Hybrid coupling method to nonlinear acoustic source and linear duct system in compressor

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ABSTRACT

In recent years, when consumers buy home appliances and cars, energy efficiency has been considered to be significant. Therefore, studies on how to increase the energy efficiency are being accelerated. A representative study is to raise the energy efficiency of the compressor which is a core part of the refrigerator using acoustic theory in suction part. However, the acoustic source characteristics in suction part of the compressor are the nonlinear and time-varying. While the acoustic propagation characteristics of suction muffler are the linear and time-invariant. Thus, couplings between the source represented in the time domain and wave propagation represented in the frequency domain are not easy. In this study, we showed how to effectively coupling. That is, in the frequency domain, the input impedance of muffler was measured. Then, the impedance changes to state space form using modal method. And a state space model is coupled to the acoustic source model in the time domain. This approach was verified through the experiment with speaker model

Keywords: Hybrid coupling, Input impedance, System identification
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1. INTRODUCTION

Energy efficiency in compressor is related to the amount of refrigerant supercharged inside of compression chamber and the work done by system. When the compressor is doing the same amount of work, the greater the amount of refrigerant to supercharge into the compression chamber is good energy efficiency. If suction muffler is well designed, energy efficiency could be improved due to the acoustic effects without an additional power source . This is called acoustic supercharging [1-5]. However, if this is not well designed, then energy efficiency can decrease.

Unfortunately, in compressor, acoustical source has nonlinear and time-varying behavior. These behaviors had already been validated experimentally [6,7]. Therefore, it is not easy to analyze the acoustic in a frequency domain only. In other words, the hybrid coupling method is needed to consider acoustic source which is represented by the time domain and propagation represented by the frequency domain [8-9].

In this paper, hybrid coupling method is introduced in detail.

2. MAIN

The refrierant flow through the suction valve is acoustical source. This phenomena is very similar to the vibration system excited by base movement. As shown in Fig.1, the velocity of base motion in vibration system is analogized with refrierant flow velocity to move the compression chamber through the suction valve. In addtion, the response of the vibration system connected to base could be analogized with sound propagating in suction muffler [10-12].

Eq. (1) Shows the equation of motion of the single degree of freedom vibration system excited by base motion.It is possible to establish the equations of motion of a sound system of single degree of freedom, such as Eq. (3) using the Eq.(2).

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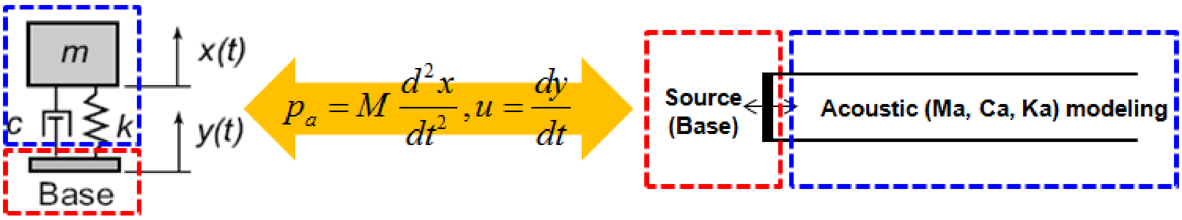


Figure 1 – Analogy vibration with acoustical system

$$M\frac{d^2x}{dt^2}+C\frac{dx}{dt}+Kx=C\frac{dy}{dt}+l \tag{1}$$

$$p_a = M\frac{d^2x}{dt^2},\quad u=\frac{d}{d} \tag{2}$$

$$\frac{d^2p_a}{dt^2}+\frac{C_a}{M_a}\frac{dp_a}{dt}+\frac{K_a}{M_a}p_a=C_a\frac{d^2u}{dt^2}+K_a\frac{du}{dt} \tag{3}$$

M_a, C_a, K_a mean the acoustical mass, damping, stiffness and p_a, u show the pressure and velocity. If single degree of freedom acoustical system is extending towards multi-degree of freedom acoustical system, acoustical mass, damping and stiffness should be assumed to be modal mass, damping and stiffness connected in parallel as shown in Fig.2.

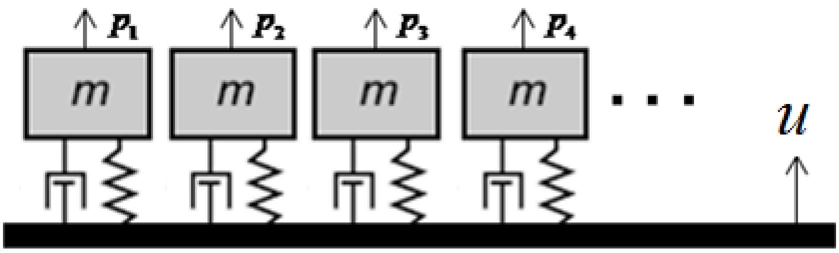


Figure 2 – Extending to multi-modal acoustical system connected in parallel

Therefore, multi-modal acoustical system connected in parallel could be expressed by Eq. (4). And Eq. (4) may be represented by the expression (5) in terms of input impedance.

$$\frac{d^2p_{an}}{dt^2}+\frac{C_{an}}{M_{an}}\frac{dp_{an}}{dt}+\frac{K_{an}}{M_{an}}p_{an}=C_{an}\frac{d^2u}{dt^2}+K_{an}\frac{du}{dt} \tag{4}$$

$$Z_{in}(\omega)=\frac{\sum_n p_{an}}{u}=\sum_n\frac{-C_{an}\omega^2+jK_{an}\omega}{\left(-\omega^2+\frac{K_{an}}{M_{an}}\right)+j\frac{C_{an}}{M_{an}}\omega} \tag{5}$$

M_{an}, C_{an}, K_{an} are n-th modal mass, damping and stffness.

The input impedance of the suction muffler could be measured by the Two Microphone Transfer Function

(TMTF) method. After then, using Peak Picking Method (PPM) and Eq.(5), M_{an}, C_{an}, K_{an} can be predicted for each mode of measured input impedance [12]. However, the PPM is available only at well-separated mode condition of the measured Frequency Response Fuction (FRF). As shown in Fig. 3, it is not easy to predict the M_{an}, C_{an}, K_{an} through the PPM method when the modes are very close to each other.

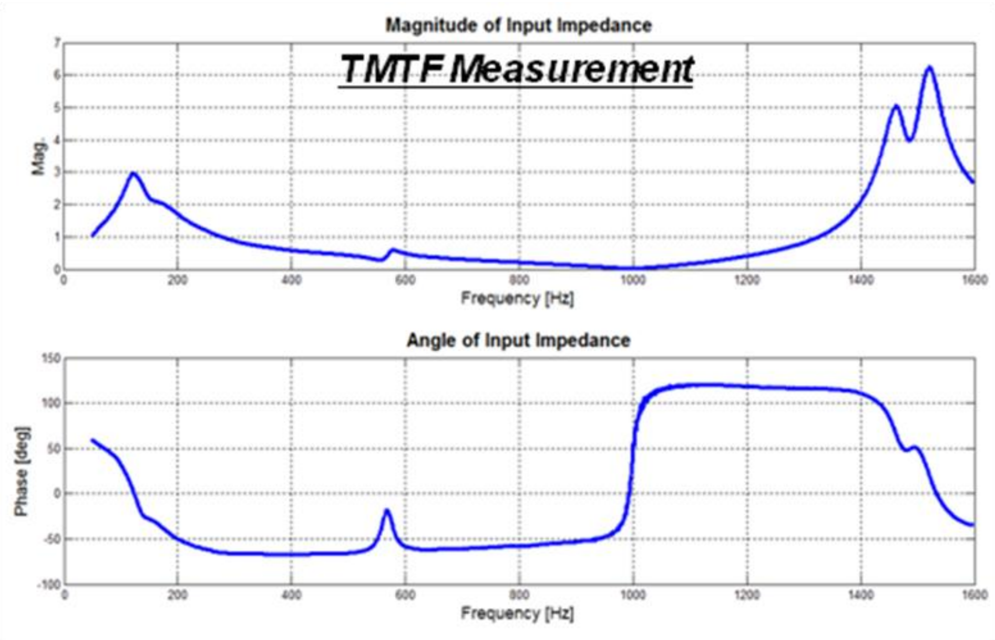


Figure 3 – Measured input impedance with not well separated mode

Therefore, in this study, Rational Fractional Polynomial (RFP) Method which is used in multi -degree of freedom modal analysis is used. When modes are coupling, multi-degree of freedom method can be approximated acurately relative to single degree of freedom method. RFP method can be expressed by Eq.(6).

$$\mathbf{Z}'(\omega_i) = \frac{\sum_{k=0}^{m} \mathbf{a_k} (j\omega)^k}{\sum_{k=0}^{n} \mathbf{b_k} (j\omega)^k} \tag{6}$$

Least squares estimation method is used to approximate the measured input impedance. The error is defined as the difference between the measured input impedance $(\mathbf{Z})$ and approximated input impedance $(\mathbf{Z}')$ such as Eq. (7). It is assumed that coefficient of the highest order term in the denominator is one.

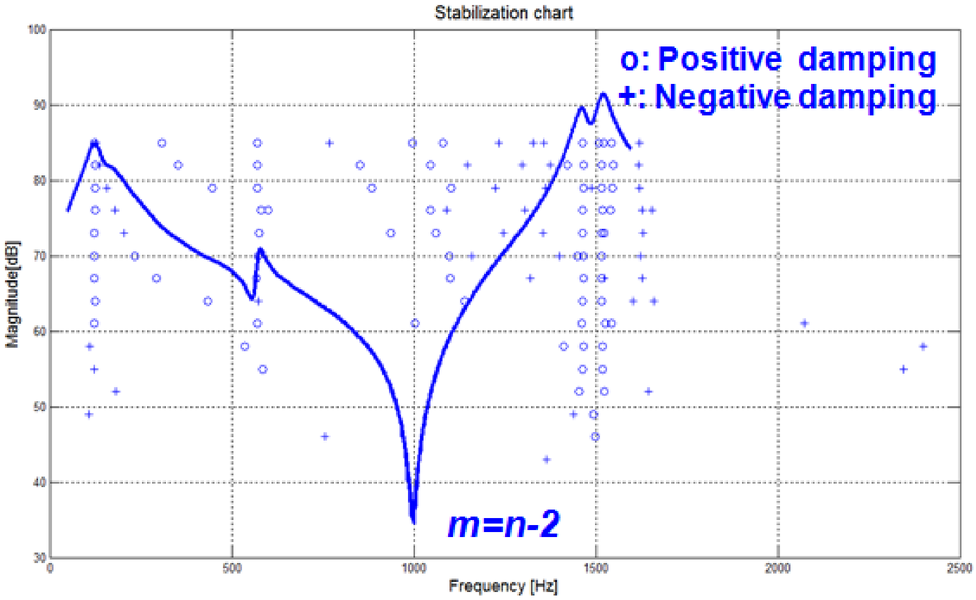
$$e_i = \sum_{k=0}^{m} \mathbf{a_k} (j\omega_i)^k - h_i \left[\sum_{k=0}^{n-1} \mathbf{b_k} (j\omega_i)^k + (j\omega_i)^n \right] \tag{7}$$

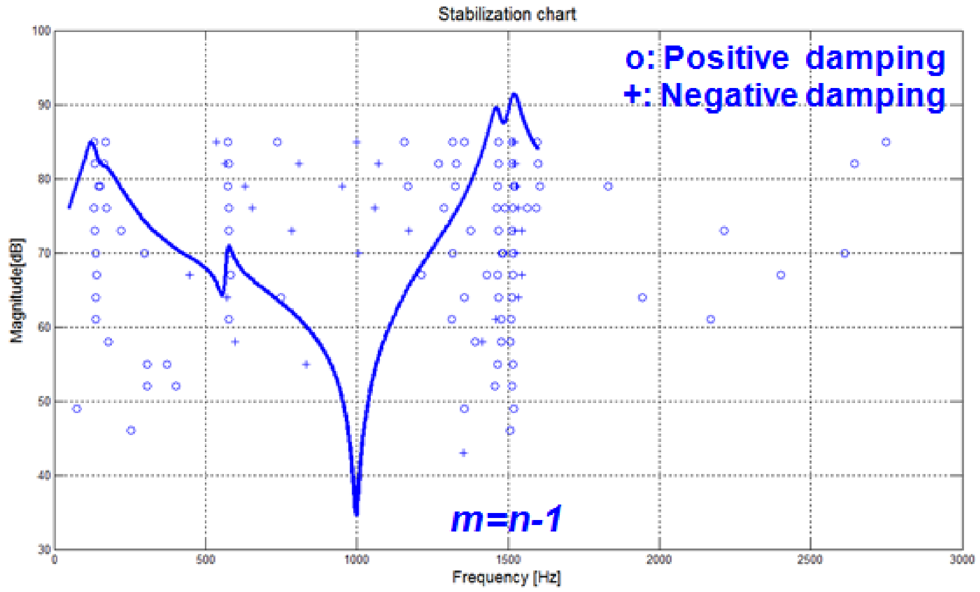
h_i is the input impedance measured at ω_i. The objective function (J) is set as power of error shown in Eq. (8).

$$J = \sum_{i=1}^{L} e_i^* e_i = \{E^*\}^t \{E\} \tag{8}$$

* means the complex conjugate and ^{t} represents transpose. Orthogonal polynomial basis can be used to the identification to avoid ill-condition of matirx. The following article is referred to more calculations [13].

Especially, it is very important to determine the order m,n of polynonimal function in Eq. (6). To determine the order of m, n, the Eq. (5) was examined. It can be seen that the order of the numerator and denominator are equal. Therefore, if the modal model parameters of input impedance are estimated, it can be seen that it should have a same order of the numerator and denominator. The Fig. 4 shows the stabilization chart which is used in modal test according to the order of m,n.





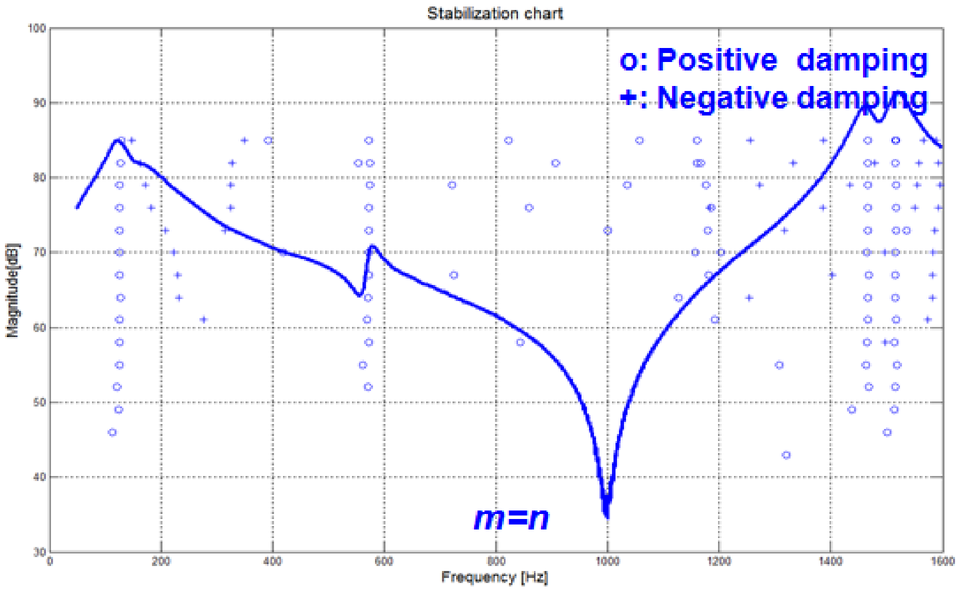


Figure 4 – stabilization chart w.r.t. order of m,n

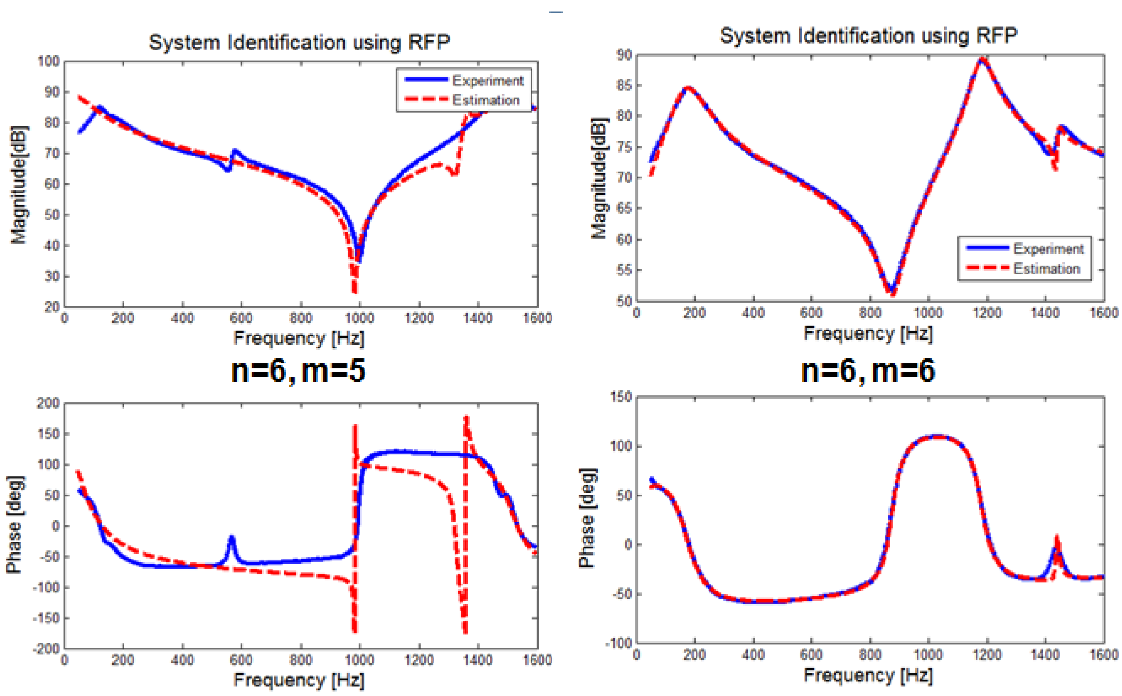


Figure 5 – Comparison accuracy of estimation model w.r.t order of m,n

When the order of numerator is less than order of denominator, unstable approximation model is created or approximation model is not accurate as shown in the stabilization chart.On the other hand, when the order of numerator is same as order of denominator, accurate approximation model is obtained as shown in Fig.4,5. Unkown coefficients $\mathbf{a}_k, \mathbf{b}_k$ can be obtained through this procedure. The obtained estimation models should be transformed to the time domain to analysis with motion of suction valve, refrigerant flow and thermodynamic model. Eq.(6) is changed to state space model. In a number of ways to convert the state space, Controllable Canonical Form (CCF) is selected as Eq.(9).

$$\begin{aligned} \dot{\mathbf{x}} &= \mathbf{A}\mathbf{x} + \mathbf{B}u \\ \mathbf{P} &= \mathbf{C}\mathbf{x} + Du \end{aligned} \tag{9}$$

$\mathbf{x}$ represents a state vecor and u, $\mathbf{p}$ expresse the speed and pressure. In addition, $\mathbf{A}, \mathbf{B}, \mathbf{C}, \mathbf{D}$ can be calculated by Eq.(10)-(13).

$$\mathbf{A} = \begin{bmatrix} 0 & 1 & 0 & \ldots & 0 \\ 0 & 0 & 1 & \ldots & 0 \\ 0 & 0 & 0 & \ddots & 0 \\ 0 & 0 & 0 & \ldots & 1 \\ -b_0 & -b_1 & -b_2 & \ldots & -b_{n-1} \end{bmatrix} \tag{10}$$

$$\mathbf{B} = \begin{bmatrix} 0 & 0 & 0 & 0 & 1 \end{bmatrix}^T \tag{11}$$

$$\mathbf{C} = \begin{bmatrix} a_0 - b_0 a_n & a_1 - b_1 a_n & \ldots & a_{n-2} - b_{n-2} a_n & a_{n-1} - b_{n-1} a_n \end{bmatrix} \tag{12}$$

$$\mathbf{D} = a_n \tag{13}$$

Experiments were carried out using the speaker source model to verify the suggested method. The predicted pressure is compared with pressure measured by a microphone. To predict pressure in front of speaker, input impedance measured in the way of TMTF method. Three microphone are used to widen a measurement frequency band. Fig. 6 shows experimental layout . And Fig. 5 is an identification result.

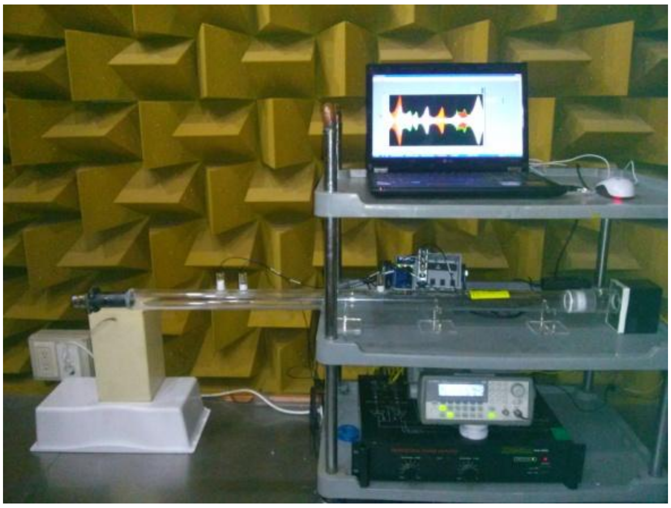


Figure 6 – Experimental layout for measuring input impedance

Identification model is transformed to state-space model to analysis in time domain using Eq.(9)-(13). The pressure is predicted using velocity input meausred by Laser Scanning Vibrometer (LSV) at speaker surfae. Comparing the pressure measured by microphone with a predicted pressure, these two results are very similar as shown in the Fig.7.

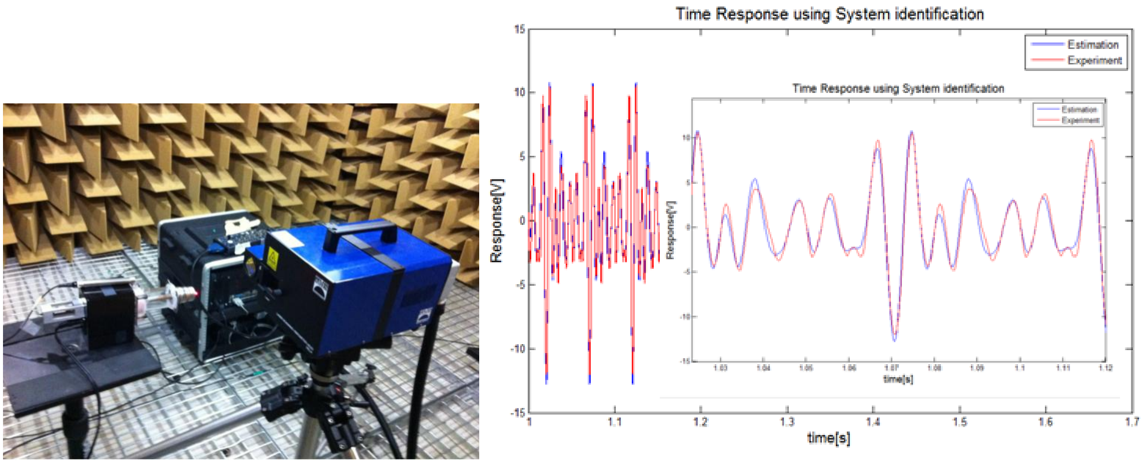


Figure 7 – Comparing the pressure measured by microphone with a predicted pressure

3. CONCLUSIONS

The procedure of the this study is that input impedance is measured by TMTF method. After, input impedance measured is estimated by RFP identification method. Approximated model is transformed to CCF form of state space to analysis in time domain. It is possible to verify the validity of the proposed method through experiment with speaker and suction muffler.

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