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Dynamic behavior of valve system in linear compressor based on fluid-structure interaction †

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Abstract

In refrigerator designs, the linear compressor is preferable to the recipro-type compressor, due to its higher energy efficiency. The linear compressor's valve system, however, causes significant noise, not only in the steady state but also in the transient state. To accurately predict the behavior of the suction and discharge valve system in both states, the interaction between the fluid flowing through the valves and the structural deformation of the valves needs to be understood. In the present study, the steady-state behaviors of the valve system were numerically analyzed using ADINA software, which takes fluid-structure interaction (FSI) into account. This computational analysis thereafter was experimentally validated. The effects of a pre-load of the conical compression spring on the dynamic characteristics of the valve system also were analyzed.

Keywords: Fluid-structure interaction; Linear compressor; Valve system; Dynamic behavior

1. Introduction

The compressor is one of the essential components of a refrigerator, together with the evaporator, condenser and expansion valve. The compressor draws in low-pressure gas-phase refrigerant from the evaporator and discharges it in the high-pressure-liquid state to the condenser. The reciprocating compressor [1, 2], in which a crank-shaft mechanism converts the motor's rotary motion into reciprocating motion to draw in and discharge refrigerant, has been the design of choice. More recently however, the linear compressor, where a linear motor directly drives a piston in a back and forth motion to change the control volume and pressure, has come into wide use. The linear compressor offers higher energy efficiency than the reciprocating compressor, in that its direct piston motion minimizes the significant transformation loss incurred by the reciprocating compressor.

The linear compressor consists of a cylinder, a suction valve, a discharge valve and a discharge cover (muffler). When the control volume is expanded by the piston movement, the discharge valve is closed and the suction valve attached to the piston is opened widely. Then, the linear compressor draws in refrigerant, which passes through the evaporator. Subsequently, as the piston compresses the control volume, the suction valve is closed and the discharge valve is opened, upon which the linear compressor discharges refrigerant to the condenser. In the linear compressor, the impact between the discharge valve and a stopper generates significant noise (Fig. 1). To reduce this, the internal behavior of the linear compressor needs to be clarified. The suction valve is required to be very flexible in order to draw in an adequately large amount of refrigerant. This causes a strong mutual influence on both the suction valve and the fluid flow. Therefore fluid-structure interaction (FSI) analysis procedures [3-7] must be considered with respect to this valve system.

The objective of this paper was to identify the dynamic behavior of the vibrating valves and the fluid flow in the linear compressor by means of both experimental and computational analyses. For the purposes of the experiments, the linear compressor specimen was manufactured in such a way that pressure and temperature data could be obtained at various stages of its operation. In the computational analysis, using the experimental data as the input parameters, a three-dimensional FSI problem was solved using ADINA software.

2. Fully coupled fluid-structure interaction (FSI)

2.1 Description of structural model

A three-dimensional structure of the linear compressor was created using Uni-graphics, CAD software based on parasolid,

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Table 1. Material peroperties in R134a linear comperssor problem.

R134a	Structure
$C_p = 0.8787\ [kJ/kg \bullet K]$ $C_v = 0.7972\ [kJ/kg \bullet K]$ $\mu = 1.18\times10^{-5}\ [kg/ms]$ $\rho = 8.651\ [kg/m^3]$ $\beta = 0.004024\ [1/K]$ $\lambda = 0.01387 [W/m \bullet K]$	$E = 2.1\times10^{11}\ [N/m^2]$ $\nu = 0.29$ $\rho = 7700\ [kg/m^3]$ $K = 8900\ [N/m]$ $\sigma_y = 1.85\times10^{9} [N/m^2]$

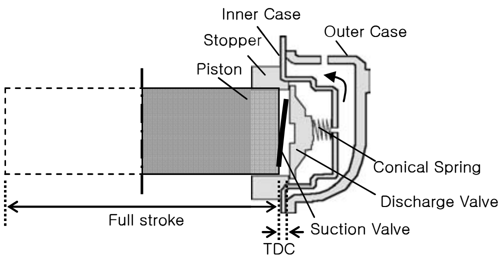


Fig. 1. Linear compressor system.

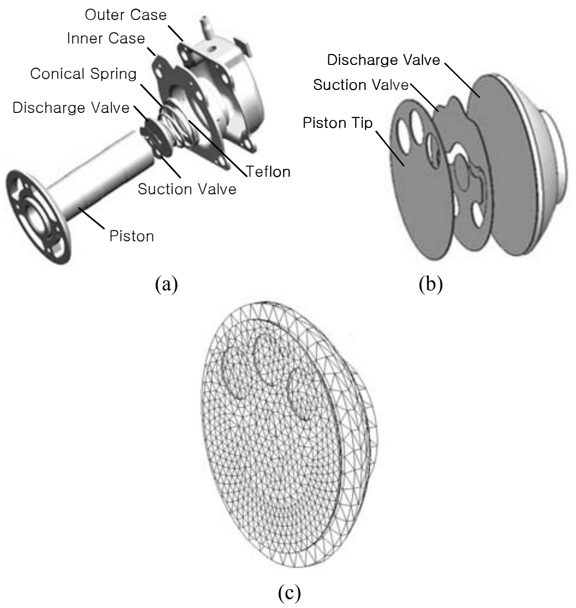


Fig. 2. Structural model: (a) Geometric entities; (b) Computational domain; (c) Mesh of FEM model.

(Fig. 2(a)) and directly loaded into ADINA-M (the ADINA system modeler). The geometry of the model was modified for analytical suitability. The model structure consisted of a piston, a suction valve attached to the piston, a discharge valve, a conical spring, a discharge cover (an inner case and an outer case) and a Teflon. The material of all of these parts was assumed to be steel (Table 1). The computational domain for the compressor movement included the functions of the piston, the suction valve and the discharge valve (Fig. 2(b)).

To minimize the size of the computational model, only the piston tip was modeled. As in a real model, the piston tip was calibrated for 60Hz displacement, which corresponds to a full stroke (Fig. 1). The conical compression spring, in order to adequately represent the stiffness effects, was modeled with a

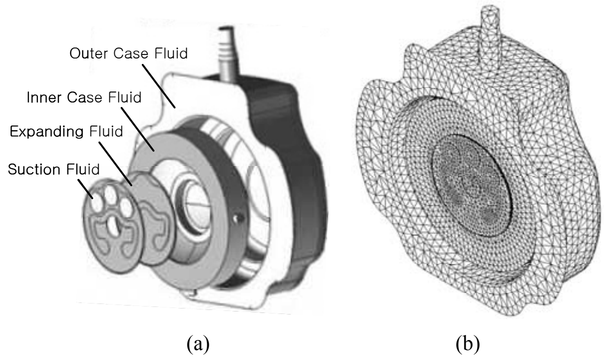


Fig. 3. Fluid model: (a) Computational domain; (b) Mesh of FEM model.

pipe element considering all of the degrees of freedom, as in real geometry. Correspondingly, the discharge valve attached to the end of the conical spring had no constraint on any of the degrees of freedom. The spring was subjected to an initial displacement by the stopper and the inner case. Accordingly, on the contact point of the discharge valve fixed to the spring, a pre-load was generated. A static force corresponding to the pre-load was exerted on the discharge valve in the direction of the restoring force. The top dead center (TDC) is the minimum distance between the piston and the discharge valve.

Tetra elements, created by the free-form meshing capabilities of the ADINA-preprocessor, were used to define the structural FEM model (Fig. 2(c)). So as to select an appropriate number of nodes per element and a proper mesh density, the natural frequencies were obtained with the ADINA and NASTRAN programs; on the basis of a comparison of the two results, an element with 10 nodes and a mesh density of 0.78 mm were selected. The total number of nodes and elements in the structural FEM model were about 13,000 and 10,000, respectively.

2.2 Description of fluid model

The computational domain of interest with regard to the fluid was refrigerant filled into the inner space of the piston tip through the discharge cover (Fig. 3(a)). This inner case fluid domain is connected to an outer case fluid domain via four small holes. R134a was used as the refrigerant (Table 1) [8]. Inlets for analysis were implemented in the form of three round-shaped perforation regions at the piston tip, as shown in Figs. 2 and 3. According to the fluid flow drawn in through those inlet regions, the suction flow rate was estimated. An outlet in the form of a middle hole at the top of the case also was defined, as illustrated in Fig. 3. The inlet- and outlet-flow boundary conditions were uniformly prescribed according to temperature and pressure data obtained from experiments conducted using Load-stand, an instrument controlling all of the performance conditions of a linear compressor, such as the stroke and the kind of refrigerant. The temperature and pressure data were estimated using ducts inserted between Load-stand and the inlet and outlet of the linear compressor. The

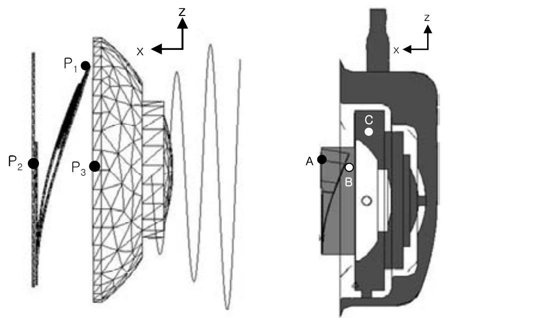
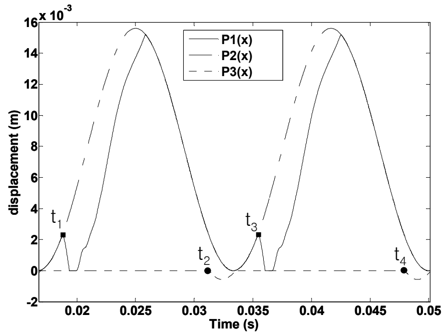


Fig. 4. (a) Displacement data acquisition points; (b) Pressure data acquisition points.

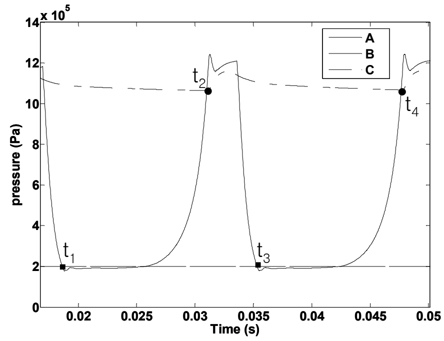
estimated boundary conditions were, at the inlet, 296K temperature and 98,000Pa pressure, and, at the outlet, 363K temperature and 961,000Pa pressure. The initial conditions, those preceding the numerical analysis, were zero velocities in all directions, a temperature of 296K and a pressure of 199,000Pa. These conditions were set empirically considering convergence. The fluid model was assumed to be laminar flow. The operating refrigerant was in the gas phase, and the fluid flow was modeled as Navier-Stokes low-speed compressible, which includes heat transfer. The suction valve was positioned between the suction fluid domain and the expanding fluid domain. The discharge valve was placed between the expanding fluid domain and the inner case fluid domain. As the piston moves back and forth, the expanding fluid domain is dilated or compressed. When the expanding fluid domain is dilated, its pressure falls to a lower level than that of the suction fluid domain. As the suction valve opens, the refrigerant flows into the compressor. In the opposite case, that is, when the pressure of the expanding fluid domain rises higher than that of the inner case fluid domain, the discharge valve opens, and the refrigerant is discharged to the outlet. Fluid-tetra elements with four nodes per element were used to generate a fluid FEM model (Fig. 3(b)). At the boundary of the FSI, the mesh density of the fluid was denser than the structural mesh. The total nodes and elements in the fluid FEM model were about 11,000 and 45,000, respectively.

2.3 Solution technique

In structural domains, the suction and discharge valves are in contact with the other structures [9]. The former touches either the piston or the discharge valve, and the latter meets the stopper. Three contact conditions were defined. In the case of the fluid domains, they are either connected or disconnected according to whether the valves are open or closed. Specifically, connection or disconnection of the two fluid domains can be defined by the condition of the gap [10], which is the interface space between a valve and the hole opened or closed by it. In the case of the suction valve, refrigerant can flow into the piston not only around it but also through an inner hole. These two passages should be treated separately. In the case of the discharge valve, because only one flow passage is created, one gap condition is sufficient. Therefore, in the present study, a total of three gap conditions were defined. Also, there were two pairs of FSI interface. One was defined by the faces of the suction valve and the piston tip with the adjacent fluid faces, and the other consisted of the faces of the discharge valve with the surrounding fluid faces. The other fluid faces were defined as rigid walls without slip. In order to solve both a structural and a fluid problem, sparse solver and Newton method of iteration scheme were used. And iterative solution procedure was employed for analysis of fluid flow with structural interactions.



(a) Valve displacement versus time (P_1: S. valve end, P_2: Piston, P_3: D. valve)



(b) Pressure versus time (A : suction hole, B : cylinder, C : inner case)

Fig. 5. Computational results for structural displacement and fluid pressure under 32N pre-load.

3. Results of analysis

The temporal behavior of valve displacement under the preload of 32N was obtained from an FSI model (Figs. 4(a) and 5(a)). P_1 is the end point of the suction valve. P_2 and P_3 are arbitrary points on the piston tip and discharge valve, respectively. Because the suction valve is attached to the piston tip, both the suction valve and the piston tip move together when the suction valve is closed. When the suction valve is open, relative displacement is generated. Owing to the valve's flexibility, eflection is created in the z-direction as well as in the x-direction. The fluid pressure distribution under the applied 32N pre-load was also calculated (Figs. 4(b) and 5(b)). The pressure at point A is inlet pressure, which is constant. Points B and C are arbitrary points of the fluid domain in the cylinder

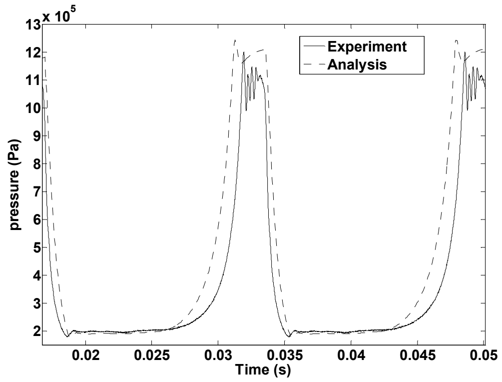


Fig. 6. Comparison of experiment and computational analysis results for pressure at point B under 32N pre-load.

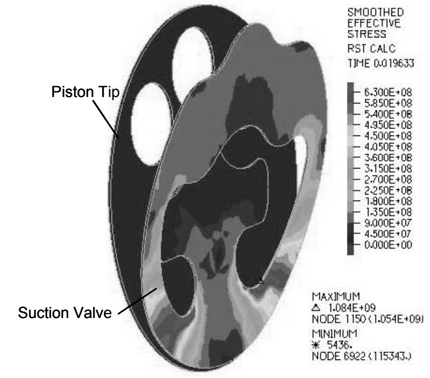
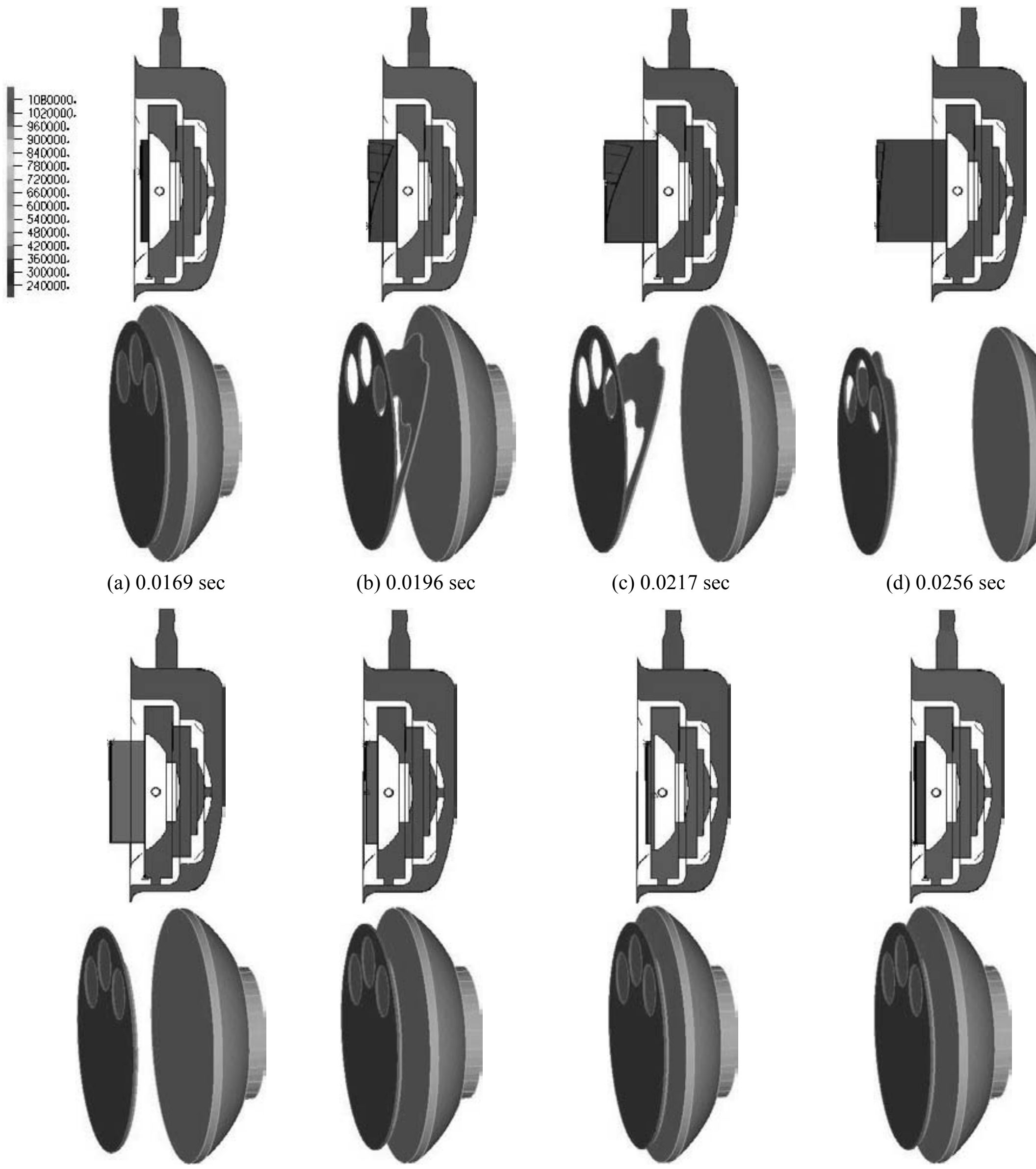


Fig. 7. Stress distribution on suction valve under 32N pre-load.



(a) 0.0169 sec (b) 0.0196 sec (c) 0.0217 sec (d) 0.0256 sec

(e) 0.0278 sec (f) 0.0304 sec (g) 0.0324 sec (h) 0.0347 sec

Fig. 8. Pressure distribution and valve deformation at various times under 32N pre-load.

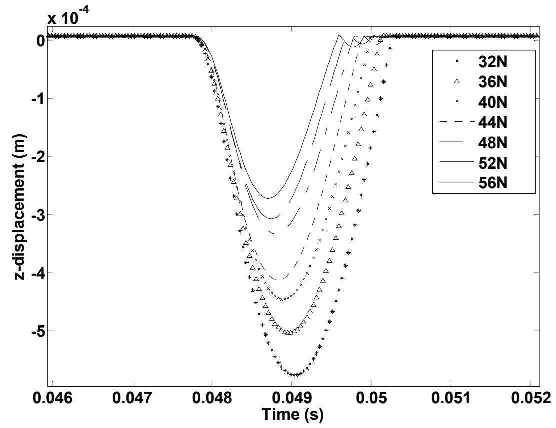


Fig. 9. Displacement of discharge valve versus time.

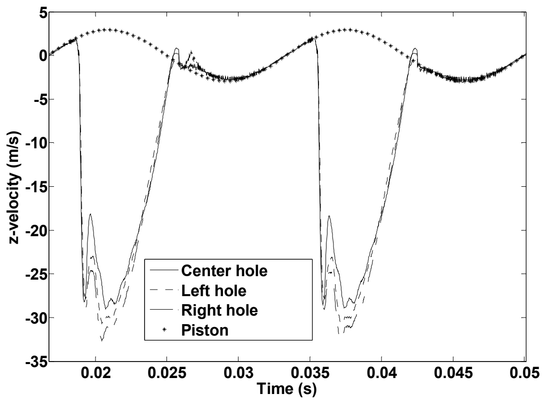


Fig. 10. Fluid suction and piston velocity versus time.

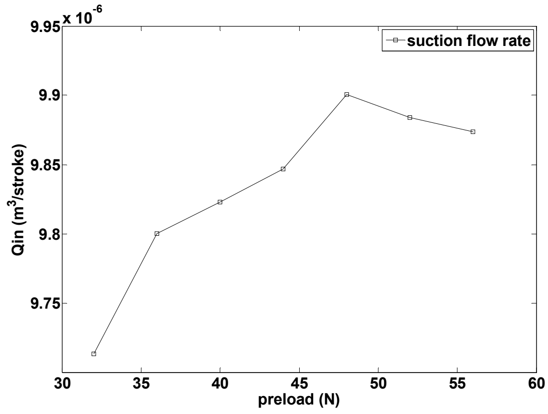


Fig. 11. Suction flow rate versus pre-load.

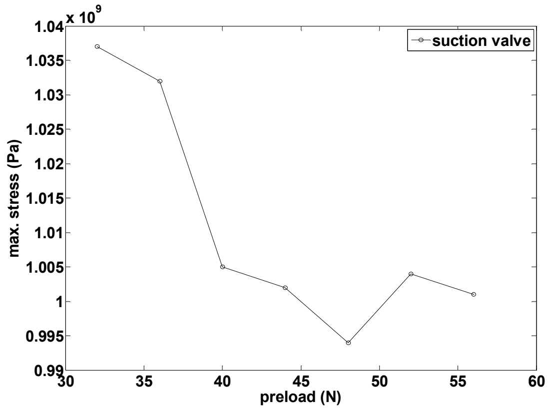


Fig. 12. Maximum stress on suction valve versus pre-load.

and in the inner case outside of the discharge valve, respectively. When the pressure at point A is higher than t point B, the suction valve should be open at times t_1 and t_3 (see Fig. 5)). Also, when the pressure at point B is greater than at point C plus the pre-load of the conical spring, the discharge valve has to be open at times t_2 and t_4 (see Figs. 5 and 8(g)). Figs. 5 and 8 show the relations between the displacements of the valves and the pressure distribution of the internal fluid. These relations, physically understandable results in themselves, verified the analysis results. The calculated solutions and experimental results for the pressure in the expanding fluid domain, under the 32N pre-load, were compared (Fig. 6). The calculated solutions were quite similar to the experimental results over the entire time range. However, the pressure fluctuations of the experimental results with the large frequency variations could not be calculated. The source of the difficulty might be faulty assumption of the laminar flow and static outlet pressure conditions. To determine the source of the pressure fluctuation, the likely external factors have to be scrutinized.

Notably, the computational analysis showed some interesting dynamic characteristics for the linear compressor. The suction valve came into contact with the discharge valve at the full stroke (Figs. 5(a) and 8(b)). During the collision and contact between valves, around 0.019 seconds and 0.036 seconds in Fig. 5(a), the displacement in x-direction of P_1, the end point of the suction valve, was not changed. And in fact, after operating the linear compressor in the experiments, it was found that the face of the discharge valve was scratched. These experimental results validated the computational analysis. In the structural model, when the 32N pre-load was applied, the suction valve underwent the maximum stress, about 1.084GPa (Fig. 7), and its safety factor was calculated at 1.71. The minimum stroke of the piston for discharge of the refrigerant to the outflow was found to be the full stroke. When the amplitude of the piston reached 85.12% of a full stroke (i.e. 13.27mm), the pressure of the expanding fluid domain was higher than the sum of the pressure of the inner case fluid domain and the pre-load of the conical spring. As a consequence, the discharge valve opened (see times t_2 and t_4 in Fig. 5(a)).

Finally, some of the pre-load effects related to the valve behavior and efficiency were investigated. Basically, when the pre-load was removed in the full stroke, the closing speed of the discharge valve after opening was slower, which caused a delay in the disconnection between the expanding fluid domain and the inner case fluid domain. Consequently, the pressure of the expanding fluid domain dropped a little more than that of the suction fluid domain. The suction valve could open only very slightly. By contrast, when the discharge valve was subjected to the pre-load, the suction valve could open considerably. The relations between the pre-load of the conical spring and displacement of the discharge valve were uncovered. Fig. 9 shows the z-direction displacement of the discharge valve corresponding to the increase of the pre-load from 32N to 56N. The maximum displacement of the discharge valve was in inverse proportion to the pre-load of the conical spring. And the timing of the opening of the discharge valve was almost the same, because the pressure of the expanding fluid domain rose sharply within a short time immediately prior to that opening, as shown in Fig. 5(b). However, the closing speed of the discharge valve was faster as the pre-load increased and multiple impacts occurred between the discharge valve and the stopper for pre-loads greater than 44N. The fluid velocity at the three inlet holes was calculated over the period of time under the 32N pre-load condition (Fig. 10). It was observed that the fluid velocity at the center hole was relatively smaller than at the other ones. When the suction valve was open as the piston expanded, each suction velocity of the three inlet holes was in the opposite direction of the piston velocity, and refrigerant could be drawn in. Therefore, the total suction flow rate could be calculated by multiplying the hole areas by the relative velocities at the three holes and the piston (Fig. 11). It could be seen that there was an optimal pre-load of the conical spring maximizing the suction flow rate. The suction flow rate under the 48N pre-load condition increased to 1.89% of that under the 32N pre-load condition. However, when the pre-load was larger than 48N, the flow rate decreased, owing to the suction valve's short "open" duration. The maximum stress on the suction valve also was analyzed according to the variation of the pre-load of the conical spring (Fig. 12). The maximum stress showed an opposite tendency to that of the flow rate. Because the suction valve contacts the discharge valve, the shorter the time that the suction valve is open, the smaller the deformation of the suction valve is. When the pre-load of the conical spring was 48N, the maximum stress on the suction valve decreased 4.15% lower than that under the 32N pre-load condition. Accordingly, 48N was deemed to be the optimum pre-load for suction efficiency and stress.

4. Conclusions

The computational analysis of the dynamic behavior of the valve system in the linear compressor took the FSIs into consideration. The numerical analysis findings showed good agreement with the experimental results, validating the accuracy of the computational analysis. The following dynamic characteristics of the valve system were predicted from the numerical simulation for the full stroke and 32N conical spring pre-load conditions. First, the suction valve contacted the discharge valve. Second, the discharge valve opened when the piston compressed about 85% of the full expanding volume. Third, the maximum stress on the suction valve was about 1.084GPa, and the safety factor was 1.71.

Finally, the effects of the increase of the pre-load from 32N to 56N were investigated as core factors impacting on efficiency and the valve behaviors in the linear compressor. The timing of the opening of the discharge valve showed no relation with the pre-load. As the pre-load was increased, the discharge valve closed faster. When the pre-load was greater than 44N, multiple impacts occurred between the discharge valve and the stopper. The suction flow rate under the 48N pre-load condition was increased to 1.89% of that under the 32N pre-load condition. Moreover, the maximum stress on the suction valve under the 48N pre-load condition was decreased 4.15% lower than that under the 32N pre-load condition. The 48N value, then, was deemed to be the optimum pre-load of the conical compression spring for suction efficiency and stress.

In future research, additional details of the various factors effecting the dynamic behaviors of the valve system should be determined and considered in order to move closer to the optimum linear compressor design.

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