Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

2006

CAE/CFD Application for Linear Compressor

Kwangha Suh
LG Electronics Inc.

Daenyoung Heo
LG Electronics Inc.

Hyeongseok Kim
LG Electronics Inc.

CAE/CFD Application for Linear Compressor

Kwang-ha Suh*, Dae Nyoung Heo, Hyeong Seok Kim

Digital Appliance Research Lab, LG Electronics Inc.,
327-23 Gasan-dong, Keumchun-gu, Seoul, 153-802, Korea
Tel: 82-2-818-3020, Fax: 82-2-867-3941
E-mail: phlip@lge.com

ABSTRACT

Two parts of CAE (Computer Aided Engineering) and one part of CFD (Computational Fluid Dynamics) techniques are introduced in this paper. The first part is on the hermetic compressor shell's modal analysis using stamping data, where the stamping analysis is mainly focused. The second part is noise radiation analysis using BEM. Since the weak point of BEM (Boundary Element Method) is taking too much time, several methods for radiation analysis are suggested and compared. The third part is on CFD with FSI (fluid structure interaction) technology. The object for the application of FSI is the suction valve system of linear compressor. The suction valve of linear compressor is attached on the top of moving piston, and opens when piston moves down. The motion of suction valve is very hard to simulate numerically, because of its characteristic of fluid structure interaction. It is bent by pressure difference and inertia force, and impacted with piston and discharge valve. FSI simulation program is developed in this study. Several designs of linear compressors are simulated and the results are compared with the experimental data.

1. INTRODUCTION

Since the rapid speed-up of computing power, the computer aided technology is getting widely used. The CAE is one of the catalyst to make the fast delivery from the product launching. As the demands of output lead time getting faster, the accuracy of CAE should be upgraded in order to minimize the trials & errors.
The examples shown in this paper are related the accuracy for better products and fast delivery time. The first one is the stamping analysis which is widely used in automobile companies and traditional press design. The original purpose of the stamping analysis is to design the punch and die or the stamping process. However, some of data from stamping analysis are applicable to other calculations. One of the examples is the modal analysis using the thickness data from stamping analysis. Since the modal frequencies and the mode shapes are very important information for noise and vibration engineers, the early and accurate information on hermetic compressor shell's modal data is the key of the time saving. The second example of sound radiation analysis also provides useful data in two points of views. The first trial is the two-step BEM approach to calculate the radiated noise for design. Even if the modern computing power is getting faster, the BEM still requires much calculation time, and this is the week point of BEM. The second trial is for the estimation of shell's participation factors on the noise in trouble using the existing shell, where the vibration data should be provided. It sometimes happens to redesign the compressor shell with no other ways to solve the noise problem. However, it needs to estimate using the second approach before redesign.
The valves are one of the most important parts in the development of compressors. It directly affects the efficiency and reliability problems, and also the original source of noise and vibration problems. It is essentially required to understand the physical phenomena of valve and design optimally. Therefore, the suction valve motion and flow around valve are simulated with FSI program developed in this research. Several designs of linear compressors are simulated and the results are compared with the experimental data.

2. Stamping Analysis

2.1 Overall Process for Modal Analysis using Stamping Data

The overall order of the process for modal analysis using the stamping data is stamping analysis, spring-back process analysis, mapping the stamping data to new model, and modal analysis. From the several trials and errors, it is found that the spring-back analysis is not necessary. Therefore, the spring-back analysis is not discussed in this paper.

2.2 Stamping Analysis

The hermetic compressor shells are deep-drawing products, and relatively longer time or more number of impacts than other types of stampings. The overall view of the stamping process for lower shell is shown in Fig.1.

Since the material data is important for the calculation, the full stress-strain curve should be obtained from the material specimen test, and the full stress-strain curve is used as interpolated data. The Fig.2 shows the thickness distribution from the stamping analysis. The trend is well-matched to measured data as in Fig2.

Since the noise of hermetic compressors is very sensitive to the shell's modal data, the accurate prediction of the modal data is very valuable information to noise and vibration engineers. The practical solutions of the noise problem in hermetic compressors are avoiding the resonance between the moving or connecting parts and the shell. The accuracy of the modal analysis is shown in Table 1. The %error means the accuracy of modal frequencies, where the value calculated as the simulated results minus the experimental results divided by the experimental results.

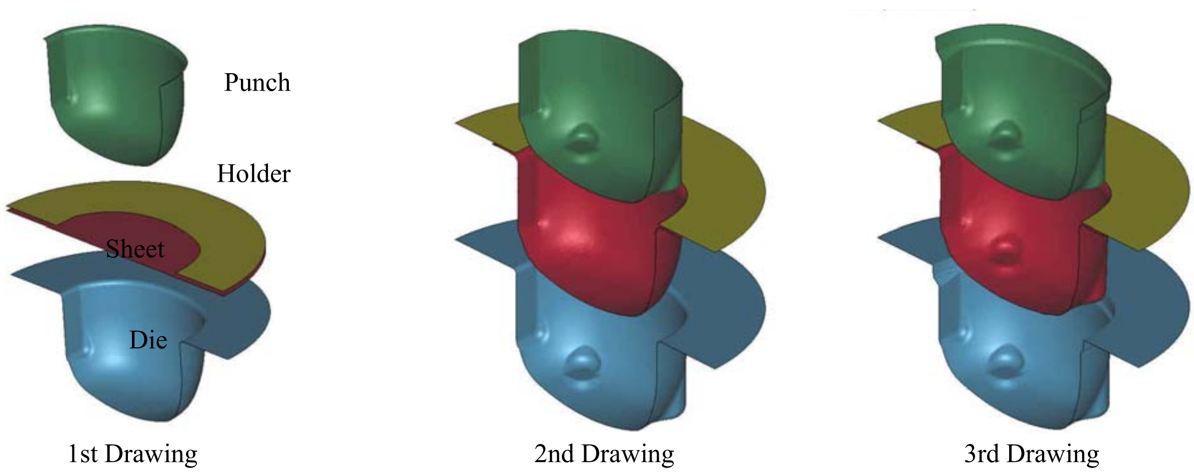


Fig.1 Overall Stamping Process

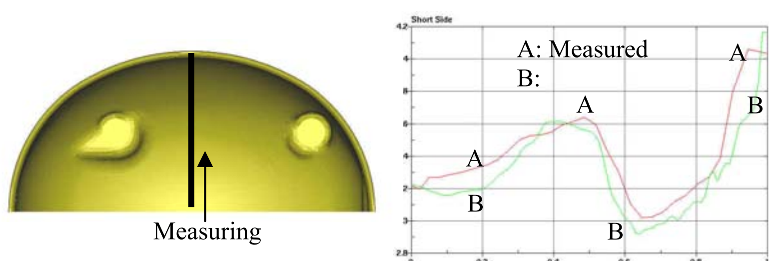


Fig. 2 Final Thickness

Mode #	%error T=const	%error T≠const
1st	9.9596	−1.4153
2nd	12.0355	1.3150
3rd	11.9709	3.9850
4th	13.6896	2.1906
5th	11.6327	−0.6032
6th	3.7507	6.4085

Table 1 The accuracy of modal

3. Radiation Analysis

3.1 The Procedure

The procedure of radiation analysis is complicated and is taking much calculation time. Since the BEM needs much calculation time, the time saving method is suggested as in Fig. 4. The usual way of calculation is as in the Fig.3 because the engineers want to describe the details of geometry to increase the accuracy of modal analysis. However, it takes too much time to get the results in the limited time of development. The Fig. 4 explains that the BEM mesh does not have to get the very fine mesh, but it needs as fine as to describe the wave length of the frequencies that we are concerning. There-fore, the BEM mesh can be much coarser than FEM mesh and the geometry details can be simplified as in Fig. 4. The overall procedure in Fig.3 and 4 is calculating modal analysis, importing the modal data to BEM model, calculating the BEM coupled interior, importing the surface data to BEM uncoupled exterior, calculating BEM uncoupled exterior, and finishing the field point process. The difference between Fig.3 and Fig.4 is whether or not the FEM mesh and BEM mesh are the same. For the Fig. 4 which is using the different mesh, the FEM data is interpolated to the BEM mesh. On the other hand, the usage of Fig. 5 is somewhat different. The Fig.4 is for the design of new shell in the development process, but the procedure in Fig.5 is for the estimation on noise and the total calculation time is shorter. The procedure is getting experimental data , calculating modes, calculating BEM uncoupled exterior, and doing the field point process.

3.2 Two Step BEM Approach

The acoustic source should be defined for the calculation of BEM coupled interior. In order not to make the problem complicated, the known source is used for the calculation and the experiment. In the SYSNOISE, the cylindrical source is used due to the characteristics of the inserted small speaker. The Fig. 6 shows the test and simulation results. Since the trends of tested and simulated noise patterns are very similar, it can be good enough to use a design tool. The directional notation is shown in the Fig. 4 and 5, and the measured point is 30cm from the shell surface.

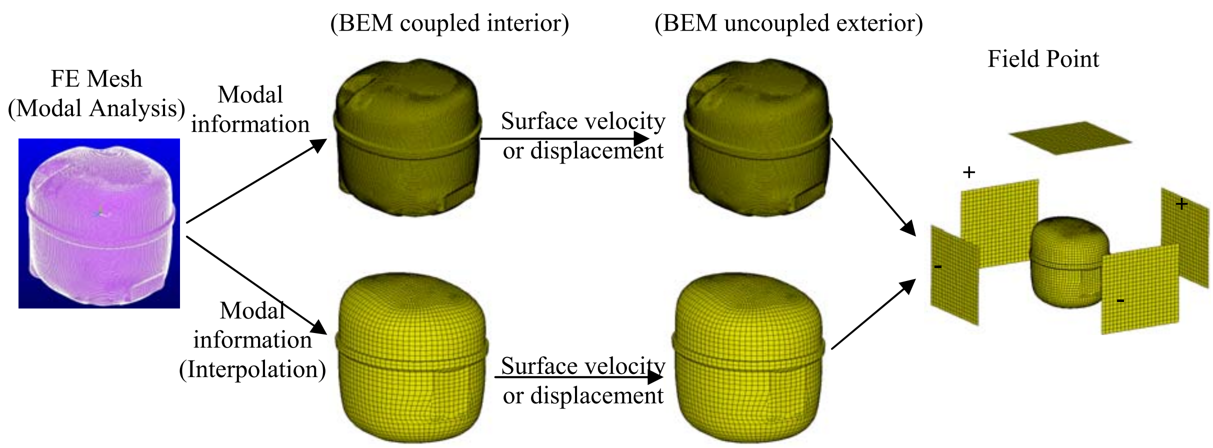


Fig. 3 The normal procedure (above) (FE mesh = BE mesh) / Fig. 4 Interpolated Data (FE mesh ≠ BE mesh)

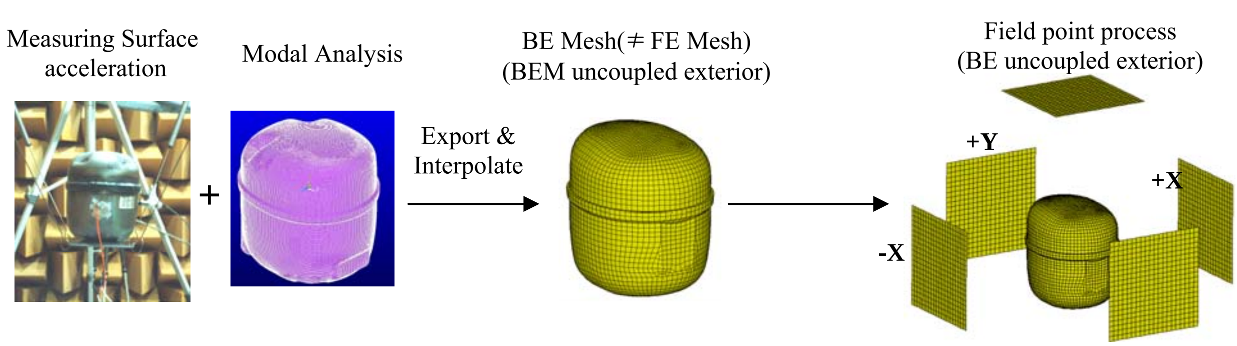


Fig. 5 The procedure to calculate the radiated noise using experimental data (FE mesh ≠ BE mesh)

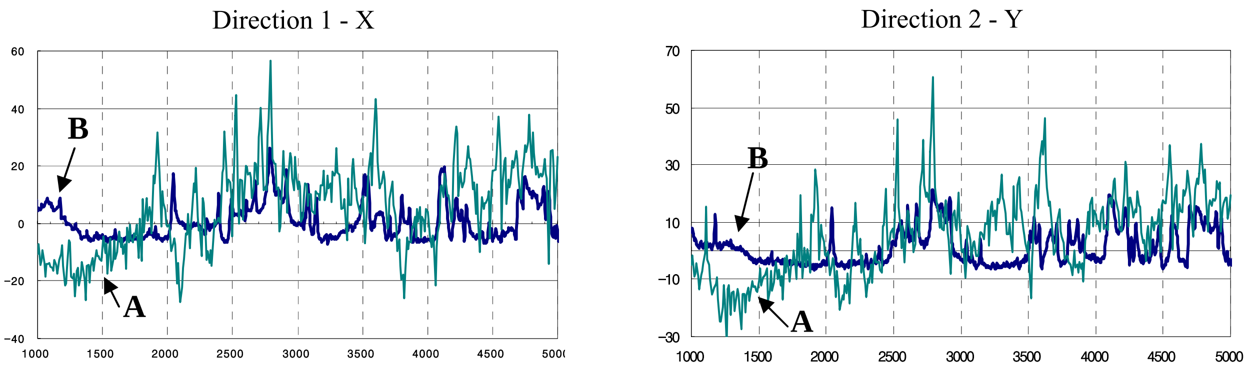


Fig6. Two Stop approach test and simulation results (A : simulation, B : test)

3.3 The Hybrid Approach : Modal Expansion Technique

If the surface information is measured, the acoustic normal velocity can be calculated. This modal expansion technique can be described as simple linear combination as follows:

$$u_{ij} = \sum_{k=1}^{m} a_k \, \phi_{ijk} \tag{1}$$

In most of cases, the number of m unknowns is more than the number of measured known, called underdetermined. Using the modal analysis results and measured data, the others data can be extrapolated from Eq.(1) using singular value decomposition. Once the modal participation factors, a_k , are known, the normal velocity to calculate the radiated noise is derived from the below Eq(2):

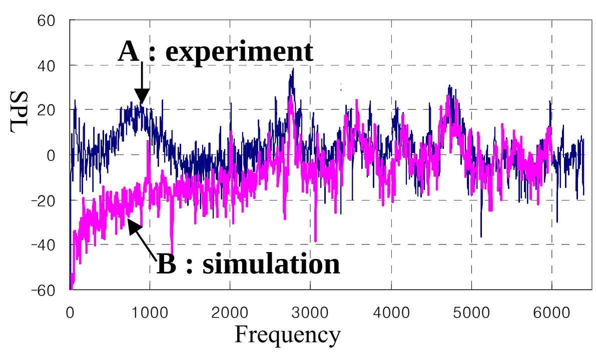


Fig. 7 X direction noise by acoustic excitation (In Air)

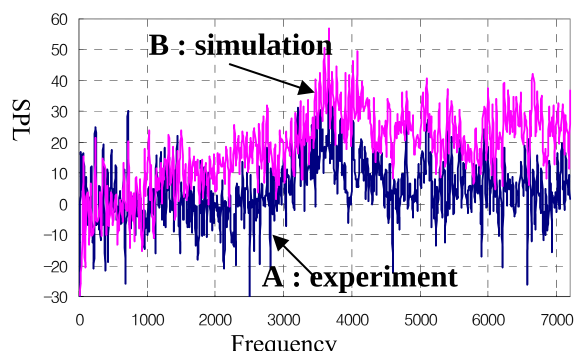


Fig. 8 X direction noise in real operating(R600a)

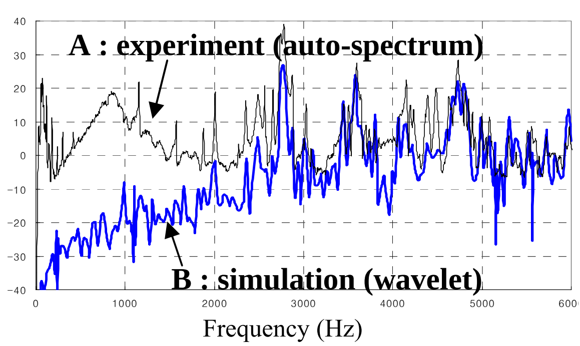


Fig. 9 Wavelet data by acoustic excitation

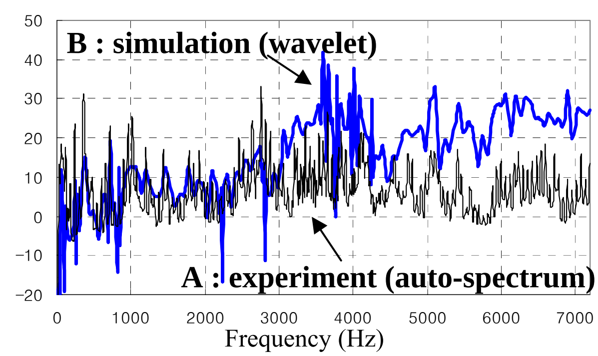


Fig. 10 Wavelet data and averaged experiment data (R600a Real Operating)

$$v_{n,i} = \sum_{k=1}^{m} (i\omega a_k) \cdot \phi_{n,ik} \tag{2}$$

This method uses the strong points of FEA and experiment, but needs the real products for the experiment. Therefore, this method can not be applicable in the design phase. When the vibration data is measured, the data should be instantaneous data because there is no way of handling the averaged data in the SYSNOISE. The Fig.7 shows the comparison of simulation results and experiment results in simple case with acoustic excitation as done in the two step BEM. The differences in the low frequency range from 0 to 2kHz is due to the acoustic cavity resonances. The Fig.8 shows the results in real operating conditions. Since the acoustic medium is R600a, the cavity modes are shift down and hard to find in the figure. However, the mechanical part causes the forced vibration induced noise making the differences in the high frequency ranges. Since the experimental data for SYSNOISE and the sound pressure level data are instantaneous ones, the data seems to be a little scattered. Since it always used the averaged data in real noise measurement, the simulation data is modified using wavelet function as in Fig.9 and Fig.10.

4. FSI Analysis for Suction Valve

4.1 Numerical Methods

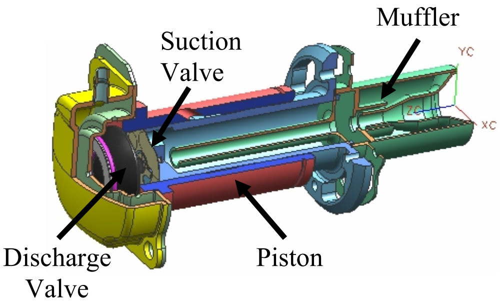


Fig. 11 Suction and discharge Part

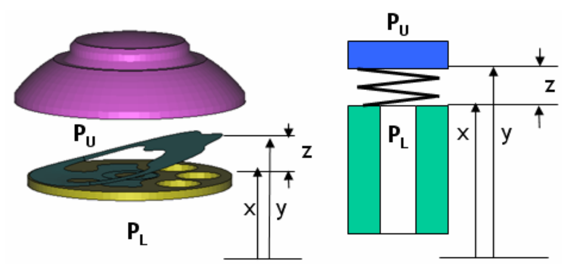


Fig 12 Schematic diagram of suction valve and defined variables

Fig. 11 shows the inside of linear compressor. The suction valve is attached on the top of moving piston, and opens when piston moves down. In one cycle, it impacts several times with piston or discharge valve when it opens or closes. Simplified valve motion, which is frequently used for simple program predicting compressor efficiency, could be achieved from *m-k* system shown in Fig. 12. The governing equation is as following.

$$m\ddot{z} + c\dot{z} + kz = \Delta P A_p - f_s - m\ddot{x} \tag{3}$$

z means opening displacement of valve, and x is the displacement of piston. m, c, and k each stands for effective mass, damping coefficient, and elastic coefficient. ΔP is the difference between pressures acting on topside and backside of valve, and A_p is the effective area which ΔP is acting on. F_s is stiction and initially installed force.

This equation shows that the valve motion is strongly coupled with flow around it. Additionally, suction valve frequently impacts with piston and discharge valve while its operation. Because of these characteristics, simplified *m-k* modeling produces huge errors and causes inaccuracy. It is the most difficult problem for developing the predicting program of compressor efficiency. Thus, FSI program, which is able to simulate strongly coupled phenomena with flow and structure, is essentially required to predict valve motion compressor efficiency.

Although FSI programs have been developed by many software companies, most of commercial programs can not solve serious structure deforming problem or impact problem. Thus, FSI program, which is able to simulate strongly coupled phenomena of fluid and structure, is developed in this research. This program can be applied to all FSI problems including serious structure deforming problem or impact problem. k-ε turbulent model and unsteady implicit solver are used for fluid, and unsteady explicit solver is used for structure.

4.2 Simulation Result

Four cases shown in Table 2 are simulated. The resonance frequency of muffler A is 120Hz, and its port diameter is

Φ5.4. Those of muffler B are 120Hz, Φ4.6, and muffler C are 90Hz, Φ4.6. The stiffness of suction valve a, b, c is 91, 143, and 157.

Table 2 Simulation Cases

Case	Muffler	S/Valve
1	A	a
2	B	c
3	A	b
4	C	c

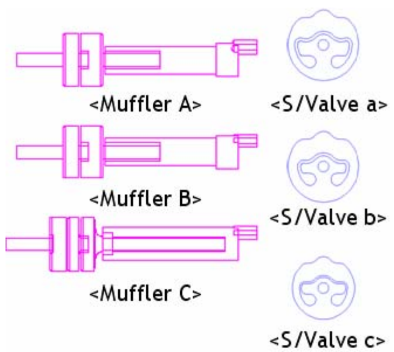


Fig. 13 Muffler and suction valve geometry

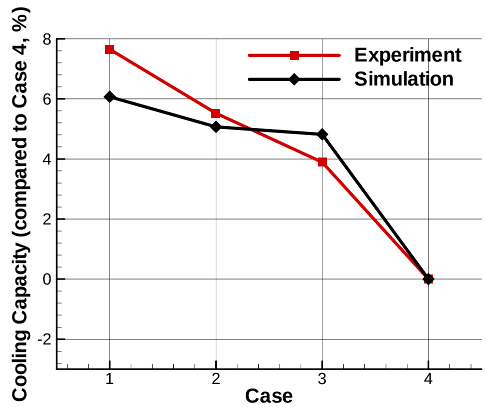


Fig. 14 Comparison of experiment and simulation result

In order to compare the accuracy of numerical simulation, the cooling capacities which are non-dimensionalized by case 4 are shown on Fig. 14. There are 1 or 2 percent errors between experiment and simulation because of some assumptions used for numerical simulation, such as initial condition, no leakage assumption, adiabatic condition, etc. This figure shows that the simulation result follows same trend as the experiment, and it has enough accuracy to identify physical phenomena inside of compressor.

Fig. 15 shows the motion of suction valve. Time scale is non-dimensionalized by one cycle (T) which is 1/60 second. The motion cycle of case 1 is longer than the others because its stiffness and resonance frequency are the smallest. Case 2 and case 4 have same suction valve, and their motions are nearly coincidence. This means that the valve motion dominantly depends on valve characteristics, and is scarcely affected by muffler. The valve of case 1 mainly opens and closes 2 times, and the other cases 3 times. At the first close, valve does not impact with piston and reopen because of the suction flow effect. But it impacts with piston and rebounds at the other closes. This is the same result as the experiment. The most important parameters affecting valve reliability are valve opening displacement and impact velocity. These parameters are experimentally achieved by gap sensor while compressor is operating. This experiment is not easy and its result shows poor accuracy. Thus this simulation program can be used effectively for reliability test.

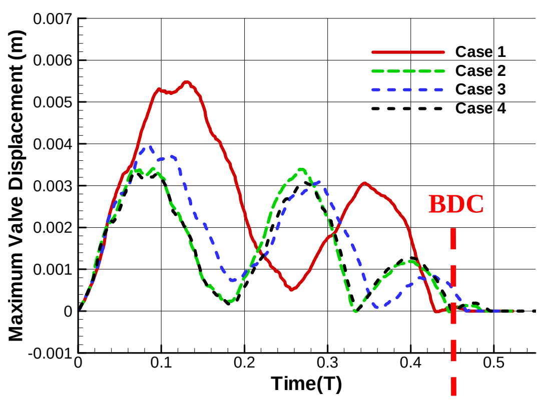


Fig. 15 Valve motion of each case

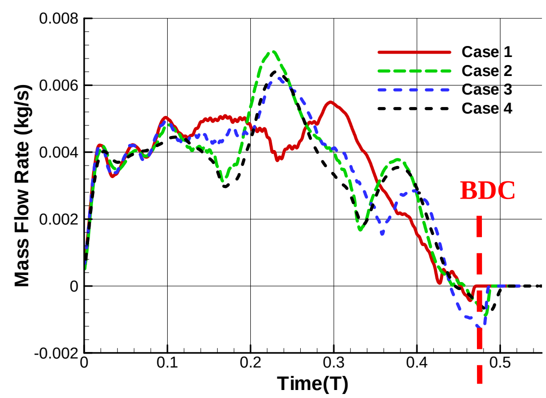
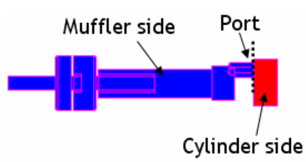
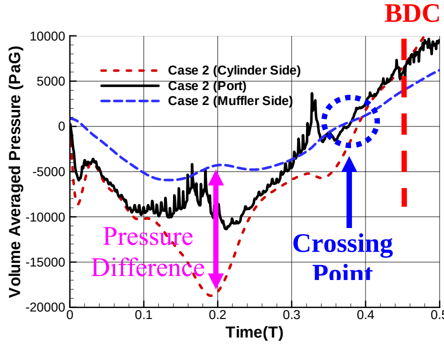


Fig. 16 Mass flow rate of each case

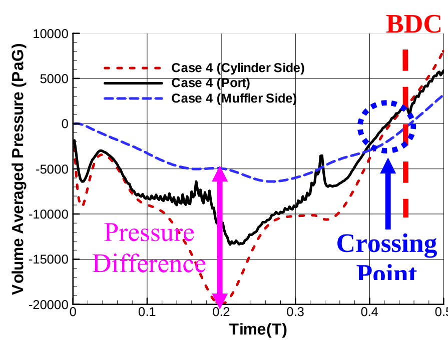
Fig. 16 shows mass flow rate according to time. Although high opening displacement increases mass flow rate, mass flow rate is similar when valve opening displacement is larger than 2, because the effective valve height corresponding to port area is 1.15~1.35. And Fig. 15 shows valve rebounds after impact with piston and backflow occurs if valve stays open near BDC. This back flow decreases cooling capacity and also acts as an important loss to efficiency.



(a) Schematic diagram

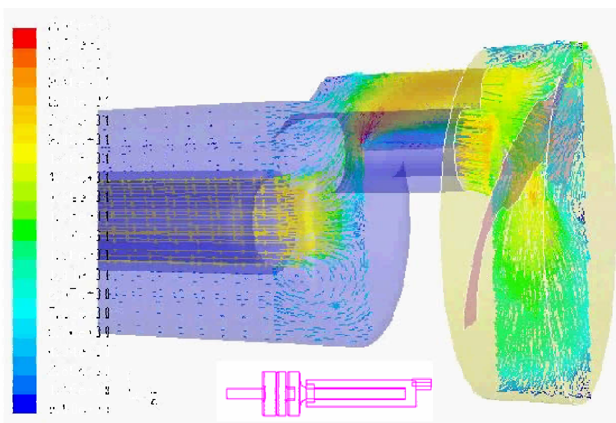


(b) Case 2 (Muffler B, 120Hz)

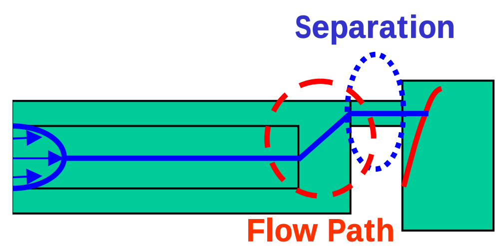


(c) Case 4 (Muffler C, 90Hz)

Fig. 17 Comparison of pressure difference at port



(a) Case 4 (Muffler C, 90Hz)



(b) Loss occur because of poor flow path

Fig. 18 Flow visualization near port

Fig. 17 shows the difference between the pressure of front and behind of port. Case 2 and case 4, which have same suction valve c, are compared in order to identify the effect of muffler. The muffler resonance frequency of case 2 is 120Hz, and that of case 4 is 90Hz. The most important difference is acoustic supercharging effect. Black line, the pressure at port, shows about 7000PaG in case 2 and 2000PaG in case 4 at BDC point. Thus case 2 has better efficiency than case 3, because higher pressure at BDC induces higher density and higher density increases higher cooling capacity. And crossing point, which is the point that the pressure of cylinder side starts to be higher than suction pressure, appears faster in case 2. This is the same result as the theory that acoustic supercharging effect is the highest when the resonance frequency of muffler is double of operating frequency, 60 Hz. The pressure difference of case 4 is higher than that of case 2, and this means that the flow loss of case 4 is higher.

Fig. 18(a) shows the velocity vectors of case 4. In case 4, the long inner pipe of muffler prevents smooth flow. Fig. 18(b) shows that the sudden change of flow path at the end of inner pipe and the flow separation inside of port induces flow loss. Inclined port can be adopted in order to decrease this loss. This is verified by simulations and experiments.

5. CONCLUSIONS

The modal analysis using thickness data from stamping analysis shows higher accuracy than the modal analysis of uniform thickness. Especially, the first and second modal frequencies, which are critical to the noise, are within

1.5% error. The two step BEM approach is reviewed for the usage in design phase and shows the characteristics in overall frequency. The hybrid approach is applicable to estimate the shell's participation factors to the noise. The modal expansion technique shows better accuracy than two-step BEM except the low frequency ranges, 0 to 2.0kHz, due to acoustic cavity resonance. The detailed treatment for acoustics is listed in reference. The FSI program, which is able to simulate strongly coupled phenomena of fluid and structure, is developed in this research in order to simulate the suction valve of linear compressor. The simulation results shows good agreement with experiment and this validates the accuracy of program. Some physical phenomena of compressor are analyzed from simulation result. It is expected that this program could be the alternative tool of difficult experiment and popularly used for compressor development.

REFERENCES

Kwang-Ha Suh, Hyuk Lee, Won-Sik Oh, 1998, An Analysis of the Hermetic Reciprocating Compressor Acoustic System*, Int Comp. Eng. Conf. at Purdue*, pp361-368

SYSNOISE *Rev 5.5* User's Manual

K-H Suh, B-C Lee, H. Lee 1998, Hermetic Reciprocating Compressor Cavity Resonance, *Proceedings of Internoise 98*, Christchurch, 1998.

H. K. Lee, G. Y. Song, J. S. Park, E. P. Hong, W. H. Jung, K. B. Park, 2000, Development of the Linear Compressor, *Proceedings International Compressor Conference*, Purdue University, West Lafayette, Indiana, U.S.A.

G. Y. Song, H. K. lee, J. Y. Yoo, J. K. Park, Y. H. Sung, 2000, Development of the Linear Compressor for a Household refrigerator, *Appliance Manufacturer Conference & Expo*, Cincinnati, U.S.A.

H. K. Lee, J. T. Heo, G. Y. Song, K. B. Park, S. Y. Hyeon, Y. H. Jeon, 2001, Loss Analysis of Linear Compressor, *International Conference on Compressors*, I Mech E, City University, London, UK

K. B. Park, E. P. Hong, H. K. Lee, 2002, Development of a Linear Motor, *Proceedings International Compressor Conference*, Purdue University, West Lafayette, Indiana, U.S.A.

K. B. Heo, H. K. Lee, C. W. Lee, M. K. Hwang, J. Y. Yoo, Y. H. Jeon, 2003, Control of Linear Compressor, 2003, *International Conference on Compressors*, I Mech E, City University, London, UK

Werner Soedel, 1984, Design and Mechanics of Compressor Valves, Ray W. Herrick Laboratories, School of Mechanical Engineering, Purdue University, West Lafayette, Indiana, U.S.A.

Werner Soedel, 1992, Mechanics, Simulation and Design of Compressor Valves, Gas Passages and Pulsation Mufflers, Ray W. Herrick Laboratories, School of Mechanical Engineering, Purdue University, IN, U.S.A.

Werner Soedel, 1972, Introduction to Compressor Simulation of Positive Displacement Type Compressors, Ray W. Herrick Laboratories, School of Mechanical Engineering, Purdue University, IN, U.S.A.