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Citation	Cryogenics , 52 (7-9) : 410 - 415
Issue Date	2012
DOI	10.1016/j.cryogenics.2012.04.001
Self DOI	
URL	http://ir.lib.hiroshima-u.ac.jp/00046726
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Relation	

Visualization of oscillating flow in a double-inlet pulse tube refrigerator with a diaphragm inserted in a bypass-tube

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Abstract

The double-inlet pulse tube refrigerator that has a diaphragm inserted in a bypass-tube, which enabled it to transmit a pressure oscillation whereas to obstruct a DC gas flow, was manufactured and tested. The oscillating flow behavior inside of the refrigerator was studied by using a smoke-wire flow visualization technique. It was found that if the diaphragm was optimized, the performance would be improved more than that of the refrigerator with a bypass valve due to the increase in the P-V work of the gas and the decrease in the convective heat loss caused by a secondary flow.

Keywords: Pulse tube; Flow visualization; Thermoacoustics; Fluid dynamics

1. Introduction

A double-inlet pulse tube refrigerator has a bypass tube with a bypass valve to adjust flow impedance [1]. The bypass tube produces a closed loop fluid path (shortly, loop path) through the regenerator and pulse tube as seen in Fig. 1. In general, if a closed loop path exists in such oscillating thermoacoustic systems as double-inlet pulse tube refrigerators and thermoacoustic engines, a DC gas flow is inherently induced in the loop path as a secondary flow of a main oscillating flow [2]. It is a subsidiary caused by an asymmetrical mass flow in the loop path during one cycle of the main oscillating flow. Because the DC gas flow leads to a convective heat loss, the performance of the systems is seriously degraded [3,4]. The convective heat loss should be reduced by decreasing the DC gas flow in order to improve the performance [5,6,7].

Swift GW et al. introduced a barrier method of a limp rubber balloon to suppress DC flow in the loop path in a thermoacoustic Stirling refrigerator [8]. The balloon blocked streaming flow through the loop path whereas it was acoustically transparent. They demonstrated that the method was effective in suppressing of DC flow. Hu et al. developed the barrier method from the rubber balloon into a membrane suppresser for DC flow in double-inlet pulse tube coolers where an elastic membrane was installed on a way of a bypass tube together with a bypass valve [9]. They reported that they succeeded in further temperature reduction by using the diaphragm configuration. Although a diaphragm configuration will suppress a DC gas flow, it might change the fluid dynamic behavior of the oscillating flow in a refrigerator because gas doesn't flow through a bypass tube so that it is hard to readily conclude that the cooling performance of the refrigerator is improved. There are few studies on the double-inlet pulse tube refrigerators with a diaphragm configuration, and thus very little is known about the fluid dynamic behavior of the oscillating flow in the refrigerators.

The objective of this study is, therefore, to clarify the influence of the diaphragm configuration on the flow behavior in the double-inlet pulse tube refrigerator by visually observing the flow behavior and to discuss qualitatively the relation of the flow behavior to the performance of refrigerator for an evaluation of the effectiveness of the diaphragm configuration.

2. Experiment

Fig. 1 shows a schematic of the experimental double-inlet pulse tube refrigerator used for the visualization study. The pulse tube (16 mm in diameter and 250 mm in length) was made of a transparent plastic tube. The regenerator (35 mm in diameter and 160 mm in length) was composed of a pile of #100 mesh stainless-steel screens with a wire diameter of 0.1 mm. The reservoir was a plastic vessel with a volume about 6 times larger than the pulse tube and was connected to the pulse tube via an orifice valve. A bypass tube was connected the inlet of the regenerator to the hot end of the pulse tube via either a bypass valve for the valve configuration or a diaphragm unit for the diaphragm configuration. Needle valves were used for the orifice and bypass valves, of which valve openings were measured in the unit of turn-divisions as arbitrary units. Fig. 2 shows details of the diaphragm unit, which was composed of two flanges with a cone-shaped hollow and a circular diaphragm made of polyethylene film with a thickness of 12 μm. The diaphragm was fixed between the flanges. Three diaphragm units, D25, D50 and D80, with a different diameter were used. In addition, tests were conducted by using diaphragms by piling up two or more films to investigate the influence of the rigidity of diaphragm. The size of diaphragm controls the gas displacement volume, and the number of films piled up controls the rigidity of the diaphragm that influence the rate of the displacement of gas volume.

The pressure oscillation was generated by alternatively introducing pressurized air of about 0.2 MPa into a rotary valve and partly releasing it to the atmosphere. Thermocouples were inserted into the pulse tube at the cold and hot ends to measure the gas temperature difference between the cold and hot ends, and the performance of the refrigerator was evaluated based on the gas temperature difference. In general, a larger temperature difference results in better performance. Three pressure transducers were installed near the inlet of the regenerator, at the hot end of the pulse tube, and in the reservoir as indicated in Fig. 1. The smoke-wire made of a 0.1-mm-diameter tungsten wire was located about midway in the pulse-tube.

Experimental condition suitable for smoke-wire visualization was determined based on the results of the preliminary and previous experiments using the pulse tube refrigerator with the valve configuration [10]. The frequency of the gas oscillation was 6 Hz, and the amplitude of the pressure wave defined as the compression ratio between the high and low pressures at the inlet of the regenerator was 1.2. After paraffin was coated on the smoke-wire surface, the wire was put inside of the pulse tube and the lead wires were connected to a high voltage pulse generator. After the pulse tube refrigerator was steadily operated under a specified experimental condition, the smoke-line was emitted. The displacement and the deformation of a smoke-line were recorded for more than five cycles by using a high-speed video camera at a frame rate of 400 frames/sec. Experiment was performed for five kinds of pulse tube refrigerators, which were an orifice pulse tube refrigerator, a double-inlet pulse tube refrigerator with the valve configuration and three double-inlet pulse tube refrigerators with the diaphragm configurations of D25, D50 and D80. Here after, these five pulse tube refrigerators were abbreviated as the orifice refrigerator, the double-inlet (or “double” in the figures) refrigerator, and the diaphragm D25, D50 and D80 refrigerators.

3. Results and discussion

3.1. Performance of refrigerators

Prior to visualization experiments, the thermal performance of the five pulse tube refrigerators was examined on the basis of the gas temperature difference between the cold and hot ends. For the orifice refrigerator, the gas temperature difference was measured while opening the orifice valve little by little with the bypass valve closed. The result was that the temperature difference was maximized at 15 turn-divisions of the orifice valve opening. Based on this result, the opening of the orifice valve was fixed at 15 turn-divisions all through the experiments. Similarly, the temperature difference for the double-inlet refrigerator was measured while increasing the opening of the bypass valve with the opening of the orifice valve fixed at 15 turn-divisions. The maximum temperature difference was reached at the opening of the bypass valve of 5 turn-divisions. These maximum gas temperature differences were adopted as the representative temperature differences of the orifice and the double-inlet refrigerators, respectively. For the operation under the diaphragm configuration, the bypass valve was replaced with the diaphragm unit. Fig. 3 shows the comparison of the measured gas temperature differences. The diaphragm D50 refrigerator had a larger temperature difference than that of the double-inlet refrigerator, but both the diaphragm D25 and D80 refrigerators had smaller temperature differences than that of the double-inlet refrigerator. Moreover, the temperature difference of the diaphragm D80 refrigerator was even smaller than that of the orifice refrigerator. This result indicates that there is an optimum diameter of diaphragm to maximize the performance of refrigerator, and that the diaphragm configuration with the optimum diameter potentially improves the performance of refrigerator compared with that of the double-inlet refrigerator. However, this result also suggests that if the size of diaphragm is far from the optimum size, the refrigerator with the diaphragm configuration would have far inferior performance to that of the refrigerator with the valve configuration.

The effect of the thickness of diaphragm on the performance was also investigated by using the diaphragm D50 refrigerator. The thickness was changed by piling up the polyethylene film with a thickness of 12 μm. Fig. 4 shows the result that the gas temperature difference decreases with the increase of the number of films. This result suggests that the performance of the refrigerator is better for smaller thickness of diaphragm in the range of the present experiment. It is still uncertain whether a film with still thinner thickness than that of the film used here further improves the performance.

3.2. Visual observation of oscillating flow

The fluid dynamic behavior of oscillating flow in the five refrigerators was studied by flow visualization, and the difference between the valve and the diaphragm configurations was investigated while focusing on the secondary flow and the P-V work of the gas. In this study, the secondary flow was defined as induced flows except a main oscillating flow in the pulse tube refrigerator, such as an acoustic streaming [11], the DC flow and so on. The secondary flow was investigated by observing the change in the relative position of the smoke-line before and after a cycle and the P-V work by observing the displacement oscillation of the smoke-line.

Figs. 5a and 5b show typical visualization results of a smoke-line during first two cycles of oscillation after smoke-line emission for the double-inlet and the diaphragm D50 refrigerators, respectively. The arrows indicate the position of the smoke-wire and the direction toward the hot end, respectively. The smoke-line was emitted at the moment when the flow direction changed from expansion to compression in one cycle and thus the bulk movement of the gas was momentarily stagnant. The frame 1 shows the smoke-line just after the emission from the smoke-wire. The frames

4, 7, 10 and 13 show the smoke-line at the turning point where the smoke-line changes its direction of motion. The shape of oscillating smoke-line gradually changed as evident from the comparison between smoke-lines for successive two cycles, such as the frames 4 and 10 at the turning point on the hot end side or 7 and 13 at the turning point on the cold end side.

Figs. 6a ~ 6e show the change in the smoke-line shape at the turning point on the hot end side of every two cycles (1st, 3rd and 5th cycles) for the five refrigerators. With the lapse of cycle, the shape of the smoke-line was deformed from the original shape of straight line to a concave parabolic line toward the hot end for the orifice, the double-inlet and the diaphragm D25 refrigerators, while to a convex parabolic line for the diaphragm D50 and D80 refrigerators. These deformations of smoke-lines suggested that the oscillating main flow in the pulse tube was accompanied by a secondary flow.

The recorded smoke-lines were converted to digital images by using an image-processing system and the change of the displacement of smoke-line along the tube was obtained. Based on the results of Fig. 6, the velocity profile of the secondary flow was estimated from the change in the relative position of the smoke-line before and after a cycle at the turning point on the hot end side under the assumption that the change was caused only by the secondary flow. Here, the estimation was performed on the basis of those of the 3rd and 5th. The obtained velocities were thus an average velocity for two cycles. The smoke-line had a certain extent of width changing with time and moreover, the change in the relative position was not sufficiently large for the secondary flow. Therefore, the obtained velocity had the intrinsic uncertainty and the error was estimated roughly 50 % or less. The result is shown in Fig. 7. Negative and positive velocities in the result correspond to those flowing toward the cold and the hot ends, respectively. For the diaphragm refrigerators, the direction of the velocity in the core region changed from negative to positive with the increase of the area of diaphragm, while the velocity changed oppositely near the wall. These results indicate that there is a possibility that the area of the diaphragm at which the velocity can be adjusted to zero everywhere in a cross section so that the convective heat loss due to secondary flow should reduce to zero. It is, moreover, seen that the velocity near the wall for the diaphragm D50 refrigerator is reduced to less than that of the double-inlet refrigerator though the velocities in the core region are of the same level of about 6 mm/s. This fact suggests that the diaphragm refrigerators are more advantageous than the double-inlet refrigerator with respect to the reduction of the convective heat loss induced by the secondary flow. The cause of this difference in the velocities near the wall can be attributed to whether the DC gas flow exists or not. For the double-inlet refrigerator, it was already reported that the DC gas flow is induced and the overall secondary flow is formed as a superposition of the DC gas flow on the acoustic streaming, and that under the optimized opening of the bypass valve, the velocity of the overall secondary flow decreases to almost zero in the core region while it increases near the wall [10]. In the case of the diaphragm D50 refrigerator, the DC gas flow is not induced owing to the diaphragm unit and accordingly the secondary flow only results from the acoustic streaming so that the velocity of the secondary flow near the wall decreases as a result of a decrease of the velocity in the core region. The secondary flow for each case of different diaphragm thickness was also derived as above, though the result is not presented here. It was found that the velocity of the secondary flow increased with the increase of the number of films. This result means that the convective heat loss due to the secondary flow increases with the increase of the thickness.

The P-V work of the gas in the pulse tube was examined on the basis of the pressure-volume (P-V work) diagram. The P-V diagram was considered on the basis of a control volume of which boundaries were defined by the fixed boundary between the regenerator and the pulse tube, by the wall of the pulse tube on the side, and by a movable boundary at the top of the control volume [1]. The change of the movable boundary during a cycle was obtained from the change of the displacement of smoke-line. Here, the P-V work for the gas on the center axis of the pulse tube was discussed as a representative value. The variation of the gas volume (V) during a cycle was calculated on the fundamental assumption that the gas volume was calculated as a product of unit area and the displacement of oscillating gas along the center axis. From the variations of the gas volume and the pressure during a cycle, the P-V diagrams were plotted for the five pulse tube refrigerators as shown in Fig. 8. The result indicates that the shape of the P-V diagram for the diaphragm refrigerators strongly depends on the area of diaphragm, and that for the diaphragm D25 refrigerator is quite similar to that of the orifice refrigerator while it is different from those for the diaphragm D50 and the double-inlet refrigerators.

Based on each P-V diagram, the magnitude of the P-V work can be estimated from the closed area of the diagram. Fig. 9 shows the variation of the P-V work together with those of the maximum displacement of oscillating gas, and the phase difference between the pressure and the displacement oscillations for each refrigerator, where the P-V work and the displacement are normalized by those of the orifice refrigerator. It is seen that the P-V work of the diaphragm refrigerators once increases with the increase of the diaphragm area, peaking at D50, and then decreased. This result indicates that there is an optimum area of diaphragm at which value the P-V work becomes the maximum, around which the phase difference increases with the increase of the area of diaphragm though the displacement decreases. Moreover, the diaphragm D50 refrigerator had the largest P-V work among the five refrigerators, but those of diaphragm D25 and D80 refrigerators were even smaller than that of the double-inlet refrigerator. The P-V work of the diaphragm D50 refrigerator was as much as 42 % larger than that of the double-inlet refrigerator. The reason for the largest P-V work of the diaphragm D50 refrigerator is thought that the phase difference is increased to 87° from 40° of the double-inlet refrigerator though the displacement decreases by 13 % than that of the double-inlet refrigerator. Moreover, the results in Figs. 8 and 9 show that the diaphragm configuration has the large influence on the P-V diagram. One explanation for this influence may be that a pressure oscillation transmitted through the diaphragm acts like the secondary piston of the warm expander pulse tube refrigerator or the two piston Stirling machine [12].

The effect of the thickness of the diaphragm on the P-V work was also investigated by using the diaphragm D50 refrigerator. Fig. 10 shows the result of the P-V diagrams for four kinds of thicknesses. The shape of the P-V diagram changes from a round shape for one sheet of film to long and flat ovals with the increase of the number of films. Fig. 11 shows the magnitude of the P-V work plotted against the number of films, where the P-V work and the displacement are normalized by those for one sheet of film. The result reveals that the P-V work monotonously decreases with the increase of the number of films due to the decrease in the phase difference though the gas displacement increases. These results of the P-V work and the secondary flow indicate that the performance decreases with the increase of the thickness as shown in Fig. 4. In addition to these results, it was experimentally confirmed that in the limiting case of the hardest diaphragm, a sheet of thin aluminum foil with a thickness of 12 μm, the performance was almost reduced to that of the orifice refrigerator.

The result of this visualization experiment indicates that if an appropriate diaphragm is used instead of a bypass valve, the performance is improved more than that of the refrigerator with the valve configuration. It is considered that the performance improvement is attributed to the increase in the P-V work due to the increase in the phase difference and to the suppression of the convective heat loss induced by a secondary flow. Furthermore, the gas flow losses in the pulse tube may be reduced by the decrease in the displacement of the oscillating gas flow in the pulse tube, though its contribution is not quantitatively evaluated here. We would like to add here before concluding the present paper that judging from small temperature difference between the cold and hot ends, the present experimental condition is not optimized in such points as the use of air as the working gas and low average gas pressure. Therefore, more conventional pulse tube refrigerators must be tested under practical operating conditions to confirm the effectiveness of the diaphragm configuration. Moreover, further investigation will be needed for the optimized selection of diaphragm materials with respect to the refrigerator performance and the durability of the diaphragm.

Summary

Oscillating flow in the double-inlet pulse tube refrigerator with the diaphragm DC gas flow suppressor was visually investigated to evaluate the effectiveness of this configuration. Results show that the optimum configuration of diaphragm exists and that if the appropriate diaphragm configuration is selected, the performance of the refrigerator is improved more than that of the refrigerator with a valve configuration due to the increase in the P-V work and the decrease in the convective heat loss caused by the secondary flow.

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Figure captions :

Fig. 1. Schematic of the experimental apparatus. A bypass valve or a diaphragm unit was alternatively replaced.

Fig. 2. Schematic of the diaphragm units and specifications.

Fig. 3. Comparison of the gas temperature difference between the hot and cold ends for the five kinds of refrigerators.

Fig. 4. Effect of the thickness of diaphragm on the gas temperature difference between the hot and cold ends. The thickness is expressed by the number of films.

Fig. 5. Typical smoke-wire visualization results of the double-inlet refrigerator and the diaphragm D50 refrigerator.

Fig. 6. Comparison of smoke-lines at the hot end turning point for three representative cycles (1st, 3rd and 5th cycles) for the five kinds of refrigerators.

Fig. 7. Velocity profiles of the secondary flow estimated from the visualization result for the five kinds of refrigerators.

Fig. 8. P-V diagrams for the five kinds of refrigerators.

Fig. 9. Variations of P-V work, displacement and phase difference for the five kinds of refrigerators.

Fig. 10. Variation of the P-V diagram with the increase of the thickness of diaphragm. The thickness is expressed by the number of films.

Fig. 11. Influence of the thickness of diaphragm on the P-V diagram, the displacement and the phase difference.