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VIBRATION AND NOISE REDUCTION OF HOUSEHOLD REFRIGERATOR

USING MODAL COMPONENT SYNTHESIS TECHNIQUE

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ABSTRACT

Using modal component synthesis method, a new design approach to suppress the transmitted noise in household refrigerator is presented. And this method is applied to design new type household refrigerator. As the result, the acoustic radiation efficiency of bed plate has decreased by 10 dB(A) and noise level has been suppressed more than 3 dB(A) compared with the conventional product.

INTRODUCTION

Recently obtaining a comfortable life environment, it is important problem to reduce the noise of the house appliances. To the household refrigerator, the strong demand for suppressing the noise will rise and it is an important factor to decide the commodity value. While the needs for saving resources and energy have been becoming serious, there exists a need to improve the performance and quality of household refrigerator. A new light weight and large size refrigerator has been developed. However as the result of these demands of design, refrigerator noise and vibration increased and new type design method has been required in order to consider the noise reduction in design process. With the decrease of the total noise, the fluctuating tone which is masked behind the broad band random noise has been becoming annoyance. The transmitted noise through mechanical pass which is caused by rigid body motion of compressor is the important factor of fluctuating tone. Authors would like to propose a new design approach for transmitted noise reduction and this concept is verified in practical design process of a new type household refrigerator.

NEW DESIGN APPROACH FOR HOUSEHOLD REFRIGERATOR

Noise and Vibration of Household Refrigerator

The household refrigerator sound spectra without fan is shown in Fig.1. The behavior of generating noise is divided into three frequency bands from its generation mechanism; low frequency band (under 800Hz), middle frequency band (800Hz-2.5kHz), and high frequency band (above 2.5kHz). The noise in high frequency band is produced by the mechanical transmission between the compressor mechanism and the shell. For this type noise, optimizing the clearance in the sliding portion and the stiffness of the component parts is attempted to reduce [1]. With regard to middle frequency band, it is generated by fluid noise. Some papers are presented about this region noise and it is applied to select the suitable place of discharge valve port and to optimize the exit shape of the muffler [1,2]. The noise in low frequency band is separated into harmonics of magnetic noise in the motor (250Hz-800Hz) and transmitted noise through mechanical path in a refrigeration system (under 250Hz). The motor noise is tried to be reduced by decreasing the magnetic permeance variations. These noises mentioned above, are able to be reduced by improvement and optimization of compressor itself. However, the transmitted noise through mechanical pass from compressor has to be evaluated as a total system in which almost all kind of mechanical components are included. Recently this type noise has been important for discrimination either the product is allowable one or not. But the design approach for total system in consideration of transmitted noise has not been established and the paper about such concept has not been reported yet. For this reason, authors would like to propose a new design approach as follows.

A Method for Transmitted Noise Reduction of Household Refrigerator

Fig.2 shows the mechanical system of a household refrigerator, which mainly consists of compressor, supporting mount, bed plate, refrigerant pipings and body. Simplified one degree of freedom model was usually used to design the vibration isolation system. However evaluating the vibration transmission of household refrigerator accurately, multi degrees of freedom model is needed to investigate the complex movement of each component including rigid body motion of compressor with 6-degrees of freedom. In order to predict the dynamic characteristics of mechanical system, the modal component synthesis technique which builds up a system model using individual components is one of the most usable method [3,4]. Fig.3 shows an analysis model to evaluate the vibration transmission of household refrigerator. The compressor is assumed to be a rigid body with 6-degrees of freedom. The supporting mount is treated as a spring with translational spring constant for each direction X, Y and Z. The finite element model are used to simulate the dynamic characteristics of bed plate, suction pipe and discharge pipe. The connection of compressor with pipings is rigid at its degree of freedom for connection. The supporting mount is used to connect of compressor with bed plate. With respect to bed plate, the welded point with body is fixed by 6-degrees of freedom and the caster is treated as free except for X and Z direction. It is assumed that refrigerant pipings have rigid connection with body. On the other hand in designing the mechanical system of household refrigerator it have to be considered a diversified discipline of design variables shown in Fig.4. There is a strong relationship and interaction between a number of design factors and components. And the design process is a try and error sequence of choices among a number of alternations in which each decision is affected by compromise between a number of demands and constraint. Therefore it is difficult to choose the optimum combination of each component by using iterative process for the analysis model shown in Fig.3. Accordingly We choose the constraint conditions related to the transmitted noise reduction and they are underlined in Fig.4. Next an assumption is introduced as follows. Since this system is regarded as the weak coupling with each component, it is assumed that the natural frequency and the mode shape of each component after combined do not change so much with each component alone. On the assumption mentioned above, the hierarchical design approach is proposed as follows.

At the first stage of this design method, each component is separated into several components according to the tolerance of criteria in design improvement. Whenever the design step has been progressed, each component is built up to evaluate the dynamic characteristics of mechanical system by using modal component synthesis technique. Each component of combined system is improved iteratively according to the modification index as far as to satisfy the several constraint conditions. In the case of household refrigerator, the design process could be divided into five stages as expressed in Fig.5. These criteria are determined as follows.

(1) Evaluation of bed plate

Since the bed plate has the large tolerance of design modification, it is treated at the first stage. The following three criteria are applied to decide the bed plate design.

1. In order to attenuate the vibration the following equation is derived.

$$f_1 > \sqrt{2} f_s (1+\alpha) \qquad (1)$$

where f : natural frequency of bed plate,

f_s: the second harmonics of power source frequency,

α : uncertainties of natural frequency of bed plate.

2. There is another criterion in which the mode shape of bed plate should not be corresponded with excitation mode of compressor, consequently not to amplify the compressor vibration. For that purpose, the mode shape of bed plate has to be controlled so that the antinode place may not coincide with the place where the compressor is mounted.

3. After two criteria previously mentioned would be satisfied, the minimum weight of bed plate is selected as the additional criterion.

Until these three criteria are satisfied, the modification of bed plate is repeated. In this stage, strain energy distribution and sensitivity analysis are applied for changing the design parameters. At the first step of design modification, the weak parts of bed plate are detected by strain energy distribution. After the weak parts pointed out by strain energy distribution, the specific parts or specific design valuables are modified by using sensitivity analysis. Strain energy distribution is calculated during an eigen value analysis for each mode. Strain energy of the bed plate is written by using the i-th modal vector$\{\phi\}_i$, stiffness matrix[K] and strain energy SE;

$$SE = \frac{1}{2}\{\phi\}_i^T[K]\{\phi\}_i \qquad (2)$$

Some parts of bed plate which have the large strain energy distribution are sensitive for modification of natural frequency and mode shapes. After the dominant parts have been selected, the deviation of natural frequency with respect to change in design variables is calculated by using sensitivity analysis. The calculation of sensitivity for natural frequency is expressed by using mass matrix [M], stiffness matrix [K], the i-th modal vector $\{\phi\}_i$, natural frequencyω_i and the j-th design parameterP_j as follows [5];

$$\frac{\partial\omega_i}{\partial P_j} = -\frac{\omega_i^2 m_i' - k_i'}{2\omega_i m_i} \qquad (3)$$

where $m_i = \{\phi\}_i^T[M]\{\phi\}_i$

$m_i' = \{\phi\}_i^T\frac{\partial[M]}{\partial P_j}\{\phi\}_i$, $k_i' = \{\phi\}_i^T\frac{\partial[K]}{\partial P_j}\{\phi\}_i$

By using this equation, the specific design parameter which is required to accomplish the objective natural frequency is predicted by the equation as,

$$P = \frac{1}{\frac{\partial\omega_i}{\partial P_j}}(\omega_{obj} - \omega_i) + P_j \qquad (4)$$

where P : prediction value of the modification of the j-th design parameter P_j

ω_{obj}: objective natural frequency.

And the eigen value analysis is carried out again by using new design parameter P. This process is repeated until the criteria are satisfied.

(2) Evaluation of bed plate and compressor with supporting mount

In this stage, bed plate, compressor and supporting mount are treated. The compressor and the supporting mount are considered as a pair of components. The criterion of this stage is defined by force transmissibility as follows;

$$\beta < \beta_s \qquad (5)$$

where β : force transmissibility between compressor and bed plate

β_s: threshold value of β.

When the force transmissibility is evaluated, the dynamic stiffness of bed plate has to be taken into consideration. Therefore in this design process, the system which contains bed plate, compressor and supporting mount is assembled by using modal component synthesis technique. And the dynamic characteristics of this system are predicted. If this system does not satisfy the criterion, the most dominant component is detected by using strain energy distribution. In this stage, the strain energy is calculated not for each element but for each component. And the component which has the largest strain energy is modified as shown in Fig.5.

(3) Evaluation of bed plate, pipings and compressor with supporting mount

The discharge and suction pipings are welded to the compressor shell directly. The rigid body motion of compressor has to be considered in the stage of piping design. In other words, there is a relationship and interaction between the several components considered in this stage. For that reason, the dynamic characteritics of mechanical system are predicted by using modal component synthesis technique. In this stage, the constraint forces at the place where each pipe is connected with body are minimized with consideration of mode shape which is close to power source frequency. This procedure is repeated for the feasible place until the constraint force minimization is satisfied.

( ) Response analysis and sound pressure prediction

By these stages previously mentioned, the simulation model of mechanical system of household refrigerator is generated. Next stage is the dynamic response analysis for excitation force imposed from the compressor. The acceleration response of the bed plate is applied to evaluate the total system of refrigerator as follows:

$$A < A_s \qquad (6)$$

where A : acceleration response of bed plate at the place where the mounts are supported

A_s: threshold value of A.

The relation between bed plate acceleration and sound pressure radiated from refrigerator body is denoted by

$$20\log_{10}(A) = k\cdot SPL \qquad (7)$$

where k : constant depend on the body shape and excitation frequency

SPL: sound pressure level from refrigerator body

The constant k of equation (7) is estimated by exciting the bed plate at a certain frequency and measuring the sound pressure at 1 m back and 1 m height of refrigerator in an anechoic room for many type bodies. Therefore, if an objective for the sound pressure level is decided, threshold value A_s can be decided from equation (7). These previous stages from ( ) to ( ) are repeated until the equation (6) will be satisfied. The vibration of bed plate is transmitted to the body which is the final noise radiator of the household refrigerator. Substituting the final acceleration into equation (7), the final noise of refrigerator can be predicted.

As mentioned above, the proposed new design approach is able to give the objective system at last with eliminating the weak point of each component in every stage. This design method is applied to produce a new type household refrigerator.

PRACTICAL APPLICATION AND VERIFICATION OF THIS DESIGN APPROACH

As a typical example using this design approach, the development procedure of a new type refrigerator is described. In here, design goal of this development is that the transmitted noise should be suppressed under 17dB(A) in low frequency band. Each stage for this design approach is presented as follows.

( ) Evaluation of bed plate

The natural frequency of the bed plate has a certain amount of frequency variation. In this case, the value is estimated about ±10 Hz, so α might be set to 0.1. Therefore f_1 has to be made greater than 190Hz. At first, the simulation model of conventional type bed plate is developed in the computer to evaluate the weak point. The transfer function of conventional type bed plate is shown in Fig.6. This figure shows us that the first natural frequency of bed plate (around 120Hz) has to increase by 70Hz to 190Hz. The strain energy distribution in each element of bed plate model is calculated. The result of calculation is shown in Fig.7. From this figure, it is pointed out that the part indicated by arrow is the dominant elements to increase the first natural frequency. Next, the sensitivity coefficient of the first natural frequency with respect to second moment of dominant area shown in Fig.7 is calculated. In consideration of the result of sensitivity analysis, the new type bed plate, in order to increase the some weak parts of the bed plate was strengthened 20 times as large as conventional type in term of second moment of area. However the thickness of bed plate was not changed. As the result, the first natural frequency of new type bed plate was able to increase to 194Hz effectively.

( ) Evaluation of bed plate and compressor with supporting mount

It is possible to achieve the isolation by choosing a natural frequency of suspension much lower than the excitation frequency. In most case, the ratio of the excitation frequency ω to natural frequency ω_n is selected between 2 and 4, namely ω/ω_n =2~4. If this ratio is selected 3 in the case of one degree of freedom model, the force transmissibility β becomes 0.13. So the threshold value β_s was selected as 0.1. At first, the initial value of stiffness constant is decided so as to satisfy β for one degree of freedom model. Next, the force transmissibility of this system which contains bed plate, compressor and supporting mount is evaluated by using modal component synthesis technique. The force transmissibility of compressor input to bed plate output at the supporting point of compressor is shown in Fig.8. Fig.8 shows this system is satisfied with the threshold value β_s above 50 Hz. The first natural frequency of bed plate in this system decreased by only 1 Hz from that of the bed plate alone (194 Hz). And the influence for mode shapes of bed plate from this combination process was recognized as negligible thing.

(7) Evaluation of bed plate, pipings and compressor with supporting mount
The finite element models have been developed for the suction pipe and the discharge pipe. All components treated in this stage are coupled together to evaluate the dynamic characteristics of mechanical system by using modal component synthesis technique. The calculation of the constraint force is repeated for each allowable point which is connected with body. The frequency of the rigid body modes influenced, but the frequency of the 6-th mode increased by only 1.2 Hz. The natural frequency of bed plate ω_s and force transmissibility β above 50 Hz was not changed. By this verification, the assumption mentioned previously was proved to be satisfied.

(7),(') Response analysis and sound pressure prediction of total system
The relation between bed plate acceleration and sound pressure from the refrigerator body is estimated by exciting the bed plate in an anechoic room. The typical data of this measurement is shown in Fig.9. This figure shows that the acoustic radiation efficiency (1/k) for horizontal direction is higher than vertical direction. Since the objective value of sound pressure in low frequency is 17 dB(A), A_s is decided 50 mm/s^2 as the threshold value with reference to horizontal direction as shown in Fig.9. Acceleration response of mechanical system applied from the rolling piston rotary type compressor is calculated. In here, the total forces and moment which are needed for this simulation are calculated by using same method as presented reference [6]. The excitation force is calculated for one of the standard operating condition given in Table 1 using Freon-12 as the refrigerant. The acceleration response of bed plate has been performed for the first and second frequency of power source. The result of analysis is shown in Table 2. For reference, the measured value of new type refrigerator is expressed in same table. This results allows us that both acceleration of horizontal direction and vertical direction have almost same level for the first frequency of power source. As the result, the threshold value was satisfied for both directions.

The new design approach presented in this paper is able to be used on the assumption that the objective system has the weak coupling with constructed components together. The unit of household refrigerator satisfied this condition. However, the further investigation will be needed for the system which has a strong coupling with each component.

The new type refrigerator was produced with the result mentioned above. Finally two experimental data measured before and after the optimization of total system are shown in Fig.10 and Fig.11. It is apparent from these figures that the acoustic radiation efficiency of bed plate has decreased by 10 dB(A) (Fig.10) and the noise level has been suppressed more than 3 dB(A) compared with the conventional type product (Fig.11). After this modification in the region of low frequency, 23dB(A) was realized as the total sound pressure level.

CONCLUSION

A new design approach to suppress the transmitted noise in household refrigerator was presented. And this method was applied to design the new type household refrigerator. Consequently following conclusive remarks could be proposed.
(1) A new design approach was proposed. The concept of this approach is as follows. At the first stage, each component is separated into several components according to the tolerance of design modification. Next, each component is improved iteratively by using evaluation method as far as to

satisfy the several criteria which are designated in every stage.
(2) To realize this approach, the modal component synthesis technique including the strain energy method and sensitivity analysis was applied with design modification criteria in each stage. Additionally acoustic radiation efficiency was defined and used to evaluate the transmitted noise from the refrigerator.
(3) By using this newly proposed approach, a new type household refrigerator is practically designed. And the noise level was suppressed more than 3 dB(A) compared with the conventional product.

Consequently, this approach contributed to develop the extremely low noise household refrigerator.

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Table 1 Analysis conditions

Suction pressure	($\times 10^4$ Pa)	6.1
Discharge pressure	($\times 10^4$ Pa)	88.5
Power source frequency	(Hz)	60.0
Number of revolutions	(rpm)	3525

Table 2 Acceleration response of bed plate

Frequency (Hz)	Analysis		Measurement	
	Horizontal	Vertical	Horizontal	Vertical
60	32.3	27.4	31.4	31.4
118	4.9	7.8	5.9	10.8

(mm/sec^2)

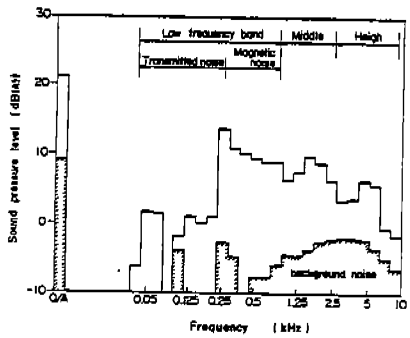

Fig. 1 1/3 octave band refrigerator sound spectrum

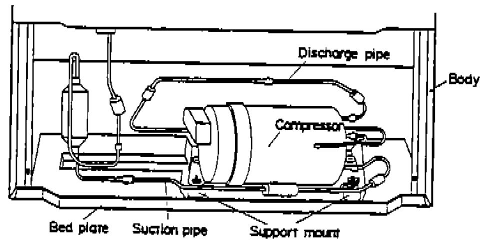

Fig. 2 Mechanical system of household refrigerator

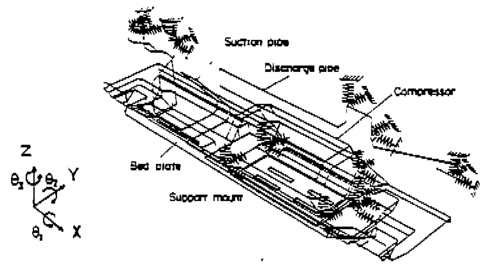

Fig. 3 Analysis model for modal component synthesis technique

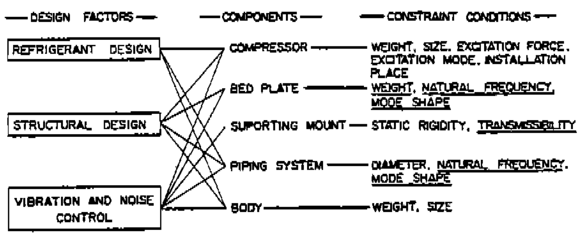

Fig. 4 Relationship among the design factors, the components and the constraint conditions of household refrigerator

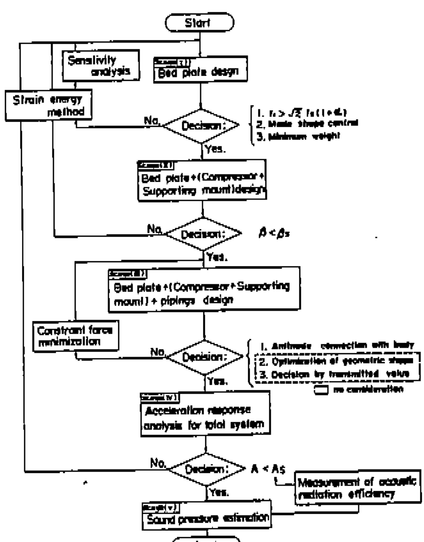

Fig. 5 The transmitted noise reduction flow of household refrigerator

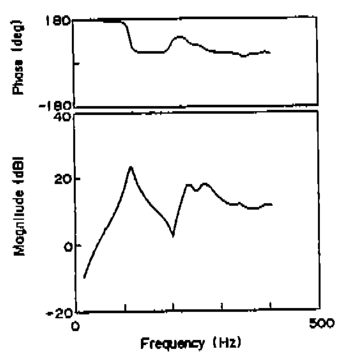

Fig. 6 Transfer function of household refrigerator (at bed plate)

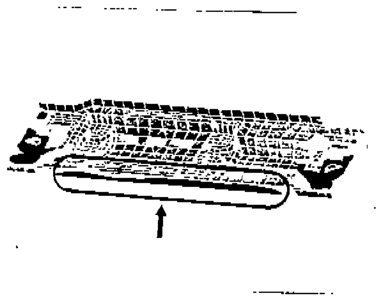

Fig. 7 Strain energy distribution of bed plate

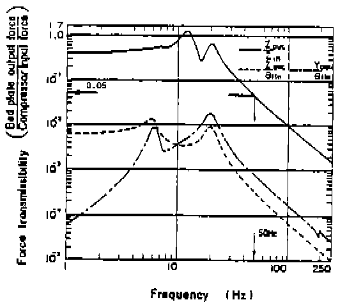

Fig. 8 Force transmissibility between compressor and bed plate

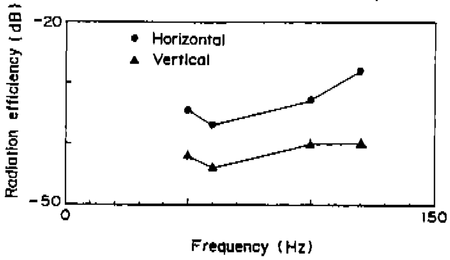

Fig. 9 Radiation efficiency of household refrigerator

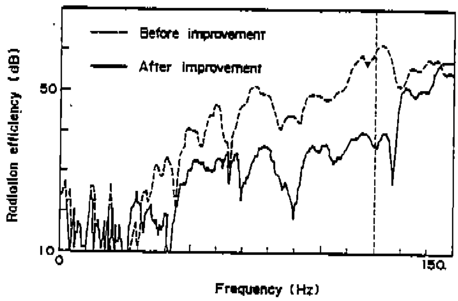

Fig. 10 Comparison of radiation efficiency between befor and after improvement

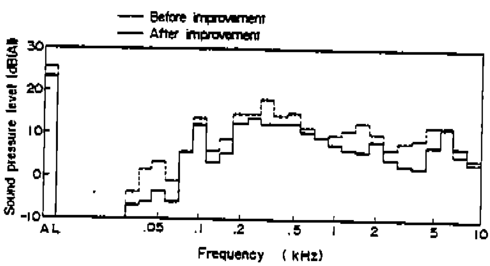

Fig. 11 Comparison of noise between befor and after improvement by 1/3 octave band analysis