AIN SHAMS UNIVERSITY

FACULTY OF ENGINEERING

AN INVESTIGATION INTO NOISE AND VIBRATIONS OF AIR CONDITIONING UNITS

THESIS

SUBMITTED IN PARTIAL FULFILMENT OF THE REQUIREMENTS FOR THE DEGREE OF MASTER OF SCIENCE IN MECHANICAL ENGINEERING.

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SUMMARY

Air conditioning noise may have adverse effects on the human comfort. Long time exposure to such a noise may cause headaches and fatigue and hence may impair the efficiency of the human being.

The local air conditioner "Koldair" has a high noise level, as compared with the standard specifications, due to its **present design.** While the maximum allowable noise level is about 55 dBA in the living rooms or offices according to the standard specifications, the air conditioner "Koldair " noise level reaches 60 dBA at high fan speed at a distance of 1 m from the unit.

In the present work an experimental study was carried out for measuring vibrations and noise at different operating conditions of each component of the air conditioner in different directions to determine the main sources of the vibrations and noise in the air conditioner.

It was found out that the main sources of vibrations are respectively:

1. The compressor in vertical-and horizontal directions due to the rotating and reciprocating out of balance as well as due to gas pulsations in the system.
2. Unbalance of the axial-and centrifugal flow fans.
3. Vibrations due to the fan motor.

On the other hand the main sources of noise in the air conditioner were found to be **due to:**

1. The axial flow fan, due to air pulsation caused by the blade movements as well as due to pressure-and velocity gradients.
2. The compressor due to its moving parts and due to the electromagnetic hum-and gas pulsations.

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3. The centrifugal flow fan due to air pulsation similar to the axial flow fan but to a much less extent.

4. The fan motor due to electromagnetic hum.

5. The turbulent flow in the air duct (aerodynamic noise due to sharp corners).

Design improvements were carried out to modify the compressor and the fan motor mountings, and to provide acoustic insulation to reduce the vibration-and noise transmission.

The new design of the air conditioner components incorporates the following modifications:

1. Replacement of the original rubber mounting of the compressor by coil springs and rubber washers to reduce the vibration transmission to the drip pan.

2. Use of flexible multi loop copper tubes instead of the original rigid input-and discharge lines.

3. Use of external muffler on the discharge line to absorp gas plusations.

4. Mounting of the fan motor on a rubber pad for better vibration isolation.

5. Lining the air ducts with felt, and providing them with curved bends instead of sharp corners.

6. Providing complete enclosure around compressor and the fan motor using fiberglass for the isolation of noise.

Due to these modifications the overall noise level of the air conditioner attained 53 dBA at high fan speed instead of 60 dBA (original condition with the unit erected in the wall). At low fan speed the noise level was about 50 dBA, instead of 56 dBA.

The expenses of the overall modifications amount to about 3 % the price of the air conditioner.

Further recommendations are:

1. Replacement of the original axial-and centrifugal flow fans by others having larger diameters and large blade areas, which give at the lower fan speed the same discharge rate delivered originally at the higher fan speed. The fans are to be made of plastic materials instead of aluminium and steel, to reduce the out of balance.

2. Fabrication of the drip pan by forging instead of welding to attain higher rigidity and hence lower vibration level.

3. Using a rotary-instead of a reciprocating type compressor which results in lower vibration-and noise levels.

4. Removal of the nuts of the compressor mounting before the erection of the air conditioner in the wall.

Nomenclature

Symbol	Units	Definitions
C	-	Spring index ratio
D	-	Damping ratio
D_o	mm	Outer diamter of spring
d	mm	Wire diameter of spring
F	N	Force
f	HZ	Exciting frequency
fn	HZ	Natural frequency
G	N/mm^2	Modulus of rigidity
i	-	Number of turns of spring
k	N/m	Stiffness
Lp	dBA	Sound pressure level
Lpt	dBA	Total sound pressure level
Lpl	dBA	Background sound pressure level
Lwl	dB	Sound power level (Background)
Lwt	dB	Total sound power level
m	kg	Mass
P	N/m^2	Sound pressure
r	-	Frequency ratio $= \frac{f}{fn}$
S_s	kp/mm^2	Shear stress
T	-	Transmissibility
w	watt	Overall sound power
δ_s	mm	Static deflection.

CONTENTS

INTRODUCTION

Air conditioning units are used for providing a comfortable environment for human beings throughout the year in the presence of sensible heat gains in summer and sensible heat losses in winter. They are widely used in living rooms, offices, hotels, hospitals... etc.

However the human comfort may be disturbed by the noise caused by the air conditioning units which may lead to fatigue and nerval stress to human beings who are exposed for a long time to such a noise.

The locally manufactured "Koldair" air conditioning unit is found to produce a high noise due to its present design as compared with modern air conditioning units, which may affect its marketing and may cause accumulation of the manufactured units in stores.

The aim of the present investigation is therefore to identify the probable sources of vibrations and noise in the Koldair unit and to introduce the possible modifications to reduce its noise level down to the standard specified limits at the lowest possible expenses.

The first stage of the work incorporates experimental measurements and analysis of the vibrations caused by the individual components of the Koldair units. The second stage incorporates noise measurements in the space around the unit as mounted in the wall and as uncovered in bench tests. The third stage includes the introduction of the possible modifications to reduce the vibration-and noise levels of the air conditioning unit. Vibrations and noise measurements were repeated after each modification step. Useful recommendations are presented for further development of the Koldair unit.

CHAPTER 1

REVIEW OF LITERATURE

Vibration-and noise sources

A ccording to previous publications /1/, ..., /9/ vibration and noise of the air conditioners may be produced by several sources such as compressor, fan motor, unbalance of axial-and centrifugal flow fans, paths by which compressor noise and vibrations are transmitted, and aerodynamic noise/1,2,..,9/. Ingalls /2 /found that the noise causes associated with the refrigenating system are:

1. Out of balance of the compressor

   For the ventilator the out of balance tone of 49 HZ and its first harmonic were a cause of discomfort. In these cases the energy of the out of balance tone was radiated from the wall of the ventilator. It was found that the 49 HZ level could reach up to 57 dB measured 3m directly in front of the ventilator in the test chamber. Panels resonance was caused all too readily by partial bridging the isolation provided by the rubber feet which is caused in practice by the following factors:

   a. Compression of the narrow rubber washer at the top of the feet because the spacers were too short. This increased the effective stiffness of the feet.

   b. Side forces acting on the compressor from the copper tube connecting it to the evaporator could be sufficient to compress the rubber between the spacer and the compressor base plate and again modify the stiffness.

c. Contact between the tubes leaving the compressor and the nearby case work.

2. Shell resonance of the compressor dome

This was audible as a narrow band noise in the 2000 HZ octave as with apure tone it was undesireable because it was just audible above the back ground broad band noise. Ingalls / 2 / discussed the design factors based on large hermetic compressors, but pointed out that more research on the subject is still needed.

3. Fan motor noise

The only source of electrical noise was the induction hum at twice mains frequency (100 HZ) caused by torsional oscillations, the noise level could be reduced by modifying the winding.

4. Boiling of the refrigerant leaving the capillary tube

It was found that noise created by boiling was acceptable if:

a. There were no burrs on the cut end of the capillary.
b. That the capillary tube entered centrally into the evaporator entry.

Wilson/ 1/ found out that the sound generated by the compressor may be radiated directly from the steel case (case radiation) or indirectly by exciting other components of the system (structure borne noise). The major paths by which compressor noise is transmitted to the system are:

a. The air surrounding the compressor (case radiation).

b. The refrigerant within the tubing (discharge pressure pulse).

c. The tubing (discharge and suction lines)

d. The mounting feet (vibration)

The control of compressor noise in the application is generally limited to the treatment of these transmission paths. Webb / 4 / mentioned that compressor noise generally can be classified under:

1. Bearing noise: are generally caused by excessive bearing to journal clearance poor surface finish, nicks, scratches, burrs or other foreign material and improper lubrication.

2. Induction motor noise: is a significant factor in the overall noise. Motors having an enclosed bar rotor may produce asignificant amount of lower frequency noise, which is definite "pulsating" or "hunting" sound occurring at harmonics of the electric line frequency and covering the range from fundamental line frequency through about 400-500 HZ.

That noise due to the eccentricity of the air gap between the rotor and the stator (variable air gap) may cause extremely noticeable pulsating noises. Hermetic motors using the open or exposed bar rotor may produce aminimum of the low frequency noise.

Open bar type rotors are more acceptable than motors with closed bar type rotors for use of refrigerators.

3. Gas noise: The refrigerant gas stream is a prolific source of noise.

Also the sources of noise such as:

1. The movement of impeller blades of the fan at high speed

   Having the same number of blades results in a high frequency noise due to sudden air movement at the blade tips.

2. Magnetic hum, all motors produce 100 HZ hum with minor harmonic.

Vibration and noise treatment

Wilson / 1 / showed that the control of compressor noise in practice is limited to the treatment of the following transmission paths:

1. Reduction of mounting vibration

There are three types of vibrations associated with the compressor.

a. Unbalance (rotational, reciprocating).
b. Torque reaction (load generated).
c. Flexural case vibration (audible frequency range).

The first two forms of vibrations occur at low frequency and appear primarily at the first two harmonics. The internal suspension is usually adequate to reduce the vibration to acceptable levels. The third type of vibrations occurs at higher frequencies. It is this vibration that generates the case radiation. The compressor internal spring suspension is ineffective here as much of this energy is transferred to the housing through the refrigerant, the discharge tube, and oil reservoir. Synthetic rubber isolators are applied to the compressor mounting feet and are effective in reducing the transmission of high frequency noise to the unit base. The allowable static deflection of a rubber isolation was limited to about 10% of its free (unload) height. Coil springs may be used to achieve a greater transmission loss at low frequencies.

All comprressor noise is at one or more harmonics of the pumping rate so that frequency alone is not definitive. However, the noise from various causes does tend to be limited to certain frequency range as follows:

a. Case radiation 200 to 10000 HZ.

b. Discharge noise 10 to 1000 HZ.

c. Tubing 55 to 1000 HZ.

d. Mounting 55 to 350 HZ.

2. Reduction of line transmission

The discharge and suction lines are a possible path for conducting compressor noise to the system. The transmission line in this case is the wall of the tube. The desired result of the tube design is flexibility so that the tubing can be lengthened by providing a coil in the horizontal plane around the compressor.

3. Reduction of discharge pulse

The "discharge pulse" is a pressure fluctuation in the superheated refrigerant leaving the compressor. The pressure pulse is confined to the refrigerant gas in the discharge piping and ordinarily does not contribute directly to the case radiation. Its contribution to noise in the refrigerantion system normally occurs because pressure waves in the refrigerant excite the condenser tubing causing vibration of the tubing, the attached unit base and the cabinet pannels. Reduction of the pressure pulse was accomplished by inserting a muffler in the discharge line between the compressor and condenser. The best location for the muffler is usually as close to the compressor outlet as possible to minimize the excitation of the discharge as well as the condenser tubing.

4.Reducing case radiation

The sound waves generated at the surface of the compressor housing are radiated outward and:

a. Reflected about within the walls of the compartment and radiated out through openings in the compartement.

b. Transmit some energy to the compartment panels (cause vibration of the panels) or

c. Some lose energy in the form of heat due to absorption.

The compressor case radiation must be done within the enclosure before the sound is radiated out of the cabinet. This is accomplished by absorption . Acoustical absorbing materials such as fiber glass or an open cell plastic foam is applied to the compressor side of the enclosure panels. Baffles may be placed between the compressor and cabinet opening to reflect a portion of the sound back into the enclosure. The absorption coefficients fall off rapidly for frequencies below 1000 HZ. The primary limitation on the use of absorbing materials is the total area to which the material can be applied. IF 10% or more of the cabinet panel area is open for ventilation the reduction in compressor case radiation is limited to about 3-5 dB (A).

A disadvantage of completely enclosing the compressor is the loss in the performance that results due to the increase of the ambient compressor temperature which increases the temperture of the return refrigerant in the housing. The increased refrigerant temperature entering the cylinder results in less mass flow of refrigerant without an equal decrease in power consumption. The performance loss may range from 1% to 5%. An increase in the condenser surface can be used to reduce the discharge pressure to recover the performance loss by a change in the refrigeration system.

Clark / 2 / showed that sound absorbing foam was added around the air passage and also in the fan casing surrounding the fan blades, (the foam was retained in the latter instance by a strip of perforated aluminium). He also showed that noise reduction was achieved by improving the fan effectiveness as follows.

a. Increasing the solidity of the blading to the extent that the blades occupied virtually all the fan area as viewed along the axis.

b. Increasing the fan diameter.

c. Increasing the depth of the air space behind the heat exchangers.

d. Improve the efficiency of the fan and the flow path so that the same volume of air could be passaged at a lower fan speed.

e. Ensuring that the compressor was freely supported on its rubber mounts (remove nuts).

f. Investigating the flow from the capillary tube into the evaporator entry.

Richardson / 3 / mentioned that the enclosure introduces additional degrees of freedom into the fan isolation system, that obviously complicates the dynamics of the complete system. It is necessary to ensure that the enclosure did