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SEPARATION AND IDENTIFICATION OF SIMULATED DIESEL ENGINE NOISE USING INDEPENDENT COMPONENT ANALYSIS AND TIME-FREQUENCY ANALYSIS

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ABSTRACT

In this paper the separation and identification of diesel engine noise is investigated using independent component analysis (ICA) and time-frequency analysis (TFA). The separation and identification of engine real noise is important for noise reduction and faults diagnosis. This study can be considered as a guide for engine real noise separation and identification. A diesel engine noise is mathematically modeled and a single cylinder engine noise is simulated. The simulated noise sources are exhaust valve close, inlet valve close, fuel injection, combustion, piston slap, exhaust valve open and inlet valve open. The individual simulated engine noise sources are mixed to represent seven mixed sources signals. The mixed sources signals are considered as acquired signals by seven channels microphone array. The mixed sources signals are separated using ICA. The separated independent components (ICs) are analyzed by short time Fourier transform (STFT) in time-frequency plane. The time is then converted to crank angle in order to identify engine noise sources. As predicted, the results show that the separated engine noise sources well correlated with the engine design specifications.

KEYWORDS

Diesel engine noise, blind source separation, independent component analysis, short time Fourier transform and time-frequency analysis.

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NOMENCLATURE

AE	Acoustic emission
ABDC	After bottom dead centre
ATDC	After top dead centre
BBDC	Before bottom dead centre
BTDC	Before top dead centre
c	Damping constant
c_{oil_c}	Cylinder wall oil damping constant
c_{oil_cr}	Crnkshaft oil damping constant
c_{oil_v}	Valve damping constant of
F	Force
FFT	Fast Fourier transform
FPICA	Fixed-point independent component analysis
F_g	Gas force
F_i	Inertia force
f	Frequency
H	Mixing matrix
ICA	Independent component analysis
ICs	Independent components
k	Stiffness constant
k_a	Rocker arm stiffness
k_{aiv}	Inlet valve rocker arm stiffness
k_{aev}	Exhaust valve rocker arm stiffness
k_c	Cylinder wall stiffness
k_h	Cylinder head stiffness
k_s	Spring Stiffness
k_{sev}	Exhaust valve spring stiffness
k_{siv}	Inlet valve spring stiffness
m	Mass
m_{cr}	Crank mass
m_{ev}	Exhaust valve mass
m_i	Reciprocating mass
m_{iv}	Inlet valve mass
m_p	Piston mass
m_r	Rotating mass
$m_{r.pp}$	Reduced mass of connecting rod to piston pin
$m_{r.cr}$	Reduced mass of connecting rod to crank pin
m_v	Valve mass
Q	Piston lateral reversible force
s	Noise source
STFT	Short time Fourier transform
t	Time
TFA	Time frequency analysis
TFTB	Time-frequency analysis toolbox
τ	Time
ω_d	Damping natural frequency
ω_n	Natural frequency
ζ	Damping factor
X	Time frequency expansion
x	Displacement and mixed signal
$\dot{x}$	Velocity
$\ddot{x}$	Acceleration
y	Displacement and estimated components
z	Displacement
$\dot{z}$	Velocity
$\ddot{z}$	Acceleration
W	Demixing matrix

1. INTRODUCTION

The process of noise generation by an engine is very complex. Multiple excitations of the engine structure occur as a result of the combustion in the engine cylinders. Those excitations generate vibration in the engine structure which travel to the surface of the structure and generate sound waves which radiate into the air [1].

A technique for source identification on the cylinder liner was developed by [2] based on wave arrival time. Experiments with simulated acoustic emission (AE) sources were used to characterize the signal propagation to chosen locations on the liner surface. Using the knowledge of the engine timing and duration of events within the diesel cycle, the technique was used to running engine. Results were applied to identification of AE events associated with piston ring/cylinder liner interaction on an in-service power generation engine. Two significant AE events associated with an oil groove and scavenging port were identified.

The same technique which based on wave arrival time is used by [3] to produce spatially located time series signal for a running engine. By this it is meant the decomposition of multi-source signal by acquiring it with an array of sensors and using source location to reconstitute the individual time series attributable to some or all of these signals. This technique is used for studying the propagation of AE waves generated from a simulated source on the surface of cast iron test block of varying geometry. It is generally observed that geometric complexity can lead to enhanced or reduced attenuation, apparently depending upon the relative effects of multiple reflections or leakage along other structural components.

The goal of this paper is to introduce a theoretical study on diesel engine noise separation and identification. It is predicted that the simulated diesel engine noise can be separated and their events can be identified. The separation of diesel engine real noise sources can help in noise control, combustion analysis and fault diagnosis. The fast Fourier transform (FFT) analysis identifies the entire frequencies in the stationary signal and its power density. If there are two or more sources in a non-stationary signal with the same frequency, it is difficult to identify them by FFT. The same frequency of the two or more sources appears once in the power spectrum and their squared magnitude will be coincident. It is important to separate the noise sources in time domain before FFT analysis in order to identify the squared magnitude of each source in case they have the frequency. The ICA technique is applied for the separation of the simulated diesel noise source. The estimation of sources by ICA and their analysis by FFT not enough to identify what the source is in the non-stationary signal. The time-frequency analysis, as another technique, is applied for the analysis of the estimated sources to identify their occurrence time.

In this paper a single cylinder direct injection diesel engine noise induced by fuel injection, combustion, valve open and valve close are mathematically modeled. The simulated noise sources are the exhaust valve close, inlet valve close, fuel injection, combustion, piston slap, exhaust valve open and inlet valve open. The simulated sources are mixed by an assumed mixing matrix. The mixed sources are separated by ICA and each source is obtained alone in time domain. The estimated sources are analyzed by TFA and identified in the crank angle-frequency plane.

2. MATHEMATICAL MODELS OF DIESEL ENGINE NOISE

The combustion induced noise is due to the unidirectional force excitation, see Fig.1.a. The gas force F_g excites the top part of the engine structure (i.e. cylinder head) while the lower part of the structure is excited by the combined gas force and inertia force, F_g+F_i. The engine structure excitation due to combustion is modeled by White et. al. [4] and Priede [5] as a simple linear spring mass system as shown in Fig.1.b, and its vibration can be represented by the following equation:

$$m\ddot{x}+c\dot{x}+kx=F \tag{1}$$

One of the major sources of noise and vibration in internal combustion engine is the impact between the piston and cylinder wall (piston slap). When a piston collides against the inner wall of a cylinder, piston skirt contacts firstly. Then, the side thrust force exerts on the piston continuously until the whole lateral area of the piston contacts with the cylinder inner wall. The piston slap is modeled as one degree of freedom spring mass system with a damper. The damping is due to the oil film between the piston and the cylinder inner wall. The equivalent system is shown in Fig.1.c and its equation of motion is given by:

$$m\ddot{x}+c\dot{x}+kx=F=Q \tag{2}$$

Valves excitation induced noise has important contribution in the engine radiated noise. Opening and closing of inlet valve(s) and exhaust valve(s) vibrate engine structure and radiate noise, see Fig.1.a. Although, the valve mass is too small compared with the piston mass and its speed is half of the piston speed in four stroke engines, the radiated noise by it is considerable. The valve opens due to the impact force from the rocker arm on its stem end. The valve vibration during opening is due to the base excitation of the rocker arm. There are two stiffness constants due to the valve spring and the rocker arm spring which are connected in parallel. The damping occurs due to the oil between the valve groove and its stem. The vibration due to the valve opening is modeled as single degree of freedom mass spring system with a damper vibrates due to the base excitation, see Fig.2.b The equation of motion can be expressed in terms of the relative displacement of the mass z=x-y as follows [6]:

$$m\ddot{z}+c\dot{z}+kz=-my \tag{3}$$

The equation is similar to the equation

$$m\ddot{x}+c\dot{x}+kx=F \tag{4}$$

With variable z replacing x and the term $-m\ddot{y}$ replacing the forcing function F.

When a valve closes due to the spring force, an impact occurs between the valve head and the valve seat. Therefore, the cylinder head vibrates and this vibration is transferred to the engine block surface as radiated noise. Similar to the valve opening system there exist valve spring and cylinder head spring which are connected in parallel. The valve closing system is modeled as single degree of freedom mass spring system with a damper as shown in Fig.2.c. The equation of motion is given by:

$$m\ddot{x} + c\dot{x} + kx = F \quad (5)$$

Since the force applied to the above mentioned systems is an impulsive force, the response of the systems due to a unit force can be obtained according to Rao [6] as follows:

$$x(t) = \frac{e^{-\zeta\omega_n t}}{m\omega_d}\sin(\omega_d t) \quad (6)$$

where $\omega_n = \sqrt{\frac{k}{m}}$, is the natural frequency of the system.

$\zeta = \frac{c}{2m\omega_n}$, is the damping factor.

$\omega_d = \omega_n\sqrt{1-\zeta^2}$, is the damping frequency.

3. SIMULATION OF DIESEL ENGINE NOISE

The noise sources simulated as if they are radiated from a single cylinder diesel engine Deutz F1L-511 with 7.7 Kw power at 1500 rpm. Valve timing and fuel injection events are chosen due to design specifications of the engine, the combustion and piston slap events are assumed, the values are shown in Table.1. For noise simulation, the engine is assumed to operate at steady state with maximum speed which is 1500 rpm and no load. At 1500 rpm engine speed, the frequency of the crankshaft is calculated to be 25 Hz with time period 0.04 second. The engine cycle which occurs in a twice crank angle rotation has a period of 0.08 second and 12.5 Hz frequency. The engine cycle period corresponds to 720° of crank angle rotation, for a sampling rate of 0.5° the sampling rate in seconds is obtained as 0.08/1440 = 5.5555×10^{-5}. In turn the sampling frequency which is the inverse of the sampling rate equals 18000 Hz. This sampling frequency is greater than the double Nyquist frequency which is approximately 2000 Hz

The engine combustion induced noise, fuel injection, piston slap and valve events are simulated by assumed values for k, m, and c in equation (6). The simulated sources in time domain s(t) are represented in crank angle domain as shown in Fig. 3.

Table.1. Crank angle positions and frequencies of the simulated noise sources

Cycle Event	Exhaust Valve Close	Inlet Valve Close	Fuel Injection	Combustion	Piston Slap	Exhaust Open	Inlet Open
Crank Angle	32° ATDC	59° ABDC	24° BTDC	15° ATDC (Assumed)	90° ATDC (Assumed)	71° BBDC	32° BTDC
Noise Frequency [Hz]	1450	1300	950	1500	1350	1400	1300

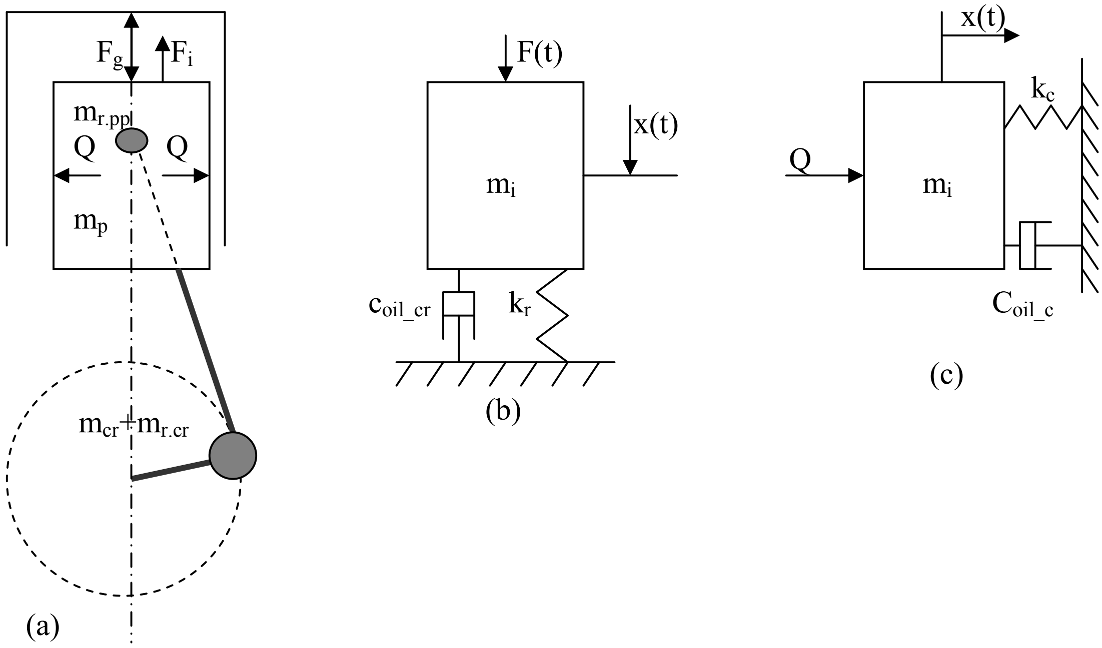

Fig.1. a unidirectional and reversible forces, (b) unidirectional force excitation, (c) reversible force excitation (piston slap)

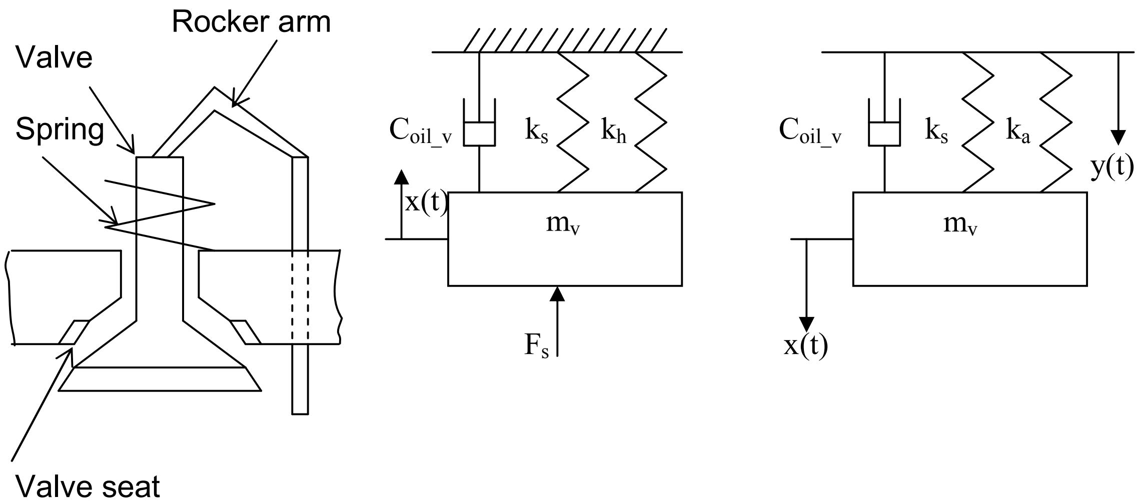

Fig.2. aValve mechanism, (b) valve closing, (c) valve opening.

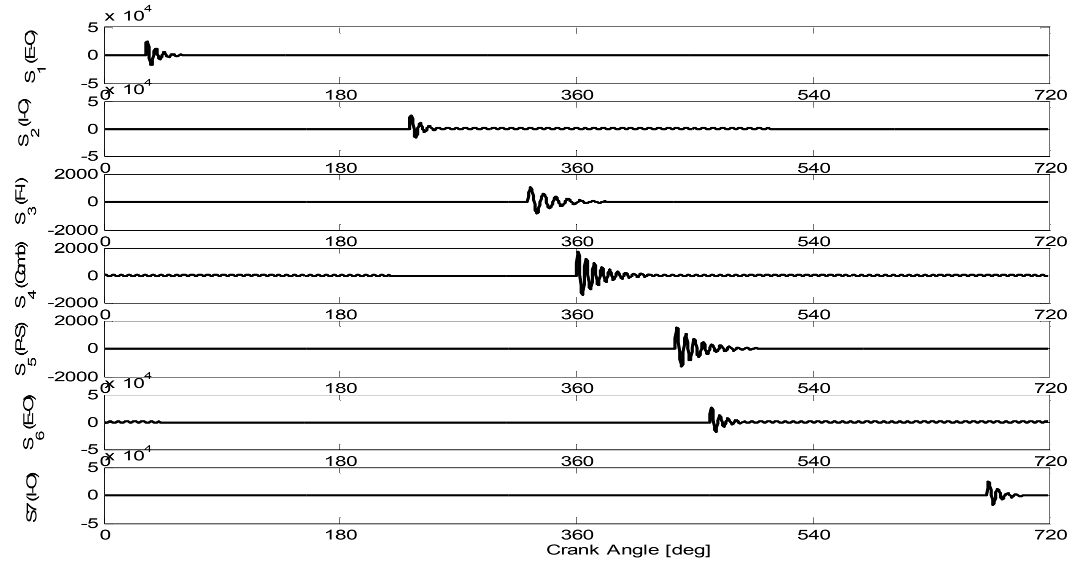

Fig.3. simulated noise sources

4. INDEPENDENT COMPONENT ANALYSIS

Independent component analysis is a signal processing technique whose goal is to express a set of random variables as linear combinations of statistically independent component variable [7]. The interesting applications of ICA are blind source separation and feature extraction.

The independent component analysis is always represented by two processes which are mixing and demixing. The mixing process as proposed by Hyvärinen et. al. [7], represents the relation between the observed and source signals. Suppose a zero-mean m-dimensional random variables denoted by $x(t) = [x_1(t), x_2(t), \ldots\ldots, x_m(t)]^T$ that can be observed, and n unknown independent components $s(t) = [s_1(t), s_2(t), \ldots\ldots\ldots, s_n(t)]$ that are the mutually statistically independent. The linear relationship between x(t) and s(t) can be expressed as:

$$x(t) = H\, s(t) \tag{7}$$

Here, H is an unknown m×n matrix of full rank, called the mixing matrix. The basic problem of ICA is then to estimate the original components $s_n(t)$ from the mixtures $x_m(t)$ or, equivalently, to estimate the mixing matrix H.

The mixing process can be explained with so called cocktail party problem [8]. Imagine that there are two persons speaking simultaneously. Two microphones were put at different locations. The microphones record two different signals $x_1(t)$ and $x_2(t)$ with amplitudes x_1 and x_2. Each of these recorded signals is a weighted sum of the speech signals emitted by the two speakers $s_1(t)$ and $s_2(t)$. The relation between the emitted signals and the recorded ones can be expressed by the following linear equations:

$$x_1(t) = h_{11}\, s_1 + h_{12}\, s_2(t) \tag{8}$$

$$x_2(t) = h_{21}\, s_1 + h_{22}\, s_2(t) \tag{9}$$

where h_{11}, $h_{12}h_{21}$ and h_{22} are the elements of the mixing matrix which depend on the distances of the microphones from speakers as shown in Fig.4 from [9,10] .

The demixing process which is similar to the mixing process can be performed by a demixing matrix W. The m observed signals x(t) are coupled to the n reconstructed signals y(t) via the demixing matrix. The relationship between the mixed signal and the reconstructed one can be given by:

$$y(t) = W\, x(t) \tag{10}$$

According to [d], for a convolutive mixture the demixing system contains n×m finite impulse response (FIR) filter of length q. Therefore, the demixing system can be expressed a an n×m matrix W(t), with its element $w_{ij}(t)$ being the impulse response from j^{th} measurements to i^{th} outputs. The reconstructed signal can be obtained as:

$$y(t) = \sum_{\tau=0}^{q-1} W(\tau)x(t-\tau) \tag{11}$$

Where τ is the time delay and $y(t) = [y_1(t), \ldots\ldots\ldots, y_n(t)]^T$.
For the case of n = m =2 the unmixing system is shown in Fig.4.

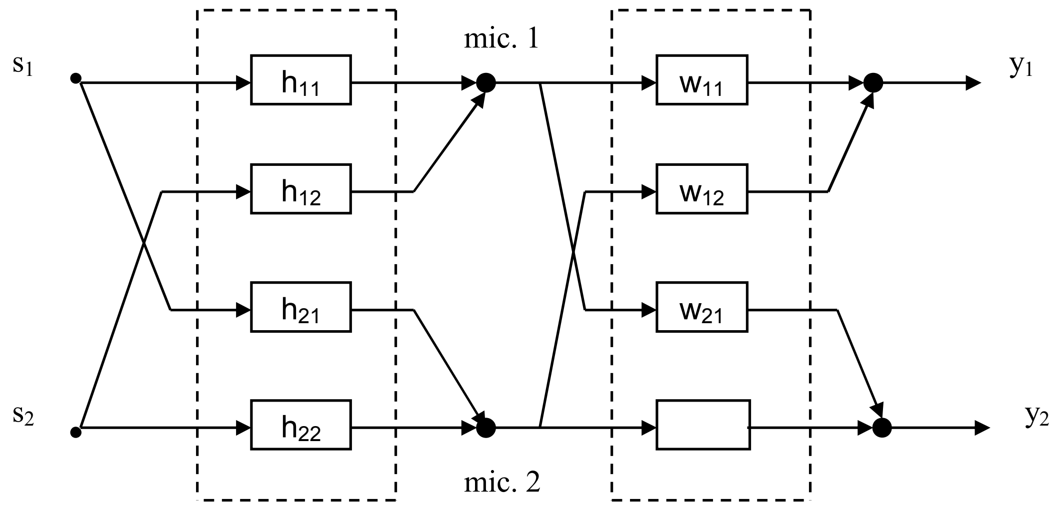

Fig.4. Scheme of Blind Source Separation

5. MIXING OF INDIVIDUAL SIMULATED NOISE SOURCES

The simulated diesel engine noise sources s(t), which are shown in Fig.3, are mixed according to the following assumed mixing matrix H:

$$H = \begin{bmatrix} 0.011 & 0.012 & 0.013 & 0.014 & 0.0160 & 0.017 & 0.0125 \\ 0.014 & 0.0168 & 0.0147 & 0.0152 & 0.013 & 0.0132 & 0.0125 \\ 0.0124 & 0.0135 & 0.0124 & 0.0124 & 0.035 & 0.01354 & 0.01324 \\ 0.0124 & 0.01245 & 0.0124 & 0.0132 & 0.0165 & 0.0135 & 0.0189 \\ 0.01245 & 0.0147 & 0.01568 & 0.01354 & 0.01354 & 0.1365 & 0.01354 \\ 0.0168 & 0.0132 & 0.0165 & 0.0178 & 0.0195 & 0.0198 & 0.0187 \\ 0.0158 & 0.0165 & 0.0198 & 0.0147 & 0.0184 & 0.01354 & 0.013542 \end{bmatrix}$$

The results of mixing are the mixed signals x(t) which are shown in Fig.5 in the crank angle domain. The mixed signals are considered as the signals that observed during real measurement.

6. SEPARATION OF MIXED NOISE SOURCES

The mixed signals x(t) are separated by fixed-point independent component analysis (FPICA) algorithm which is used by Hyvärinen et. al. [7-10]. Seven estimated components are well separated and represented in crank angle domain as shown in Fig.6.

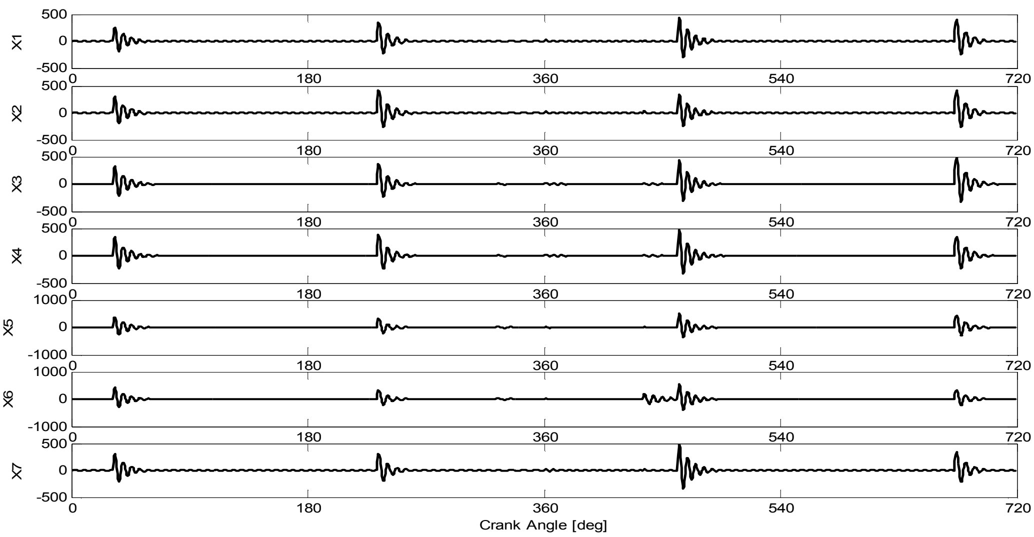

Fig.5. mixed simulated noise sources.

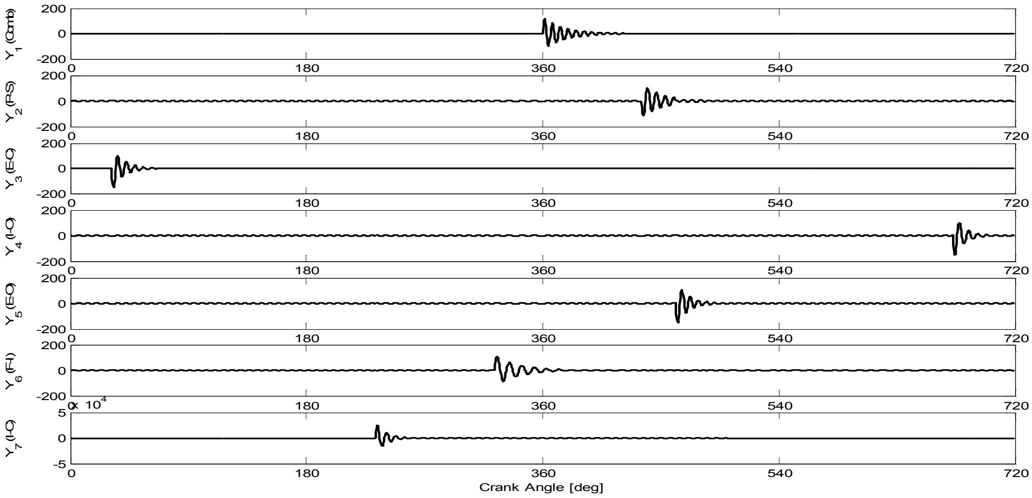

Fig.6. Estimated components by ICA

7. ANALYSIS AND DISCUSSION OF RESULTS

Analysis of the mixed signal of noise sources by fast Fourier transform as shown in Fig.7 determines the frequencies and their squared modulus, but one can not identify noise source relative to each frequency. The other problem is that, if there are two or more sources with the same frequency it is difficult to identify the squared modulus of each source, because the they coincident.

The mixed signals are separated by ICA in time domain and each estimated source appears alone as shown in Fig.6, but it is difficult to identify a noisy separated source in time domain. The estimated sources are converted to audio and each source can be listened alone, but it is difficult to identify engine noise sources by listening. Each estimated source can be analyzed by FFT to identify its frequency and squared modulus, but this information is not quite enough to identify what the source is. The sources of noise in internal combustion engine differ from each other in their occurrence relative to time (crank angle position). Each of simulated diesel engine noise sources in this paper has specific constant occurrence time except the fuel injection and combustion which they change within a range. A new technique for identification of the frequency of each noise source relative to the time (crank angle) is required.

The time frequency analysis is applied for the analysis of the noise sources which are estimated by ICA. The most familiar time-frequency analysis is the short time Fourier transform, which is used in the in this paper. The short time Fourier transform is a transform that presents a signal x(t) in a time-frequency expansion. Windowing a signal arounda time τ and then computing the Fourier transform, is the easiest way to determine the local behavior of a signal. In this way, the short time Fourier transform of a signal is given by Roberto [11] as

$$X(\tau, f) = \int_{-\infty}^{\infty} x(t)w(t-\tau)e^{-j2\pi ft}\,dt \tag{12}$$

The estimated noise signals as shown in subfigure (a) of figures (8-14) are analyzed by STFT in time-frequency plane. A MATLAB time frequency analysis toolbox (TFTB) is used for time-frequency analysis. The time is converted to crank angle by multiplying it by 6n where n is the engine speed. Contour plots on the crank angle-frequency planes are drawn for the estimated signals as shown in subfigure (b) of figures (8-14). The centre of contour represents the maximum of the squared modulus. To represent the squared modulus a three dimensional graph is plotted for separated source signals as shown in subfigure (c) of figure (8-14).

The crank angle positions and the noise frequencies of the exhaust valve close, inlet valve close, fuel injection, combustion, piston slap, exhaust valve open and inlet valve open are identified as are given in Table.1.

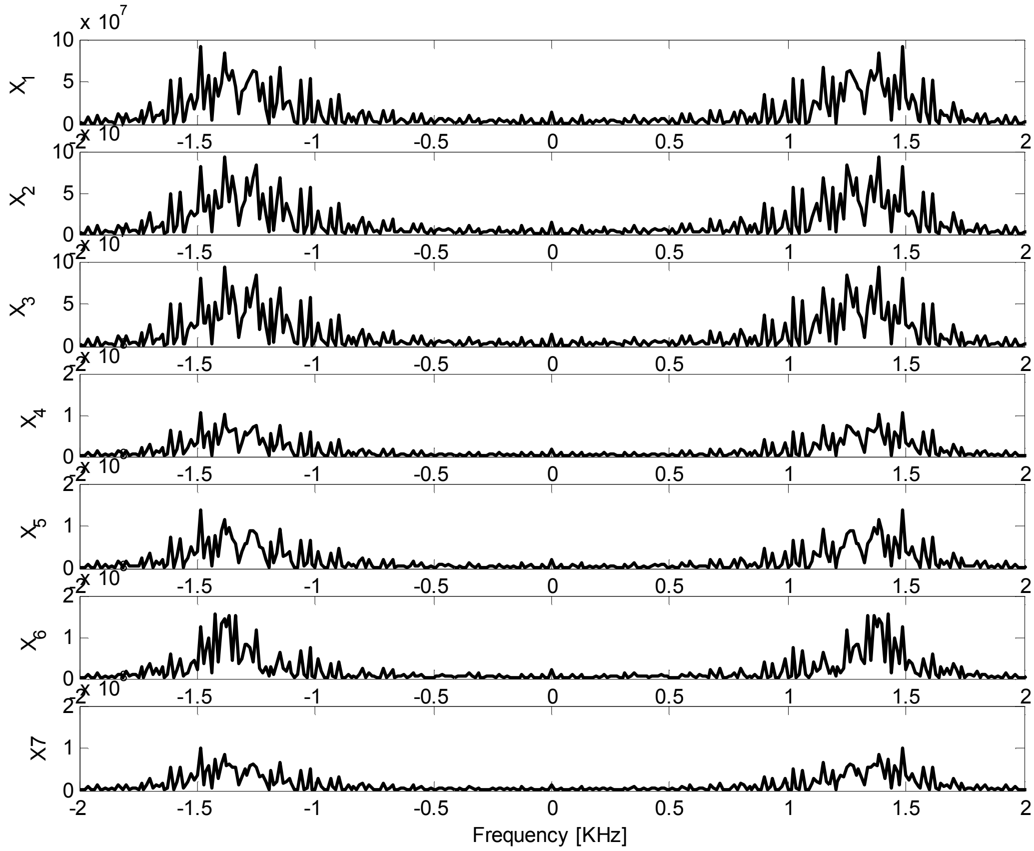

Fig.7. Power spectrum of the mixed signals

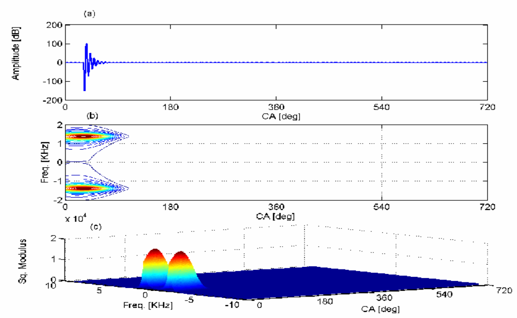

Fig.8. TFA of exhaust valve close

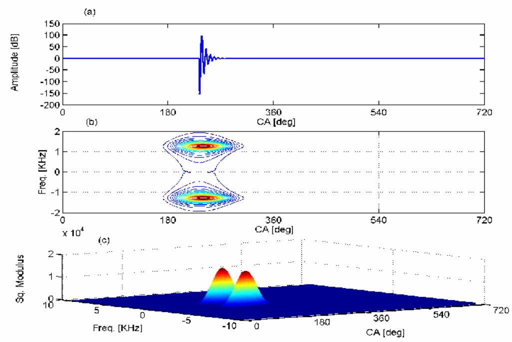

Fig.9. TFA of inlet valve close

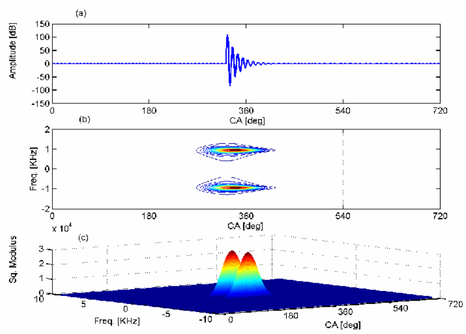

Fig.10. TFA of fuel injection

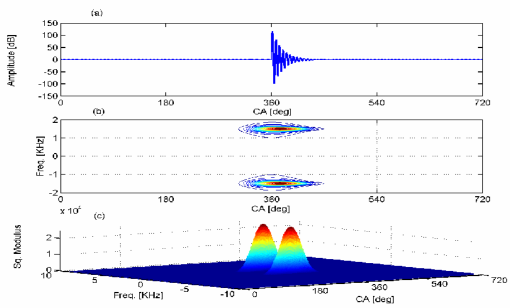

Fig.11. TFA of combustion

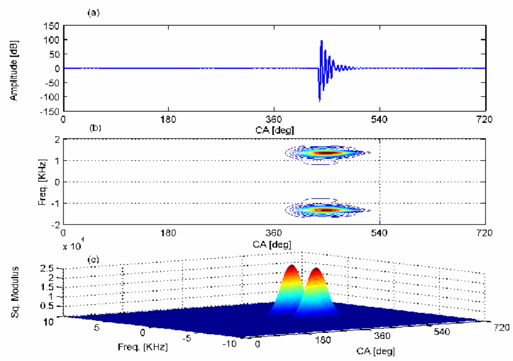

Fig.12. TFA of piston slap

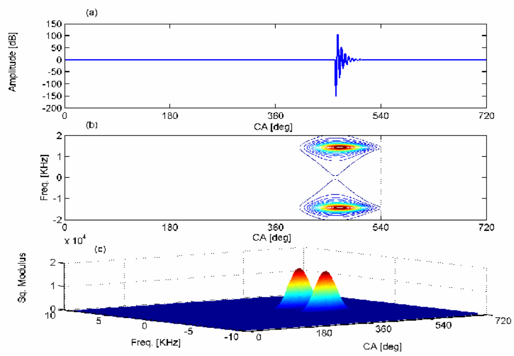

Fig.13. TFA of exhaust valve open

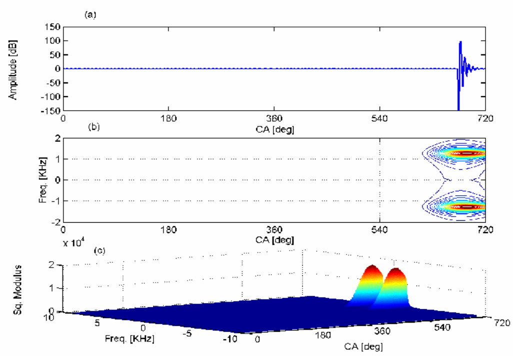

Fig.14. TFA of inlet valve open

8. CONCLUSIONS

The mixed diesel engine noise sources which are simulated as blind sources are well separated by the independent component analysis.

The separated independent components are analyzed in time-frequency plane by short time Fourier transform. The exhaust valve close, inlet valve close, fuel injection, combustion, piston slap, exhaust valve open and inlet valve open are well identified in the crank angle domain.

The theoretical results are promising so that with a multi-channel microphone array the separation of real diesel engine noise sources can be achieved using the independent component analysis.

Separation of diesel engine noise sources by ICA and identification of their occurrence in the crank angle domain by STFT can be used for fuel injection advance or retard detection, combustion analysis and valve clearance detection.

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Appendix A

Assumed values for simulation

Mass:

$m_i = m_p + m_{r.pp} = 5$ Kg
$m_{iv} = 0.27$ Kg
$m_{ev} = 0.3$ Kg

Stiffness:

$k_r = 444132198$ N/m
$k_{siv} + k_{aiv} = 18014001.95$ N/m
$k_{siv} + k_h = 18014001.95$ N/m
$k_{sev} + k_{aev} = 24901011.9$ N/m
$k_{sev} + k_h = 23213309.55$ N/m

Damping factor:

$c_{oil\_cr} = 5290$ N.s/m
$c_{oil\_iv} = 600$ N.s/m
$c_{oil\_ev} = 600$ N.s/m
$c_{oil\_c} = 5290$ N.s/m