Springer

Journal of Mechanical Science and Technology

Journal of Mechanical Science and Technology 23 (2009) 1456~1467

www.springerlink.com/content/1738-494x
DOI 10.1007/s12206-008-1128-8

Experimental analysis for reducing refrigerant-induced noise of 4-way cassette type air conditioner †

Hyung Suk Han[1], Weui Bong Jeong[2,*] Shigeo Aoyama[3] and Jin Yong Mo[3]

[1]*1st Team, Naval Sea System Center, Defense Agency for Technology and Quality, 1, Aju-dong, Geoje, Korea*
[2]*Department of Mechanical Engineering, Pusan National University, Jangjeon-dong, Kumjung-ku, Pusan 609-735, Korea*
[3]*Air-conditioning Technical Expert Group, System Appliance Division, Samsung Electronics, 416, Metan-3dong, Yeongtong-Gu, Suwon, Korea*

(Manuscript Received August 5, 2008; Revised November 15, 2008; Accepted November 24, 2008)

Abstract

Various refrigerant flow patterns can produce a range of noise types according to their cycle conditions. Consequently, the identification of flow patterns in a tube is crucial to reducing refrigerant-induced noise. Because of the obstacles involved in identifying them accurately by experiment, in this paper, these flow patterns are estimated from the flow pattern map. Working from the assumption that the refrigerant-induced noise for an air conditioner in the heating mode comes from slug flow in the condenser-outlet pipe, the reduction of refrigerant-induced noise by avoiding slug flow in a tube is examined. To fully understand the conditions under which the refrigerant-induced noise occurs, cycle simulator equipment for an outdoor unit is developed. With this cycle simulator, noise tests of 4-way cassette type indoor units are performed under the conditions that the refrigerant-induced noise occurs. Increasing the mass flux in a tube by reducing the diameter of the condenser-outlet pipe can avoid slug flow, and the refrigerant-induced noise can therefore be reduced. The results of the cycle simulator can be verified with an outdoor unit 5HP system multi air conditioner and the results are well in line with simulator results.

Keywords: Refrigerant-induced noise; Evaporator-inlet; Slug flow; Flow pattern map

1. Introduction

Many of today's air conditioner makers are currently trying to reduce their noise levels without a simultaneous reduction in heat capacity. The noise of an air conditioner can be classified by characteristics such as fan noise, motor noise, compressor-transferred noise and refrigerant-induced noise. In this study, refrigerant-induced noise from indoor air conditioner units will be discussed. There have been many previous studies on this refrigerant-induced noise.

Hirakuni [1] suggested the use of a porous metal in front of an expansion device in order to change the flow pattern from slug to bubbly flow and thereby reduce refrigerant-induced noise. Hirakuni, Simida and Yamamoto [2] dealt with the refrigerant-induced noise for the refrigerator at the capillary tube and suggested a sub-cooler in front of the capillary tube in order to attain a greater degree of sub-cooling. Umeda [3] dealt with the refrigerant-induced noise according to the inlet and outlet pipe layout of the electric expansion valve and suggested that the inlet pipe of the electric expansion valve should be laid horizontally in order to avoid slug flow. Kannon [4] dealt with the relationship between the refrigerant noise and the flow pattern of the slug flow. He verified, by experiment, that the transient pressure wave occurred at the time when the vapor fraction of the refrigerant increased. Uemeda [5] dealt with the noise related to

† This paper was recommended for publication in revised form by Associate Editor Yeon June Kang

*Corresponding author. Tel.: +82 51 510 2337, Fax.: +82 51 517 3805
E-mail address: wbjeong@pusan.ac.kr

the 2-phase flow for water-air systems and attempted to determine the relationship between the noise and pressure variation when 2-phase water-air passed through the orifice tube.

In this study, the refrigerant-induced noise of the air conditioner is discussed in relationship to the flow pattern of the refrigerant in a pipe. The flow patterns are estimated from the flow pattern map suggested by Baker [6]. Because time variant refrigerant-induced noise usually comes from the slug flow, the object of this research is to find the design factors in which the slug flow does not occur in any of the cycle conditions. The noise protection method from the refrigerant has ordinarily been performed after the problem occurs. However, this research addresses that it can be estimated at the first step of design rather than later when the noise test is done.

In this paper, the mechanisms involved in the production of refrigerant-induced noise are initially reviewed. Additionally, at the special cycle condition in which the refrigerant-induced noise occurs, noise tests are performed by the cycle simulator with different condenser-outlet pipes in order to avoid slug flow. After applying the most effective design from the test of the cycle simulator that can reduce refrigerant-induced noise, a verification test is performed with a 4way cassette type air conditioner.

2. Mechanism of refrigerant induced noise

Refrigerant-induced noise usually occurs when the state of refrigerant flowing in a pipe is 2-phase and it usually occurs at the inlet or outlet pipes of the heat exchanger. The characteristics of the sound often vary by the flow patterns of the refrigerant passing through a tube at the 2-phase condition. Fig. 1 [7] shows the typical flow patterns in a pipe for the horizontal flow,

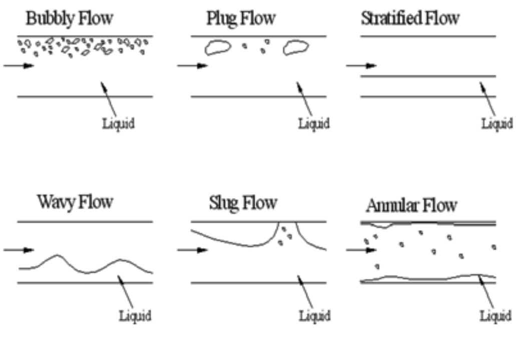


Fig. 1. Typical flow pattern in a horizontal tube.

and the flow pattern transits from the bubbly to the annular flow when the fraction of the gas is cumulated.

The governing equation of a pressure drop for separated 2-phase flow model can be explained as shown in Eq. (1) [8].

$$-\frac{dP}{dz}=\frac{4\tau}{d}+[\alpha\rho_g+(1-\alpha)\rho_l]g\sin\theta$$
$$+G^2\frac{d}{dz}[\frac{x^2}{\alpha\rho_g}+\frac{(1-x)^2}{(1-\alpha)\rho_l}],\ \alpha=\frac{A_g}{A},\ \ x=\frac{G_g}{G} \qquad (1)$$

Here, α is void fraction, A_g is the average cross-sectional area of the tube occupied by the gas phase, A is the total cross sectional area of the tube, τ is wall stress, d is tube diameter, G is mass flux of fluid, G_g is mass flux of gas, ρ_l is liquid density, ρ_g is gas density and x is mass quality.

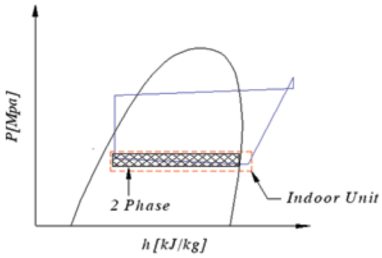
When the time gradient of the pressure drop from Eq. (1) is considered for intermittent flow such as slug and churn flow, the pressure drop should be changed with time because of the time variant void fraction. It means that the noise from time-variant pressure drop can be produced when the slug flow goes through the tube such as bended tube or orifice. When a bubble in the intermittent flow is collapsed while it is passing though a tube, its volume should be varied, which produces the radiated acoustic pressure as given in Eq. (2) [9].

$$p_a=\frac{\rho_L}{4\pi R}\frac{d^2V}{dt^2} \qquad (2)$$

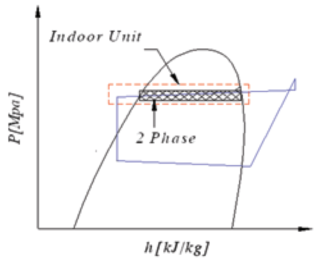
Here p_a is radiated acoustic pressure, R is the distance from the cavity center to the measuring point, ρ_L is density of liquid, V is time-varying volume of the cavity.

In addition, cavitation also takes place unstably when a pressure drop varies. Therefore, it can be estimated that the refrigerant-induced noise unstably occurs when the flow pattern in a tube is an intermittent flow.

When the flow pattern is annular, the void fraction is not varied as much as time. Therefore, the time gradient of Eq. (1) should be almost zero. It means that the noise from time variant pressure drop will not happen. However, if the mass flow rate is the same, volumetric flow rate of annular flow is higher than that of the slug or bubbly flow. When the annular flow is passing through the inlet or outlet pipe of a



(a) Cooling mode



(b) Heating mode

Fig. 2. P-h diagram of cooling and heating mode.

heat exchanger, then the pressure drop from the velocity should be increased even though the time-gradient of the pressure drop is not so high. Hence, when the flow pattern is annular, it can be estimated that the refrigerant-induced noise mainly occurs by cavitation but it does not vary as much as time.

The refrigerant-induced noise problem usually occurs at the low refrigerating mode for the indoor side unit. However, it is less problematic for the outdoor side unit because the noise level from the refrigerant is less than that from the fan and the compressor relatively. Therefore, in this study, only the refrigerant-induced noise of the indoor side unit will be discussed.

Fig. 2 shows the p-h (pressure-enthalpy) diagram of a general refrigerating cycle of the air conditioner. When the air conditioner is operating in cooling mode, the indoor side unit acts as an evaporator as shown in Fig. 2 (a). The 2-phase flow occurs from the inlet of the evaporator including the expansion valve outlet. In the evaporator, the transient pressure drop does not occur so much because the diameter of the pipe is constant and the route of flow is simple even though the phase of the refrigerant is 2-phase. Therefore, refrigerant-induced noise usually does not occur in the evaporator. However, the evaporator inlet uses the pipes with various dimension of diameter and the route of flow is complex. Thus, a transient pressure drop occurs in the evaporator inlet pipes and the time gradient of pressure drop is more than that in the

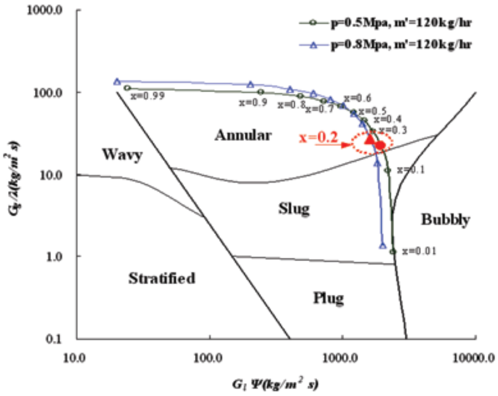


Fig. 3. Estimation of flow pattern in the evaporator-inlet pipe by Baker's flow pattern map.

evaporator. Therefore, the refrigerant-induced noise usually occurs at the evaporator inlet at cooling mode.

The flow pattern of the refrigerant at the evaporator-inlet can be estimated from a flow pattern map. The experimental results at the evaporator-inlet according to mass quality are added on the Baker's flow pattern map given in Fig. 3. The flow pattern of the refrigerant at the evaporator-inlet almost becomes an annular flow at cooling mode since the quality of the refrigerant is usually about 0.2. Referring to the above, the refrigerant-induced noise for annular flow should be proportional to the velocity of the refrigerant. Consequently, the dominant factor related to the refrigerant-induced noise at the cooling mode should be the velocity of the refrigerant. In order to reduce the refrigerant-induced noise at this state, the velocity of refrigerant should therefore be reduced, which means that the quality of the refrigerant should be reduced.

At heating mode, however, the indoor side unit acts as a condenser as shown in Fig. 2(b). The refrigerant transits from the gas to the liquid when passing through the condenser. It becomes 1-phase superheated vapor at the condenser-inlet and sub-cooled liquid at the condenser-outlet for the normal cycle conditions. However, the refrigerant flowing out from the condenser can be 2-phase at a special cyclic condition. At this time, the quality of the refrigerant is usually not so high. Therefore, the refrigerant is possibly related to slug flow at the condenser outlet. When the slug flow passes through the condenser outlet pipe, time varying abnormal noise can occur. Hence, it can be expected that the refrigerant-induced noise at the heating mode should be related to the flow pattern of the refrigerant especially slug flow.

Fig. 4 shows the noise spectrum of a 5.2kW 4way cassette type air conditioner when the state of the refrigerant at the condenser-outlet is 1-phase and 2-phase. In Fig. 4, the refrigerant-induced noise increases from 500Hz to 2.5kHz when the state of refrigerant is 2-phase at the condenser-outlet. Additionally, the time signal of the sound pressure level is very unstable when the state of refrigerant is 2-phase, as shown in Fig. 5.

If the increment of the sound pressure level comes from the slug bubbles, the frequency characteristics of the increasing noise should be strongly related to the resonance frequency of the slug bubbles. Eq. (3) shows the resonance frequency of a bubble cavity [10].

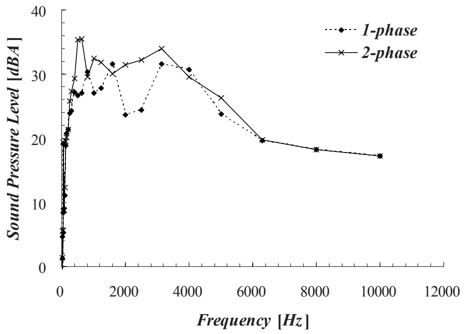


Fig. 4. Noise spectrum of 5.2kW 4way cassette type air conditioner.

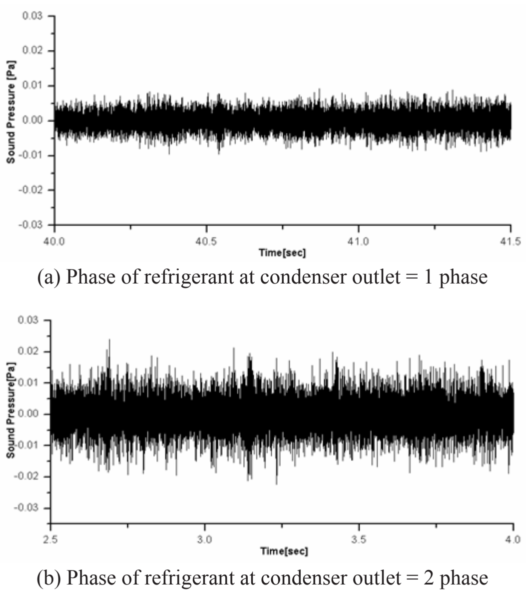


Fig. 5. Time signal of sound pressure for 5.2kW 4way cassette type air conditioner.

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

Here, R_0 is the radius of a bubble, f_n is the resonance frequency of a bubble, p is the hydrostatic pressure of surrounding liquid, κ is the ratio of specific heat with constant pressure to that with constant volume. Fig. 6 shows the variation of bubble radius according to its resonance frequency. There must be many bubbles with different radii in the evaporator-inlet pipe. Assuming that the radii of slug bubbles are distributed from 50% to 150% of the radius of pipe (radii=4.76mm, thickness=0.7mm), it can be seen that the frequency range at which sound pressure level increases is from 500Hz to 2.5kHz.

The frequency range at which sound pressure level increases in Fig. 4 appears similar to that of resonance of slug bubbles in Fig. 6. Therefore, it can be estimated that the increasing noise level at 2-phase state is caused from the resonance of slug bubbles in the evaporator-inlet pipe.

The objective in this paper is, through investigating these experimental phenomena, to find the condition that does not produce the abnormal refrigerant-induced noise, even though the state of refrigerant flowing out from the condenser is 2-phase at the heating mode.

3. Estimation of flow pattern

To monitor the state of the refrigerant, a sight glass is widely used. However, it is difficult to visualize the flow pattern exactly. In this paper, the flow pattern is estimated by a flow pattern map. Because the flow direction in a heat exchanger is horizontal, all of the flows in this paper assume horizontal flow. Thus, the

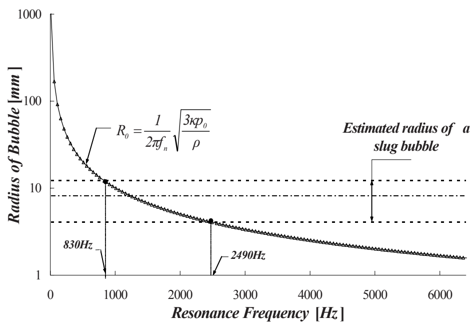


Fig. 6. Resonance frequency of a slug bubble in the evaporator-inlet pipe.

flow pattern can be estimated by Baker's flow pattern map [6], one of the most famous flow pattern maps of horizontal flow. Because the visualization of flow pattern is not conducted, the process to predict the flow pattern in this paper is assumed to be adiabatic, and it may have some uncertainties of identification of flow pattern.

Since, considering cavitation, it is so difficult to estimate the flow pattern, the effect of cavitation is neglected in this paper. According to Baker's flow pattern map, it can be found that the flow pattern varied according to the mass flux of liquid and vapor. The parameters defining flow pattern are given in Eq. (4-7).

$$G_g = \frac{m' x}{A} \tag{4}$$

$$G_l = \frac{m'(1-x)}{A} \tag{5}$$

$$\lambda = \left(\frac{\rho_g}{\rho_{air}} \cdot \frac{\rho_l}{\rho_{water}}\right)^{1/2} \tag{6}$$

$$\psi = \frac{\sigma_{water}}{\sigma}\left(\frac{\mu}{\mu_{water}} \cdot \frac{\rho_{water}}{\rho_l}\right)^{1/3} \tag{7}$$

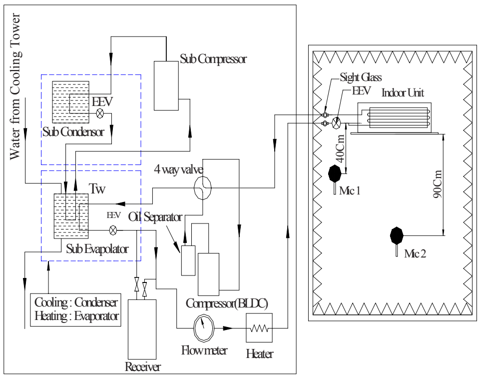
Here, m' is the mass flow rate of the refrigerant, x is quality, A is the cross sectional area of the tube, G_g is the mass flux of the gas, G_l is the mass flux of the liquid, ρ_g,ρ_l,ρ_{water} are the density of the gas, liquid and water, μ_g, μ_l, μ_{water} are the viscosity of the gas, liquid and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

In this paper, the flow pattern by Baker map is identified under the following conditions.

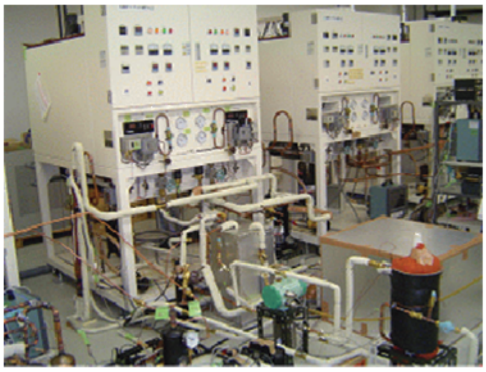
(1) The range of the pressure of refrigerant is from 0.5 to 0.8Mpa at cooling mode and from 1.2 to 2.0Mpa at heating mode.

(2) The range of mass flow rate in a tube is from 100kg/hr to 120kg/hr.

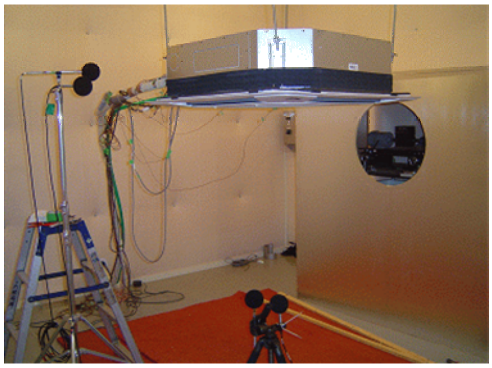
(3) The range of quality is from 0 to 0.2.



(a) Schematic diagram of he outdoor unit simulator



(b) Outdoor unit simulator



(c) Semi anechoic chamber and indoor unit

Fig. 7. Photographs of the simulator.

4. Experiments

To clarify the variations in noise for various conditions of outdoor units at the heating mode, the cycle simulator of an outdoor unit is applied for a 5.2kW 4way cassette type indoor unit using R22 refrigerant. The cycle simulator is the test equipment, which acts as an outdoor side unit. It can realize various types of cycle conditions of outdoor side units. With this simulator, the noise test for an indoor side unit is performed at the heating mode at which the noise problem occurs. Fig. 7 shows the schematic diagram and picture of this simulator.

Samsung Yokohama Research Institute in Osaka developed this simulator, and all of the tests in this study were performed at the Samsung Yokohama Research Institute.

This simulator can control the cycle conditions by changing the BLDC compressor frequency, the open-step of the electric expansion valve, the refrigerant weight in the receiver tank, the heater power, the water temperature in the cooling tower, and the indoor side air temperature.

In the previous paragraph, it was found that the flow patterns such as slug and plug flow cause time-variant refrigerant-induced noise at the heating mode. Therefore, it is very important to find the conditions in which slug or plug flow does not occur at any 2-phase conditions in a tube. In this paragraph, these conditions will be found experimentally by using the cycle simulator at special cycle conditions.

4.1 Modification of pipe diameter

When the air conditioner is operating at heating mode, the indoor unit serves as a condenser. It transfers the refrigerant from super-heated vapor to sub-cooled liquid at normal cycle conditions. But, in some special conditions, the capacity of the heat exchanger may not be enough to condense the refrigerant completely, as discussed in paragraph 2. At this time, the refrigerant flowing out from the condenser can be 2-phase fluid and it produces abnormal noise when passing through the condenser outlet pipe.

Fig. 8 shows the variation of the flow pattern according to the different pipe diameter at the condenser-outlet. In Fig. 8, it can be seen that the flow pattern varies according to the pipe diameter even though it has the same mass flow rate. According to the flow pattern map, it can be supposed that the flow pattern of the refrigerant at the current condenser-outlet pipe (d=9.52mm) is slug flow if the quality is 0.2-0.4.

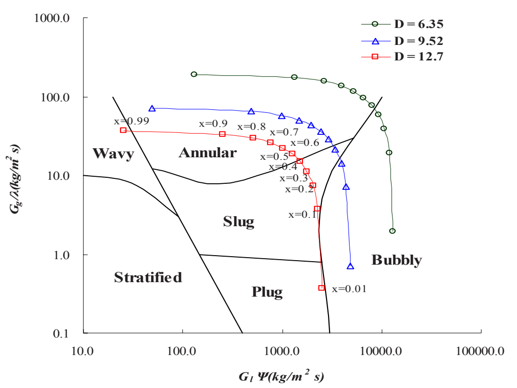
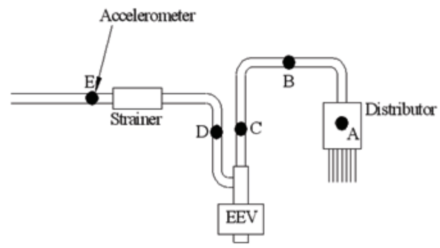


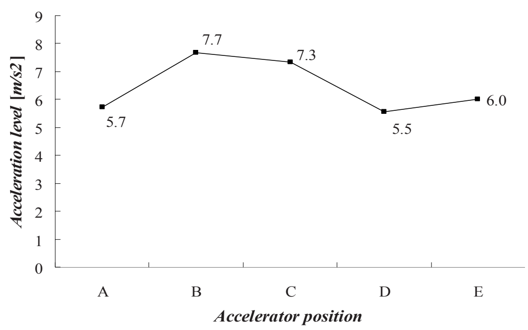
Fig. 8. Variation of flow pattern with pipe diameter(mass flow rate = 120kg/hr, pressure = 2.7Mpa).

To find the location where the vibration level is serious for a conventional condenser-outlet pipe at the heating mode, the accelerations were measured on its various positions as shown in Fig. 9(a). The accelerations were measured with B&K type 4393 accelerometer and Pulse 3560C. When the vibration test was performed, the vibration level was higher around positions "B" and "C" rather than around the other positions, as shown in Fig. 9(b). Here, the root mean square value of the acceleration at axial, radial and tangential direction is used as the measure of acceleration level when pipe diameter is 9.52mm.

When the flow pattern map in Fig. 8 is considered, the slug flow at the condenser-outlet pipe should be avoided by applying a smaller pipe diameter than that of the current one. When the vibration test results in Fig. 9 are considered, it can be estimated that the noise from slug flow occurs seriously at the pipe between the distributor and the electric expansion valve. Thus, the pipe diameter from the distributor to the electric expansion valve is changed from 9.52mm to 6.35mm, as shown in Fig. 10. Additionally, in order to transfer slug flow to bubbly, a strainer is also installed between the distributor and the electric expansion valve. Fig. 10 shows the strainer applied in this paper. For the strainer, porous metal with a pore size of 60 ppi (pore per inch), is applied.



(a) Vibration measuring position of the condenser-outlet pipe



(b) Acceleration level of each position

Fig. 9. Acceleration measured at condenser-outlet pipe.

Through the noise test with the cycle simulator, changing the pipe diameter and adding the additional strainer can reduce the refrigerant-induced noise, as shown in Fig. 11. Table 1 shows the cycle data when performing the noise test. In Table 1 and Fig. 11, Test 1 represents a test with a conventional indoor unit for condenser-outlet pipe, and Test 3 with a modified indoor unit applying 6.35mm pipe and adding an applying 9.52mm pipe for condenser-outlet pipe, Test 2 with a modified indoor unit applying 6.35mm pipe additional strainer for the condenser-outlet pipe. In Fig. 11, the noise components at a range of the frequency from 1kHz to 4kHz are reduced considerably when the diameter of the pipe is reduced from 9.52 to 6.35mm and the strainer is applied. Specially, the time varying abnormal refrigerant noise disappears when 6.35mm pipe is applied, as shown in Fig. 12. Through these noise test results, reducing the diameter of the tube at the condenser-outlet is very effective in reducing refrigerant-induced noise even though the state of the refrigerant at the condenser outlet is 2-phase.

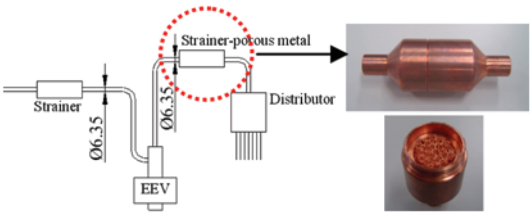


Fig. 10. Strainer with porous metal.

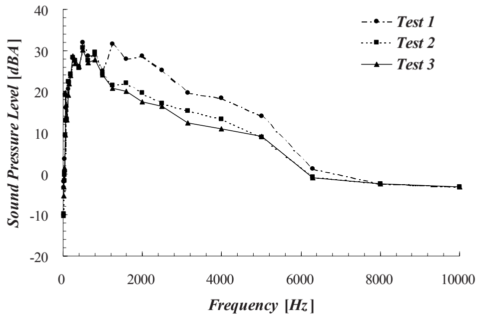


Fig. 11. 1/3 octave spectra with different diameter of condenser-outlet pipe.

Table 1. Cycle conditions for the noise test for different pipe structure.

Factor(unit)	Value		
	Test 1	Test 2	Test 3
Discharge pressure(kPaG)	2059	2094	2054
Suction pressure(kPaG)	495	511	486
Mass flow rate(kg/hr)	118.75	120.84	118.59
Sub-cooling (℃)	0	0	0
Refrigerant phase at condenser-outlet	2	2	2

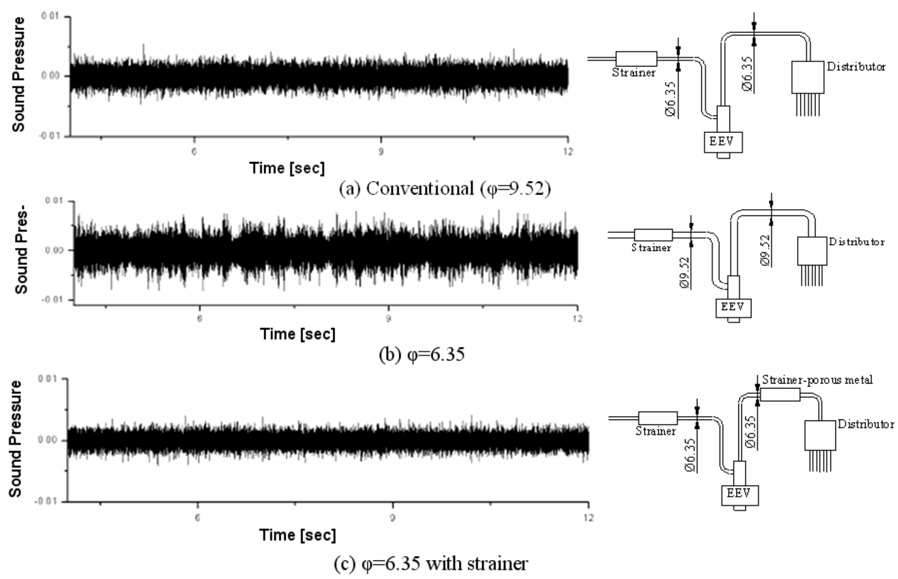


Fig. 12. Time signals of acoustic pressure with different diameter of condenser-outlet pipe.

4.2 Modification of mass flow rate and quality at the condenser-outlet

To verify whether the reduction of the noise in the previous paragraph is effective at various cycle conditions, a noise test was performed with the cycle simulator at different mass flow rates and qualities.

It is possible to change the mass flow rate of the refrigerant by changing the frequency of the BLDC compressor and the open-step of the electric expansion valve for the simulator.

It is also possible to change the quality of the refrigerant at the condenser-outlet by changing the refrigerant weight in the receiver of the simulator. A flow meter is used with the simulator in order to measure the mass flow rate, and the state of the refrigerant can be monitored by sight glass. The quality at the condenser-outlet can be estimated as shown below.

If the p-h diagram can be moved parallel to the horizontal direction as shown in Fig. 13, the capacity of the heat exchanging should be constant at the condenser (indoor unit) for the heating mode. The heat exchanging capacity can be approximately measured with the temperature difference between the air-inlet and outlet temperature of the indoor unit from the following equation.

$$Q_1 = m'_{ref}\,(h_2 - h_1) \tag{8}$$

$$Q_2 = m'_{air}\,C_p\,(T_{inlet} - T_{outlet}) \tag{9}$$

$$Q_1 = Q_2 \tag{10}$$

In Eqs. (8-10), Q_1 is the heat exchanging capacity at the refrigerant side, Q_2 is the heat exchanging capacity at the air side, $m`_{ref}$ is the mass flow rate of the refrigerant, $m`_{air}$ is the mass flow rate of the air, h_1 is enthalpy at the condenser-outlet, h_2 is enthalpy at compressor discharge, C_p is the specific heat of air at constant pressure and T_{inlet}, T_{outlet} are the air temperature of inlet and outlet for the indoor unit.

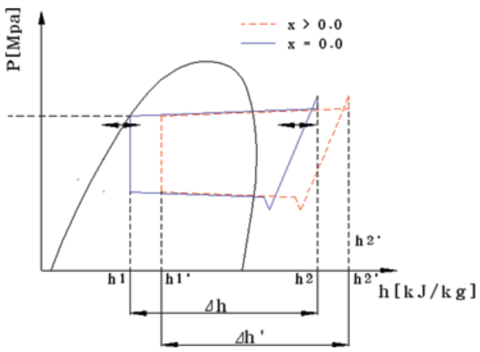


Fig. 13. Parallel movement of the p-h diagram to the enthalpy axis.

Hence, at first, the average temperature difference between the air-inlet and outlet of the condenser is measured by using a thermocouple array when the state of the refrigerant at the condenser-outlet is 1-phase. At this time, because the refrigerant at the condenser-outlet is either a saturated or a sub-cooled liquid, the enthalpy can be known. In addition, the enthalpy at the discharge of the compressor can also be known with the temperature and pressure at the discharge pipe.

Therefore, the heat exchanging capacity can be determined from Eq. (4). When h_2 is moved to the right in Fig. 13, which denotes h_2' and the temperature difference between the air-inlet and outlet of the condenser is the same as that measured at first, the enthalpy difference h_2'-h_1' should be the same as h_2 - h_1 in Fig. 13. Therefore h_1' and the quality of the refrigerant can be calculated from Eqs. (11-13) when the state of the refrigerant at the condenser-outlet is saturated 2-phase state.

$$h_2 - h_1 = h_2' - h_1' \tag{11}$$

$$h_1' = h_2' - \frac{Q_2}{m_{ref}} \tag{12}$$

$$x = \frac{h_2' - h_f}{h_g - h_f} \tag{13}$$

Here, h_f is enthalpy of saturated liquid, h_g is enthalpy of saturated gas, and x is quality at the condenser-outlet.

Table 2 shows the cycle conditions for noise tests at different mass flow rates and qualities. In Table 2, tests 4, 5 and 6 denote the tests when the quality at the condenser-outlet is 0.0, 0.06 and 0.13, respectively, for 120kg/hr mass flow rate. Tests 7, 8 and 9 denote the tests when the quality at the condenser-outlet is 0.0, 0.06 and 0.16, respectively, for 100kg/hr mass flow rate.

Fig. 14 shows the variation of overall sound pressure level with different mass flow rate and quality. Fig. 15 shows the variation of the 1/3 octave band spectrum. In Figs. 14 and 15, the sound pressure level is growing when the quality increases. But the time-

Table 2. Mass flow rate and quality for the noise test.

Factor(unit)	Noise test for 2-phase flow					
	Test 4	Test 5	Test 6	Test 7	Test 8	Test 9
Target mass flow rate	120	120	120	100	100	100
Quality at condenser out	0.0	0.06	0.13	0.0	0.06	0.16
The phase of the refrigerant at condenser out	1	2	2	1	2	2

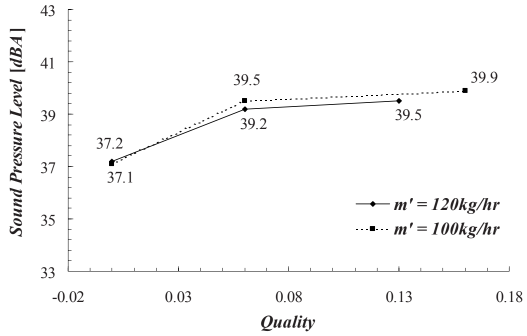


Fig. 14. The variation of overall sound pressure level with different mass flow rate and quality.

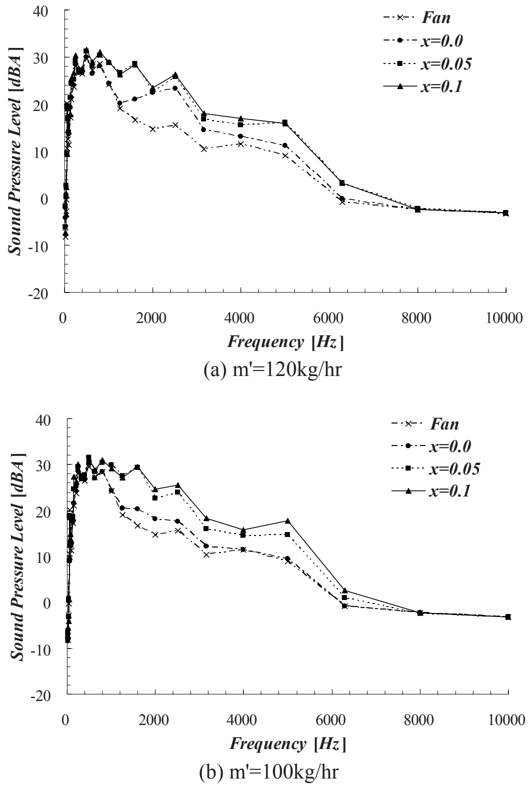


Fig. 15. The variation of sound pressure level spectra with different mass flow rate and quality.

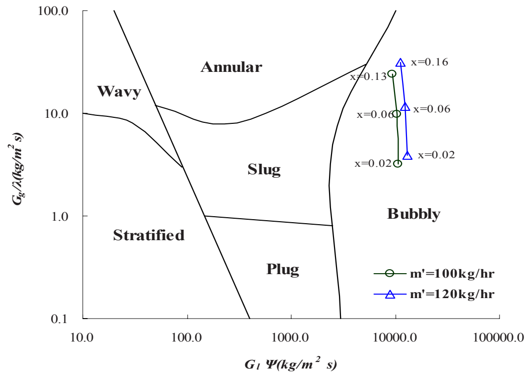


Fig. 16. Flow pattern with different mass flow rate and quality.

varying abnormal noise does not seriously occur in this test.

Fig. 16 shows the flow pattern estimated from the flow pattern map at this cycle condition with the Baker map. From Fig. 16, the slug flow does not occur. Therefore, according to the Baker map and noise test, it can be verified that a reduction of the pipe diameter should be effective in avoiding slug flow and it can reduce time variant refrigerant-induced noise.

4.3 Verification of simulator results with outdoor unit of 5HP system multi type air conditioner

To verify the results from the simulator in paragraphs 4.1 and 4.2, a noise test of a 4-way cassette type indoor unit was performed with the outdoor unit of 5HP system multi type air conditioner as shown in Fig. 17.

Two indoor units of the 4-way cassette type were established in the anechoic chamber, as shown in Fig. 17. They were applied with a different type of condenser-outlet pipe, which has a different pipe diameter as shown in Table 3. Noise tests were performed at different test conditions for cooling and heating modes. When the air conditioner was operating at heating conditions, the time varying abnormal noise from the slug flow occurred for the conventional one. But, the same condition (the improved one), which reduced the pipe diameter, did not produce this kind of noise. Table 4 shows the sound pressure level at different environmental conditions. In Table 4, the sound pressure level for the 2-phase condition is reduced by 1.3dB at the heat overload condition.

Table 3. Specification of evaporator-inlet pipe for test samples.

Type	Outer diameter of condenser outlet pipe(mm)		Additional strainer
	In Flare-EEV	EEV-Distributor	
Conventional	9.52	9.52	×
Improved	6.35	6.35	○

Table 4. Noise test results.

Temperature condition, ISO standard (Indoor / Outdoor)	Sound pressure level [dBA], 1.5m under the indoor unit			
	Conventional		Improved	
	2 phase	1 phase	2 phase	1 phase
20℃, 58.4%RH / 7℃, 86.7%RH (Heat standard)	38.4	35.6	37.7	36.3
27℃, 25.1%RH / 24℃, 55.3%RH (Heat overload)	38.2	37	36.9	36.6
21℃, 52.8%RH / 21℃, 52.8%RH (Cooling low)	39.1	37.3	40	38.5
27℃, 46%RH / 35℃, 24.4%RH (Cooling Standard)	39.8	38.3	40.4	38.8

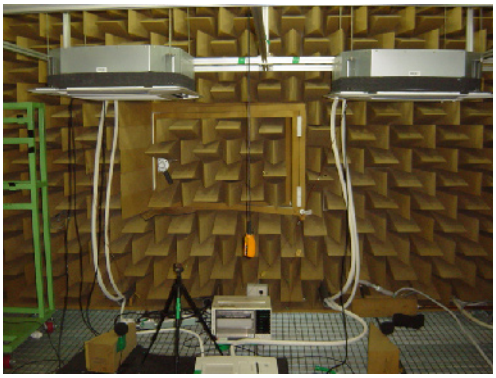


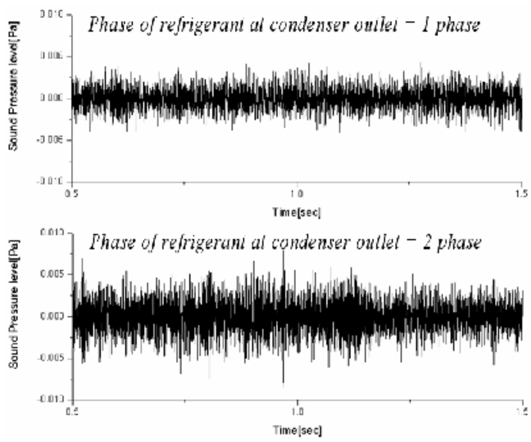
Fig. 17. Noise test for verification of simulator results with 4-way cassette.

Fig. 18 shows the time signal of sound pressure level when the state of the refrigerant at the condenser-outlet is 1-phase and 2-phase at heat overload conditions. In Fig. 18, the refrigerant-induced noise of the improved one is not varied so much from that of the conventional one according to the phase difference.

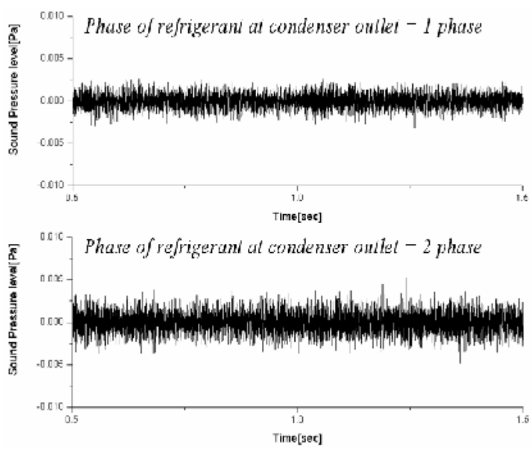
But when the air conditioner was operating at cooling conditions, the noise level was increased in direct opposition to that at the heating condition for the sample of reduced diameter.

Table 5. Cooling and Heating performance for reducing diameter of condenser-outlet pipe.

Mode	Items	Conventional	Reducing diameter
Cooling	Total Capacity(kW)	13.06	13.11
	Total Input(kW)	5.6	5.59
	COP(kW/kW)	2.33	2.37
	EEV open step (Full step =480step)	158	162
Heating	Total Capacity(kW)	15.76	16.16
	Total Input(kW)	5.43	5.66
	COP(kW/kW)	2.9	2.86
	EEV open step (Full step =480step)	479	479



(a) Conventional indoor unit



(b) Improved indoor unit

Fig. 18. Time signal of the sound pressure level for conventional and improved indoor units.

This was because the velocity of the refrigerant in the cooling mode was increased by reducing the diameter. Nonetheless, there was no abnormal noise.

Consequently, the reduction of refrigerant-induced noise by reducing the diameter of the condenser-outlet pipe can be verified, and it can be determined that the results of the simulator are very well in line with those of the air conditioner.

Because reducing the diameter can make the pressure drop increase, performance and reliability can both be problems. Nevertheless, because the electric expansion valve controls the cycle condition compensating for the pressure drop caused by reduction of the diameter, it was no problem when the performance and reliability tests were conducted. Table 5 shows the test results for the performance at cooling and heating mode. As shown in Table 5, EEV open step is increased about 4 steps for the diameter reduced sample compared to conventional one at cooling mode. To verify the reliability at cooling and heating overload conditions, a performance test is performed at the condition that the length of the connection pipe between indoor and outdoor unit is increased (70m) and the height between indoor and outdoor unit is increased also(30m). This test is conducted to check whether the air-conditioner unit has some problems for operating at high pressure drop condition. Through this reliability test, it can be found that there is no problem for operating even though condenser outlet pipe is reduced from 9.52 to 6.35mm.

5. Conclusions

The variation of noise for an air-conditioner at the heating mode was measured and evaluated by changing the diameter of the condenser-outlet pipe and adding a strainer in order to avoid slug flow. Through these tests, the following results were obtained.

(1) Refrigerant-induced noise could be estimated from the flow pattern map at a given cyclic condition.

(2) Noise test results are well coincident with those expected by the Baker map.

(3) Increasing of slug flow at the condenser-outlet could cause the refrigerant-induced noise.

(4) Applying a smaller diameter of pipe could reduce the refrigerant-induced noise by reducing the slug flow at the condenser-outlet.

(5) An additional strainer between the distributor and expansion valve could transit the flow pattern from slug to bubbly, which derived the reduction of the refrigerant-induced noise.

Acknowledgment

This research was financially supported by the Ministry of Education, Science Technology (MEST) and Korea Industrial Technology Foundation (KOTEF) through the Human Resource Training Project for Regional Innovation.

References

[1] S. Hirakuni, Noise Reduction Technology Caused by Refrigerant Two-Phase Flow for Room Air-Conditioner, *Japanese J. Multiphase Flow*, 18 (1) (2004) 23-30.

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

[3] T. Umeda, Reduction of Noise Caused by Gas-Liquid Two-Phase Refrigerant Flow Through an Expansion Valve, *JSME*, 59 (557) (1993) 243-248.

[4] T. Kannon, Study on Noise Caused by Slug Flow through a Capillary Tube, *JSME*, 63 (611) (1997) 2392-2397.

[5] T. Umeda, Noise Caused by Gas-Liquid Two-Phase Flow with Single Large Gas Bubble through an Orifice (1st Report, Experimental Study Using Air-Water Two Phase Flow), *JSME*, 60 (574) (1994) 56-63.

[6] O. Baker, Design of Pipe Lines for Simultaneous Flow of Oil and Gas, *Oil and Gas Journal*, 53 (1954) 185-190.

[7] P.B. Whalley, *Two Phase Flow and Heat Transfer*, Oxford University Press (1999).

[8] D. Chisholm, *Two-phase Flow in Pipelines and Heat Exchangers*, George Godwin, London (1983).

[9] M. Strasberg, Gas Bubbles as Source of Sound in Liquids, *Journal of the Acoustic Society of America*, 28 (1) (1956) 20-27.

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

[11] H. S. Han, Aoyama, Shigeo, Analyzing for Refrigerant Induced Noise for Split Type Air Conditioner Indoor Unit, *KSNVE*, 16 (3) (2006) 240-246.

Hyung-Suk Han received a B.S. degree in Production and Mechanical Engineering from Pusan National University in 1996. He then went on to receive his M.S. and Ph.D. degrees from Pusan National University in 1998 and 2007, respectively. Dr. Han is currently a Senior Researcher at Defense Agency of Technology and Quality, Pusan, Korea. He is currently serving as a Co-Researcher of Noise and Vibration Analysis Laboratory in Pusan National University. Dr. Han's research interests are in the area of the mechanical applications of noise and vibration including refrigerant-induced noise.

Wei-Bong Jeong received a B.S. degree in Mechanical Engineering from Seoul National University in 1978. He then went on to receive his M.S. and Ph.D. degrees from KAIST in 1980 and from Tokyo Institute of Technology in 1990, respectively. Dr. Jeong is currently a Professor at the Mechanical Engineering at Pusan National University in Busan, Korea. He is currently serving as an Academic Director of the Korean Society for Noise and Vibration Engineering. Dr. Jeong's research interests are in the area of the measurement and signal processing of noise and vibration, finite/boundary element programming of noise and vibration, fluid-structure interactions and acoustic-structure interactions.