International Journal of Air-Conditioning and Refrigeration Vol. 17, No. 2 (2009) 37~44

Reduction of the Refrigerant-Induced Noise from the Transition of Flow Pattern by Decreasing Tube Diameter

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(Received March 12, 2009; Revision received May 9, 2009; Accepted June 15, 2009)

Abstract

It is well known that a refrigerant-induced noise is caused by two-phase flow in the indoor unit of a heat pump air-conditioner. Especially when the flow pattern in a pipe is intermittent flow, the irregular noise occurs frequently. But it is very difficult to avoid this kind of the noise for the application of air-conditioner. Therefore, in this research, the flow patterns at two-phase flow state in a pipe of the indoor unit for the air-conditioner are researched using cycle simulator at typical cycle conditions. In order to find the relationship between refrigerant-induced noise and flow pattern, the noise patterns are investigated with respect to the estimated flow pattern from the various flow pattern maps. Base on the estimations of the flow patterns by those maps, the refrigerant-induced noise is evaluated as decreasing tube diameter, which can transit the flow pattern from slug to annular flow.

Key words: Air-conditioner, Refrigerant-induced noise, Two-phase flow, Flow pattern, Cycle simulator

1. Introduction

The refrigerant-induced noise from the indoor unit of the air-conditioner is one of the serious noise problems when it is developed. Since the refrigerant-induced noise varies according to the cycle conditions, it takes many expense and time to reduce this kind of noise. Moreover, because the noise level from the fan is growing down at a low cooling or heating mode recently, the contribution of the refrigerant-induced noise to the overall noise is growing up. Therefore the reduction of it is very urgent. Many of the researches have been conducted in these day for reducing this kind of noise.

Umeda[1] dealt with the layout of the inlet and outlet pipes of the electric expansion valve for avoiding the slug-flow that produced the noise. He experimentally verified that the flow pattern became slug when the direction of flow of the refrigerant into the expansion valve was vertical. Therefore, he suggested that the pipe layout of the expansion valve inlet should be laid horizontally. Hirakuni[2] dealt with the refrigerant-induced noise of the refrigerator at the capillary tube. He found that the noise increased when the flow pattern was slug-flow in front of the capillary tube. He recommended the addition of a dryer in front of the capillary tube for reducing refrigerant-induced noise. Since the dryer can separate the refrigerant into liquid and vapor, the flow pattern can transform to annular flow and the refrigerant-induced noise can decrease. But the researches for basic mechanism of it are not conducted sufficiently until now.

In this research, the producing mechanism of the refrigerating noise is dealt with the theories of two-phase flow and bubble dynamics. Based on these theories, the variation of the flow patterns are inferred with various flow pattern maps according to the variation of the mass flux by changing tube diameter. And then, the variations of the refrigerant-induced noises from the indoor unit of air-conditioner are evaluated with cycle simulator.

2. The mechanism of the refrigerant-induced noise

In the beginning state of the research for two-phase flow, homogeneous flow model was used. Homoge-

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neous flow model is assumed that it is one phase flow that liquid and gas are well mixed. But in theses day, the separated flow model is widely used that can consider the relative velocity between liquid and gas. The governing equations of the separated flow model are continuity equation, energy equation and momentum equation as given in Eqs. (1)-(3) respectively.

$$W_f + W_g = W = const \tag{1}$$

$$-\frac{dp}{dz}(W_g v_g + W_f v_f) = W\frac{dE}{dz} + \frac{d}{dz}\left[\frac{1}{2}(W_f u_f^2 + W_g u_g^2)\right] + W_g \sin\theta \tag{2}$$

$$-\frac{dp}{dz} = \frac{1}{A}\frac{dF}{dz} + [(1-\alpha)\rho_f + \alpha\rho_g]g\sin\theta + \frac{W^2}{A}\frac{d}{dz}\left[\frac{1}{A}\left(\frac{x^2 v_g}{\alpha} + \frac{(1-x)^2 v_f}{1-\alpha}\right)\right] \tag{3}$$

Here, W is the mass flow rate, p is the pressure, z is the distance in the axial direction, v is the specific volume, E is the internal energy, u is the velocity, ρ is the density, a is the void fraction($a=W_g/W$), x is the vapor quality($x=A_g/A$), θ is the angle of the pipe, A is the cross-sectional area of the pipe, F is the shear stress on the pipe due to the friction, and subscripts f and g denote liquid and gas, respectively.

Eq. (3) can be rewritten as given in Eqs. (4)-(7) separating the total pressure drop into the friction, gravity and acceleration terms respectively.

$$-\frac{dp}{dz} = -\left(\frac{dp}{dz}F\right) - \left(\frac{dp}{dz}z\right) - \left(\frac{dp}{dz}a\right) \tag{4}$$

$$-\left(\frac{dp}{dz}F\right) = \frac{1}{A}\frac{dF}{dz} \tag{5}$$

$$-\left(\frac{dp}{dz}z\right) = [(1-\alpha)\rho_f + \alpha\rho_g]g\sin\theta \tag{6}$$

$$-\left(\frac{dp}{dz}a\right) = \frac{W^2}{A}\frac{d}{dz}\left[\frac{1}{A}\left(\frac{x^2 v_g}{\alpha} + \frac{(1-x)^2 v_f}{1-\alpha}\right)\right] \tag{7}$$

From Eqs. (6)-(7), it can be estimated that the time gradient of the pressure drop should be increased when the flow pattern is intermittent flow such as slug and plug flow when the average vapor quality is constant. It is because that void fraction is changed as time goes on ($a= a(t)$) when the slug and plug bubbles go through a typical position of the pipe even though the mass flow rate is constant in adiabatic condition. However, the time gradient of the pressure drop should be low when the flow patterns in a pipe are annular and bubbly because the void fraction is constant in a pipe.

Therefore, considering the time gradient of the pressure drop according to the flow pattern of two-phase flow, the refrigerant-induced noise should occur from the two-phase flow, especially when the flow pattern is intermittent flow. And also it can be estimated that the refrigerant-induced noise is increased when the bubbles in a two-phase flow are generated and destructed.

Related to the generation and destruction of the bubbles, Strasburg[3] experimentally found that the bubble noise occurred when the volume of the bubble was changed. He suggested an equation for the bubble behavior that was represented by the volume terms, as shown in Eq. (8).

$$\frac{\rho}{4\pi R_0}\ddot{V} + R\dot{V} + \frac{kP_0}{V_0}(V - V_0) = P_0 - p_e(t) \tag{8}$$

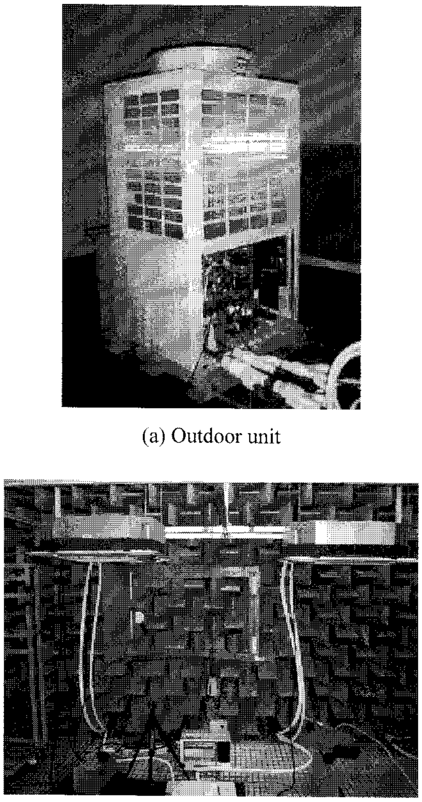
Here, P_0 is the static pressure of the bubble at the initial volume of the bubble (V_0), $p_e(t)$ is the instantaneous external pressure, R_0 is the initial diameter of the bubble, V is the volume of the bubble, k is the specific heat and R is the resistance coefficient that varies in a complex manner with the bubble size and frequency.

In a special case when the unsteady bubble goes through the orifice or bending pipe, the irregular refrigerant-induced noise can occur more seriously because of the volume- variations of the bubbles and cavitation.

Therefore, in this research, the refrigerant- induced noise is investigated experimently with producing mechanism of the refrigerant-induced noise estimated from theories of two-phase flow and bubble dynamics. The two-phase flow conditions are generated by the cycle simulator equipment and the patterns of the bubbles are estimated by the flow pattern map.

3. Cycle simulator

In a multi-type air-conditioner for a building as shown in Fig. 1, the pressure and temperature in the indoor unit are controlled to keep the regulated value. However, there are some cases where the refrigerant cycle changes suddenly due to the operating conditions of the indoor unit. In this case, the refrigerant-



(a) Outdoor unit

(b) Indoor units

Fig. 1. Multi-type air-conditioner.

induced noise occurs because the degree of sub-cooling cannot be retained sufficiently at the inlet of the EEV (Electric expansion valve) for the indoor unit.

As commented at the previous section, it is well known that a refrigerant-induced noise is caused by the two-phase flow, especially when the flow pattern is intermittent flow. However, it is very difficult to evaluate the noise in these cases using an actual set at a site or a laboratory.

Therefore, the cycle simulator equipment shown in Fig. 2 was developed in order to control the cycle conditions of the indoor unit. The pressure and the flow rate could be adjusted by the compressor, condenser and EEV.

The most important purpose developing cycle simulator is making the refrigerant to two-phase flow state at a typical position of the indoor unit.

Because the heat exchanger of an indoor unit is evaporator at the cooling mode, the water-cooling part becomes condenser. The phase of the refrigerant becomes liquid state at the exit of the condenser

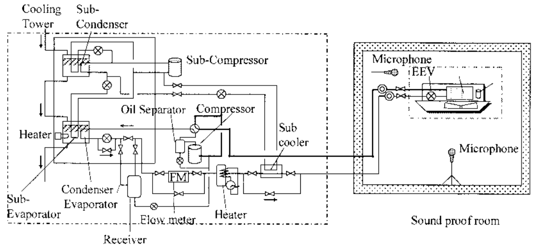


Fig. 2. Cycle Simulator.

controlling its temperature same as the set-up value. And then, it flows into the flow-meter which measures the mass flow rate of it. After then, it is heated properly by the heater so that it becomes two-phase state of the expected quality.

The thermodynamic cycle of the simulator at heating mode is opposite to that at the cooling mode. Because the refrigerant-induced noise at the heating mode frequently occurred when the refrigerant was become two-phase state at the exit of the indoor unit unusually, the cycle simulator was designed to make these conditions for the indoor unit.

When the refrigerant at the exit of the indoor unit is two-phase, the flow rate cannot be measured. Therefore the refrigerant is cooled by the sub-cooler once and then, the flow rate is measured after the refrigerant becomes liquid. The evaporating pressure and the super heat are regulated by the EEV. The refrigerant flow rate is regulated by the operating frequency of the compressor. However, because the vapor quality at the exit of the indoor unit cannot be measure directly, the estimation method of the exit quality is devised as followed.

The air flow rate of the indoor unit is measured initially and the heating capacity of the indoor unit is calculated by the temperature difference between the suction and the discharge of the air side. The enthalpy of the refrigerant at the exit of the heat exchanger can be known by means of the refrigerant flow rate assuming that the capacity calculated by the air-side data is same as the capacity calculated by the refrigerant side. As a result, the vapor quality at the exit of the indoor unit can be known without direct measuring.

Since the vapor quality was measured from indirect measuring in this research, some uncertainties could exist compared to the real vapor quality.

With this cycle simulator, the noise test for the indoor unit of the multi-type air-conditioner was performed. The indoor unit was 4way cassette type and the temperature and humidity conditions in a test room were heat overload condition according to the ASHRAE standard(27℃, 25.1% RH) where the refrigerant-induced noise was most serious commonly.

4. Estimation of the flow pattern

In order to investigate the relationship between two-phase flow and noise pattern, the flow pattern in a pipe should be identified. For these estimations, many of flow pattern maps are being developed until now. Because of the effect of gravity, the flow patterns are different as the pipe layout. Therefore, the flow pattern maps are separated to the horizontal and vertical pipes representatively. Even though the heat exchanger of the indoor unit of the air-conditioner in this research cannot be classified to horizontal or vertical pipe exactly, it is assumed that the pipe layout is consist of horizontal one only because the area of horizontal flow is more than that of vertical flow at the inlet and outlet of the heat exchanger. In fact, the acceleration on the horizontal pipe is more than that on the vertical one when the accelerations on the pipes were measured.

Therefore, in this research, the flow patterns for the two-phase flow are estimated with Baker map[4], Hashizume map[5] and Titel- Duckler map[6] which are representative flow pattern maps for the horizontal flow. The estimated flow patterns from them will be compared to the acoustic noise from the experiment in the next section.

4.1 Baker 's flow pattern map

Baker map is one of the most popular flow pattern maps. This map is used for condenser design, oil pipe design and so on. Baker defined the flow patterns with mass flux and modified coefficient as shown in Eq. (9).

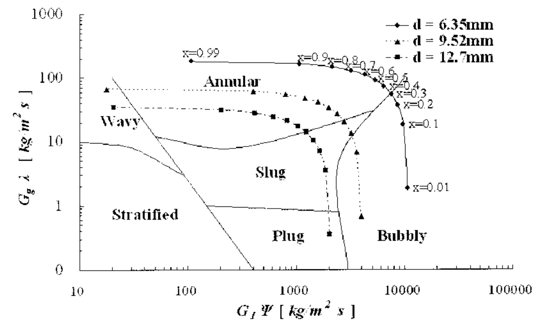
$$G_g = \frac{Wx}{A}, \quad G_f = \frac{W(1-x)}{A} \tag{9}$$

$$\lambda = \left(\frac{\rho_g}{\rho_{air}}\frac{\rho_f}{\rho_{water}}\right)^{\frac{1}{2}}, \quad \psi = \frac{\sigma_{water}}{\sigma}\left(\frac{\mu_f}{\mu_{water}}\frac{\rho_{water}^2}{\rho_f^2}\right)^{\frac{1}{3}}$$

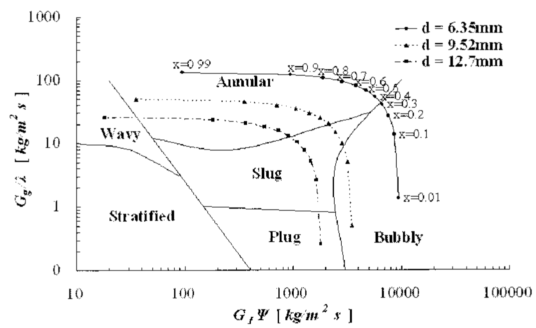
Here, W is mass flow rate of the refrigerant, x is quality, A is cross sectional area of the tube, G_g is mass flux of the gas, G_f is mass flux of the liquid, ρ_g, ρ_f, ρ_{water} are the density of the gas, liquid and water, μ_g, μ_f, μ_{water} are the viscosity of the gas, liquid and water, and σ_{water}, σ are the surface tension of the water and liquid.

The indoor unit dealt in this research is operating with R22 refrigerant at 1.88-2.1MPa pressure and 85-110kg/hr mass flow rate at heat operating condition. The flow pattern estimation by Baker map is as shown in Fig. 3. In Fig. 3, the flow patterns are depicted for applying 6.35, 9.52 and 12.7mm pipes in the condenser-outlet pipe respectively. Here, the diameter is the outside one and the thickness is 0.7mm. In Fig. 3, it can be known that the flow pattern of the pipe is estimated to be a slug flow when the vapor quality is 0.1-0.3 for 9.52mm pipe. When the pipe diameter is increased to 12.7mm, it can be known that the range having slug flow is increased. However, the flow pattern is estimated to be annular flow at the all of the range of vapor quality when the diameter of the pipe is reduced to 6.35mm.

Through the estimation results from Baker map, it is expected that the slug flow can be avoided by reducing pipe diameter and the irregular noise can be reduced simultaneously.



(a) W=110kg/hr, p=1.88MPa



(b) W=85kg/hr, p=2.1Mpa

Fig. 3. Flow pattern estimation with Baker map.

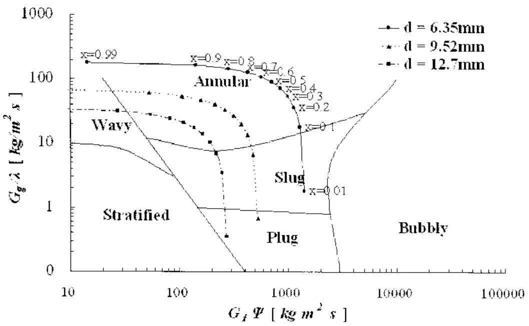
4.2 Hashizume's flow pattern map

Hashizume made flow pattern map with R134a and R22 refrigerant by experiment. This flow pattern map followed Baker map basically. But Hashizume applied modifying property correction factor for liquid term as shown in Eq. (10). This flow pattern map was the first one which observed the refrigerant not air-water.

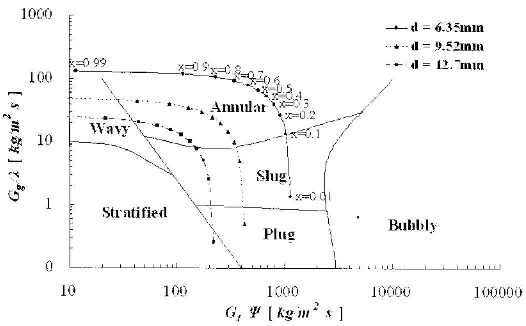
$$\psi = \left(\frac{\sigma_{water}}{\sigma}\right)^{\frac{1}{4}} \left(\frac{\mu_f}{\mu_{water}} \frac{\rho_{water}^2}{\rho_f^2}\right)^{\frac{1}{3}} \tag{10}$$

Fig. 4 shows the estimation results of the flow pattern evaluated by Hashizume map in the same conditions as section 4.1. In Fig.4, the flow pattern in a pipe is slug flow when the vapor quality is under 0.15 for 9.52mm pipe.

However, the slug flow occurs only when the vapor quality is very low(under 0.1). These results are different to those from Baker map. But it can be known that the areas having slug flow according to the variation of vapor quality are reduced on the flow pattern map when the diameter of the pipe is reduced.



(a) W=110kg/hr, p=1.88MPa



(b) W=85kg/hr, p=2.1Mpa

Fig. 4. Flow pattern estimation with Hashizume map.

Through the estimation results from the Hashizume map, it can be estimated that the slug flow and consequent irregular noise should be reduced when the diameter of the pipe is reduced.

4.3 Titel and Dukler's flow pattern map

Titel and Duckler suggested generalized flow pattern map based on the two-phase flow theory considering the properties of the operating fluid and the geometries of the pipe. The horizontal axis of this flow pattern map is martinelli parameter as shown in Eqs. (11) and (12)[8], and the vertical axis is froud number as shown in Eq. (13). If the flow pattern is estimated to be wavy or separated flow, it is subdivided to wavy and separated flow by variable K as given in Eq. (14). And if the flow pattern is intermittent or bubbly flow, it is subdivided to bubbly and intermittent flow by variable T as given in Eq. (15).

$$X = \left[\frac{(dp/dz)_f}{(dp/dz)_g}\right]^{1/2} = \left(\frac{1-x}{x}\right)^{0.875} \left(\frac{\rho_g}{\rho_f}\right)^{0.5} \left(\frac{\mu_f}{\mu_g}\right)^{0.125} \tag{11}$$

$$\left(\frac{dp}{dz}\right) = \frac{2fG^2}{\rho d} \tag{12}$$

$$\text{if } Re < 2000 - Laminar: \quad f = \frac{16}{Re}$$

$$\text{if } Re > 2000 - Turbulant: \quad f = \frac{0.079}{Re^{1/4}}$$

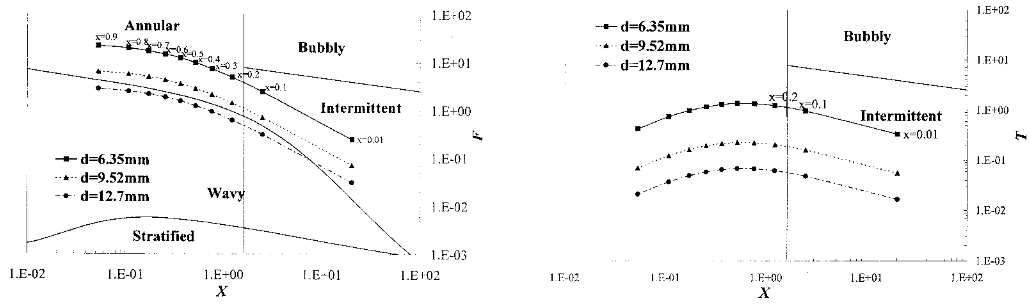
$$\left(Re = \frac{Gd}{\mu}\right)$$

$$Fr = \frac{G_g}{[\rho_g(\rho_f - \rho_g)\cdot d \cdot g]^{1/2}} \tag{13}$$

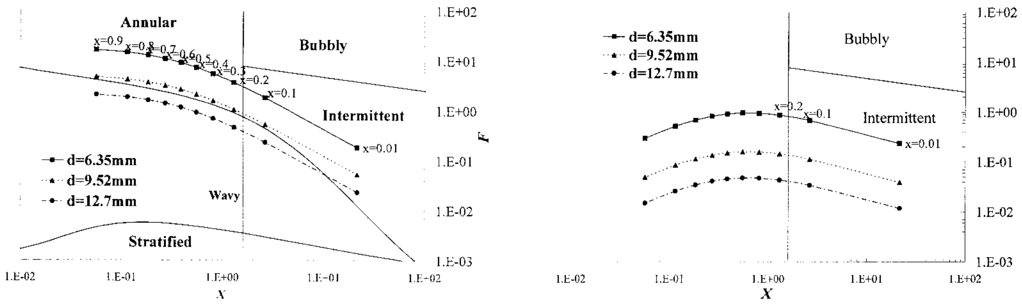
$$K = Fr \cdot \left[\frac{G_f \cdot d}{\mu_f}\right]^{1/2} \tag{14}$$

$$T = \left[\frac{|(dp/dz)_f|}{g \cdot (\rho_f - \rho_g)}\right]^{1/2} \tag{15}$$

Here, X denotes Matinelli parameter, Fr denotes Froud number, Re is Reinold number, f is friction factor, $(dp/dz)_f$ is the frictional pressure gradient if the liquid in two-phase flow is flowing alone in a tube, $(dp/dz)_g$ is the frictional pressure gradient if the gas in two-phase flow is flowing alone in a tube, d is tube diameter and g is acceleration due to gravity.



(a) W=110kg/hr, p=1.88MPa



(b) W=85kg/hr, p=2.1Mpa

Fig. 5. Flow pattern estimation with Titel and Dukler map.

Fig. 5 is the estimation results of the flow pattern evaluated by Titel and Duckler map in the same conditions as section 4.1. However, the estimations of flow patterns are different from those of Baker and Hashizume map. It can be estimated that the flow pattern in a pipe is not varied even though the diameter of the pipe is varied. Consequently, it is estimated from the Titel and Duckler map that the refrigerant-induced noise from the flow pattern is not varied as the variation of the pipe diameter.

After reviewing the estimations of the flow patterns with various flow pattern maps, it can be known that the irregular refrigerant-induced noise can be reduced by reducing pipe diameter from the Baker and Hashizume flow pattern map. However, it can be estimated by Titel and Duckler flow pattern map that the irregular refrigerant-induced noise is almost same even though the pipe diameter is reduced.

The noise estimation by the flow pattern map had some uncertainties because the estimations of flow patterns with various flow pattern maps are a little different to one another. In order to overcome these uncertainties for estimating refrigerant-induced noise with flow pattern map, more studies are necessary in the future.

In order to compare the noise pattern and the estimations of the above flow pattern maps, the noise test was performed using cycle simulator at cycle conditions where the flow patterns are estimated.

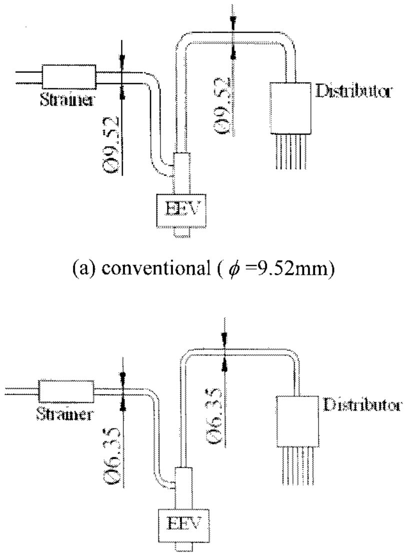
5. Experimental verification

In order to investigate the noise pattern estimated from various flow pattern maps in previous section, the noise test was performed for the indoor unit of air-conditioner which the condenser-outlet pipe was 9.52 and 6.35mm respectively as shown in Fig. 6.

Table 1 is the measured sound pressure level of the indoor unit. The sound pressure levels in Table 1 include the refrigerant-induced noise as well as fan noise. Through removing fan noise from the overall sound pressure level from Eq. (16), the refrigerant-induced noise can be calculated.

$$L_R = 10\log(10^{\frac{L_T}{10}} - 10^{\frac{L_F}{10}}) \tag{16}$$

Here, L_R is refrigerant-induced noise(dB), L_T is overall sound pressure level(dB), L_F is fan noise(dB).



(a) conventional (ϕ =9.52mm)

(b) improved (ϕ =6.35mm)

Fig. 6. Schematic diagram of the condenser-outlet pipe.

Table 2 shows the calculated sound pressure level of the refrigerant from Eq. (16). From Tables 1-2, it can be found that the overall sound pressure level is reduced about 0.7-1.3dB and the refrigerant-induced noise is reduced about 2.3-5.6dB when the diameter of the pipe is reduced from 9.52 to 6.35mm.

Figs. 7(a) is the narrow band spectrum of the sound pressure level according to the variation of the diameter for the condenser-outlet pipe.

However, because the frequency of the refrigerant-induced noise is broadband, it is difficult to identify where the noise reduces or increases in the measuring frequency range. Therefore, 1/3 octave band spectrum is shown in Fig. 7(b) additionally. In Fig 7(b), it can be found that the noise from 1.25kHz to 5kHz is reduced dominantly when the diameter of the pipe is changed from 9.52 to 6.35mm.

The frequency characteristics as above can be explained with the resonance frequency of the bubble suggested by Minnaert[7] as given in Eq. (17)

$$R_0 = \frac{1}{2\pi f_n}\sqrt{\frac{3k\tilde{p}}{\rho_L}} \tag{17}$$

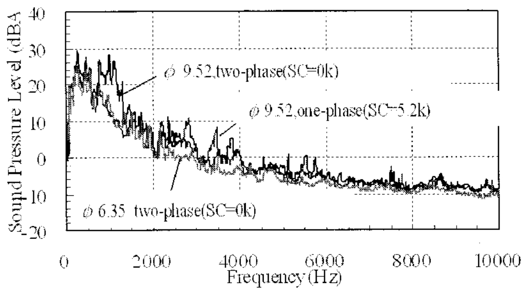
Here, R_0 is the radius of a bubble, f_n is the natural frequency of a bubble, k is the ratio of specific heat at constant pressure to constant volume and $\tilde{p}$ is the hydrostatic pressure of surrounding liquid.

Table 1. Total sound pressure level.

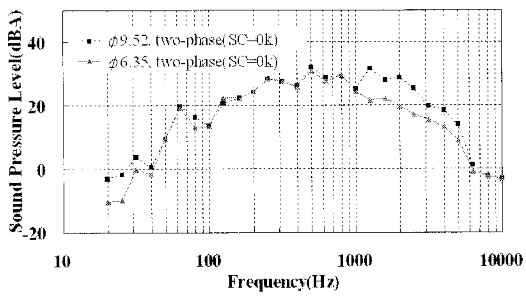
Pressure [Mpa]	Mass flow rate [kg/hr]	Total sound pressure level [dBA]			
		ϕ =9.52mm		ϕ =6.35mm	
		2-Phase	1-Phase	2-Phase	1-Phase
1.88	110	38.4	36.1	37.7	36.3
2.1	85	38.2	36.7	36.9	36.6

Table 2. Calculated sound pressure level for the refrigerant.

Pressure [Mpa] [kg/hr]	Mass flow rate	Estimated sound pressure level for the refrigerant [dBA]			
		ϕ =9.52mm		ϕ =6.35mm	
		2-Phase	1-Phase	2-Phase	1-Phase
1.88	110	34.7	19.7	32.4	19.9
2.1	85	34.2	28.4	28.6	26.0



(a) Narrow band spectrum



(b) 1/3 octave spectrum

Fig. 7. Noise spectrum of the sound pressure level.

Eq. (17) can be derived with Eq. (8) in section 2 when the resistance coefficient is neglected and the volume is assumed to oscillate as given in Eq. (18).

$$V = V_0 + a\sin(\omega t) \tag{18}$$

Here, V is the volume of the bubble, V_0 is the initial volume of the bubble, a is the amplitude of oscillation, ω is the angular frequency and t is the time. If the bubble is non-spherical, R_0 is the taken as the radius of a sphere having same volume.

Assuming the increment of the noise from 1.25kHz to 5kHz for the pipe with 9.52mm diameter is come from the resonance of the bubble, the bubble size(radius) can be estimated to be in the range of 3.1-12.5mm when the pressure, density of liquid and the ratio of specific heat are 2.1Mpa, 1065kg/m^3 and 1.63 respectively. Because the size of slug bubble is generally larger than a number of times of the inner diameter of the pipe, it can be estimated that the bubbles in a pipe are slug bubbles accounting for the calculated radius of them. Therefore it can be verified that the noise increment in the frequency range from 1.25kHz to 5kHz should be come from the slug bubbles.

Consequently, when the results of the noise test are compared to the estimations of the various flow pattern maps, it can be verified that the estimation of the flow pattern from Baker and Hashizume flow pattern maps are well accordance with the noise test results. But the estimation of flow pattern from the Titel and Duckler map would not suitable for the estimation of the irregular refrigerant -induced noise of air-conditioner. Even though Baker's map is well known that it is not well accordance with the real flow pattern of the refrigerant from the previous works, it is well accordance with noise pattern in this research. Since the identification of the flow pattern is relied on the visualization, the estimation of the flow pattern could be some difference to the acoustic point of view. Because of these reasons, it cannot be declared that Baker's map is not proper to estimate the noise from 2-phase flow bubbles due to the results of the previous works. These vagueness will be studied more detail in the future.

6. Conclusion

By developing the cycle simulator, the noise from the refrigerant is measured at typical cycle conditions continuously. And also, the comparisons between the acoustic noise and flow pattern are conducted with the noise test and the estimations of the flow patterns by various flow pattern maps.

Through these results, the following conclusions can be obtained.

(1) The refrigerant induced noise of the indoor unit in heat operating condition comes from the slug bubbles in the condenser-outlet pipe.

(2) The noise pattern for the indoor unit in this research is well in accordance with the estimating flow patterns, which estimated by Baker and Hashizume map

(3) The relationship between noise and flow pattern cannot be found with Titel and Duckler flow pattern map in this research

In this research, the producing mechanism of the irregular refrigerant-induced noise mechanism could be identified with two-phase flow and bubble dynamic theories and suggested the relationship between the noise and flow pattern with the estimations of the flow pattern map. The results in this research are expected to be helpful for reducing refrigerant-induced noise for the air- conditioner.

References

[1] Umeda, T., 1993, Reduction of Noise Caused by Gas-Liquid Two-Phase Refrigerant Flow through an Expansion Valve, JSME, Vol.59, No.557, pp.243-248.

[2] Hirakuni, S., Smida, Y., Yamamoto, H., 1998, Study of Noise Reduction of Refrigerant for Capillary Tube in the Refrigerator, 32nd Conference Journal of Refrigeration and Air-Conditioning Association, Vol.4, pp.22-24.

[3] Strasberg, M., 1956, Gas bubbles as source of sound in liquids, The Journal of the Acoustical Society of America, Vol. 28, No. 1, pp. 20-27.

[4] Baker, O., 1954, Design of pipe lines for simultaneous flow of oil and gas, Oil and Gas Journal, Vol.53, pp.185-190.

[5] Hashizume, K., 1983, Flow pattern and void fraction of refrigerant two=phase flow in a horizontal tube, Bulletin of the JSME, Vol.26, 1597-1602.

[6] Taitel, Y. and Dukler, A., E., 1976, A model for predicting flow regime transitions in horizontal and near horizontal gas-liquid flow, AIChE journal, Vol.22, pp. 47-55.

[7] Minnaert, W.K., On musical air bubbles and the sound of running water, Phil. Mag. 16, pp.235-248, 1933

[8] Whalley, P. B., 1999, Two Phase Flow and Heat Transfer, Oxford University Press