Experimental Study of the Heat Transfer in Pulse Tubes

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ABSTRACT

The present study has been conducted to observe the details of heat transfer under pulsating pressure and oscillating flow in a pulse tube. An experimental apparatus was fabricated to measure the gas temperature, wall temperature, pressure, and the instantaneous heat flux inside a pulse tube. The measured gas temperature and heat flux must be corrected to compensate for their finite time constant under oscillating flow conditions. In experiments performed from 1 Hz to 3 Hz, the phase difference between the instantaneous heat flux and the gas-wall temperature difference was clearly observed. The experimental heat fluxes were compared to theoretical correlations such as the Complex Nusselt Number Model (CNNM) and the Variable Coefficient Model (VCM). In general, the absolute value of the heat flux predicted by the CNNM was greater than that of the VCM. The experiment confirmed the validity of the VCM for the instantaneous heat flux under the pulsating pressure and oscillating flow in the warm end of the basic pulse tube.

INTRODUCTION

Oscillating flow under pulsating pressure is a common phenomenon in an engineering system such as a pulse tube cryocooler, Stirling cryocooler, or G-M cryocooler. Due to the complex physics and lack of experimental data, the heat exchangers in these systems are usually designed by conventional steady-state heat transfer relations that can not predict the oscillating heat transfer phenomena properly. It is known that a phase shift exists between the instantaneous heat flux and the gas-wall temperature difference under oscillating flow and pulsating pressure conditions. The conventional Newton's law of cooling does not contain a term that explains this phase shift phenomenon. Kurzweg[1] attempted to apply the previous oscillating heat transfer data to a Stirling cycle heat exchanger. Gedeon[2] introduced a complex Nusselt number using the results of Kurzweg's work. He obtained a Nusselt number for incompressible oscillating flow and showed the existence of the phase shift between the heat flux at the wall and the gas-wall temperature difference when the oscillatory frequency was high. Kornhauser[3] showed that heat transfer analyses using the complex Nusselt number could predict the experimental data well for the Stirling engine. Jeong et al.[4] obtained two-dimensional

Cryocoolers 11, edited by R.G. Ross, Jr.
Kluwer Academic/Plenum Publishers, 2001

velocity and temperature profiles for the oscillating flow caused by pulsating pressure that was induced by piston motion. He suggested a new heat transfer relation, so called the 'Variable Coefficient Model'. Lee et al.[5] studied analytically the heat transfer of the Stirling cycle heat exchanger. He obtained the gas and wall temperature profiles when the axial gradient of the wall temperature was not constant and the oscillating flow only existed. He also examined the effect of the frequency and the maximum displacement. Jeong et al.[6] installed the heat flux sensor on the outer surface of the heat exchanger for the basic pulse tube and calculated the instantaneous heat flux at the inner wall using the measured data. In their experiment, however, the calculated heat flux at the interior wall had an uncertainty due to the capacitance effect and the complex geometry of the heat exchanger.

This paper describes the experiment of the instantaneous measurement of the heat flux and the temperature at the heat exchangers of the pulse tube under oscillating flow and pulsating pressure. The measured heat flux data were compared with the theoretical predictions that had been previously developed.

EXPERIMENTAL CONFIGURATIONS

An experimental apparatus was fabricated as shown in Fig. 1. At the inlet of the pulse tube, the stainless steel mesh (#200) was stacked to make one-dimensional flow between 1/4 inch tube and 1 inch tube. Its thickness was 4 mm. The cold-end heat exchanger was made of 1 mm thick copper tube and nicely fitted to the pulse tube. To install the heat flux gauge on the inside wall of the heat exchanger, the flange with 7 mm thickness was brazed to the heat exchanger and assembled with bolts. The warm-end heat exchanger had the same shape as that of the cold-end heat exchanger, but it had the water jacket to cool the warm-end by the cooling water as shown in Fig. 2.

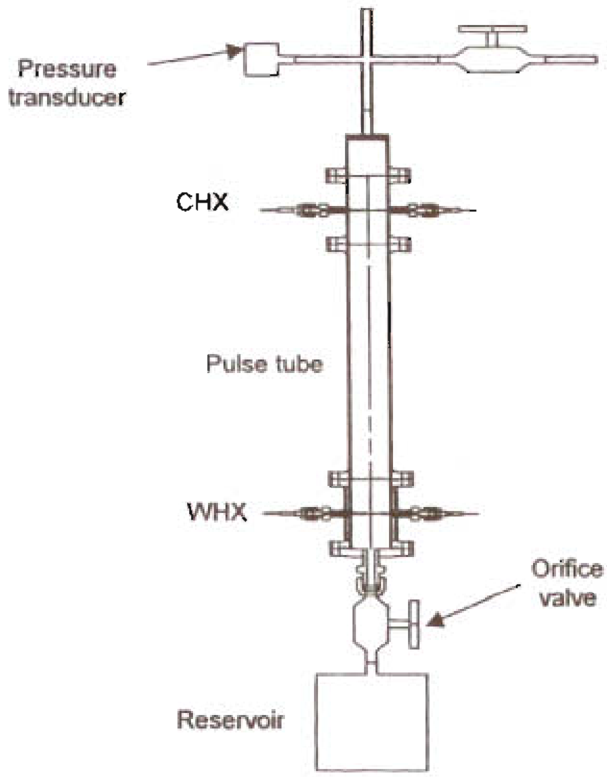

Figure 1. Schematic diagram of the experimental apparatus. (CHX: cold-end heat exchanger, WHX: warm-end heat exchanger).

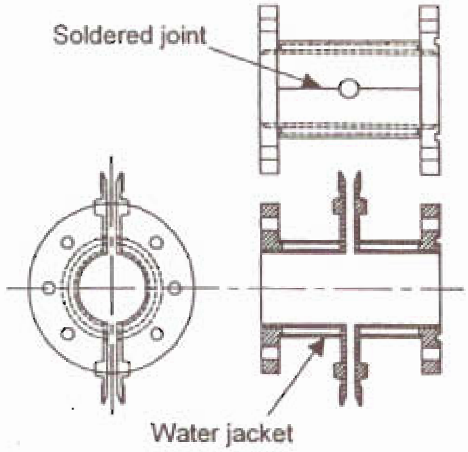

Figure 2. Warm-end heat exchanger.

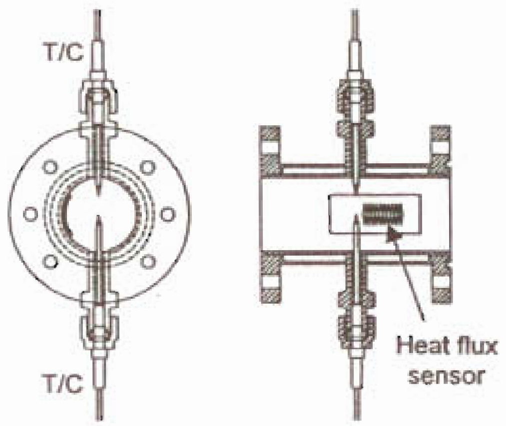

Figure 3. Sensor Installation (T/C: thermocouple).

Fig. 3 illustrates the sensor installation at the heat exchanger. A fast response thermocouple (OMEGA Model EMQSS-010E, type-E) was inserted into the heat exchanger by reducer fitting. The wire diameter of this thermocouple was 0.038 mm and its junction was exposed to the flow field for the fast response characteristics. The heat flux gauge was attached to the inner wall of the heat exchanger. It was a 75 μm thick micro-foil type sensor and its time constant for the step input was 20 ms according to the manufacturer. Thermal resistance of the heat flux gauge was very small compared to that of the convection with the helium gas. The difference of the heat flux between the case with the sensor and without the sensor was about 5 % of the heat flux value. The heat flux gauge had also the T-type thermocouple that could measure the wall temperature of the heat exchanger.

The pulse tube was made of 1 mm thick stainless steel tube with outer diameter of 25.4 mm and the length of 200 mm. Helium compressor (CTI-cryogenics Model 8200) was used to supply helium gas as working fluid. The rotary valve system provided the pulse tube with pulsating gas pressure and flow. The frequency of the pulsating pressure was adjusted by the rotational speed of the rotary valve, which was controlled by the stepper motor and the function generator.

The strain gauge type pressure transducer (Sensym Model ST2000) was installed at the inlet of the pulse tube to monitor the pulsating pressure. The signal of the heat flux gauge was so small (the order of μV) and so sensitive to the environmental noise that this signal was pre-amplified by isolation pre-amplifier (YOKOGAWA Model 313100-61E). All the experimental data were acquired by data acquisition board (Keithley Model DAS1600) and stored in the personal computer when the cyclic steady-state was reached.

EXPERIMENTAL RESULTS AND DISCUSSION

As mentioned earlier, utilizing complex Nusselt number concept, Kornhauser[3] proposed a heat transfer correlation as follows:

$$q'' = \frac{k}{D_h}[Nu_r \cdot \Delta T + \frac{Nu_i}{2\pi f} \cdot \frac{d\Delta T}{dt}] \quad (1)$$

where

k = thermal conductivity of the gas
D_h = hydraulic diameter of the pulse tube
f = oscillating frequency of pressure wave
Nu_r = real part of the complex Nusselt number

Nu_i = imaginary part of the complex Nusselt number

The complex Nusselt number in Eq. (1) was determined by using the least square method from the experimental data.[3] He also presented the complex Nusselt number for various Peclet number ($Pe_\omega = \rho C_P \omega D_h^2/4k$). On the other hand, Jeong et al.[4] has derived the following relation from the two-dimensional energy equation for the case of the fluid flow between two flat plates. The heat flux at the wall was expressed as follows:

$$q'' = \frac{k}{H}\left[K_1 \cdot \Delta T + \frac{K_2 \cdot H^2}{k} \cdot \frac{dP}{dt}\right] \tag{2}$$

where

$$K_1 = \frac{6 \cdot H}{\lambda \cdot (1-\zeta) \cdot (5-\zeta)}$$

$$K_2 = \frac{-\lambda(1-\zeta)}{H(5-\zeta)}$$

$$\zeta = \exp\left(\frac{-H}{\lambda}\right), \quad \lambda = \sqrt{\frac{3kRT_m}{P\, C_p \pi f}}$$

$$\Delta T = T_{gm} - T_{w}$$

$2H$ = distance between two flat plates
R = ideal gas constant
P = pressure of the gas

The thermal conductivity of the gas and the wall temperature were assumed to be constant when Eq. (2) was derived. When Eq. (2) was applied to the pulse tube, 2H was assumed to be the diameter of the pulse tube.[7] The fundamental difference of Eq. (2) from Eq. (1) is that the heat flux term associated with the oscillation is derived from the direct instantaneous pressure change. Although the temperature gradient term in Eq. (1) is replaced by the pressure gradient term in Eq. (2), two equations describe virtually the same physical phenomenon because the temperature is varied by the pressure change. Kornhauser's relation contained the complex Nusselt number that must be determined from the extensive experiment, but Jeong's relation included no empirical constant. Kornhauser's relation was also known as the Complex Nusselt Number Model (CNNM) and Jeong's relation as the Variable Coefficient Model (VCM).

Under the pulsating pressure and oscillating flow, the heat flux and the gas-wall temperature difference have been measured as shown in Fig. 4. In this paper, the experimental data are shown only for the case of the closed orifice (basic pulse tube type) and no load condition at the cold-end heat exchanger. There was an apparent phase shift between the heat flux and the temperature difference at 2 Hz, but the phase shift was not clear at 1 Hz. The heat flux was also calculated by Eq. (1) and Eq. (2) using the measured temperature and the pressure. The predicted flux was compared to the measured heat flux at the cold-end and the warm-end heat exchanger as shown in Fig. 5 and Fig. 6. The negative heat flux means the heat flow from the gas to the wall. At the warm-end heat exchanger, both the CNNM and the VCM can predict the heat flux reasonably well. On the other hand, Fig. 6 shows that both the results from the two theoretical relations failed to predict the experimental data at the cold-end heat exchanger. The CNNM predicted the heat flux variation pretty well, but with large amount of discrepancy. This result indicated that the complex Nusselt number suggested by Kornhauser could not be applicable to the cold-end heat exchanger. Since Kornhauser correlated the complex Nusselt number only by Peclet number (Pe_ω), his empirical correlation could not represent the general case. Therefore, it is suggested that another non-dimensional parameter, such as maximum Reynolds number (Re_{max} =

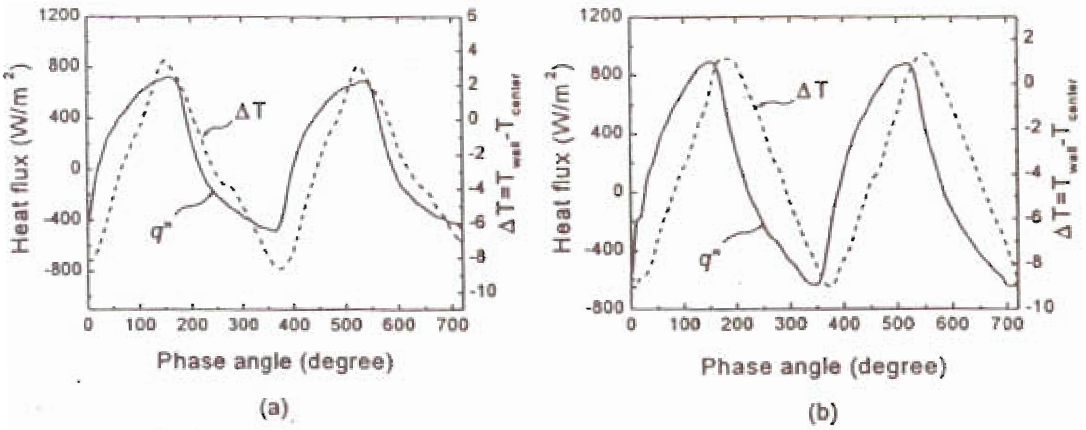

Figure 4. Heat flux and gas-wall temperature difference at the cold-end heat exchanger: (a) 1 Hz (b) 2 Hz

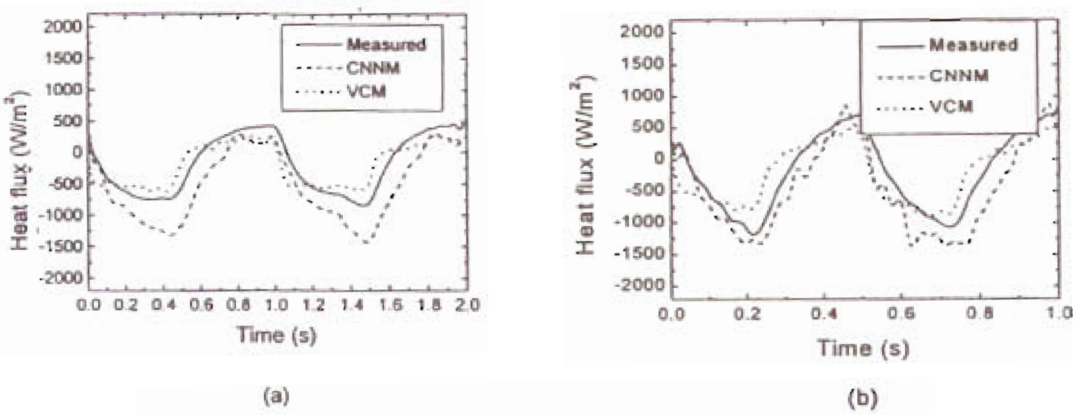

Figure 5. Heat flux at the warm-end heat exchanger: (a) 1 Hz (b) 2 Hz

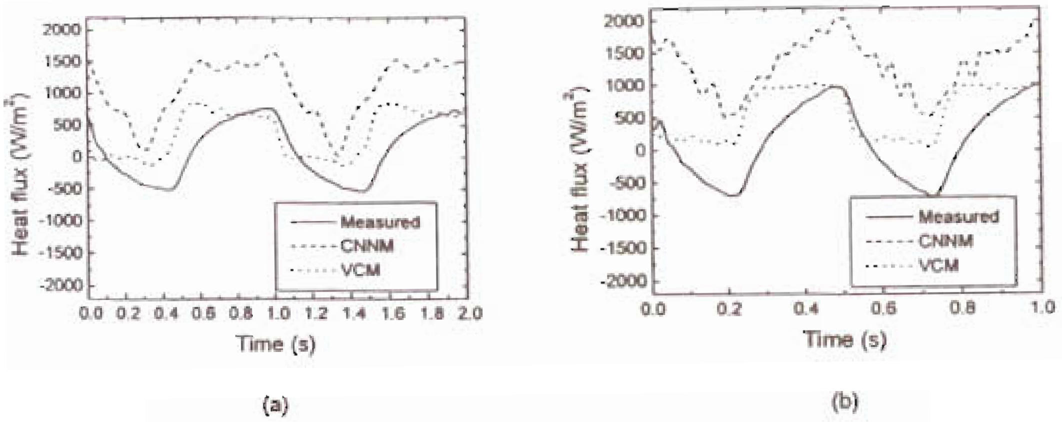

Figure 6. Heat flux at the cold-end heat exchanger: (a) 1 Hz (b) 2 Hz.

Table 1. Comparison of the integrated heat flux during one cycle

Frequency (Hz)	Integrated heat flux at the warm-end heat exchanger [J]		
	Measured	CNNM	VCM
1	0.74	1.94	0.69
2	0.39	0.93	0.40
3	0.21	0.62	0.31
Frequency (Hz)	Integrated heat flux at the cold-end heat exchanger [J]		
	Measured	CNNM	VCM
1	0.71	4.63	1.63
2	0.35	2.85	1.25
3	0.20	1.60	0.78

$\dot{m}\, D_h/\mu A$), should be included in the complex Nusselt number. Here, $\dot{m}$ is the maximum mass flow rate of the oscillating flow, μ is the viscosity of the gas and A is the cross-sectional area. The VCM did not match the variation pattern of the heat flux in one cycle, but the cyclic integrated value was more close to the measured value than that of the CNNM. Table 1 shows the comparison of these values.

The CNNM had another difficulty in its application at higher frequency. Since it contains a time derivative term of the measured temperature as shown in Eq. (1), the prediction of the heat flux by the CNNM may require one more calculation step. As shown in Fig. 7(a), a small variation in gas temperature was amplified by the time derivative operation and the heat flux from this gas temperature had a large oscillation as shown in Fig. 7(b). This inevitable error amplification could be virtually reduced by approximating the measured temperature to a smooth function like sinusoidal wave (continuous line in Fig. 7). Fig. 8 shows, thus, the predicted heat flux of the CNNM by this error reducing method.

CONCLUDING REMARKS

The experiment was performed to the heat exchangers of the basic pulse tube for the frequency of 1 – 3 Hz. From this experiment, we recognized the following results.

(1) The VCM (Variable Coefficient Model) could predict the heat flux in the warm-end heat exchanger of the pulse tube better than the CNNM (Complex Nusselt Number Model).

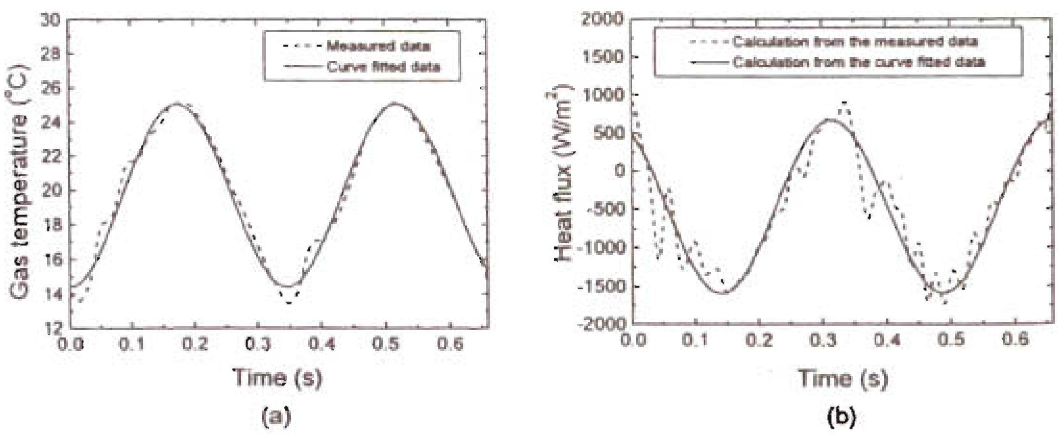

Figure 7. Heat flux from the CNNM using the curve fitted gas temperature (3 Hz, warm-end)

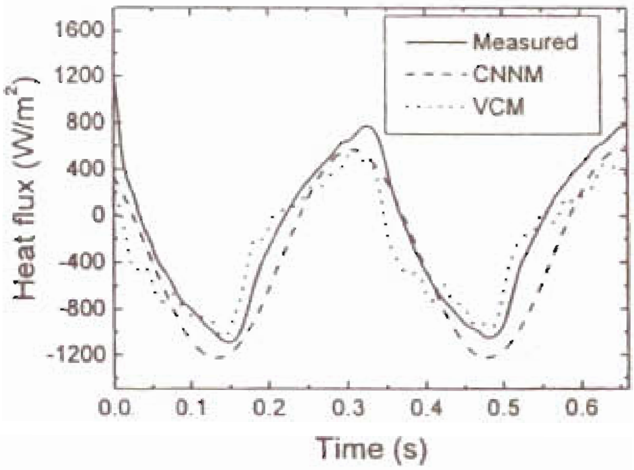

Figure 8. Comparison of the heat flux using the calculated complex Nusselt number. (3 Hz, warm-end).

(2) Neither the VCM nor the CNNM predicted the heat flux at the cold-end heat exchanger where the oscillating flow effect was noticeable. The better model is necessary for the oscillating flow with large amplitude.

ACKNOWLEDGMENT

This research was supported by the Korea Research Foundation under the contract no. 1998-018-E00164 and partially supported by the Brain Korea 21 project. The authors would also like to acknowledge the support of the Ministry of Science and Technology through the Dual Use Technology project (grant no. 99-DU-04-A-02).

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