ON THE OSCILLATING FLOW BEHAVIOR OF PULSE TUBE REFRIGERATORS*

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ABSTRACT

The behavior of working gas subjected to an oscillating flow in a confined passageway has been extensively studied and has had particular appeal from thermal engineering community to low temperature refrigerator. One particularly subtle subject of the PTR research comes from the significant quantitative dispersion between experimental data and predictions evaluated by the widely used correlations based on steady-state flows. Recent experiments of cylindrical empty tubes and regenerators under oscillatory flow conditions were intended primarily to provide such information. The present paper will assess the state-of-the-art in this subject, emphasizing the basic concepts and their relation to those of steady-state flows. We briefly summarize our previous works and present some useful results, including experimental measurements on the oscillating flow behaviors of regenerators filled with metallic wire screens and of cylindrical empty tubes, primarily DC-gas flow measurements, and a CFD model for thermal and fluid dynamic analyses of PTRs. In fact the work is in keeping with various studies, which aim to simulate the oscillating flow and heat transfer behaviors more accurately by means of experimental measurements and directing numerical modeling.

INTRODUCTION

Pulse tube refrigerator (PTR), compared to Stirling and GM machines, has considerable advantages due to no mechanical moving parts at low temperatures. This leads to much more reliable operation and longer life times with considerable low vibrations and electromagnetic interference from the cold finger. A PTR basically comprises of a pulse tube, a regenerator, a cold and warm heat exchangers, a set of connecting tubes and several valves (or capillary tube or Inertance tube) for generating proper phase angle between gas flow and pressure oscillations which, from a fundamental point of view, is the inherent dynamic characteristics for such refrigerators: operated under cyclic- or oscillatory flow conditions. In briefly, the above-mentioned components are primarily consisted of a set of cylindrical thin empty tubes with or without filled with porous media of metallic wire screens or powder acted as regenerator matrix. The significant influence of dynamic flow and heat transfer behaviors of the oscillating flow on the performance of PTR has been broadly recognized. Several argumentations are worthy being addressed here:

(1) The right condition for cooling to occur requires the regenerator has not only fine pores and wires to give a large surface area for heat exchange, but little resistance for pressure drop on the gas flow as well (Kays and London, 1964, Walker and Vasishta, 1971).

(2) It has gradually recognized that the friction factors of the oscillating flows are several times higher than that of the steady flows at the same Reynolds numbers based on the cross-sectional mean velocity (Tanaka *et al*., 1990, Yoshida *et al*., 1995, Zhao and Cheng, 1996, Helvensteijn *et al*., 1998, Ju *et al*., 1998a, Yuan *et al*., 2003).

(3) There occurs a phase difference not only between the gas flow and pressure, but also between the gas temperature and the heat flux at high frequency oscillations (Kurzweg, 1985, Gedeon, 1986).

(4) The axial heat transfer of working gas confined in empty tubes (pulse tube and connecting tubes) or in the void volume of regenerator and heat exchangers with thermally insulating walls and temperature gradient along its length can be enhanced by the oscillations at rates orders of magnitude greater than by pure gas conduction without a net transfer of mass (Kurzweg, 1985, Gedeon, 1986, Kuriyama *et al*., 1997, Ju, 2002). The enhanced heat transfer, essentially due to thermal interaction of the oscillating gas flow and its boundary layer (Swift, 1988), is caused by the large time-dependent radial temperature gradient produced by the oscillations, resulting in large quantities of periodic heat absorption-release radially and hence axially.

(5) The heat enhancement by the interaction was found to be a function of oscillation amplitude, Prandtl number, frequency, and tube radius. The oscillatory pressure drop and heat transfer are governed not only by the oscillation frequency, but also by the swept length (amplitude of the fluid displacement) of the oscillatory flow.

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The understanding of above transport mechanism is of crucial importance in the performance predictions, which in turn are fundamental to design, optimization and evaluation of PTRs. For the practicality, one of primary subjects is to predict accurately the pressure drop, phase shift and heat transfer in the PTR. Of particularly exotic question arises from the fact of significant quantitative disagreement between experimental data and theoretical predictions evaluated by the correlations based on steady-state flows, i.e. Kays and London (1964), which have been widely used. As pointed out above, there are important differences between the steady-state and oscillating flow, so that the correlations based on steady-state flows would not be able to predict accurately the gas pressure drop and heat transfer inside the regenerator and pulse tube of PTRs. Recent experiments with cylindrical empty tubes and regenerators subjected to oscillatory flows were intended primarily to provide such information. However, the experimental data available in the literature, mainly depending on the specific operating parameters and conditions under investigation, is rather limited and often quite incomplete therefore not suitable for general use.

Relatively few theoretical studies have been published on this subject due to the complexity of the compressible oscillatory flow related to thermal and energy transportations in PTRs. Accurately understanding of such transport mechanisms requires a full solution of N-S equations. Unfortunately, the complexities of these equations make an analytical solution essentially impossible and numerical simulation is commonly adopted. Owing to the partial differential equations for the 3-D coupled mass, momentum and energy conservation, evaluation of these equations is quite difficult and is numerically still infeasible. 1-D numerical models are therefore widely used (Minas, 1994, Ju *et al*., 1998b, Smith, 2001), to reduce CPU time and avoid difficulties involved with 2- or 3-D models. In a 1-D model, fluid parameters are averaged in the cross section normal to the principle flow direction and the governing equations are expressed in terms of these mean parameters. The heat transfer and pressure drop correlations in terms of cross-section averaged parameters need to be inserted into such a model. As a result, the earlier studies on the PTRs are of limited applicability in the absence of reliable experimental data for the oscillating flow resistance and heat transfer. In order to avoid complicated numerical formulation and iteration, many other methodologies, i.e. thermoacoustic theory (Swift, 1988, Xiao, 1992), linear flow network analysis (Huang *et al*., 1996, Ju *et al*., 2000), harmonic analysis (Swift and Ward, 1996, de Waele *et al*., 1998) and the method of characteristics (Xu *et al*., 1999) have been developed to study the oscillating flow behavior with time-averaged thermal transportation. The above-mentioned studies however considered the cases of nearly isothermal or adiabatic surface wall, and limited in conjunction with linearization approaches. Practically, the walls of the pulse tube and regenerator in PTRs are not adiabatic or isothermal but exposed to a nonlinear temperature distribution. Furthermore, most of above studies was limited to consider independently the oscillating flow behavior and the thermal transportation.

In this paper we briefly summarize our previous works and present some useful results. We describe in section 1 on the experimental measurements of the oscillating flow behavior of regenerators filled with metallic wire screens, of cylindrical empty tubes, and of comparison with those of steady-state flows. Primarily DC-gas flow measurements will be also outlined. An improved CFD model for thermal and fluid dynamic analyses of oscillating flow behaviors of PTRs will be introduced in Section 2, followed by a few concluding remarks.

1 EXPERIMENTAL MEASUREMENTS

1.1 Experimental System

A dynamic experimental apparatus, specifically designed for simultaneously measurements of transient pressure, mass flow rate and temperature of the oscillating gas flow at the two ends of a test section confined in a closed-loop circular system, was built and constructed. Details of experimental arrangement and measurement approaches have been discussed by Ju *et al*. (1997, 1998a), therefore, only a streamlined summary of key techniques is given here. The test section consisted of a cylindrical empty stainless steel tube (as pulse tube, connecting tube or inertance tube) or packed with metallic wire screens (regenerator and heat exchangers). The test section has two demountable flanges at its two ends directly connected to velocity straighteners, which in turn can be connected to a reservoir with an adjustable needle valve. The oscillatory flow was generated by means of two self-made linear compressors, of one having a swept volume of 10 cc with a fixed operating frequency of 50Hz, another with 2cc swept volume and its operating frequency varied from 20 to 80 Hz. The instantaneous velocity (cross-sectional mean velocity) of the oscillatory gas flow was obtained by two approaches: (a) measured directly by using a hot wire anemometer (DANTEC, 90N10), and (b) determined by measuring the transient pressure oscillations inside the reservoir. The average relative derivation between the two was about 3.35% for 95% experimental data. The wire anemometer is a commercial and generally instrument for the velocity measurements. In our experiments, two small identical hot probes (DANTEC, 55P11) were placed at the centerline of the tube between the two velocity straighterers of the test section. The hot wire probes having a response time of 20μm were made of tungsten of 5μm in diameters and must be firstly calibrated before operations at different operating temperatures. A calibration system was then set up for the calibration of the hot wire anemometer at three different temperature ranges of ambient (293-298K), mixed ice-salt (258-263 K) and liquid nitrogen (77-79K). Three small differential pressure transducers, connected to charge amplifiers, were used for the pressure measurements at the inlet and outlet of test section, and inside the reservoir. The transducer has a high natural frequency of 150 kHz. As the typical frequencies of a PTR are much smaller than the natural frequency, we can safely ignore the error in measuring the amplitude and phase delay of the pressure oscillations. Analog-to-digital conversions were carried out by an A/D conversion board, which was plugged into a personal computer. A 4-channel simultaneous sample and hold front ends were employed so that both the transient pressure and velocity voltage signals were sampled simultaneously.

The test apparatus has been used for the measurements of the pressure drops and phase shifts of the gas flow through the test section of regenerators packed with SS stacked screens (Ju *et al*., 1998a, Yuan *et al*., 2003), and of a set of empty tubes (Ju *et al*., 2003), subjected to an oscillating flow. In these tests, we studied experimentally the effects of different structure parameters (diameter and length of tubes, mesh size) and operating parameters (end setting conditions, average pressure and frequency), on the flow resistances (pressure drops) and phase shifts.

1.2 Pressure Drop and Phase Shift of Regenerators

In this case, the test section was packed of stacks of stainless-steel wire screens with five different mesh sizes of 80, 150, 250, 300 and 400. The properties for these wire screens can be found elsewhere (Ju *et al*., 1998a, Yuan *et al*., 2003). It was worthy noting that the raw data measured from the hot wire anemometer and pressure transducers required data processing including velocity transformation and correction by calibration curves, high frequency harmonic filtration, Fourier analysis, etc. Another key technique was of how to deal with the experimental data and to obtain the correlation equations in terms of appropriate characteristic parameters for oscillating flows, and compare to those of steady-state flows. In view of all of the correlations of pressure drops through packed screens were evaluated in terms of Reynolds number, we adopted the approach suggested by Zhao and Cheng (1996) in terms of two similarity dimensionless parameters for oscillatory flow, the kinetic Reynolds number, $\mathrm{Re}_h = \rho U_{\max} D_h / \mu$, and the dimensionless fluid displacement $X = X_0 / D_h = 2U_{\max} / D_h \omega$. The cycle-averaged friction factor is defined as

$$f_{mean} = \frac{\Delta \overline{P}_{mean} D_h}{\frac{1}{2}\rho (U_{\max})_p^2 L} \tag{1}$$

where $\Delta\overline{P}_{mean}$ is the cycle-averaged pressured drop in one cycle, $U_{\max}$ the amplitude (maximum cross-sectional mean velocity) of the gas velocity related to the amplitude of the fluid displacement X_0, D_h the hydraulic diameter of wire screens, L the length of the test section, and ω is the angular frequency of the oscillatory flow. The cycle-averaged friction factors in terms of Re_h and X were computed and correlated according to Eq. (1) based on experimental data. We compared $\Delta\overline{P}_{mean}$ measured from our experiments for the oscillating flow to the pressure drop ΔP_{st} of steady flow based on the data given by Kays and London (1964) at the same Reynolds numbers, as shown in Table 1. It was found that the ratios were in the range of 2~3 at ambient temperatures, of 2.5~3 at mixed ice-salt temperatures and of 5~6 at liquid nitrogen temperatures. We conclude that the lower the temperature, the larger the ratio.

Table 1: Pressure drops of oscillating flow compared with those of steady flow at three different temperatures

Temperature range (K)	Mesh size	Re_h	$\Delta\overline{P}_{mean}$ (kPa)	ΔP_{st} (kPa)	$\Delta\overline{P}_{mean} / \Delta P_{st}$
Ambient (293-298)	150	5.57	12.67	5.98	2.28
	250	5.17	17.21	7.42	2.32
	300	5.09	20.44	8.69	2.35
Mixed ice-salt (258-263)	150	63.00	32.66	10.55	3.09
	250	55.80	37.59	14.36	2.61
	300	46.20	44.00	18.20	2.42
Liquid nitrogen (77-79)	150	302.6	238.8	38.95	6.13
	250	262.5	289.7	52.59	5.51
	300	220.9	369.3	59.52	6.21

The oscillating flow behavior in a PTR demonstrated not only pressure drops but also phase shifts. There are several phase shifts, of between the input and output pressures $\Delta\theta_{P1P2}$, the input and output velocities $\Delta\theta_{U1U2}$, the inlet pressure and velocity $\Delta\theta_{P1U1}$, and the output pressure and velocity $\Delta\theta_{P2U2}$. Of which $\Delta\theta_{P2U2}$ is particularly important on the cooling performance of a PTR. Measurements showed that there exists a minimum value of $\Delta\theta_{P2U2}$, which supplies a special kinetic Reynolds number for different mesh sizes of wire screens and decreases with increase in mesh size and Re_h, but is relatively independent of *X*.

1.3 Flow Resistance and Conductance of Inertance Tube

In most cases the regenerator packed with wire screens can be equivalent to a bundle of parallel cylindrical tubes if neglecting geometric complexity in it. In view of the similarity of the friction factors for steady-state flow in a single or a bundle of parallel tubes and in a packed screen column, it is postulated that the similar characteristic parameters for an oscillatory flow in the similarity structure are also the same except of different characteristic diameters. As a result, the above correlation equations for the oscillatory flow in the regenerator can be used safely for cylindrical empty tubes (i.e. pulse tube and connecting tubes). It would be not suitable however when the tube inner diameter is much smaller than the tube length, like inertance tube, since in this case a complex impedance rather than a simple resistive impedance generated by the oscillations. Much attention has recently been given to the inertance tube used in PTRs (Gardner and Swift, 1997). The inertance tube is a long, very small internal diameter (the ratio of diameter to length is usually in the range of 10^{-3}) and adds a reactive impedance, analogous to inductance in a simple AC electrical circuit, that allows the phase shift between the pressure and mass flow in the pulse tube to be adjusted to an extent as efficiently as Stirling coolers. In brief, the inertance tube is an acoustic term connoting both flow resistance and inductance of moving gas. Studies show that use of the inertance tube is significantly beneficial for large-scale pulse tubes or at higher frequencies. Relatively few studies are available on experimental measurements of the flow resistance and flow conductance of inertance tube, particularly at high-pressure amplitudes. We briefly discuss here this situation and present experimental measurements. Considering an incompressible gas oscillating sinusoidally in a tube of length L and inner radius a ($a << L$) at constant room temperature. The tube is filled with pressurized gas (ideal gases), which moves back and forth with oscillatory pressure p at frequency ω. The flow impendence of inertance tube is complex in general and is defined as (Ju *et al*., 2003)

$$Z_A = (p_{in} - p_{out}) / A_t U_{\max} = R_A' + jX_A' \tag{2}$$

Here p_{in} and p_{out} are the transient pressures at the inlet and outlet, respectively. Under two assumptions: (1) the pressure amplitude is small, results neglecting the nonlinear inertia term; (2) the length of the tube is much smaller than the local sound wavelength of gas, the flow impendence can be simply approximated as (Ju *et al*., 2003)

$$Z_A = \frac{8\mu L}{\pi a^4}\sqrt{1+|Ka|^2/32} + j\frac{\rho\omega L}{\pi a^2}\left(1+\frac{1}{\sqrt{3^2+|Ka|^2/2}}\right) \tag{3}$$

Here $K=\sqrt{2}/\delta_V$ with $\delta_v=\sqrt{2\mu/\rho_0\omega}$ is the viscous penetration depth denoting the relevant boundary-layer thickness (Swift, 1988). The real part and imaginary part indicate the flow resistance and flow inductance, respectively,

$$R_A = \frac{8\mu L}{\pi a^4}\sqrt{1+|Ka|^2/32} \tag{4}$$

$$X_A = \frac{\rho_0\omega L}{\pi a^2}\left(1+\frac{1}{\sqrt{3^2+|Ka|^2/2}}\right) \tag{5}$$

We found that the flow resistance coefficient (friction factor) given by Eq. (1) for the inertance tube subjected to an oscillatory flow can be expressed as

$$f_{mean} = \frac{64}{\mathrm{Re}_h}\sqrt{1+\frac{|Ka|^2}{32}} \tag{6}$$

Comparing Eq. (6) to the well-known correlation $f_{st} = 64/\mathrm{Re}_h$ of the steady-state flow in a single empty tube yields

$$\beta = \frac{f}{f_{st}} = \sqrt{1+\frac{|Ka|^2}{32}} > 1 \tag{7}$$

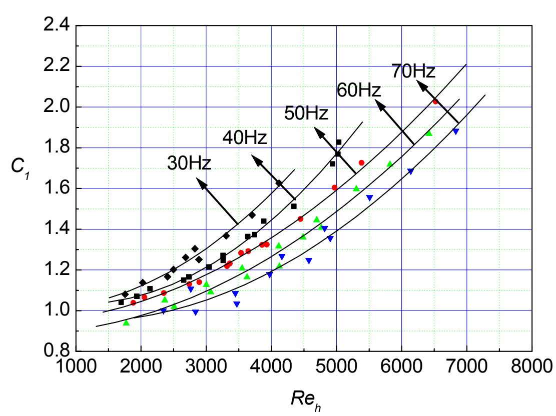
It indicates that at low-pressure amplitude the flow resistance coefficient of the inertance tube in the oscillatory flow is larger by a factor of β than that of a steady-state flow at the same Reynolds numbers. The factor β is a function of inertance tube diameters and frequency and is in the range of 1.04~1.1 for 0.6mm, of 1.2~1.35 for 1.0 mm and of 1.4~1.8 for 2mm tubes, respectively. However, the correlation given above based on the assumption of low-pressure amplitudes is usually not applicable for a practical PTR operating at high-pressure amplitudes, where the nonlinear inertia term cannot be neglected. We introduce therefore here two modification coefficients of C_1 and C_2 for considering the nonlinear effects at high-pressure amplitudes

$$R_A' = C_1 R_A;\ X_A' = C_2 X_A \tag{8}$$

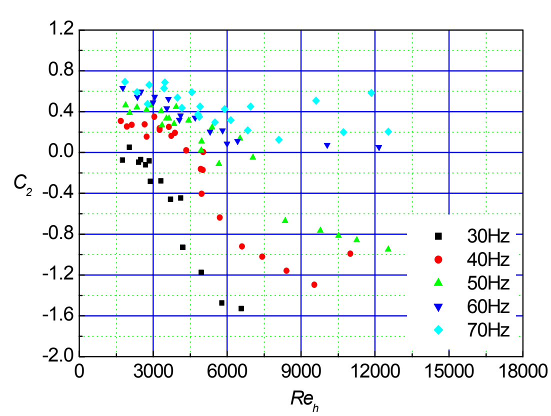
By measuring the transient pressures at the inlet and outlet and the amplitude of gas velocity through the inertance tube, together with Eq. (2) and (8) we can determine C_1 and C_2. A series of experiments have been carried out to determine the modification coefficients C_1 and C_2 for four different inner diameters of 0.6, 1.0, 1.5 and 2.0mm at various tube length ranging from 100 to 1500mm, at frequencies of 30, 40, 50, 60 and 70 Hz. The working gas was helium and the system mean pressure was 2.0MPa at room temperature. Experimental data showed that the flow resistance and inductance of inertance tubes at high-pressure amplitudes strongly depend on the operating frequency and Reynolds numbers. Both C_1 and C_2 vary with different tube diameters, but are nearly independent of the tube length. Typical data of C_1 and C_2 of the flow resistance and flow inductance of inertance tubes at tube diameter of 0.6mm, in terms of ω and Re_h, was compiled in Figure 1 (a) and (b). It is clearly that C_1 increases monotonously in a super-linear dependence with increasing Re_h and with decreasing ω, which indicates that the nonlinear effect becomes larger with the increasing of gas velocities. C_1 gradually tends to 1 when Re_h reach to zero. Therefore, the explicit solution of Eq. (4) is only applicable to small Re_h. A comparison of C_1 for different tube diameters at the same Re_h showed that the larger the tube diameter, the smaller the C_1. The reason is obvious: the velocity is small for large tube diameter at the same Re_h thereby the nonlinear effects become relatively weak. Figure 1 (b) showed that C_2 is always less than 1, which means that the flow inductance of the inertance tube at high-pressure amplitudes is smaller than that at low amplitudes. C_2 decreases with increasing Re_h, and increases with increasing ω, contrary to the tendency of C_1. However, the influence of ω on C_2 gradually becomes smaller with the increase in tube diameters. From these experiments we can see evidently the difference of the flow resistance and flow inductance of inertance tubes at high and low pressure amplitudes. It is helpful to understand the physical mechanism of inertance tubes subjected to the oscillating flow in high frequency PTR operations and to further conduct theoretical analysis.

1.4 DC Gas Flow in PTR

The double-inlet in PTRs (we named DPTR) is a passive phase shifter and gas mass flow through it depends on the pressure drops across the regenerator. So, the position and negative mass flow over each half cycle are difficult to balance in one cycle if there is a closed loop formed by the double-inlet. The net mass flow rate that flows in one direction is defined as DC-flow. Gedeon (1997) pointed out that the gas velocity and density fluctuations in phase and amplitudes due to the pressure fluctuation are the reason for the DC-flow. The normalized pressure drop will prevent the DC-flow in the orifice PTR since there is no return path. The closed loop flow comprised by the double-inlet between the hot ends of the regenerator and pulse tube in the DPTR provides the condition for the DC-flow.

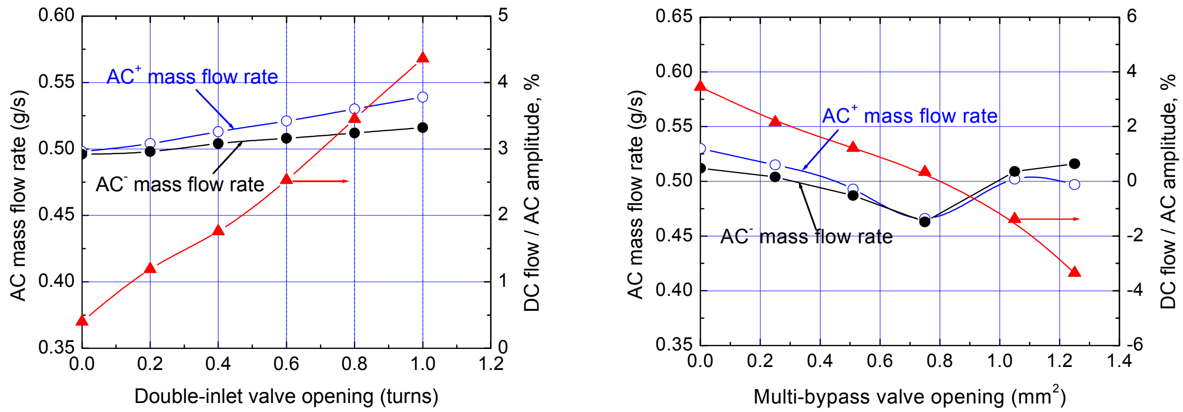


(a)



(b)

Figure 1: The flow resistance and flow inductance in terms of ω and Re_h, at tube diameter of 0.6mm



(a) (b)

Figure 2: Time-averaged mass flow rate and DC-flow rate at cold end (Pa = 0.9MPa), (a) DPTR and (b) MPTR

We measured the amplitudes and phase shifts of mass flow rate and pressure at cold end of a PTR during actual operation for different double-inlet and multi-bypass valves (Ju *et al.,* 1997). The two half-cycle time-averaged mass flow rate was evaluated from experimental data. The two half-cycle time-averaged AC mass flow rates evaluated by integrating the instantaneous velocity data for various openings of double-inlet valves were plotted in Figure 1 (a). One found that the averaged AC mass flow in the positive direction (away from the compressor) was not equal to that in the negative direction (towards the compressor). There was a net mass flow rate that flow in one direction for fixed opening of double-inlet valve. This was a kind of DC-flow phenomena that was not analogous to AC electrical system. It was natural to guess that the DC-flow was a kind of circulating flow (the ratio of DC-flow to AC-flow at was about 3.5% corresponding to the lowest temperature of PTR). The path of the circulating flow was between the cold end and the double-inlet valve. The larger the openings of the double-inlet valve, the larger the DC-flow rate. We concluded that the DC-flow introduced a loss to the DPTR by generating a direct convection gas flow and enthalpy flow between the cold and hot ends. Figure 2 (b) showed the two half-cycle time-averaged AC mass flows for different flow area of multi-bypass tubes. The DC-flow rate decreased with increasing flow area of multi-bypass. When the multi-bypass was adjusted to an optimum value (around 0.75mm^2) with respect to the lowest temperature, the DC-flow rate trended to zero. By further increasing the flow area of multi-bypass, a negative DC-flow was observed and the cold end temperature was increased, which could be higher than that at the double-inlet version. Because of the multi-bypass in the PTR, we believed there was a path of circulating flow between the multi-bypass tube and hot end, to damp DC-flow between the cold and hot ends. From our experiments, we found that the DC-flow may occur in the DPTR and MPTR if the flow area of the double-inlet and multi-bypass are not set at optimum values. The ratio of the DC-flow to the averaged AC flow is an important parameter for PTR design. We found that the multi-bypass version can reduce the DC-flow in the DPTR and improve the performance of PTRs.

2 COMPUTATIONAL SIMULATIONS

A significant progress in our understanding of oscillating flow behaviors of PTRs comes from direct numerical modeling although quantitative disagreement between predications and experimental data remained. A large number of numerical simulations have been made over a wide range of conditions for PTRs. We recently developed a mixed Eulerian-Lagrangian model (Ju, 2001), not only for simulating the time-dependence of dynamic parameters and cooling performance, but also for visualizing the entire cooling-down process of PTRs. It was a 1-D model based on the conservation of mass, energy and momentum for the oscillatory gas flow in PTRs. We used the Eulerian method, a fixed computational grid, to simulate the transient dynamic parameters in the regenerator and heat exchangers. The Lagrangian approach, a moving grid, was used to follow the exact tracks of gas particles as they move with pressure oscillation inside the pulse tube to avoid any numerical false diffusion. A variety of physical factors, including real thermal properties of helium gas, pressure drop and heat transfer in the regenerator, are taken into account. The simulation allowed a clear visualization of the transient parameter variations and of the cooling-down processes in PTRs, and gave good quantitative information on the dominant influence in the cooling performance. The numerical results were in reasonable agreement with the measured cooling capacity dependence of the temperatures.

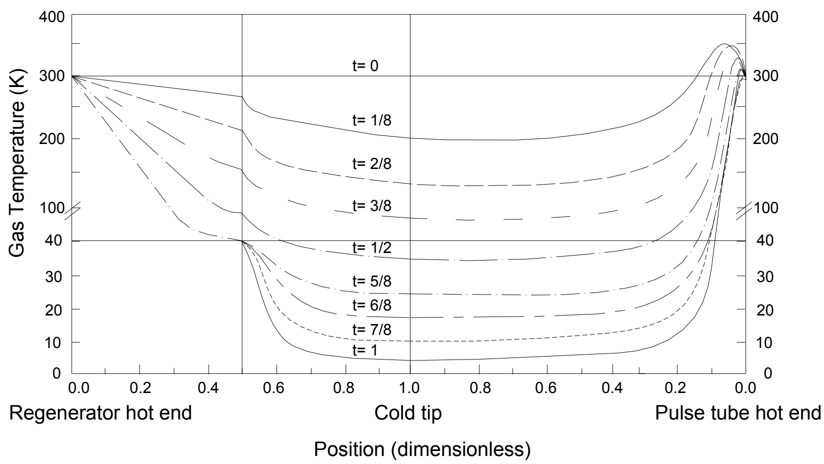


Figure 3: Time dependence of the cooling-down processes of gas temperature profiles in regenerator and pulse tube

Calculation was made for a 4K-PTR (Ju, 2001) and a hybrid GM/PTR (Ju and Wang, 2002). A typical cooling-down process of two-stage 4K-PTR from room temperature *T=300K* at initial dimensionless time *t=0* down to the lowest temperature *T=4K* at *t=1* along the length of 1st and 2nd-stage regenerators and 2nd-stage pulse tube was plotted in Figure 3. At *t=0*, all gas elements in the PTR were set at $T = 300K$, and then the temperatures drop with time. After many cycles, the PTR reached its low temperature of *T=4K* at time *t=1*. It showed nearly linear temperature profiles in the 1st-stage regenerator at time intervals of *t=1/8, t=1/4* and *t=3/8*. Non-linear temperature profiles at other time intervals and in the 2nd-stage regenerator were indicated. It also showed that about 80~90% of temperature drop was concentrated in the 40% of the pulse tube length near the hot end. Results showed that the cooling-down process and temperature profiles in the low temperature regenerator and pulse tube are significant difference from that of ideal gas. The time-dependent temperature profiles are strongly affected by the oscillating flow and heat transfer behaviors and by the real helium thermal properties.

CONCLUSIONS

Oscillating flow behavior of working gas in PTRs has potentially valuable interesting of both theoretical formula and practical design for cryogenic refrigerators. A briefly summary of our previous works and results are reported. The research is keeping with various studies and has been considerably extended by means of direct numerical simulation and experimental measurements, although our understanding of several aspects remains incomplete, quite a bit work still has to be done, and further development of theory is needed. Oscillatory flow characteristics are also of special interest to a wider thermal engineering community since their employment in advanced heat exchangers as working media for heat transfer enhancement by oscillations.

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LE COMPORTEMENT OSCILLATOIRE DES CYROGÉNÉRATEURS À TUBE PULSÉ

RESUME : Le comportement d'un gaz soumis à un écoulement pulsé dans une voie confinée a fait l'objet d'études détaillées, et présente un intérêt particulier pour les ingénieurs thermiques, pour le refroidissement à basse température. L'un des aspects subtils de la recherche en ce domaine, est le désaccord entre les données expérimentales et les prédictions basées sur les corrélations en écoulement stationnaire. Des nouvelles expériences avec des tubes cylindriques simples et avec régénérateurs ont été menées pour fournir de telles informations. Nous rappelons brièvement nos études précédentes et présentons quelques résultats utiles, dont certaines mesures du comportement oscillatoire des régénérateurs à base de grilles métalliques et des tubes cylindriques vides, principalement des mesures d'écoulement stationnaire. Nous présentons un modèle numérique pour l'analyse thermique et dynamique des cryogénérateurs à tube pulsé. Ce travail s'inscrit dans un programme qui vise à améliorer la simulation de l'écoulement oscillatoire et du transfert thermique, à l'aide de mesures expérimentales en vue d'affiner les modèles numériques.