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Minimising flow losses within the pulse tube of a Stirling pulse tube cryocooler

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Abstract. Successful operation of Stirling pulse tube cryocoolers relies on minimising flow mixing within the pulse tube. Hence, the pulse tube and the flow straighteners at either end must be designed with great care. In this study, the flow within the pulse tube of an existing Stirling pulse tube cryocooler is numerically analysed and alternative flow straightener designs are suggested. The numerical simulation have been carried out using CONVERGE CFD, a finite volume Navier-Stokes solver. The standard k-ε RANS turbulence model has been used to account for the effects of turbulence within the pulse tube.

1. Introduction

Stirling pulse tube cryocoolers (SPTCs) are small low temperature refrigerators that can provide cooling for electronic devices such as infrared sensors and superconducting devices. The pulse tube within an SPTC acts as a gas spring and replaces the cold end displacer used in traditional Stirling cryocoolers. In order to ensure successful operation, the gas within the pulse tube of an SPTC needs to remain stratified and flow mixing must be minimised. Hence, the inlet and outlet to the pulse tube must be designed with great care.

A schematic of a typical in-line SPTC arrangement is shown in Fig. 1. The assembly consists of:

- An oil free linear compressor.
- A regenerator which is typically filled with fine stainless steel wire mesh (REG in Fig. 1).
- A cold end heat exchanger which is typically filled with copper wire mesh (CHX in Fig. 1).
- A pulse tube (PT in Fig. 1).
- A warm end heat exchanger also filled with copper wire mesh (WHX in Fig. 1).
- A phase control mechanism which can either consist of a long inertance tube and a reservoir [1,2] or a phase shifter [3,4].
- A vacuum chamber around the cold head assembly, consisting of the regenerator and the pulse tube, in order to minimise convective losses.

The heat exchangers either side of the pulse tube also act as flow straighteners. A uniform flow is required inside the pulse tube as this will help maintain a stratified profile. However, these flow straighteners will introduce additional pressure drops which should be kept to a minimum as a significant pressure drop will have an adverse effect on the overall SPTC efficiency.

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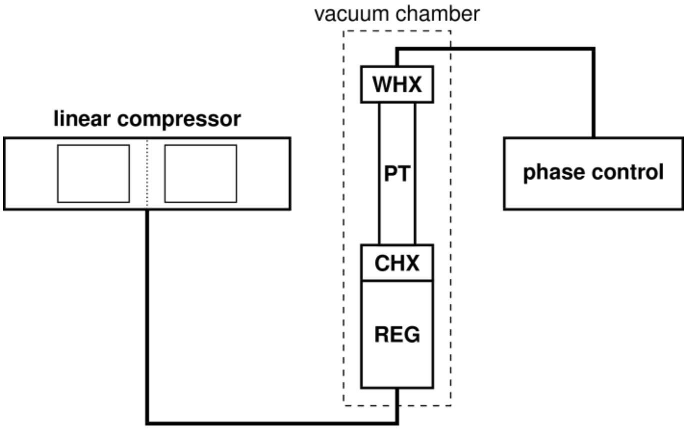

Figure 1. In-line SPTC schematic with abbreviations **REG** for regenerator, **PT** for pulse tube, **CHX** for cold end heat exchanger and **WHX** for warm end heat exchanger.

This study focuses on the design of suitable flow straighteners that minimise flow mixing inside the pulse tube without introducing significant pressure drops. Initially, the flow within an existing in-line SPTC design is numerically examined. Thereafter, alternative flow straightener designs are suggested and assessed numerically. By examining the flow and pressure drops across three designs, a suitable flow straightener design is selected. Experimental validations of the numerical work presented here will be carried out in future studies.

2. Methodology

Fig. 2 shows simplified drawings of three different pulse tube and flow straightener assembly designs. Design A is based on an existing in-line SPTC developed by the Cryogenic Engineering Group at the University of Oxford in collaboration with Honeywell Hymatic. This SPTC has a typical operating frequency of around 60 Hz and a fill pressure of 28 bar. It has a thin walled stainless steel pulse tube with an internal diameter of 7.7 mm and a height of 85 mm. The heat exchangers (i.e. flow straighteners) are filled with copper 50-mesh (wires per inch). During operation, the cold end is maintained at 80 K

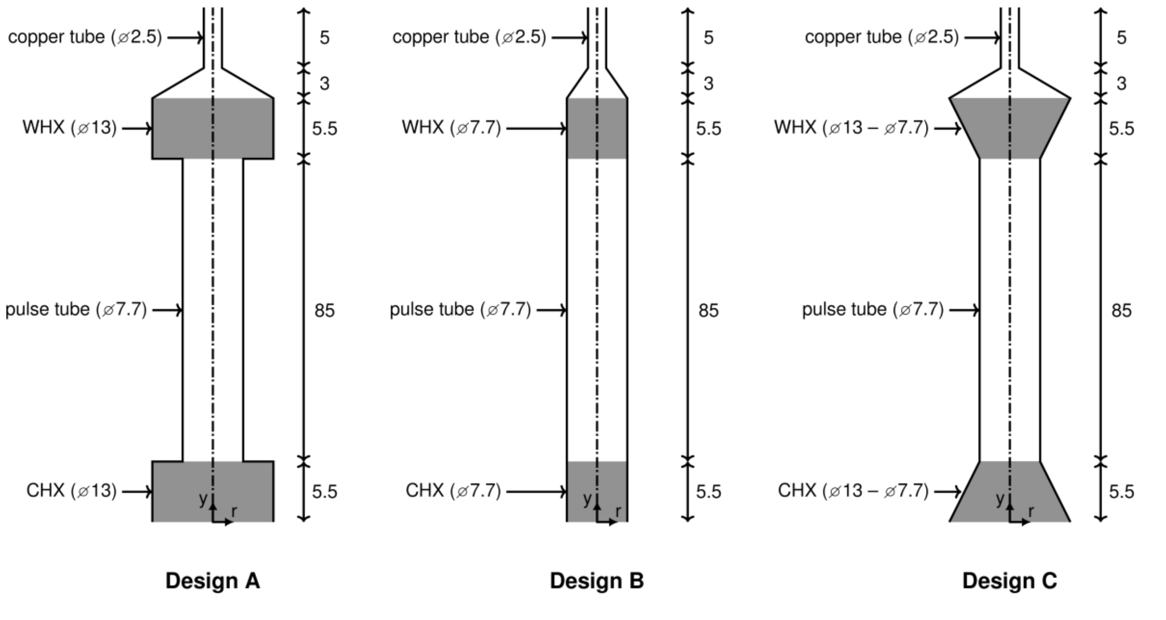

Figure 2. The three different pulse tube and flow straightener assembly designs considered. All dimensions are in millimeters. Not drawn to scale.

and the warm end is kept at ambient temperature. Designs B and C are two alternative designs under consideration where the focus has been on removing the abrupt change in cross section from WHX and CHX to the pulse tube in design A. Design B has a uniform cross section throughout and in design C tapered flow straighteners (i.e. heat exchangers) are used to gradually change the internal diameter.

2.1. Numerical model

The numerical simulations were carried out using CONVERGE CFD [5], a finite volume Navier-Stokes solver. The rotational symmetry of the pulse tube geometry meant that a 2D axisymmetric model could be used for the numerical simulations. A sinusoidal velocity, u_{WHX} m/s, was prescribed at the inlet to the warm end

$$u_{WHX} = u_0 \sin(2\pi f t), \tag{1}$$

with operating frequency f set to 60 Hz and where t is time in seconds and $u_0 = 31.7$ m/s (based on a 1.4 g/s peak-to-peak mass flow variation). Similarly, a sinusoidal pressure, P_{CHX} bar, was applied at the cold end

$$P_{CHX} = 28 + 3\sin(2\pi f t + 4\pi/3), \tag{2}$$

where a peak-to-peak variation of 6 bar and a phase difference of $4\pi/3$ was assumed. The temperatures at the top and bottom boundaries were fixed at 300 K and 80 K, respectively. All other external boundaries were assumed to be adiabatic. Furthermore, the standard k-ε RANS turbulence model was used to account for the effects of turbulence within the pulse tube. In order to establish cyclic steady state, the numerical simulations were run for 20 cycles.

3. Results

The velocity magnitude during maximum flow from the warm end (WHX) to the cold end (CHX) and vice versa are shown in Fig.3 & 4. In order to look at the differences in greater detail, velocity profiles at three different location along the pulse tube are shown in Fig.5 (flow direction WHX to CHX) and Fig.6 (flow direction CHX to WHX). Based on these results, the current inlet design (i.e. design A) leads to the greatest mixing, while design B leads to the most uniform profile. Design C is an improvement on design A, but does not perform as well as design B.

Based on the velocity profiles, design B is the clear favourite. However, the reduced cross-section in design B will lead to a greater pressure drop across the flow straighteners. In order to examine this in detail, the averaged pressures at various locations along the pulse tube for all three designs were calculated and the results are shown in Fig.7. The increase in pressure drop in design B is by no means insignificant, however given that the peak-to-peak variation in pressure within the SPTC is of the order of a few bars, an increase in pressure drop in the order of 10 mbar might not have a significant effect on the overall cryocooler efficiency. In order to be able to quantify this, experiments will be carried out in the future to see if the advantages of the uniform flow in design B outweigh the potential drop in efficiency due to this increase in pressure drop. If the additional pressure drop is not shown to have a big effect on the overall efficiency, then design B will be the design of choice.

4. Conclusions and Future Work

The flow within the pulse tube of an existing SPTC has been numerically analysed. With the aid of the numerical simulations, a new inlet/outlet has been designed that will reduce flow mixing within the pulse tube. The next step will be to experimentally validate the results of the simulations.

Acknowledgements

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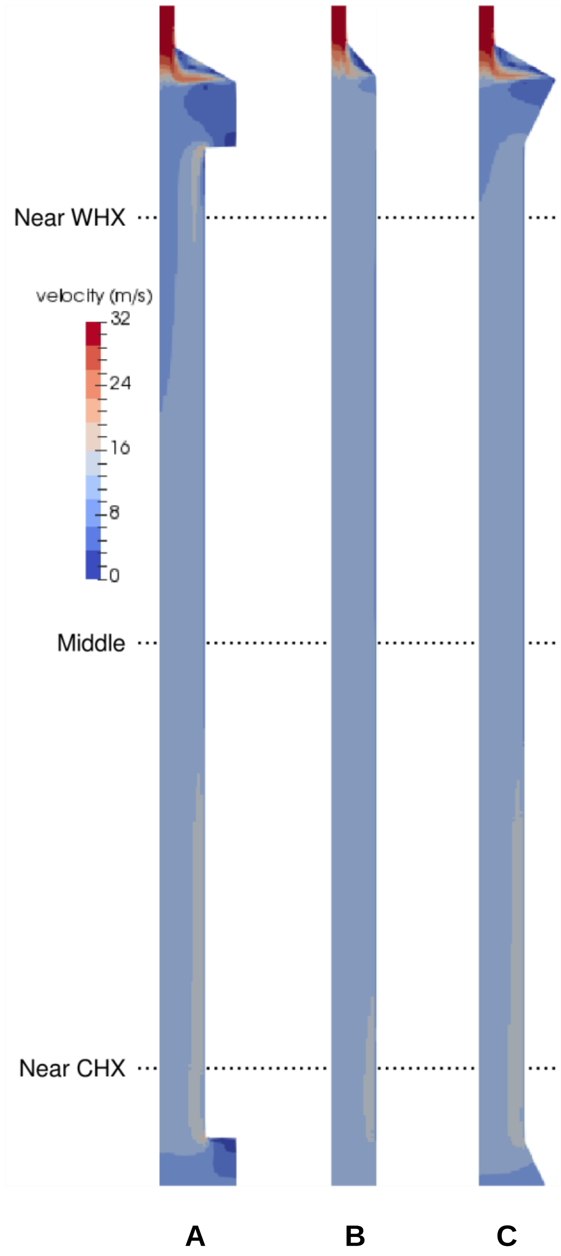

Figure 3. Velocity magnitude during maximum flow from WHX to CHX.

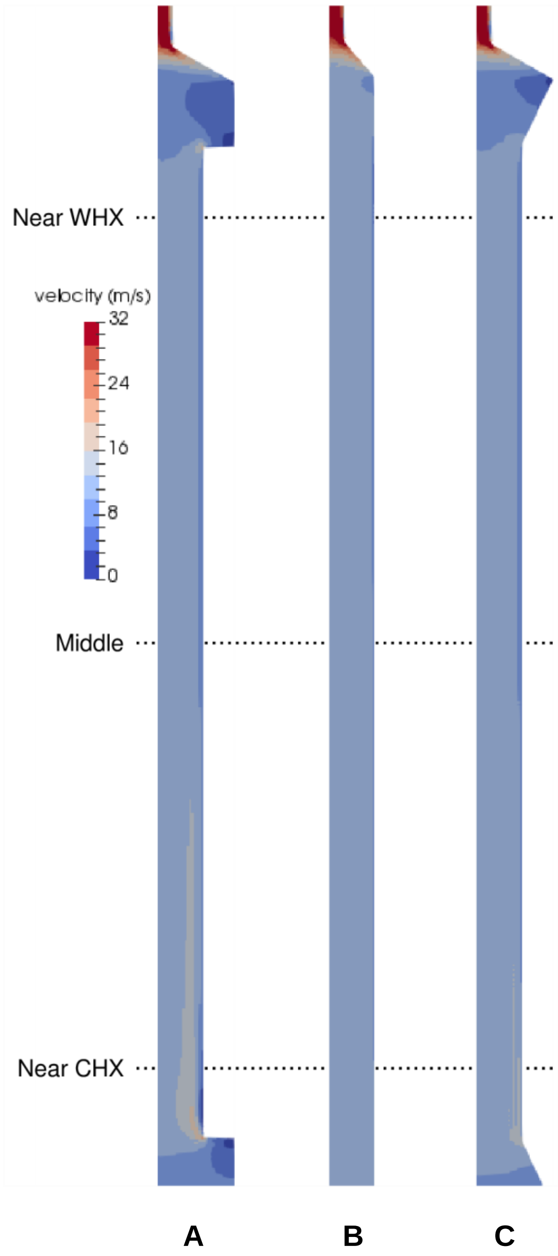

Figure 4. Velocity magnitude during maximum flow from CHX to WHX.

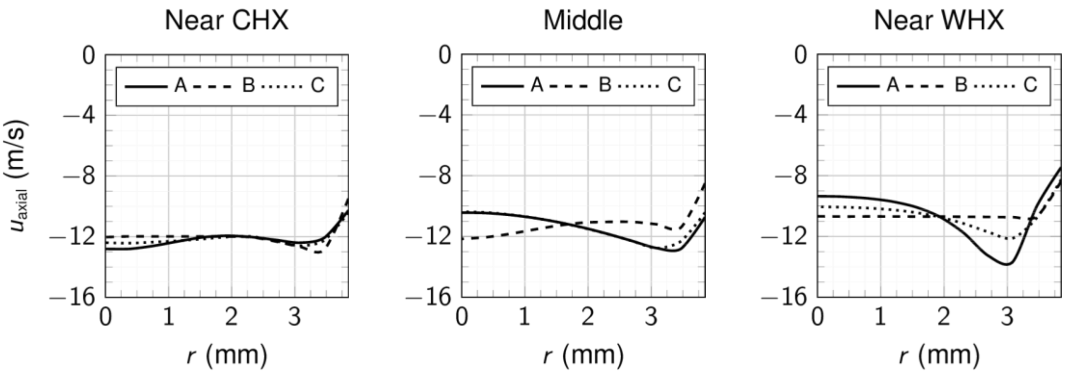

Figure 5. Cross-section velocity profiles at three different locations along the pulse tube during maximum flow (WHX to CHX). The locations are shown in Fig. 3.

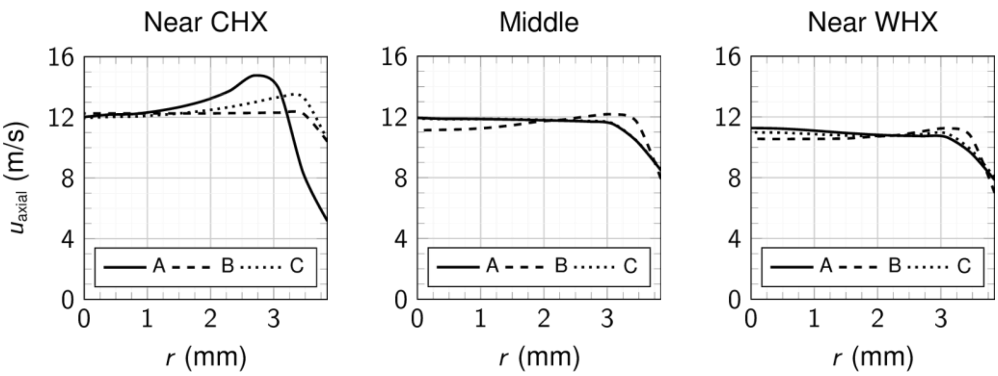

Figure 6. Cross-section velocity profiles at three different locations along the pulse tube during maximum flow (CHX to WHX). The locations are shown in Fig. 4.

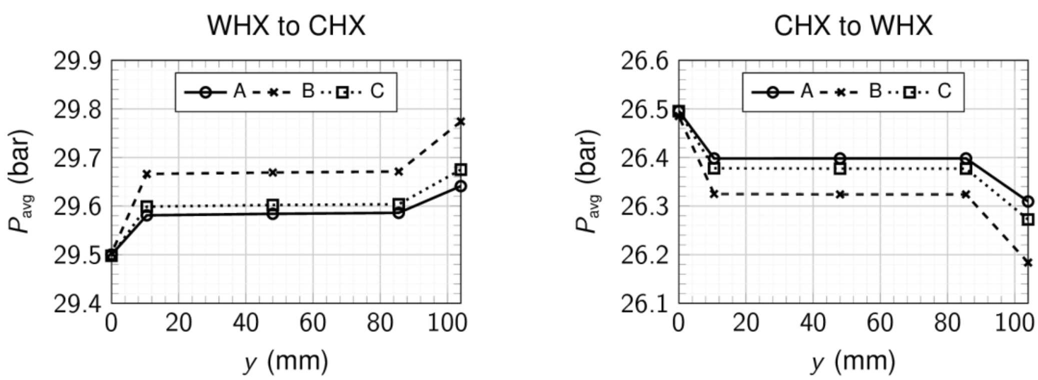

Figure 7. Average pressure along the pulse tube for all three designs during maximum flow. In all three designs, CHX starts at y=0 and ends at y=5.5 mm and WHX starts at y=90.5 mm and ends at y=96 mm.