MBIE for funding, the University of Canterbury, and Fabrum Solutions Ltd for commercializing the technology.

REFERENCES

1. Gedeon, D. Sage Modelling Software. at <http://sageofathens.com/>

2. Caughley, A. J. & Wang, C. Development of a Diaphragm Pressure Wave Generator for Cryocoolers. *Cryocoolers 15* 309–315 (2008).

3. Caughley, A. J., Haywood, D. J. & Wang, C. a Low Cost Pressure Wave Generator Using Diaphragms. *Adv. Cryog. Eng.* **53B,** 1122–1129 (2008).

4. Caughley, A. J., Emery, N. & Glasson, N. D. Diaphragm Pressure Wave Generator Developments At Industrial Research Ltd. in *Adv. Cryog. Eng.* 695–702 (2010).

5. Wang, C., Caughley, A. J. & Haywood, D. J. Development of a Low Cost High Frequency Pulse Tube Cryocooler. *Adv. Cryog. Eng.* **53B,** 1555–1561 (2008).

6. Emery, N., Caughley, A., Glasson, N. & Meier, J. Co-Axial pulse tube for oxygen liquifaction. *Adv. Cryog. Eng.* **57A,** 183–189 (2012).

7. Emery, N., Caughley, A., Glasson, N., Tucker, A. & Gschwendtner, M. Development of a High Frequency Pulse Tube. in *Cryocoolers 16*, 175–182 (2011).

8. Emery, N. *et al.* Co-axial pulse tube development. *Cryocoolers 17* **1,** 135–141 (2012).

9. De Waele, A. T. A. M. & Liang, W. Basic dynamics of split Stirling refrigerators. *Cryogenics.* **48,** 417–425 (2008).

10. Caughley, A., Tucker, A., Gschwendtner, M. & Sellier, M. Novel diaphragm based Stirling cryocooler. in *Adv. Cryog. Eng.* **667,** 667–674 (2012).

11. ANSYS® CFX Release 13.0, ANSYS, Inc.

12. Caughley, A., Sellier, M., Tucker, A. & Gschwendtner, M. CFD Modelling of a Diaphragm Stirling Cryocooler. in *Cryocoolers 17* 283–292 (2012).

13. Gedeon, D. Sage User ' s Guide. (2013).