Study of Free Piston Vuilleumier Heat Pump*

(Performance Characteristics of Prototype Machine during Self-Excited Vibration)

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A prototype of a free piston Vuilleumier heat pump (FPVM) was designed and fabricated in order to study the basic FPVM performance characteristics during self-excited vibration. The FPVM has a cooling capacity of 3.2kW and a cooling COP of 0.6 during self-excited vibration. Dependence of the dynamic motion of the displacers and the FPVM performance on the mean gas pressure, hot gas temperature, and hot and cold water temperatures was investigated experimentally and analyzed using a simple 2nd order isothermal simulation method combined with linear analysis. The performance characteristics of the FPVM when a hot and a cold displacer were, and were not, connected by a spring were examined and discussed. There was no qualitative difference between the FPVM's. It was found that the simulation results agreed well qualitatively with the experimental results and that the simulation method was suitable for analyzing the FPVM performance characteristics and designing a practical machine.

Key Words: Vuilleumier Cycle, Self-Excited Vibration, Performance Test, Numerical Analysis, Heat Pump, Air Conditioning, Free Piston, Coefficient of Performance

1. Introduction

A Vuilleumier heat pump (hereafter abbreviated as VM) which uses no CFC's is a thermally activated heat pump driven by combustion heat of natural gas. To reduce the size and structural complexity of the VM, the authors have developed a free-piston VM (hereafter abbreviated as FPVM) which uses a vibration system as the displacer drive⁽¹⁾. FPVM's exhibit complicated operating characteristics because the strokes of the two displacers and the phase angle between them vary with the operating frequency, depending on the spring constant and displacer masses as well as the operating conditions, including the operating temperature. Thus the clarification of the FPVM characteristics requires analysis by numerical simulation and experimental analysis based on performance testing of an actual FPVM system.

There are two operating modes available for the FPVM; the self-excited operation without motor vibration, and forced motor vibration for one of two displacers. In their previous studies, the authors derived the basic equations for the dynamic behavior of the displacers during self-excited operation and forced vibration from linear analysis⁽²⁾, and established the performance characteristics of forced motor vibration by experiment using an FPVM prototype and simulation analysis⁽³⁾. On the other hand, the self-excited operation of actual FPVM systems have only been reported by Schulz et al.⁽⁴⁾ and the present authors⁽¹⁾. In these reports, Schulz employed a prototype in which the hot and cold displacers were connected with a spring, while the authors used a system without such a connection. Although Schulz⁽⁴⁾ and Kuhl et al.⁽⁵⁾ showed the dynamic behavior of the displacers during operation and the dependence of performance upon the operating parameters such as the working gas temperature, they did not make any qualitative or quantitative comparisons of these experimental results by simulation analysis.

Regarding the effect of the mechanical spring coupling developed by Schulz, the present authors

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showed the difference between the energy transmission mechanisms of the vibration system, and used simulation analysis to elucidate the dependence of the performance characteristics on the coupling spring constant⁽²⁾. However, this subject has not been investigated using an actual FPVM system.

This paper presents the results of experiments conducted to establish the effect of this coupling. Using an FPVM prototype, the effect was examined using the gas spring connection between the hot and cold buffer spaces. Self-excited operating characteristics with and without this connection effect were experimentally elucidated, and the test results were compared with the results of simulation analysis. Unlike the simulation analysis of the previous report⁽²⁾, which primarily concerned the coupling effect due to a mechanical spring, the experiment presented here employed coupling established by a gas spring only. The simulation analysis also examined the difference between the mechanical and gas spring couplings.

2. Equipment and Experimental Method

2.1 Prototype FPVM

The configuration of the prototype FPVM used is

Table 1 Main specifications

Heater type	bare pipe
Heat exchanger type	shell & tube
Regenerator	packed wire screens
Linear motor type	moving coil
Hot displacer stroke(max.)	55mm
Cold displacer stroke(max.)	60mm
Hot displacer diameter	80mm
Cold displacer diameter	90mm
Hot displacer rod diameter	28mm
Cold displacer rod diameter	30mm
Displacer mass	$m_H=2.8\text{kg}$ $m_C=3.3\text{kg}$
Spring constant	$k_H=46.6\text{kN/m}$ $k_C=18.0\text{kN/m}$
Buffer space volume	$V_{BH}=1.60 \times 10^{-3} \text{ m}^3$ $V_{BC}=0.45 \times 10^{-3} \text{ m}^3$

Fig. 1 Schematic view of free piston Vuilleumier heat pump machine

illustrated in Fig. 1, and the main specifications are listed in Table 1. The hot displacer is connected to the case in the buffer space via a coil spring and a displacer rod. The cold displacer is connected to a displacer rod fixed to the case, within itself via a coil spring. The cold displacer contains a buffer space. The linear drive motors are of the moving coil type, which is almost free from lateral forces. The coils are fixed to the displacer rods, and the yokes to the buffer cases. The hot displacer is guided by a sliding ring attached to it and a guide ring inside the yoke of the linear motor. The cold displacer is guided by two sliding rings attached to its cold and warm sides, respectively, so that it can perform linear movement inside each cylinder. Each displacer is provided with a PTFE resin gas seal. The hot displacer rod is gas-tight with a floating resin seal fixed to the case, while the cold displacer rod's gas seal is fixed to the displacer side. The heater has been configured so that many bare pipes are arranged in close contact around a circle. The heat exchangers are of the shell and tube type, and are arranged in annular form around the displacer, together with the regenerator. The regenerator is made of packed stainless steel wire. The combustor is a diffusion combustor with a corrugated air preheater.

In the prototype FPVM configuration in Fig. 1, the broken lines show the connecting pipe between the hot and cold buffer spaces to couple the two displacers via a gas spring. This pipe was installed only when it was required as a gas spring coupling. The connecting pipe has an inner diameter of 30 mm and a length of 490 mm. When the buffer spaces are connected, the total buffer volume is $2.40 \times 10^{-3} \text{ m}^3$. The gas spring constants affecting the respective displacers vary depending on whether or not the buffer spaces are connected. In the hot displacer vibration system, the gas spring constant represents only a small portion of the system's total spring constant. In contrast, the gas spring constant in the cold displacer vibration system represents a substantial portion. Therefore, the total spring constant for non-connected operation is larger than that for connected operation. To achieve self-excited operation in the FPVM, it is necessary that both displacer vibration systems have almost the same natural frequency⁽²⁾. To achieve this when the buffer spaces were not connected, the previous mass of the cold displacer, 3.3 kg, was changed to 3.6 kg. In addition, it is necessary to prevent the displacers from colliding against the cylinders when testing the dependence on the operating parameters for buffer space connection. To this end, the hot displacer diameter was reduced from 28 mm to 26 mm to reduce the excitation acting on the hot displacer, thus limiting

the stroke. In this experiment, additionally, the mass of the hot displacer was changed to 3.1 kg and that of the cold displacer to 3.4 kg.

According to the linear analysis in the previous report⁽²⁾, both gas-spring and mechanical-spring couplings have the same effect on the energy transmission mechanism of the FPVM vibration system. Schulz's FPVM⁽⁴⁾, which arranges two displacers in series in one cylinder and couples them via a mechanical spring, is expected to be useful in reducing the size of the FPVM system. On the other hand, the authors' FPVM, which achieves coupling with a gas spring, puts no limitations on the arrangement of either displacer, thus proving greater freedom in FPVM system design. Other advantages of the gas-spring coupling are: it can also be used in the Schulz type FPVM and the design of its elements, including the linear motors and springs, is easier to optimize.

2.2 Experimental method and conditions

The average working gas pressure in the FPVM was measured using a strain gauge pressure sensor installed in the warm space of the hot section. The prescribed pressure was maintained by regulating the amount of gas filled. The average working gas temperature in each working space was measured using a sheath alumel-chromel thermocouple with a 1 mm outer diameter. The hot gas temperature was controlled by regulating the flow rate of the natural gas fuel. The temperature was measured using alumel-chromel thermocouples spot-welded to the outer walls of the heater tube and hot cylinder, and copper-constantan thermocouples spot-welded to the outer wall of the cold cylinder. The inlet temperature and flow rate of the circulating warm and cold water supplied to the FPVM were kept constant using a constant-temperature water feeder. The temperatures of the warm and cold water at the inlet and outlet of the heat exchanger were measured using a thermistor, and the flow rates were measured using an oval type flow meter. Warm water was passed from the warm heat exchanger in the cold cylinder directly to that in the hot cylinder. Instantaneous gas pressures in the hot space, two warm spaces, cold space, and two buffer spaces were measured using piezoelectric pressure converters. The displacement of the hot displacer was detected via the displacer rod by using a laser displacement meter from outside the buffer space through a sight glass attached to the lower flange of the buffer space. The displacement of the cold displacer was measured using a laser displacement meter through a sight glass attached to the warm space flange. The coil terminals of the displacer driving linear motors were opened when the motors are not used, in order to reduce the load during

displacer movement. Table 2 shows the standard test conditions. Except for those changed for the experiment, all operating parameters were fixed at the values shown. Power to the linear motors was supplied by amplifying the sine waveform from the transmitter using a power amplifier. As in our previous reports^{(2),(3)}, the phase angle between the two FPVM displacers, α , is represented by the delay of the hot displacer motion relative to cold displacer motion. For both displacers, it is assumed that the upper motion in Fig. 1 is positive.

3. Experimental Results

3.1 Basic operating characteristics of self-excited operation

Figures 2 and 3 show examples of measurements of the displacer displacements x_H and x_C during the self-excited operating mode of the prototype FPVM and their power spectrum obtained by fast Fourier transform (FFT) analysis. These results were

Table 2 Test conditions

Working gas	Helium
Mean pressure	10MPa
Hot gas temperature	973K
Cold water inlet temperature	285K
Hot water inlet temperature	313K
Water flow rate	$0.143 \times 10^{-3} \text{ m}^3/\text{sec}$

Fig. 2 Motion of displacers

Fig. 3 Power spectrum of displacer motion by FFT

obtained for the operation mode in which the two buffer spaces were not connected. The operating frequency was 16.5 Hz. These figures show that the motions contain only low levels of higher order vibrations. Thus the displacers move in sine waves. Figure 4 shows the working gas pressure fluctuation and its power spectrum for the same operation. It can be seen that the working gas pressure fluctuation also has a sine waveform, with low levels of higher order vibrations.

Table 3 shows the performance characteristics for when the hot and cold displacer vibration systems are coupled by a spring and when they are not. The

Table 3 FPVM performance

Item	FPVM with connecting pipe			FPVM without connecting pipe
	Hot displacer rod diameter	26mm		28mm
Motor input	0W	0W	93W	0W
Frequency	15.7Hz	15.5Hz	15.4Hz	16.5Hz
Cold displacer stroke	56.9mm	49.3mm	55.5mm	45.9mm
Hot displacer stroke	51.6mm	39.2mm	49.7mm	32.7mm
Phase angle	88deg.	91deg.	98deg.	92deg.
Gas temperature in cold space	275K	277K	275K	280K
Gas temperature in warm space	327K	324K	329K	320K
Cooling capacity	3236W	2347W	3206W	1512W
Heating capacity	7462W	5408W	7457W	3795W
Indicated work in cold space	4595W	3350W	4479W	2250W
Indicated work in buffer space	-90W	-42W	-51W	6W
COP _{intc}	110W	89W	105W	93W
COP _{intH}	1.51	1.77	1.51	1.78
COP _{ic}	2.47	2.76	2.53	2.76
COP _{iH}	0.75	0.76	0.76	0.65
COP _c	1.72	1.76	1.77	1.64
COP _H	0.60	0.58	0.62	0.53
COP _H	1.37	1.34	1.44	1.34

(a) Pressure

(b) Power spectrum by FFT

Fig. 4 Working gas pressure

internal COPs(COP_{intH}, COP_{intc}), indicated COPs (COP_{iH}, COP_{ic}), and net heating/cooling COPs(COP_H, COP_c) are defined by the following equations:

$$\text{COP}_{\text{intc}} = W_c / W_H \quad (1)$$

$$\text{COP}_{\text{intH}} = W_M / W_H \quad (2)$$

$$\text{COP}_{\text{ic}} = Q_c / Q_{\text{IN}} \quad (3)$$

$$\text{COP}_{\text{iH}} = Q_M / Q_{\text{IN}} \quad (4)$$

$$\text{COP}_c = Q_c / Q_F \quad (5)$$

$$\text{COP}_H = Q_M / Q_F \quad (6)$$

$$Q_{\text{IN}} = Q_M - Q_c + W_H + W_c - W_M \quad (7)$$

where Q_F is the fuel input based on the higher caloric value. Here W_c , W_H , and W_M represent the indicated work in the cold, hot, and warm spaces, respectively; Q_c is the cooling capacity; and Q_M is the heating capacity. All these parameters are represented by the same sign. Also COP_{intH} and COP_{intc} indicate the COP of a gas cycle not containing the heat loss in the heat exchangers, and COP_{iH} and COP_{ic} represent the COP containing the heat losses in the heat exchangers but not the heat loss in the combustor. Comparison of the results for the same combination of diameters of the hot and cold displacer rods reveals that by connecting the buffer spaces, the cooling capacity rises more than double, from 1.5 kW to 3.2 kW, and COP_{ic} and COP_c are 0.75 and 0.60, favorable levels comparable with those of a conventional VM fitted with a crank mechanism⁽⁶⁾. Although COP_{intc} is reduced from 1.78 to 1.51, this is because of the increases in the amount of heat exchanged in the heat exchangers, which raised the warm space gas temperature and lowered the cold space gas temperature. For the warm space gas temperature in Table 3, the upper and lower lines correspond to the hot and cold sides. When the diameter of the hot displacer rod is reduced, the stroke and cooling capacity, Q_c , during self-excited operation becomes smaller, but the COP changes only slightly. In this case, by driving the hot displacer with a motor input of 93 W, the cooling capacity increases to 3.2 kW, providing a COP_c of 0.62. For work in the buffer space in Table 3, the upper and lower lines correspond to the work done by the hot and cold displacer rods. If the diameter of the hot displacer rod is reduced from 28 mm to 26 mm during connected operation, the excitation acting on the hot displacer becomes smaller, while at the same time the coupled spring constant drops. During self-excited operation with a rod diameter of 28 mm, the work in the hot buffer space is negative, which means that because of the gas-spring coupling, work of 90 W has been transmitted from the cold displacer. During operation using a 26 mm rod, if the system is excited to obtain a comparable stroke, the coupling spring transmits less work, namely 51 W.

3.2 Comparison of energy flow

Figure 5 shows the energy flows in the prototype FPVM when the two displacer vibration systems are coupled with a spring and when they are not. When the buffer spaces are not connected, i.e., when there is no gas-spring coupling (Fig. 5(a)), the hot and cold displacers are operated by generating mutually independent displacer driving work. Displacer driving work is consumed as sliding losses associated with the

($f=16.5$ Hz, $S_H=32.7$ mm, $S_C=45.9$ mm, $\alpha=92$ deg.)
(a) FPVM without connecting pipe

($f=15.7$ Hz, $S_H=51.6$ mm, $S_C=56.9$ mm, $\alpha=88$ deg.)
(b) FPVM with connecting pipe

Fig. 5 Energy flow

sealing and guide rings and work loss in the buffer spaces. In the prototype FPVM, the work loss in the cold buffer space appears as heat emitted to the cold space and in the cold section of the warm space. On the other hand, when the vibration systems are coupled via a gas spring (Fig. 5(b)), the 121 W of work generated by the cold displacer is partly consumed by driving the cold displacer, with over 70 percent being transmitted to the hot displacer via the gas spring of the buffer spaces. As indicated in the figure, the work transmitted to the hot displacer is partly consumed by driving the displacer and the rest is recovered as compression work on the hot side. When the buffer spaces are not connected, the sum of the work losses in the buffer spaces is approximately 100 W. When the buffer spaces are connected, however, it is only 20 W despite the increases in the displacer strokes and capacities. By coupling the hot and cold displacers using the connecting pipe, most of the compression work generated in the cold buffer space can be recovered as work for driving the hot displacer, thereby significantly reducing the work loss in the buffer space.

3.3 Characteristics of self-excited operation

This subsection shows how the displacer motion and performance of vibration systems with and without spring coupling, depend on the working gas pressure and operating temperature during the self-excited operation of the prototype FPVM. The results of simulated calculation are also presented. When the buffer spaces were connected, the 26 mm-diameter displacer rod was used. In Figs. 6, 7, 8, and 9 (a) and (b) indicate the system with and without coupling. The solid lines indicate test results and the broken lines simulation results.

Figure 6 shows the dependence of the displacer motion and of FPVM performance on the average working gas pressure, P_m . As P_m rises, the phase angle, α , between the hot and cold displacer strokes, S_H and S_C , increases and the operating frequency, f , becomes higher. According to linear analysis, f and α during self-excited operation are expressed by the following equations⁽²⁾.

$$f = \sqrt{(C_H K_C + C_C K_H) / (m_H C_C + m_C C_H)} / 2\pi$$

$$= \sqrt{\{(m_H / C_H) \omega_H^2 + (m_C / C_C) \omega_C^2\} / (m_H / C_H + m_C / C_C)} / 2\pi \quad (8)$$

$$\tan \alpha = 2\pi f (m_H C_C + m_C C_H) / \{m_H m_C (\omega_H^2 - \omega_C^2)\} \quad (9)$$

$$\omega_H = \sqrt{K_H / m_H}, \quad \omega_C = \sqrt{K_C / m_C} \quad (10)$$

$$K_H = k_H + \left\{ -\frac{P_0 A_H A_{RH}}{V_0} \left(\frac{A_H - A_{RH}}{A_H} - \frac{T_M}{T_H} \right) \right\}$$

$$+ x \frac{P_0 A_{RH}^2}{V_{B0}} \quad (11)$$

$$K_C = k_C + \left\{ -\frac{P_0 A_C A_{RC}}{V_0} \left(\frac{A_C - A_{RC}}{A_C} - \frac{T_M}{T_C} \right) \right\}$$

Fig. 6 Dependence of FPVM performance on P_m

$$+ \kappa \frac{P_0 A_{RC}^2}{V_{B0}} \quad (12)$$

$$V_0 = T_M \Sigma V_i / T_i \quad (13)$$

Where A_c and A_h are the cross-sectional areas of the cold and hot displacers, A_{RC} and A_{RH} are the cross-sectional areas of the cold and hot displacer rods, V_{B0} is the buffer space volume, and κ is the specific heat ratio. When the buffer spaces are not connected, V_{B0} 's in Eqs. (11) and (12) are replaced by the hot-side and cold-side buffer space volumes V_{BH0} and V_{BC0} , respectively. Regardless of whether or not the buffer spaces are connected, the gas spring constant of the working space acting on the hot displacer and occurring in the second term on the right side of Eq. (11), is negative. Similarly, the gas spring constant of the working space acting on the cold displacer and occurring in the second term of the right side of Eq. (12) is positive. As P_m becomes higher, the hot-side total spring constant K_H becomes smaller, while the cold-side total spring constant K_C becomes larger. Therefore, the natural frequency of the hot displacer alone, ω_H , becomes lower, the natural frequency of the cold displacer alone, ω_C , becomes higher, and α becomes larger. According to the simulation calculation, if P_m rises from 8.5 MPa to 10 MPa, when there is no connection, ω_H falls by 0.4 Hz, while ω_C rises by 0.8 Hz. On the other hand, when there is a buffer space connection, ω_H falls by 0.3 Hz, while ω_C rises by 0.6 Hz. Because both displacers of the prototype FPVM have

Fig. 7 Dependence of FPVM performance on T_H

much the same masses m_H , m_C , and damping resistances c_H and c_C , it is thought that f rises as P_m increases, whether the buffer spaces are connected or not. In Figs. 6 (a) and 6 (b), W_c and Q_c become larger with an increase in P_m . With increase in P_m , COP_{iH} and COP_{iC} gradually fall, while COP_{iH} and COP_{iC} slowly rise. The reduction in COP_{iH} and COP_{iC} can be explained by an increase in the amount of heat exchange in heat exchangers with the increase in P_m , which leads to a lower cold space gas temperature and a higher warm space gas temperature. On the other hand, the increases in COP_{iH} and COP_{iC} are caused by performance-independent heat losses in the heat exchangers, such as heat conduction losses, and becomes smaller relative to the capacity. There is no qualitative difference between the P_m dependence characteristics exhibited when the two displacers are and are not coupled via a gas spring. Also the experimental results show qualitatively good agreement with the simulation results.

Figure 7 shows the dependence of displacer motion and performance on the hot gas temperature T_H . When the buffer spaces are not connected, self-excited operation is achieved for T_H levels over 902 K. When the buffer spaces are connected, it is achieved at an even lower temperature, 813 K, despite the smaller diameter of the hot displacer rod. In both cases, as T_H increases, S_H , S_C , α become larger and f also gradually increases. The T_H dependence of f shown in Fig. 7 (a)

differs from the experimental results by Kuhl et al.⁽⁵⁾. According to their test data, f gradually drops with increase in T_H . As T_H becomes higher, it follows from Eqs. (11) and (12) that K_H decreases, K_C increases, ω_H decreases, and ω_C increases. The difference may be explained as follows: the present prototype FPVM is greatly affected by an increase in ω_C ; however, the system used by Kuhl et al. was more greatly affected by an increase in ω_H . In Fig. 7, W_C and Q_C become larger as T_H becomes higher. COP_{intH} and COP_{intC} show only slight changes, while COP_{iH} and COP_{iC} become greater as T_H rises. These results reveal that T_H is an effective parameter for controlling the FPVM capacity during self-excited operation. When the buffer spaces are connected, by reducing T_H from 973 K to 813 K, Q_C can be reduced by around two thirds, from 2.5 kW to 0.9 kW. Compared with the significant change in Q_C , COP_{iC} is reduced only slightly, from 0.77 to 0.64. On the other hand, when the buffer spaces are not connected, as T_H drops from 973 K to 902 K, Q_C falls from 1.5 kW to 0.9 kW. Because self-excited operation is possible in a narrower temperature range, the capacity can change through a narrower range than it can during connected operation.

Figure 8 shows the dependence of the displacer motion and performance on the inlet temperature of cold water, T_{WC} . As T_{WC} rises, S_H , S_C and α fall, and f also becomes lower. Thus W_C and Q_C become smaller. Also COP_{intH} , COP_{intC} , COP_{iH} and COP_{iC} increase. S_H and S_C become smaller because, as T_{WC}

Fig. 8 Dependence of FPVM performance on T_{WC}

risers, the working gas pressure fluctuation, which is generated by the cold displacer motion and which excites the hot displacer, becomes smaller. Because K_H and K_C drop with rise in T_{WC} , ω_C and ω_H become smaller, and f becomes lower. In addition, it is thought that α drops because in proportional terms the reduction in ω_C is larger than the reduction in ω_H . The dependence of the displacer motion and performance on T_{WC} is less significant when the buffer spaces are connected.

Figure 9 shows the dependence of displacer motion and performance on inlet temperature of hot water T_H . When the buffer spaces are not connected (Fig. 9(a)), as T_{WH} rises S_H increases, while S_C scarcely changes. In contrast, when the buffer spaces are connected (Fig. 9(b)), it will be found that as T_{WH} rises S_H changes extremely little and S_C becomes smaller. As T_{WH} rises, the working gas pressure fluctuation, which is generated by the cold displacer motion and which excites the hot displacer, becomes smaller. Therefore, in both cases, the stroke ratio S_C/S_H becomes smaller as T_{WH} rises, because K_H and K_C increase with rise in T_{WH} , ω_C and ω_H become larger, and f becomes higher. It is thought that α becomes larger because in proportional terms the increase in ω_C is larger than the increase in ω_H . When the buffer spaces are not connected, it can be observed that as T_{WH} rises W_C and Q_C increase gradually. When the buffer spaces are connected, however, W_C gradually increases, whereas Q_C drops because of the increase in

Fig. 9 Dependence of FPVM performance on T_{WH}

Fig. 10 Dependence of FPVM performance on K_L

heat loss at high capacity. Also COP_{inH} , COP_{intc} , COP_{iH} and COP_{ic} become smaller as T_{WH} rises.

Compared with the experimental results, the simulation calculation evaluated f at a higher level and S_H , S_C and α at lower levels. However, the simulation results, including heating/cooling capacity and COP's, show good qualitative agreement with the experimental results.

4. Effect of Gas-Spring Coupling

The structure of the prototype FPVM makes it difficult to change the buffer volume so as to change the gas spring constant. Therefore the effect of gas-spring coupling on FPVM performance was examined by simulation calculation. Figure 10 shows the dependence of FPVM performance on the coupling spring constant K_L . The broken lines indicate the results for coupling via the gas spring alone, the alternate long and short broken lines the results for coupling via a mechanical spring alone, as in Ref. (2). Calculations were carried out for a hot displacer rod diameter of 26 mm by varying the gas-spring constant using V_{B0} as a parameter. Thus, $K_L=0$ differs from the present experimental condition for the vibration system without a gas-spring coupling. From the following equation, the coupling spring constant of the gas spring⁽²⁾ is 2.6 kN/m for $V_{B0}=2.40 \times 10^{-3} \text{ m}^3$.

$$K_L = \kappa P_0 A_{RH} A_{RC} / V_{B0} \quad (14)$$

Here W_L is the work which the coupling spring transmits from the cold displacer to the hot displacer. Coupling by mechanical spring was calculated for the case in which the gas-spring K_L is 0. It was revealed that both the gas-spring and mechanical-spring coupling have almost the same effect on FPVM

performance and on the displacer motion except for α . Mechanical-spring coupling adds the mechanical-spring K_L to K_H and K_C in Eqs. (11) and (12)⁽²⁾, and thus K_L acts similarly as the spring constant of both displacer vibration systems. Because both vibration systems have nearly the same mass, changes in ω_H and ω_C were almost the same, and α changed extremely little. With the gas-spring coupling, however, the cold displacer rod has a larger diameter consequently, if K_L is changed by changing V_{B0} , then K_C shows a greater ratio of change than K_H . This is why α increased gradually as K_L increased.

5. Conclusion

The operating characteristics in terms of displacer motion and FPVM performance were experimentally investigated by using a prototype, and the results were compared with our simulation analysis results. We have obtained the following conclusions:

(1) The dependence of the prototype FPVM's displacer motion and its performance on the working gas pressure, hot gas temperature, and hot/cold water temperature were clarified by experiment and simulation calculation.

(2) There is no qualitative difference between the characteristics exhibited when the hot and cold displacers are and are not coupled via a gas spring. The dependence of the displacer motion and performance on the working gas pressure is greater when there is a coupling. However, its dependence on the cold water temperature is greater when there is no coupling.

(3) The FPVM's cooling capacity and cooling COP are significantly larger when the hot and cold displacer vibration systems are coupled with a spring than they are when the vibration systems lack this coupling.

(4) The newly developed prototype achieved a favorable performance with a cooling capacity of 3.2 kW and a cooling COP of 0.6. When the hot displacer rod diameter was reduced and a motor drive was used, it provided a cooling capacity of 3.2 kW and cooling COP of 0.62 (motor input: 93 W).

(5) Compared with the experimental results, the simulation calculation, which combined linear analysis and second order isothermal simulation, evaluated the operating frequency at a higher level and the strokes and phase angle at lower levels. Qualitatively, however, the simulation results showed good agreement with the experimental results. This also applies to the performance parameters. Thus, it has been established that the simulation analysis is valid for the performance analysis of actual FPVM's.

(6) The effects of coupling by a gas spring alone

and a mechanical spring alone on FPVM performance were investigated by simulation calculation. The gas and mechanical springs have different effects on the spring constants of both hot and cold displacer vibration systems. They also show a difference in the dependence of the phase angle on the coupling spring constant.

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