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A Study on the Improvement of the Sound Quality of the Interior Noise of A/T Vehicle in Idle State
공 회전시 자동변속기 차량의 실내소음 음질 개선에 관한 연구

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ABSTRACT

본 논문은 자동변속기를 탑재한 차량에서 에어콘(air condition)을 작동시키고, 공 회전시 기어의 변속을 "D"단에 두었을때 실내에서 발생하는 이상음의 원인 규명 및 해결 방법에 관한 연구 결과를 논하고저 한다. 이 이상한 소음의 원인을 규명하기 위하여 실린더 내부의 연소 압력, 메인 베어링캡(main bearing cap)의 진동, 엔진 마운팅 보스의 진동 및 차량의 실내소음을 동시에 측정하여 분석하였으며 이 결과에 의하면 이상음의 원인은 크랭크샤프트(crank shaft)의 굽힘진동이 파워플랜트(power plant)를 가진하여 진동을 증가시키고, 이 진동이 마운팅 보스를 통하여 차량의 차체에 전달되며, 차체의 진동에 의해서 발생하는 고체 전달음(structure-borne noise)이었다. 또한 이상음의 주기는 주파수 성분은 200-400Hz 이었다. 이 이상음은 크랭크 샤프트의 댐퍼 풀리의 질량을 저감하여 크랭크 샤프트의 동특성을 개선 함으로서 해결 가능하고, 혹은 점화시기를 지연하여 연소 압력을 낮춤으로서 해결 가능하다.

주요기술용어 : Subjective Evaluation(주관적 평가), Objective Evaluation(객관적 평가), Transmisson Path(전달 경로), Aluminum Pulley(알루미늄 풀리), Crankshaft(크랭크샤프트)

1. INTRODUCTION

Recently the demand for cars with a comfortable noise and vibration level is strongly increasing. Not only during acceleration but also at idle speed, the noise level in the passenger compartment is animportant determinant of occupant comfort. On the other hand, the improvement of subjectively perceived sound quality is gaining importance more and more.(1)(2)(3)*

In general, the vibration level of the engine and power train (ENG+T/M) is strongly influenced by the vibration behavior of the crank shaft.(7)(8)(9)(10)* The three dimensional vibration behavior, especially bending vibration, is excited by the inertia and gas forces of individual cylinders resulting in impulse forces on the main bearings of the engine. These impulse forces bring about

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the impulse vibration on the power train to induce the structure-borne noise in the passenger compartment. Hence, it is very important to control effectively the crank shaft system to reduce structure-borne noise in the passenger compartment.

Over the years much research has been carried out with the aim of reducing the compartment noise level and improving the sound quality during acceleration.(2)(4)(5)(6)* In most previous studies, dominant attention has focused on sound quality control only during acceleration by applying new mechanisms(like flexible flywheel, dual-mode damper pulley) to the crank shaft system.(5)(6)*

This paper describes the method of improving the sound quality at idle speed by increasing the stiffness of the pulley-side bending vibration of the crank shaft and by optimizing combustion forces. The former was done by applying an aluminum pulley to the crank shaft system, the latter by controlling the ignition timing of the engine.

2. BACK GROUND OF THE STUDY

In the passenger compartment of a car, anuncommon sound with half-order interval of engine rpm occurred when the transmission was in "drive" range and the air conditioning was operating and the vehicle speed was idle. The condition did not occur when the transmission was in natural(N) and the air conditioning was not operating, Another special phenomenon was that the more ignition timing was advanced, the more severe was the roughness of the uncommon sound. Therefore to resolve this problem experimentally, the objective analysis as well as a subjective evaluation were performend on the problem vehicle which has an automatic transmisson(after this, "rough sound" is written instead of "uncommon sound").

3. EXPERIMENTAL VEHICLE AND SUBJECTIVE EVALUATION

Table 1 shows the specifications of the engine used in the experiment.

First, in order to find out the conditions under which the rough sound becomes more severe or less severe and to evaluate the degree of harshness of the sound, the subjective sound evaluation in compartment was performed on the problem vehicle under the 5 conditions shown Table 2.

According to the subjective sound evaluation results from the test conditions shown in Table 2, the degree of harshness of the rough sound depends on combustion forces(controlled by spark timing) and the load of the air conditioner.

Nest, to identify that the combustion force of

Table 1 The Specifications of The Test Engine

Engine Type	In Line 4 Cylinder
Valve System	SOHC 12 Valve
Firing Order	1−3−4−2
Bore × Stroke	77.5mm × 83.5mm
Total Piston Displacement	1495cc
Compression Ratio	10.0
Base Spark Timing	BTDC 10°
Maximm Torque	13.4Kg.m/4500 rpm
Maximm Power	92 PS/5500 rpm

Table 2 Conditions of Vehicle for Noise and Vibration Test

NO	TEST CONDITIONS OF VEHICLE			NOISE EVALUATION (INSIDE OF VEHICLE)
	GEAR	A/C STATE	IGN. TIMMING	
1	"N"	OFF	BTDC 10°	No HARSHNESS
2	"D"	OFF	BTDC 10°	No HARSHNESS
3	"D"	ON	BTDC 3°	SLIGHT
4	"D"	ON	BTDC 10°	MEDIUM
5	"D"	ON	BTDC 15°	HARSH

which cylinder seriously effects the rough sound, the subjective sound evaluation was estimated by removing the ignition plug of each individual cylinder in turn. From this kind of test, it was determined that the the rough sound in passenger compartment disappeared with no firing of #1 cylinder.

Therefore, it is identified that the rough sound abruptly increased in synchronization with firing of #1 cylinder.

4. OBJECTIVE EVALUATION AND TEST EQUIPMENT SET UP

Normally, the sound quality and vibration at idle speed is govemed by the 2nd order component frequency of engine rpm. However this rough sound was an impulse sound modulated by high frequency components. So, in order to identify the frequency bands of the noise, the transmisson path of vibration and the excitation force causing the vibration, under the same conditions as Table 2, the following measurements and analysis were done in the time domain as well as in the frequency domain.

1. the interior sound pressure measurement at the front seat.
2. the acceleration measurement at the engine mount position
3. the acceleration measurement at the body of the vehicle
4. The acceleration measurement at the main bearings of the engine
5. the gas pressure measurement of each cylinder

Besides tests under engine running conditions corresponding to those shown in Table 2, a rig test also should be conducted to identify the transfer functions of structures in vibration transmisson path and the dynamic characteristics of the crank shaft system by the impact hammering test. Fig. 2 shows the measurement set up for the transfer function analysis of structures and the crank shaft system.

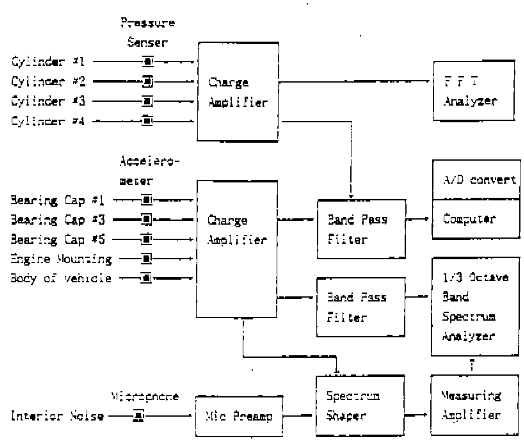

Fig.1 The Measurement Equipment Set Up

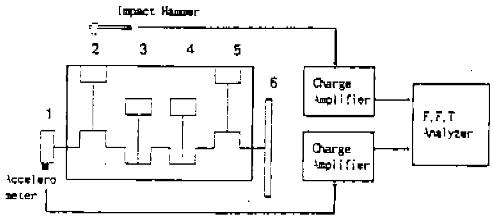

Fig.2 The Apparatus for Transfer Function Measurement.

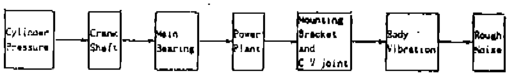

Fig.3 The Paths of Structure Borne Noise

5. INTERIOR NOISE AND VIBRATION MEASUREMENT OF THE VEHICLE

To obtain information about the frequency band of the rough sound, a microphone has been installed at the front seat in the passenger compartment. The sound pressure signal detected by the microphone has been transmitted into the 1/3 octave band frequency spectrum analyzer(B&K-2134) and the audio sound through the audio amplifier and spectrum shaper which has a 1/3 octave band pass filter function. Tests were conducted under No. 3 test conditions of Table 2.

It was observed that the signals displayed on the monitor of the frequency spectrum analyzer has 5~8dBA variation(impulseness) in 200~400 Hz band. Table 3 shows the variation of the values of interior sound pressure at idle rpm. By more careful observation, it was discovered that the sound pressure levels in the 200~400Hz band jump up abruptly and jump down with the half-order interval of engine rpm in synchronization with the ignition of cylinder #1. In order to confirm the correlation between the filtered signal and the rough sound, a sound comparison test was conducted. The object of this kind of test is to listen and to compare both sounds with each other subjectively. One sound i the signal filtered by the spectrum shaper and amplified by the audio amplifier ; another sound is the real sound in the passenger compartment. The sound color of both sounds was nearly similar. In addition, the acceleration of the vehicle was sharply increased and decreased by about 6~10dBA, in synchronization with the ignition of cylinder #1.

Therefore, it was objectively confirmend that the rough sound in the compartment was structureborne nosie by vibration of vehicle. That is, the rough sound is the wave energy which can generate the sound in the 200Hz~400Hz band of 1/2 order frequency of engine rpm and the transmisson path of nosie and vibration is shown in Fig. 3

6. GAS PRESSURE MEASUREMENT

To identify the relation between the rough sound and the gas pressure of cylinder #1. Each engine cylinder was equipped with piezoelectric pressure transducers. The pressure signals were amplified and transmitted into the FFT analyzer (HP3562A). It was triggered by the ignition of cylinder #1, A/D converted and averaged for 100 cycles (1 cycle is 720 degrees of crank angle) in the time domain and in the frequency domain. Fig. 4 to Fig. 8 show averaged cylinder pressures

Table 3 The Sound Pressure Variation at the Problem Frequency

Center Freq (Hz)		200	250	315	400
Interior Noise (dBA)	Max.	60.	53.	53.	47.
	Min.	52.	48.	46.	42.

of each cylinder under the test conditions of Table 2. Table 4 gives the information about the maximum gas pressure levels of each cylinder.

In these thest results, since the gas pressures of #1 cylinder is only slightly higher than those of the other cylinders, this rough noise was not generated by any other special combustion or unreasonable firing of #1 cylinder. However, by comparison between Fig. 5 and Fig. 7, as the load of the air conditioner increases the gas pressures of each cylinder also increase over nearly the whole frequency band. Especially in the 200~400 Hz band the gas pressure increase level is higher than that of other frequency bands. The pressure increase ratio vs crank angle has been shown in the time domain analysis. It also increases with the load of the air conditioner.

Another well known phenomenon is that the gas pressures and the rate of gas pressure increase(dP/dO) have increased with advancing of spark timing. Therefore, only by retarding the spark timing, without any other hardware change of engine, ti is possible to reduce the interior noise level and vibration level and to improve the sound quality. However there is some limit of spark timing retard needed to maintain the idle stability and to overcome the friction fo the engine needed to run at idle speed.

Table 4 The Specifications of The Test Engine

Cylinder No. Test Con.S/Timing		#1	#2	#3	#4
"D" Range Air Conditioner "ON"	3°	11.11	9.63	7.82	10.61
	10°	16.05	14.56	12.38	14.82
	15°	18.69	16.81	15.38	17.00
"D" Range Air Conditioner "OFF"		8.76	7.45	6.89	9.32
"N" Range Air Conditioner "OFF"		5.17	5.18	4.92	6.13

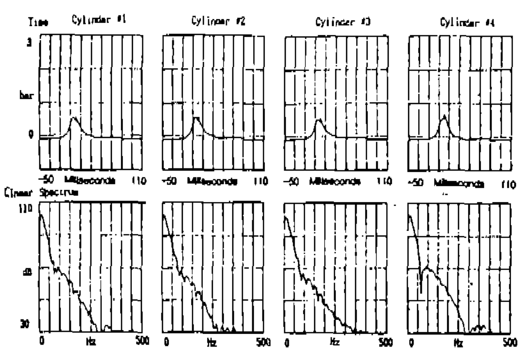

Fig.4 The gas pressure of each cylinder at idle speed : "N" range and air conditioner "OFF", spark timing is BTDC 10°

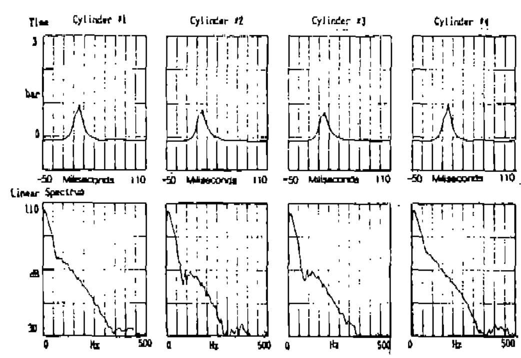

Fig.5 The gas pressure of each cylinder at idle speed : "D" range and air conditioner "OFF", spark timing is BTDC 10°

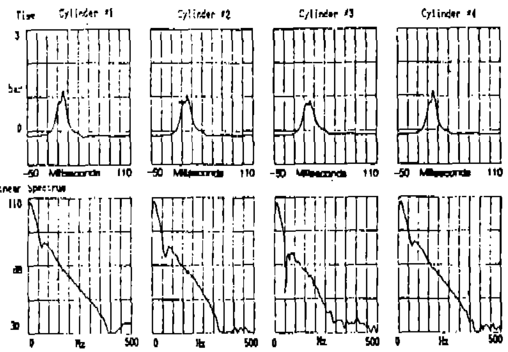

Fig.6 The gas pressure of each cylinder at idle speed : "D" range and air conditioner "ON", spark timing is BTDC 3°

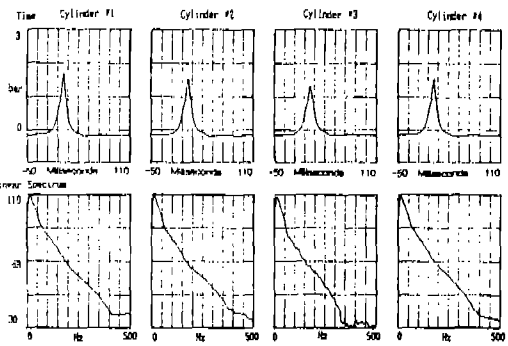

Fig.7 The gas pressure of each cylinder at idle speed : "D" range and air conditioner "ON", spark timing is BTDC 10°

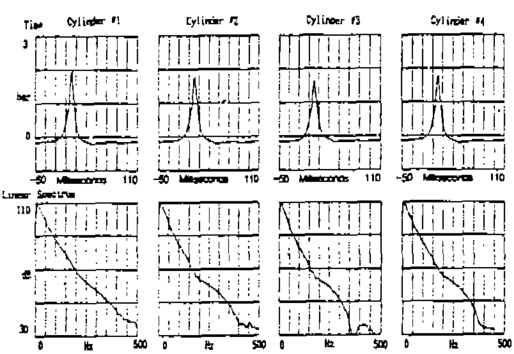

Fig.8 The gas pressure of each cylinder at idle speed : "D" range and air conditioner "ON", spark timing is BTDC 15°

7. THE VIBRATION MEASUREMENT OF TRANSMISSON PATHS

been shown in Fig. 3. In the middle of it, if there is some resonance of the structures the vibration amplitude of the structure and the rough sound in compartment will be strongly influenced by it. In order to find out whether there is any resonance phenomenon in the transmisson paths the piezoeletric acceleration pick up was mounted at several locations on the engine(#1, #3, #5 main bearing caps and the front engine mount) and the acceleration was measured. The measured signals were amplified, A/D converted and transmitted into the computer. This data was analyzed, synchronized with the other data(gas pressure and interior noise) in the time domain. The analysis of the time domain gives a lot of information, because by spectrum analysis of data from the previous section, the impulse signal data are modulated with the 200~400Hz band and it has decayed with strong damping Fig. 9 to Fig. 11 Show the comparison of signals among the sound pressure in compartment, acceleration of the engine mount, acceleration on the main bearing caps and gas pressure of #1 cylinder in the time domain. Each signal has been trigged by the ignition of #1 cylinder and have been recorded for 1 cycle(720 degrees of crank angle). All signals were filtered through a band pass filter in 200~400Hz band except for the gas pressure. Advancing the spark timing t0 BTDC 15°(original is BTDC 10°) increases the vibration and noise at each measuring position. In contrast, retarding the spark timing to DTDC 3° decreases it. In addition, since the vibration of #1 main bearing is much higher than those of other main bearings, it was assumed that there is a severe resonance in crank shaft system in middle of 200~500Hz band.

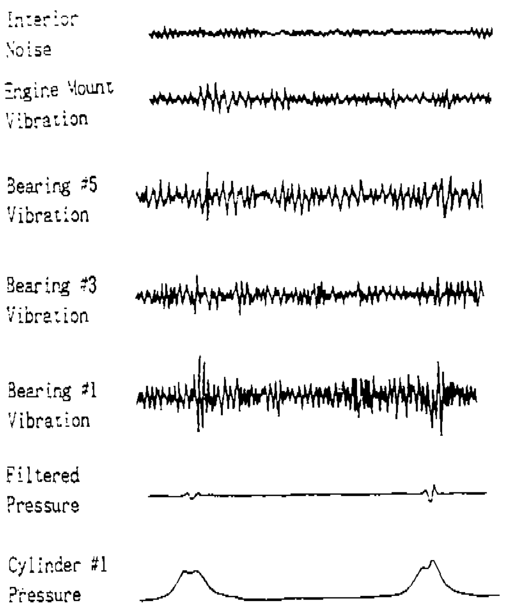

Fig.9 The comparison among noise, vibration and cylinder pressure synchronized with the ignition of #1 cylinder at idle and spark timing of BTDC 3° : original damper pulley.

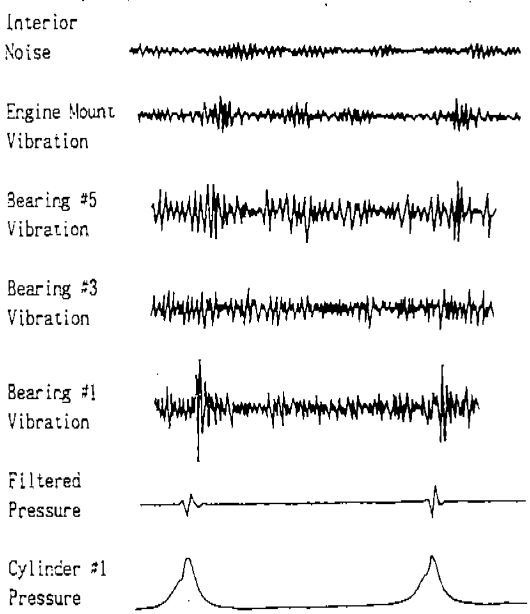

Fig.10 The comparison among noise, vibration and cylinder pressure synchronized with the ignition of #1 cylinder at idle and spark timing of BTDC 10 ° : original damper pulley.

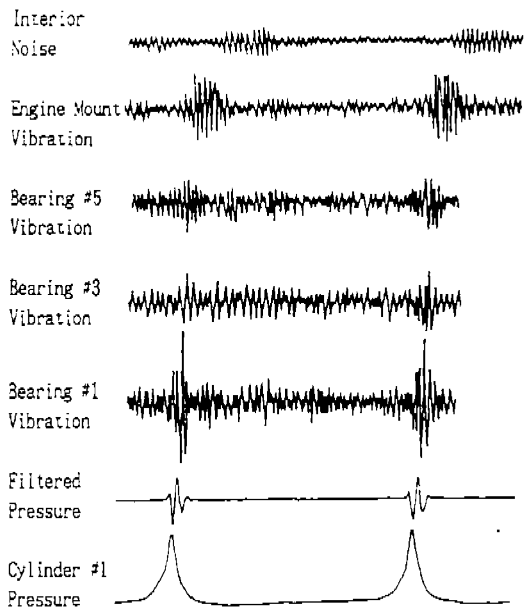

Fig.11 The comparison among noise, vibration and cylinder pressure synchronized with the ignition of #1 cylinder at idle and spark timing of BTDC 15 ° : original damper pulley.

8. THE DYNAMIC CHARACTERISTICS TEST

To identify the vibration behavior of the crank shaft. A modal analysis of the crank shaft was conducted using the hammering test(Fig. 2). By this test, it was found that there is a resonance of the crank shaft at 418.7Hz, and a transfer function of the crank shaft is shown in Fig. 12. Fig. 13 shows the mode shape of crank shaft at this resonance frequency. From this test result, the crank shaft vibrates with the shape of front pulley side bending in 418.7Hz. Therefore it was concluded that the rough sound in compartment is induced by structural vibration of the body, and this behavior(bending mode) of the crank shaft influences the engine mount vibration resulting in the body vibration. Finally, the behavior of the crank shaft bending vibration and its resonance phenomenon are the main factor causing the rough sound and discomfort in the compartment. To resolve this problem, the bending mode and resonance frequency of the crank shaft system should be modified or changed.

9. APPLICATION THE ALUMINUM PULLEY AND MASS REDUCED CAST PULLEY

To change the dynamic characteristics of the crank shaft bending vibration, there are several met-hods. However considering cost, mass production, durability, and weight, the best method should be decided. In this study, the bending resonance frequency of the crank shaft front end side was increased by mass reduction by application of an aluminum pulley. Fig. 12 shows the comparison of the transfer function between the original cast pulley and the aluminum pulley and Table 5 shows the mass and the resonance frequency of each pulley. The resonance frequency of the cast damper pulley is 418.9Hz and that of aluminum pulley is 670Hz. Therefore, with the application of an aluminum pulley, there is some benefit in the transfer function from 200Hz to 400Hz. It means that the vibration of the structure and the noise in the compartment will be reduced. The interior sound pressure level and the vibration levels of the engine mount and the main bearings and the gas pressure of #1 cylinder are shown in Fig. 14 and Fig. 15 with application of the aluminum pulley. In this case, the filtered signals were not affected very much by the spark timing swing except for the gas pressure of #1 cylinder. Finally to confirm the result of this objective test, the subjective evaluation of the rough sound was done in the compartment of the problem vehicle. The rough sound was not heard in the compartment of the problem car with the application of aluminum pulley. Table 6 gives some information about the peak soud pressure level, gas pressure level and acceleration level of each of the transfer paths for the application of the aluminum pulley and cast iron pulley.

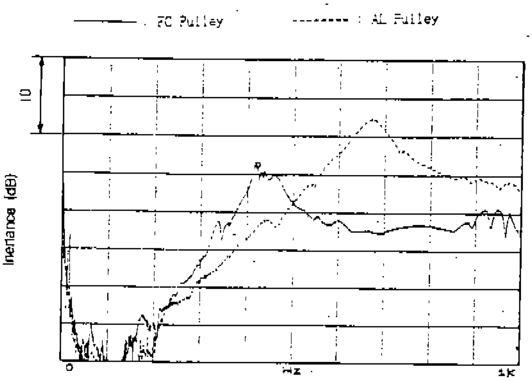

Fig.12 The Transfer Function Comparison Between AL Pulley and FC Pulley

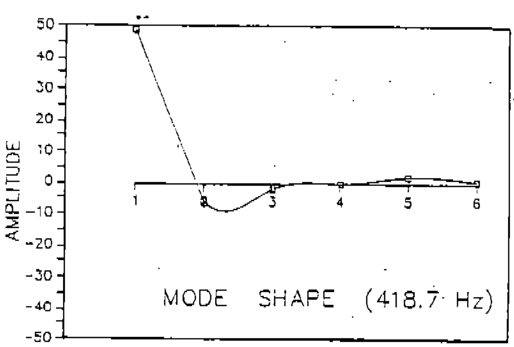

Fig.13 The Mode Shape of Crank Shaft

Table 5 The Mass and Resonance Frequency of Each Pulley

	Original Pulley	Aluminum Pulley
Mass	2100g	882g
Resonance Frequency	418.7Hz	675Hz

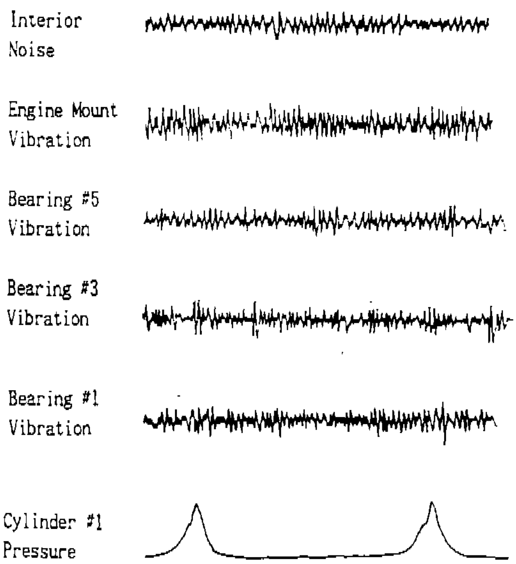

Fig.14 The comparison among noisee, vibration and cylinder pressure synchronized with the ignition of #1 cylinder at idle and spark timing of BTDC 10 ° : aluminum pulley

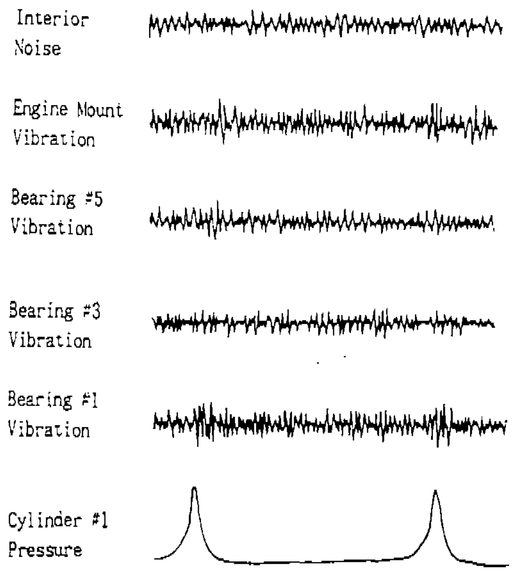

Fig.15 The comparison among noisee, vibration and cylinder pressure synchronized with the ignition of #1 cylinder at idle and spark timing of BTDC 15 ° : aluminum pulley

Table 6 The Noise and Vibration Data Comparison Between AL Pulley and FC Cast Pulley

Measuring Range : 200~400Hz
Vibration Units : m/s^2
Interior Noise Units : dBA

	A1 PULLEY	ORIGINAL DAMPER PULLEY			
Spark Timing	BTDC 10° A/CON ON	BTDC 10° A/CON OFF	BTDC 3° A/CON ON	BTDC 10° A/CON ON	BTDC 15° A/CON ON
BR'G #1	1.25	1.45	2.31	2.98	3.72
BR'G #3	1.15	1.17	1.18	1.94	1.96
BR'G #5	1.05	1.27	1.22	2.02	2.24
ENG Mount	1.14	2.0	4.17	4.4	4.5
IN Noise	60.0	60.5	61.04	63.06	64.01

10. CONCLUSION

(1) The rough sound in the passenger compartment is impulsive sound which is modulated with strong damping and has wave energy of the 200~400Hz frequency band. It has 1/2 order interval of engine rpm and is synchronized with the ignition of #1 cylinder of the engine.

(2) The vibration behavior of the front side pulley causes bending of the crank shaft and does significantly influence the rough-

ness of the sound in compartment of car.

(3) The rough sound in compartment disappears with the application of an aluminum pulley which increases the resonance frequency of the crank shaft front bending vibration by about 61.2%.

(4) The rough sound is also reduced by decreasing the combustion force by using spark timing retard.

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