

CFD Modeling of Reciprocating Flow around a Bend in Pulse Tube Cryocoolers

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ABSTRACT

Coaxial and U-shaped Pulse Tube Cryocoolers involve a change in the direction of the flow as it proceeds from the regenerator through the cold heat exchanger to the (buffer) pulse tube. This rather sharp U-turn in the flow causes non-uniformities over the cross section which may have an adverse effect particularly in the buffer tube, e.g. creating eddies leading to undesirable mixing. Flow straighteners are sometimes introduced to deal with this phenomenon.

This study is concerned with a CFD model of the reciprocating flow around a 180° bend, as applicable to U-shaped Pulse Tube Cryocoolers. Cases with and without flow straighteners (FS) have been considered. Special attention is paid to the role of flow straighteners in terms of convective heat loss and pressure loss. It is shown that the flow characteristic in the cold head of the pulse tube cryocooler is critical, and is of greater significance in miniature cryocoolers. The pressure drop in the flow around a sharp bend vs. the one in a straight tube is studied. Both the pressure drop and the convective heat transfer are affected by the Valensi and Reynolds numbers characterizing the reciprocating flow. The CFD software makes it possible to visualize the flow and map out the temperature and velocity fields as they vary in time, as well as to observe local phenomena such as vortices and flow reversal.

INTRODUCTION

In order to study the parasitic losses in coaxial and U-shaped pulse tubes cold heads, it is essential to analyze the flow and the temperature distributions in oscillating flow around a bend with and without flow straighteners. The first work in this field was conducted by the cryogenic group of Fuji in 2002.¹ They succeeded in improving the cooling power and the efficiency of PT Cryocooler (PTC) just by adding flow straighteners in the cold end of the tube. The Fuji group performed a test with continuous (steady state) air flow around a bend with and without flow straighteners and succeeded in visualizing the stream lines. The test showed much improvement in uniformity of the velocity field due to the addition of flow straighteners to the test rig. The improvement of the velocity field has led to improvement in the cooling power and efficiency of the cryocooler.

Garaway and Grossman² have developed a miniature U-shaped Pulse Tube employing an external pressure oscillator. They discovered a non-uniform temperature at the cold end of the tube.

When operated in the ambient air (outside the Dewar), ice from condensed humidity has formed on the outer side of the tube (away from the center of the U-shaped cold head) while none was formed on the inner side. This was a clear experimental indication that the flow in the cold end of the tube was not uniform.

An analytical solution for oscillating flow in a channel was developed³, assuming incompressible, laminar flow between two plates. The results portrayed the velocity and temperature distributions and the dependence of the pressure drop and the convective heat losses on the Reynolds and Valensi numbers.¹ These are the dimensionless parameters governing the flow in the oscillating flow considered in the present study.

Many studies have been conducted concerning flow conditioners in metering device application. A. Erdal⁴ did work with a CFD tool and has shown that this technique can help to improve flow conditioner and metering technology.

This pioneering work is aimed at analyzing the velocity and temperature field of oscillating flow around an 180° bend. By using CFD, it is possible to calculate characteristics such as pressure drop and heat losses as functions of Reynolds and Valensi numbers. It will be shown that adding flow straighteners leads to a significant improvement in the uniformity of the flow field, resulting in a more uniform temperature field and improved performance.

PROBLEM DESCRIPTION

General

Figures 1a and 1b show two channels separated by a partition of negligible thickness causing the flow to make an 180° turn around a sharp edge. The width of each channel is $2a$ and its length is L . The “Traveling length” of the flow is small compared to the length of the channel. The sharp edge end is maintained at a constant temperature of 77 K and the far ends are maintained at 300 K. All other “walls” are insulated. The flow is periodic and the fluid is assumed incompressible. We use in this problem Helium as an ideal gas at 30 bars.

Software and solution characteristics

The CFD software in use was Fluent 6. The solver was 2-D segregated. Time domain was characterized as unsteady flow. The equations to solve were flow and energy (Navier-Stokes). The most useful method to solve the equations was: (a) Pressure – “Second order”, (b) Pressure and

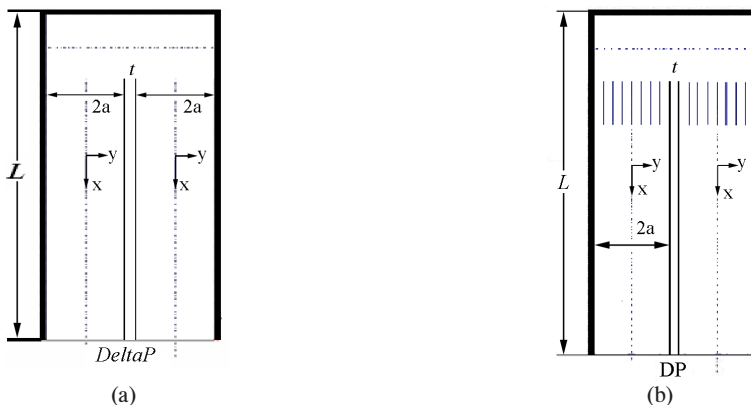


Figure 1. (a) U-turn without flow straighteners, (b) U-turn with flow straighteners.

¹ Reynolds number: $Re = 2a\rho u/m$; Valensi number: $Va = \rho a^2 \omega / \mu$. Here $2a$ [m] is the channel width, u [m/sec] is the mean flow velocity, ω [sec^{-1}] is the oscillation frequency, ρ [kg/m^3] is the density, and μ [Ns/m^2] is the dynamic viscosity.

velocity coupling – “SIMPLEC”, (c) Momentum – “First order upwind” and (d) Energy – “First order upwind”. The convergence criterion was: 1E-6. The convergence also was tested by summing 1000 nodes and comparing this value to the previous cycle at the same time. The difference used to test convergence between two consecutive cycles is found in Figure 2.

Solution method

A series of simulations were first conducted to determine the friction factor under varying conditions. Ten different cases were studied with six different values of the Valensi number and four different values of the Reynolds number. Varied parameters included: mean velocity, frequency, density and the channel length. The results were presented in dimensionless form, as generally done in steady flow problems.

Second, three case studies with three different Reynolds numbers were analyzed. Case A is the reference case with Reynolds value of 900. Case B has twice the Reynolds number (Re=1800). Case C, has half the value of the Reynolds in case A (Re=450). In all the cases the value of the Valensi number was kept constant.

Case B was studied with and without flow straighteners. Case C was studied with flow straighteners.

In all cases, the ratio (β) between the total to the convective heat transfer from the hot end and the cold end has served as the major criterion of the efficiency of cold head design.

RESULTS

Friction Factor

The friction factor (f) was studied in the range of Valensi numbers 50 to 2500 and Reynolds numbers 500 to 3600. The results, when expressed in dimensionless form, show that the product $f \cdot Re$ falls on the same line for various operating conditions (see Figure 3). For example, points 7&10 are with the same Valensi number but with different frequency and density rates.

In the range of Valensi numbers 250 to 2500 the product of the friction factor and Reynolds number in our problem of flow around an 180° bend with a sharp edge coincides with that of the analytical solution for oscillating flow in a straight channel.³ For Valensi numbers smaller than 250 the two cases begin to deviate from each other: the smaller Va , the larger the deviation because of high traveling length compared to the channel width and the influence of the sharp edge on the flow down the channel.

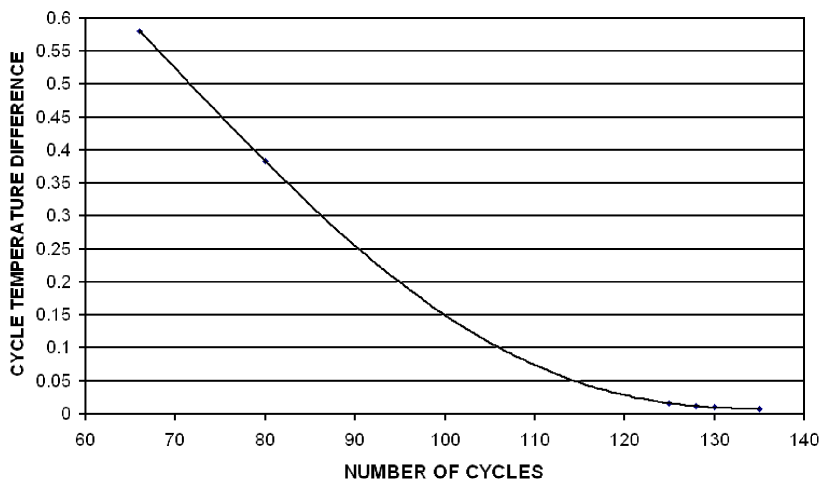


Figure 2. Convergence verification

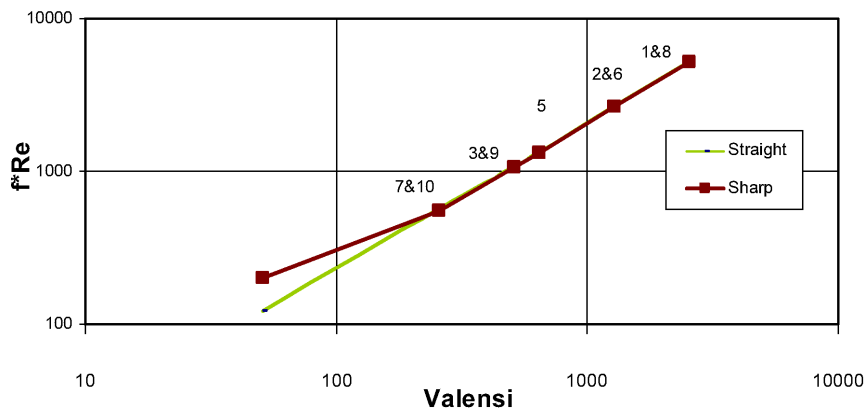


Figure 3. Friction factor in flow around an 180° bend with a sharp edge vs. straight channel.

Case study A

In case study A the frequency was set to 50 Hz, the Reynolds number was 900, the Valensi number was 1233, the channel length (L) was 100 mm and the channel width (2a) was 5 mm. The traveling length is 3.5 mm.

Figure 4a shows the temperature distribution along the tube in the center of the channel at different times in the cycle (expressed in radians – portions of the full 2 cycle). Figure 4b is an exploded view near the cold end. The temperature profile along most of the tube is found to be practically linear at all times; however, in the last 10% of the tube length near the bend the profiles are definitely not linear.

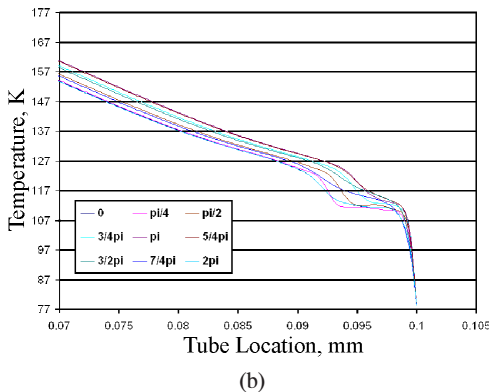
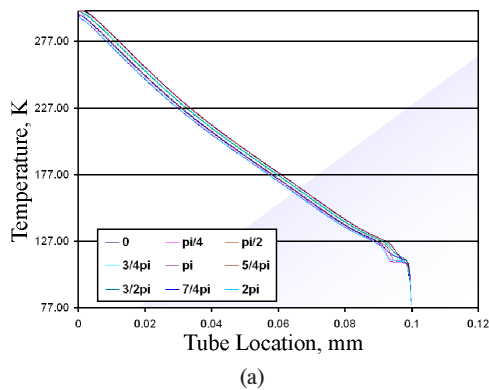


Figure 4. (a) Temperature profile - case A, (b) Expanded view.

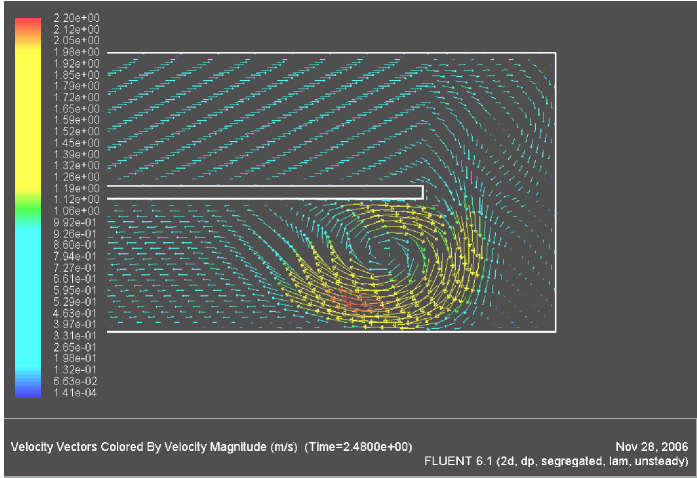


Figure 5. Velocity vector distribution in the channel at the beginning of the cycle – Case A.

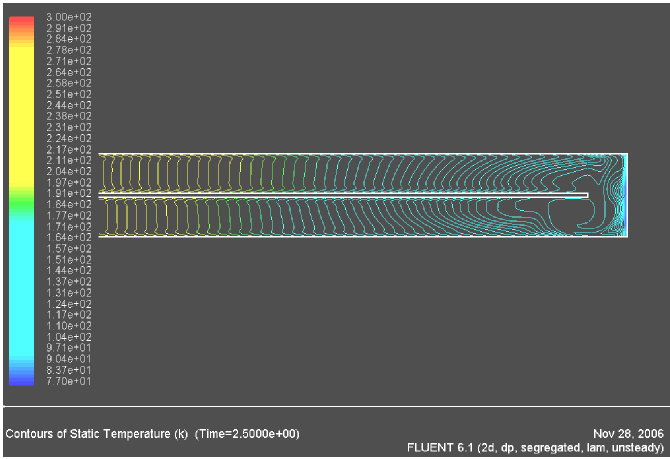


Figure 6. Temperature distribution at the beginning of the cycle – Case A.

Figure 5 describes the velocity vector distribution in the channel, clearly showing a vortex developing near the end of the channel. The size of the vortex varies with time along the cycle; the current plot refers to the beginning of the cycle, at which time the vortex is largest in size. Figure 6 describes the corresponding temperature distribution at the beginning of the cycle. A small gradient is visible at the cold end, spread all around the bend due to the enhanced mixing.

Case study B – without flow straighteners

In case study B the frequency was set to 50 Hz, the Reynolds number was 1800 (twice that of case A), the Valensi number was 1233, the channel length (L) was 100 mm and the channel width (2a) was 5 mm. The traveling length is 7 mm.

Figure 7a shows the temperature distribution near the cold end. The temperature profile along the tube is linear at all times; however, in the last 20% of the tube length near the bend the profiles are definitely not linear. Figure 7b shows the temperature map at the beginning of the cycle. It is clear that near the sharp edge a low temperature gradient zone is created.

Figure 8 describes the velocity vector distribution in the channel, clearly showing a vortex developing near the end of the channel, as in case A. Here again, the size of the vortex varies with

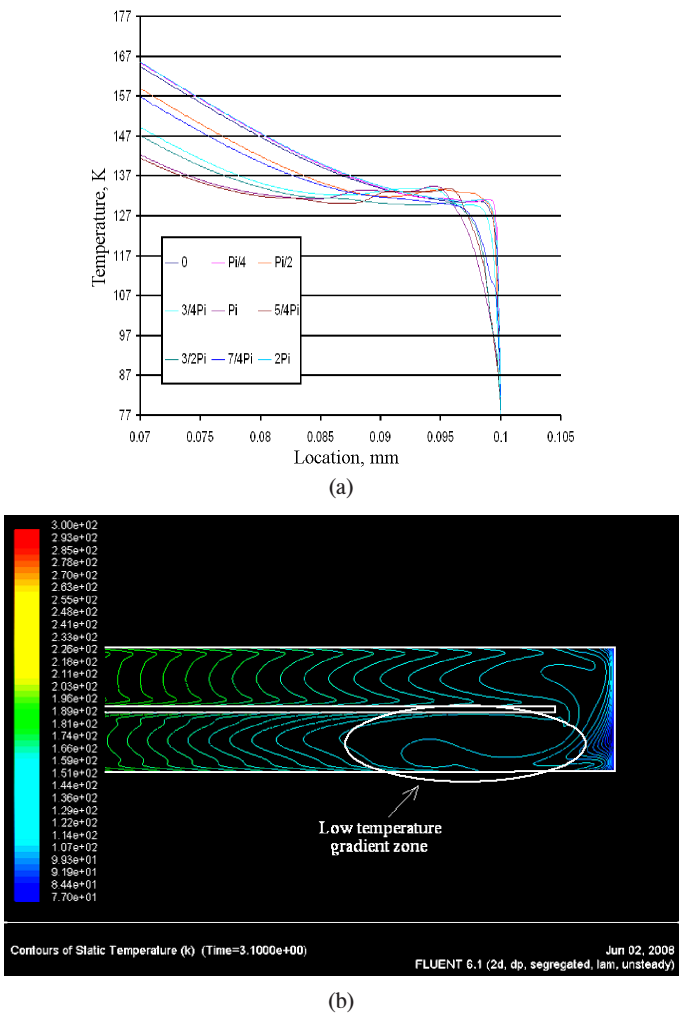


Figure 7. (a) Temperature profile (expanded) - case B without flow straighteners, (b) Temperature map at the beginning of the cycle.

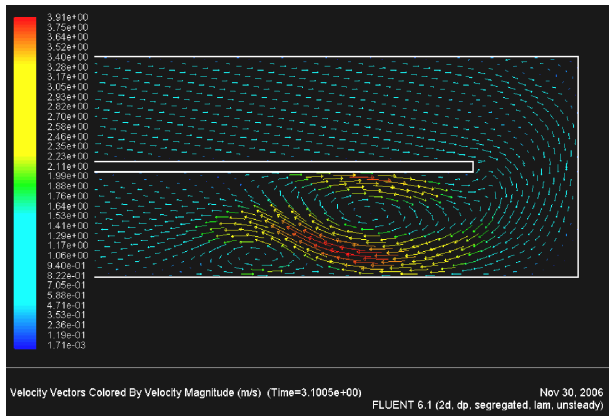
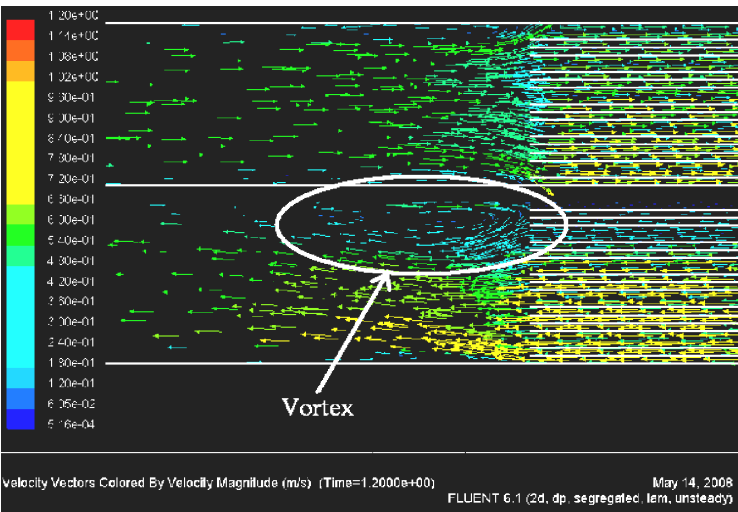


Figure 8. Vortex at the beginning of the cycle – Case B.

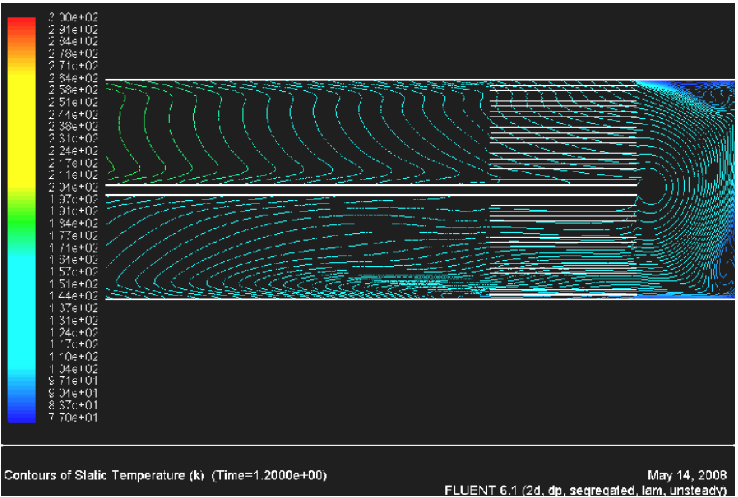
time along the cycle; the current plot refers to the beginning of the cycle, at which time the vortex is largest in size. This vortex is larger than the one in case A. A second small vortex is starting to develop beside the major vortex.

Case study B – with flow straighteners (fs)

This case study is the same as case study B but this time flow straighteners were added. The flow straightener's length is 7 mm (the same length of the traveling length) with an aperture of 0.25 mm. The aspect ratio is 28. In this case much improvement was achieved. Figure 9a demonstrates that a vortex after the flow straightener is observed. The temperature gradient is also different from that in the case without flow straighteners: see Figure 9b compared to Figure 7b. The temperature gradient near the sharp edge in this case is much higher than the temperature gradient without flow straighteners.



(a)



(b)

Figure 9. (a) Vector map at the beginning of the cycle – Case B with fs, (b) Temperature distribution at the beginning of the cycle – Case B with fs.

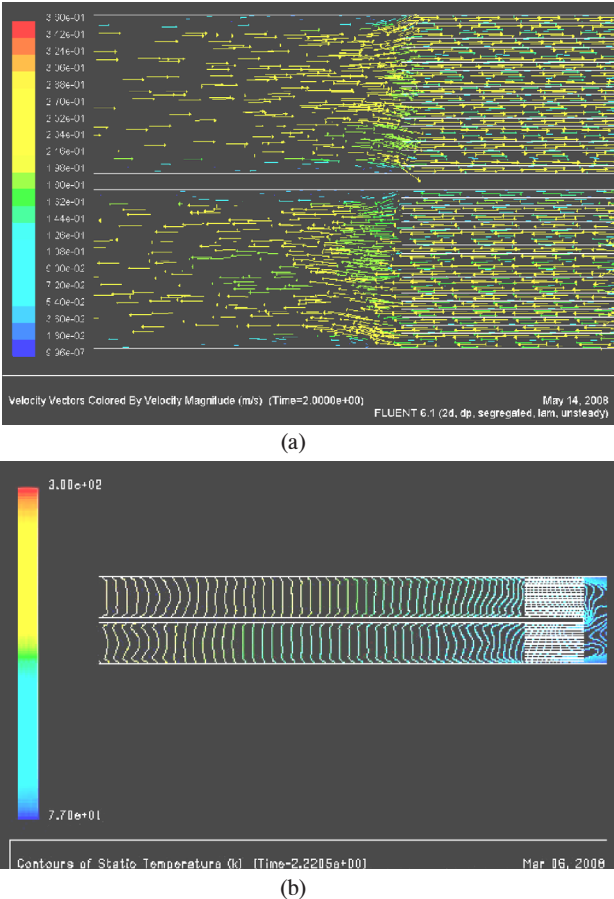


Figure 10. (a) Velocity vector map beginning of the cycle – Case C with fs, (b) Temperature distribution at the beginning of the cycle – Case C with FS.

Case study C – with flow straighteners

In case study C the frequency was set to 50 Hz, the Reynolds number was 450 (half that of case A), the Valensi number was 1233, the channel length (L) was 100 mm and the channel width (2a) was 5 mm. The traveling length is 1.75 mm. No vortices were observed (see Figure 10a). Figure 10b shows the temperature distribution with a well observed gradient compared to case A without flow straighteners (Figure 6).

Ratio between the total and the conductive heat transfer (β).

The most important parameter in this study is the ratio between the total heat transfer and the conductive heat transfer (b). The higher the convective portion, the higher the heat loss from the hot

Table 1: Case Results

Reynolds Number	λ	β_{fs}	β_{nfs}	β_{pp}
1800	1	141	159	134
900	2	32	40	42
450	4	8.4	19.5	20.7

to the cold end and the lower the efficiency of the device, was studied as a function of the Reynolds and Valensi numbers, the traveling length and the geometry of the flow straighteners. A new parameter λ is introduced: this is the ratio between the flow straightener length and the traveling length. Table 1 summarizes the results of all cases studied in this work. It is evident that at low Reynolds number and high λ the efficiency of the flow straightener is better than in the parallel plate solution. At high Reynolds number and low λ this efficiency is low.

CONCLUSION

This is the first known work conducted on oscillating flow around a 180° bend with and without flow straighteners. At Valensi numbers higher than 250 the friction factor in the bend with sharp edge configuration is similar to the parallel plate's configuration. This work shows that flow straighteners can reduce heat losses in pulse tube cold heads. The effectiveness of the flow straightener depends on the ratio between the traveling length and its length. More work can be done in investigating the heat losses at different Valensi numbers and at different aspect ratios.

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