NOISE SOURCE IDENTIFICATION
AND ADOPTION OF PROPER NOISE CONTROL STRATEGIES
ON WHEELED TRACTORS

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NOISE SOURCE IDENTIFICATION AND ADOPTION OF PROPER NOISE CONTROL STRATEGIES ON WHEELED TRACTORS

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I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

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ABSTRACT

NOISE SOURCE IDENTIFICATION AND ADOPTION OF PROPER NOISE CONTROL STRATEGIES ON WHEELED TRACTORS

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This thesis is aimed at identifying the noise sources of a wheeled tractor to reduce the noise levels below the legislative limits by controlling noise sources through proper methodologies.

The study focuses firstly on identifying the noise sources of a wheeled tractor by using proper noise source identification techniques. These techniques can be summarized as sound intensity measurements, sound power level determination studies and spectral analysis of the noise data acquired in the tests. Simple sound intensity mapping techniques are used and the intensity contour maps are generated to identify the noise sources.

Most important and effective noise sources are identified and the critical noise sources are focused to apply appropriate noise control strategies not only at the prototype production stages but also at the early design stages.

Consequently, upon consideration of both structure-borne and flow-induced noise, the pass-by noise level and the operator’ s ear noise levels of the tractor are reduced by nearly 3 dB (A) through application of proper noise control strategies.

Keywords: Sound Intensity, Noise Source Identification, Tractor, Noise Control Strategies

ÖZ

TEKERLEKLİ TRAKTÖRLERDE GÜRÜLTÜ KAYNAĞI TANIMLANMASI VE UYGUN GÜRÜLTÜ KONTROL STRATEJİLERİNİN UYGULANMASI

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Bu tez çalışmasında, uygun yöntemler aracılığıyla gürültü kaynaklarının idare edilmesi ile birlikte gürültü düzeyinin yasal sınırlarının altına indirilmesi için bir tekerlekli traktörün gürültü kaynaklarının tespit edilmesi amaçlanmaktadır.

Bu araştırma ilk olarak uygun gürültü kaynağı belirleme teknikleri kullanılarak tekerlekli traktörün gürültü kaynaklarının tespit edilmesine odaklanmıştır. Bu teknikler, ses yeğinliği ölçümleri, ses gücü seviyesi belirleme çalışmaları ve testlerle elde edilen verilerin izgesel analizleri olarak özetlenebilir. Ses yeğinliği haritalama teknikleri kullanılarak oluşturulan ses yeğinliği haritaları yardımıyla gürültü kaynaklarının tespit edilmesi sağlanmıştır.

Araştırma sonunda belirlenen en etkili ve önemli gürültü kaynakları arasından sadece deneme üretimi sürecinde değil; aynı zamanda tasarım sürecinin ilk evrelerinde de uygun gürültü idaresi stratejilerini uygulamak için kritik gürültü kaynaklarına odaklanılmıştır.

Sonuç olarak, uygun gürültü idaresi stratejilerinin uygulanması sayesinde traktör şanjmanı ve üst inşa yapısı kaynaklı ve hava akışı sevkli kaynaklı gürültülerin göz önünde tutulması sağlanmış; böylece hem traktör geçişi sırasında yaydığı gürültü seviyesinde hem de operatör kulak hizası gürültü seviyesinde yaklaşık 3 dB (A) azalma elde edilmiştir.

Anahtar Kelimeler: Ses Yeğinliği, Gürültü Kaynağı Belirleme, Traktör, Gürültü Kontrol Yöntemleri

To my dear family

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LIST OF ABBREVIATIONS

L_P	: Sound Pressure Level (dB)
L_W	: Sound Power Level (dB)
L_{WA}	: A-Weighted Sound Power Level (dB (A))
L_{pA}	: A-Weighted Sound Pressure Level (dB (A))
P_{ref}	: Threshold of Hearing (Pa)
P_i	: A-Weighted Acoustic Pressure for i^{th} Frequency
P_{rms}	: A-Weighted Acoustic Pressure for Whole Frequency Range
L_{pA}	: A-Weighted Sound Level for Whole Frequency Range
L_{pA_i}	: A-Weighted Sound Level for i^{th} Frequency
P	: Instantaneous Pressure at a Point (Pa)
$\vec{U}$	: Particle Velocity Vector at a Point (m/s)
$\vec{I}$	: Sound Intensity Vector (W/ m²)
W	: Sound Power (W)

S : Surface Area (m²)

S_i : Partial Surface Area for the i^{th} Intensity Grid (m²)

N : Total Number of Intensity Grids of Measurement Area

I_n : Sound Intensity (W/ m²)

I_{ref} : Reference Sound Intensity (W/ m²)

$\overline{I_{ni}}$: Average Sound Intensity for the i^{th} Partial Surface of Measurement Area (W/ m²)

L_{I_n} : Sound Intensity Level (dB)

$\overline{L_{I_n}}$: Average Sound Intensity Level (dB)

W_i : Partial Sound Power (W)

W_{ref} : Reference Sound Power (W)

S_0 : Unit Surface area (m²)

$\overline{L'_{pA}}$: Average A-weighted sound pressure level of sound source for the reference surface area

L'_{pAi} : A-weighted sound pressure level as dB for i^{th} microphone location

N : Total Number of Microphone locations

FFT : Fast Fourier Transform

CHAPTER 1

INTRODUCTION

Agriculture is simply the production of food and goods through farming. Besides, agriculture covers the development of the quality of these food and goods. Furthermore, the outcome from agriculture can be considered as the transportation and marketing of these products. Agriculture is one of the most significant evolvements that directed to the rise of civilization with the utilization of farm machines and implements.

There are several agricultural machines used on today's farms such as harvesters, tractors, cultivators, ploughs, etc. Although other multi-functional machines may have taken the harvesting task away from tractors, they still can do the majority of work on farms these days. They can be used to pull implements for tilling the ground, seeding, transportation, etc. Tractors are considered in this study from noise control point of view. There are two types of tractors; crawler tractors and wheeled tractors. A crawler tractor is a vehicle that runs on continuous tracks instead of wheels. Crawler tractors are generally used on rocky lands with higher inclinations where pneumatic tires may have short bursts and may not possess sufficient grip on ground. On the other hand, a wheeled tractor, can be seen in Figure 1.1, has pneumatic tires for the front and rear axles.

Wheeled tractors are the main concern of noise control studies for this thesis work.

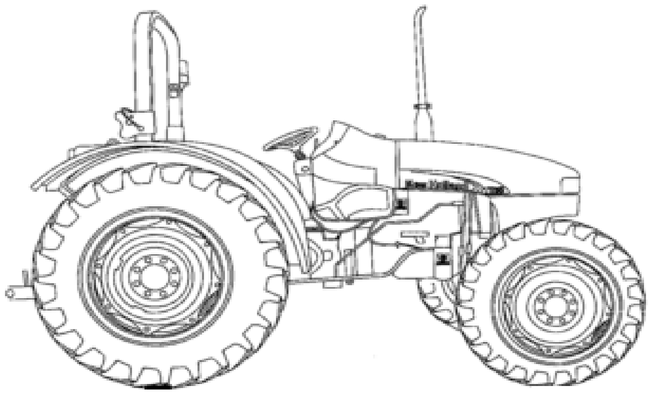


Figure 1.1 A Wheeled Tractor

Tractors have always been the most necessary farm machine. As tractors got larger, producers started to install cabinets to save drivers from undesired elements and extending the hours the machine could be used throughout the year.

Excessive noise radiated from tractors is one of these elements. As farm and its machinery evolve, increasing pressure has been coming from worldwide markets to reduce or decrease the noise. It has already been investigated and proved that excessive noise exposure was unhealthy and harmful for humans.

Legislations have started to enforce new laws and directives to protect human health. New legislations and directives were published for more strict applications. Tractor manufacturers must comply with the European Noise Directives 77 / 311 / EEC [1] and 74 / 151 / EEC [2]. Decreasing the noise level below the legal requirements has always been a challenging task or issue for almost all tractor manufacturers.

There are a few ways to deal with this issue. One of the most powerful tools that can be used in modern acoustics today is the noise source identification. By identifying where the noise sources may be located and their spectral characteristics, it then becomes much easier to understand, to reduce or to control the overall noise level of a tractor. This is the typical aim of noise vibration harshness programs. Over the years many different techniques have become available as technology moves forward. When tractors are specifically considered, majority of noise comes from the engine. Other sources can be the transmission box, hydraulic components, cooling fan and exhaust pipes, etc.

Vibrations from all these sources can cause related components to produce structure-borne noise. Noise source identification software will be used to develop noise maps showing the most effective portions of tractor in this research. Special care will also be exercised to improve the sealing or air tightness especially in the areas where cables and harnesses enter.

Within this research, the flow induced noise will be studied because the bigger portion of noise level comes not only from the exhaust and its pipe but also from the cooling fan of radiator.

The objective of this study is to apply the noise source identification methods to find out related noise sources; to complete the study by using noise control strategies on tractors as well as to minimize the noise level radiated by recently designed tractors.

The outline of the study is organized in six chapters. Chapter 2 summarizes the literature survey of noise source identification techniques and noise control applications on tractors and in automotive industry applications.

Chapter 3 introduces the methodologies, giving the equations necessary for calculations to compare results and findings and presenting the project tractor with its potential noise sources. Chapter 4 explains the experiments, experimental studies, instruments used during tests in detail and related ISO standards to describe how to calculate sound power levels, tests and the instruments to be used throughout the tests. In addition, Chapter 5 reveals all the experimental results from tests and measurements, tabulates the results and plots intensity maps, shows the modifications on tractors and also comments on data. Moreover, tabulates the calculation results found by using measured data in equations.

And finally, Chapter 6 summarizes the studies, tests and explains how the modifications were conducted on tractors and compares the data found after modifications and recommends future works.

CHAPTER 2

LITERATURE SURVEY

Tractors have been used all around the world with an increasing trend as agriculture has grown rapidly worldwide for a couple of decades. Despite their usage, increasing pressure has been coming from worldwide markets and also from the public to cut down or minimize the noise emitted during their operations. It has already been proved that excessive noise exposure was unhealthy and harmful for humans. Fallon [3] studied the relation between farm workers' hearing losses and the noise emitted by tractors. It was stated that farm workers' hearing loss is proportional to the amount and duration of noise. Moreover, an important relation between hearing loss and ages of people was determined among people older than 35 years of age who are exposed to farm machinery by Solecki [4].

Agricultural and its applications in Japan have grown enormously since 1955 together with tractors and other farm machineries. These new technological developments have shown that an increase in noise exposure that is not only annoying, but also damaging to human health.

Miyakita et al [5] tried to decide, if Japanese tractor users and farmers are at risk for hearing loss due to noise in comparison with people working at the offices, and by evaluating the present conditions regarding occupational noise levels among people involved in agriculture. The results proposed that particularly male farmers have a high prevalence of hearing loss in the higher frequency ranges.

Daily noise exposure levels ranges from 81.5 to 99.1 dB (A) for tea harvesting and processing, and from 83.2 to 97.6 for sugar cane harvesting. Taking into account long extended working hours and excessive noise from farm machineries, it is concluded that farmers are at risk for noise-induced hearing loss. These findings clearly points out that a strong need for implementation of hearing conservation programs among agricultural workers exposed to machinery noises. [5]

Important differences were found between male farmers and male office workers in the percentage of subjects with hearing levels more than 40 dB at 4 kHz. For example, in their 40 s, the rate was 16.4% among farmers and 9.6% among office workers. When the time interval is 60 s, the rate became 50.3% for farmers and 29.9% for office workers. For hearing levels at 1 kHz, no considerable differences were obtained among male subjects. [5]

In female subjects, no significant differences were found between farmers and office workers except among the 50-year-olds at 1 kHz test frequency. The results prove that Japanese farmers, particularly male farmers, are at risk from noise-induced hearing loss when compared with office workers. From the results, it was clearly realized that tractor or other agricultural machinery manufacturers must pay necessary attention to noise control applications. [5]

Another investigation was conducted by Dewangan et al [6] to determine the noise generation during stationary condition and the noise level at operator's ear level of tractors of 18.7 and 26.1 kW power and hand tractors of 4.6 and 6.7 kW. The sound pressure level at operator's ear level, in dB (A), was found as around 92 dB (A). This value does not comply with legislative limits. Two models of the 2-wheel drive tractors and two models of hand tractors were selected for the study.

In order to monitor the sound propagation characteristics in terms of sound pressure level (SPL), grid points were marked on the experimental open field, ranging rods and measuring tape at a grid spacing of 1 m x 1 m. The test tractors were kept at the centers of the grid lines with their engines operating at full throttle. [6]

Finally, the maximum sound pressure level produced by tractors and hand tractors during stationary condition at rated engine speed was found to be 92 dB (A) and 94 dB (A), respectively. The fact that the SPL of hand tractors is higher than that of tractors by 2 dB (A) was due to the absence of an engine shroud. The SPLs were higher for field operations corresponding to the implements requiring higher drafts for field operations. [6]

Considering all these side effects during farm mechanization process, to protect human health and to minimize the effects, governments started to enforce new laws and legislations.

There are two legislations about the noise level of wheeled tractors used in fields or places open to public. 77 / 311 / EEC [1] Maximum Permissible Noise Level at operator’s ear position and 74 / 151 / EEC [2] Maximum Permissible Noise Level of tractor’s Pass-By Noise.

Regarding tractors, manufacturers must comply with these legislations. The legislation 77 / 311 / EEC [1] ensures that the tractor drives’ noise level perception must not exceed 86 dB (A) and requires the measurement of maximum noise level at operator’s ear position during tractor movement. The test requirements can be summarized as;

- The test field must be in an open and sufficiently silent location. For example, an open space of 50-metre radius and a central part with a radius of at least 20 m. [1]

- The surface of the track must not cause excessive tire noise. [1]
- The weather must be fine and dry with little or no wind. [1]
- The tires must be inflated to the pressure recommended by the tractor manufacturer, the engine, transmission and drive axles must be at normal running temperature. [1]
- The microphone must be located 250 mm to the side of the central plane of the seat, the side being that on which the higher noise level is encountered. [1]
- The microphone diaphragm must face forward and the centre of the microphone shall be 790 mm above and 150 mm forward of the seat reference point. [1]
- Noise must be measured at the maximum engine speed using sound-level meter in the gear-range combination giving the speed nearest to 7,5 km/h at the rated rpm. The tractor must be unladen when measurements are being made. [1]

The second legislation is 74 / 151 / EEC [2] guarantees that the maximum noise level of a tractor during operation affecting surroundings must not exceed 89 dB (A) and requires the measurement of noise level while tractor is passing by in front of a microphone. The test requirements can be summarized as;

- The test field must be in an open and sufficiently silent location. For example, an open space of 50 meter radius and a central part with a radius of at least 20 m. [2]
- The surface of the track must not cause excessive tire noise. [2]
- The weather must be fine and dry with little or no wind. [2]
- The microphone shall be placed 1,2 meters above ground level at a distance of 7,5 meters from the path of the tractor's centre line. [2]

- Two lines respectively 10 meters forward and 10 meters rearward of microphone axis line shall be marked out on the test track. Tractors shall approach one line at a steady speed. The throttle shall then be fully opened as rapidly as practicable and held in the fully opened position until the rear of the tractor crosses the other line. The throttle shall then be closed again as rapidly as possible. [2]

With the inception of these two directives in Europe, it has always been a challenging task for the tractor manufacturers to obey the limits. Noise source identification techniques have been introduced and noise control techniques have been implemented.

In the literature survey, there are several sources on the tire-road interaction noise. However, this type of noise can be considered as a noise source mainly for vehicles run at speeds higher than 70 km/h. Since maximum speeds for tractors are presently much lower than 70 km/h, these sources found in literature survey will not be shared and discussed in this thesis. The maximum speeds for tractors are between 30 km/h and 45 km/h.

2.1 Noise Sources Identification

Noise source identification is the first step of noise control studies. To solve noise problem on a machine, it is essential that noise sources should be identified and described thoroughly. To decide which design changes are proper to decrease the noise radiated by a machine, one needs to characterize the noise in the sense of:

- Spectral content
- The location of the dominant sources
- The relative importance of dominant sources
- In depth description of noise generation mechanisms

Noise source identification occupies a number of tools to manage such a characterization, such as spectral methods, sound intensity measurements, coherent output energy analysis, etc.

Internal combustion engines on tractors are considered generally as the main noise source. The power source of a tractor, the engine, should be investigated first in terms of noise and vibration, because it has an importance to meet the required comfort level by farmers.

JunHong and Bing [7] investigated the mechanisms of engine front noise generation by exterior noise diagnosis analysis of an in-line-six cylinder direct injection diesel engine diesel engine by intensity techniques. Sound level measurements were carried out at 1m height from the engine front, rear and right sides while the engine was running at 2200 rpm and full load. It was found that most of the sound energy was emitted from the front side of the engine.

The peaks in the noise spectra taken at the front side were independent of engine rotational speed, which indicated that structural resonance frequencies were the primary causes of the engine front noise. As a detailed noise diagnosis of front side, sound intensity measurement was performed according to the standards ISO 9614-1 [8], ISO 9614-2 [9] and ISO 9614-3 [10]. From the results of the intensity measurements, noise sources were identified as oil sump, front timing gear cover and front pulley. A structural modification was considered based on former noise generation system analysis. A laminated damped steel oil pan was designed and produced. The noise from the front timing gear cover was reduced by use of the double shinned approach. Furthermore, the head cover and intake manifold were isolated by using rubber material. Therefore, the noise level of the modified engine was 115.2 dB (A) and about a 3.5 dB (A) noise reduction was achieved. [7]

Q. Leclére et al [9] studied the low frequency amplitude modulation of the noise generated by a diesel engine operating at idle. Modulated vibrations were transmitted to the frame mainly by one of the engine mounts. The combustion was the first potential source to be suspected and inspected. However, as the article stated the origin of amplitude modulation is not related to the combustion process.

Thus, secondary potential sources have to be inspected on the diesel engine such as; the timing system, the diesel pump, the power-assisted steering pump, the alternator, the air-conditioning compressor and the oil pump. Spectral analysis tools are applied on multi-channel measurements to identify the sources. A sensor is placed on each potential noise and vibration source. A virtual source analysis shows that several uncorrelated sources are contributing to the operating response, particularly on frequencies for which a high amplitude modulation is observed. This virtual source analysis has been implemented, showing that although the phenomenon was poorly coherent with the combustion while it was coherent with the diesel pump. [9]

A conditioned spectral analysis has confirmed these suspicions. A practical study has been carried out to verify that the diesel pump and the whole injection circuit were involved in the amplitude modulation. Added masses appropriately placed on the injection circuit at the engine mount and strongly attenuate the amplitude modulation. [9]

Talotte et al [12] used noise source identification techniques to reduce railway noise. Reduction of noise at a source might be more challenging than the use of noise barriers but this requires a thorough understanding of the source mechanisms. The paper shows a critical survey of the identification and modeling of railway noise sources.

Besides, the paper summarizes the current knowledge of the physical source phenomena as well as the potential for noise reduction. The study is concerned with improvements to source modeling, especially for aerodynamic noise, investigation of other sources and development of more advanced models for predicting railway noise in the environment. Source identification on trains has been more than simply measuring pass-by levels with a single microphone. More advanced methods have proved useful information which involves either microphone arrays or a combination of different sensors. Use of microphone arrays, combination of different sensors and coherent output power techniques were described. [12]

Microphone arrays were used to localize the aerodynamic noise sources emitted from the sideways and upper part of the trains. Different microphone array configurations like star and spiral arrays were used for different frequency ranges. To differentiate the vehicle and track noise, diagnosis techniques had been developed. Apart from microphone arrays, coherent output power technique was used to visualize the aerodynamic sources on a high-speed train called TGV. [12]

In another study, Murat İnalpolat [13] investigated sound radiation and power flow characteristics of plates which constitute the bodies of common engineering applications like cars and household appliances. Two-microphone sound intensity measurement with a probe utilizing side by side configuration is used to analyze the near-field radiation characteristics of a square steel plate excited by a shaker at its midpoint. Three different vibro-acoustical measurement techniques were used in an integrated manner and results obtained are compared with those obtained from analytical models developed. Structural intensity was used to identify the power flow patterns on the plate. All measurements were repeated for externally damped configuration of the same plate and results were analyzed.

A number of noise source identification techniques have been developed over the years. Under appropriate conditions several different available techniques can be applicable and provide useful information.

However, to solve the more complicated noise source identification problems for instance those involving multiple noise sources; such as internal combustion engines or complete vehicles, most existing techniques are inadequate; so development of more sophisticated source identification methods are necessary.

In recent years, the coherence function technique, among several others, has gained increasing attention. The most important feature of this technique is its use of a multiple-input linear system to model an acoustic or vibration system with several sources. Wang and Crocker [14] published their paper to explore further the possibility of using the coherence function technique for noise source identification in a multiple-source acoustic environment when other techniques may be difficult to use or not provide sufficient information.

Assuming that most of the diesel engine noise was due to the combustion pressure excitation of the engine structure, the noise generation of a six-cylinder, naturally aspirated, direct-injection, v-type Cummins diesel engine was studied. The noise generation system of the engine was considered as a multiple-input, linear system. The inputs of the system were the cylinder pressure signals measured respectively, by six pressure transducers; the output was the engine noise measured by a microphone at about 1 m away from the engine. [14]

Two different approaches for noise source identification based on theory for multiple-input systems have been investigated. In the first approach the concept of frequency response function was employed. In the second approach the concept of coherent residual spectral densities was used. [14]

In order to gain physical insight into the methods, an experimental investigation was conducted with an idealized system designed to simulate a multiple noise source environment. [14]

Results of the experiments showed that, in cases when strong measurement contamination existed, neither approach was able to give good estimates of the spectra of the noise sources. Therefore, whether or not the present techniques are applicable depends not only on the degree of source coherence but on the extent of measurement contamination. [14]

Sung-Chon Kang and Jeong Guon Ih [15] dealt with the issues of the identification and localization of noise sources using the sound intensity method for a reactive field. For these purposes, a three-dimensional model structure similar to the engine room of a passenger car was assumed. The model contains complicated noise sources distributed within the small space, including narrow and reflecting planes constructed with rigid boxes.

For this model, the near field acoustic intensity is calculated by scanning over the upper plane opposite to the bottom by using the acoustic boundary element method. It was observed that the application of sound intensity method without proper care in this situation can yield the detection of fake sources. Therefore, the sound intensity scanning over the engine room upper, with its hood open, may indicate the false positions or components as noise sources. The field reactivity has to be checked and the care should be attended in this type of measurement using the sound intensity methods. [15]

Dumbacher et al [16] studied acoustic array techniques as an alternative method for noise source identification. The basic theory of array procedures for nearfield acoustical holography and an inverse frequency response function technique are given. Experimental evaluation was provided for tire-road interaction noise identification.

The sound pressure measurements from the microphone array can be done by a number of suitable techniques to provide an estimate of the location and magnitude of sound sources. In the automotive industry, such an estimate is useful for identifying airborne and structure-borne sources such as wind noise, tire noise and sheet metal parts' vibration to noise contributions on an engine dynamometer, and pass-by noise source localization. [16]

The array techniques evaluated in the paper include nearfield acoustical holography, temporal array methods, and an Inverse frequency response function method. [16]

The above mentioned studies generally dealt with the engine noise. Ki-Sug Oh et al [17] studied on axle noise and its effects. After having reduction in engine noise, the axle has become an important noise source in sport utility vehicles due to the reason that these vehicles are generally used off-road.

The study introduces both experimental and analytic analyses for reducing the whine noise generated by axles of the vehicle. The test vehicle has a 5-cylinder diesel engine. The vehicle is driven from 40 km/h to 130 km/h to measure the interior noise and vibration due to the fact that whine noise occurs in this speed range. The overall sound levels of vehicle interior noise were measured between 65 dB (A) and 70 dB (A). The suspension system of the vehicle has nine vibration transfer paths from the axle system to the chassis. Through these paths rubber vibration isolation mounts were installed between the axle system and the respective link and between the link and the chassis frame. To recognize the transfer path of the interior noise caused by axles, a vibration path analysis, modal analysis and operational deflection shape analysis are systematically accomplished. The vibrations at several points of the axle are measured by using accelerometers attached to the axle system to describe the transfer path of the vibration from the axle system to the car body. [17]

Structural changes were conducted to reduce the noise generated by axles. Finally, the stiffness of the axle system was changed by stiffening the rib on the carrier cover and increasing the thickness of the carrier to reduce the vibration and radiated noise of the axle system. [17]

Another noise control study including spectral analysis methods was completed by Yalçınkaya [18] for Backhoe Loaders having an attenuation of 2 dB. The thesis aimed at controlling the noise emitted by the Backhoe Loader with noise source identification methodology including sound power level determination tests, coherent output power tests and sound intensity measurements. Comparative evaluation of results of spectral analysis of exterior noise emitted by the machine with possible noise sources reveals the possible noise source frequencies as cooling fan blade passing frequency, hydraulic pump operational frequency and engine firing frequencies. Changing the design of cooling fan and applying proper vibration isolation parts helped to decrease overall sound power level by 2 dB.

2.2 Noise Control Strategies

There are basically three ways in any noise control study;

- Noise control at the source of the sound
- Noise control along the path through which the sound travel
- Noise control at the receiver of the sound

The 3rd item was not a proper way for this thesis study because the aim of the study is to reduce the noise levels of the machines. Therefore, the 1st and 2nd items were used in the thesis. Noise and Vibration in dynamic systems can be decreased by a number of means. These can be generally classified into active, passive, and semi-active methods.

Active control includes the use of certain active elements like speakers, actuators, and microprocessors to produce an ‘‘out-of phase’’ signal to electronically mask or cancel the disturbance. Active and semi-active noise control applications are not generally used in automotive industry due to their complexity and high cost levels. Due to the fact that passive noise control applications are simple to assemble and economical, they are commonly used on tractors or other vehicles.

The most commonly used passive control ways for air-borne noise include the use of absorbers, barriers, mufflers, silencers, etc. For reducing structure-borne vibration and noise, several methods are also available. Sometimes, by changing only the system’s stiffness or mass to modify the resonance frequencies can decrease the unwanted vibration as long as the excitation frequencies do not change.

However, the vibrations need to be isolated or dissipated by using isolator or damping materials. For example, Rao [19] described the application of passive damping technology using visco-elastic materials to control noise and vibration in automotive industry. Special damped laminates and spray paints suitable for mass production and capable of forming with conventional techniques are now manufactured in a continuous manner using advanced processes. These are widely used in the automotive industry in some applications to reduce noise and vibration.

The use of damping treatment in the automotive and aerospace industries is made possible by the advancements in production processes that are cost-effective and suitable for high volume production. [19]

Ghosh et al [20] studied to control the noise level of a diesel engine by developing a new exhaust muffler, since exhaust as a noise source might be the single biggest contributor to the overall noise from the engine.

For the same power rating, diesel engines are noisier than gasoline engines, since the combustion characteristics of diesel engines produce more harmonics than the slower combustion of gasoline. An unmuffled gasoline engine emits exhaust noise in the range from 90 to 100 dB (A), while an unmuffled diesel engine under identical conditions emits exhaust noise in range from 100 to 125 dB (A). This work aims at improving a new muffler for the vehicle having a diesel engine and then comparing the results in terms of both acoustic performance and engine performance with respect to an unmuffled exhaust system of a vehicle. [20]

A new design, by Ricardo with the application of the engine performance simulation program, was taken up, and this was modified in certain aspects to be fit the engine used in the study. The muffler was produced and data were recorded for evaluation of the acoustic performance as well as the engine performance and for purposes of comparison with the existing muffler as well as with the unmuffled exhaust system. The new muffler was found to be better than the existing one in terms of both acoustic performance and engine performance. [20]

With the new muffler, the maximum noise reduction was 19.3 dB (A) and the maximum brake thermal efficiency was 39 per cent, while with the existing muffler the corresponding values were 14.5 dB (A) and 37.4 per cent. This work finally experimentally shows that the results from software can be changed and applied to an alternative design. [20]

Besides the previous mentioned study, Ghosh et al [21] aimed at predicting the noise level by mathematical modeling the exhaust muffler and validating the analytical results with the experimental results of engine. Since the pressure drop in an exhaust muffler plays an important role for the design and development of a muffler, the prediction of pressure drop by mathematical modeling will be very useful for the design and development of muffler.

It was observed that brake thermal efficiencies without muffler and existing muffler were little higher than that of modified muffler. The brake thermal efficiency with modified muffler is little less than the without muffler, because of the higher pressure drop in case of modified muffler in comparison to without and existing mufflers. When sound level was 116.8 dB (A) by without muffler, at that time we get sound level 96.4 dB (A) within existing muffler, 82.2 dB (A) within the new modified and fabricated muffler and finally 77.047 dB (A) from mathematical modeling. It was interesting to note down the variations between the sound levels variation measured by modified and fabricated muffler and theoretical muffler is 4.3%. This was simply due to the reasonable assumptions taken in the mathematical modeling. [21]

The prediction of intake system noise of an internal combustion engine is one of the important issues regarding the noise attenuation of an internal combustion engine. In the study of Jeong-Guon Ih et al [22], noise source parameters of an engine intake system during running-up conditions were measured by using the direct method employing two external loudspeakers, turned on simultaneously, and three microphones for the separation of upstream and downstream wave components. Predicted insertion loss and radiated sound pressure level using the measured source parameters were compared with those of measured data and predicted data employing several idealized source models which have been adopted for the calculations.

Even for the frequencies of firing harmonics, it was shown that the prediction precision was enough for the practical application in the acoustical development of automotive intake system. The measured source parameters were compared with the idealized source models such as constant pressure source, constant volume-velocity source, and anechoic source. Idealized source models yielded poor prediction of the insertion loss and radiated sound pressure loss in comparison with those predicted with measured impedance. [22]

Considering these results in mind, it was suggested that, for the purpose of approximate calculation of the overall trend of insertion loss and radiated sound pressure loss of the intake system, anechoic source model and measured source impedance with a cold engine condition could be good alternatives. [22]

Another noise attenuation application for tractors or heavy-duty vehicles can be accomplished by cutting down the hydraulic pump noise.

Tractors are equipped with external gear pumps in order to feed the hydraulic systems to operate and lift the implements necessary on the field work. These external gear pumps basically suck the oil at one side and send it to the other side with the required properties, pressure, flow, etc. The external gear pumps have two meshing gears generating required vacuum at one side. These gears are mainly spur gears due to cost and easy operational reasons.

However, it had already been shown that using helical gears instead of spur gears decreased considerably the overall noise level of hydraulic pump. Cesare Angeloni [23] published an article in Machine Design Magazine about gear pumps noise reduction. Conventional, high-pressure gear pumps are noisy because they trap and compress fluid between gear teeth as it rotates. The result is a sharp pressure rise that generates noise especially above about 1,500 rpm. Continuous-contact pumps feature helical gears that do not trap fluid as they rotate, as is the case with conventional gear pumps. This minimizes pressure ripple and gives high efficiency and quiet operation at speeds to 5,000 rpm. The Continuous-contact pump design eliminates compressed oil between gears, yielding smooth pressure changes, high efficiency, and quiet operation at speeds to 5,000 rpm. Besides, noise levels range from 52 to 68 dB (A) at 2,750 rpm, based on ISO 4412 testing.

An experimental thesis work was completed by Fatma Ceyhun Şahin [24] in 2007. Thesis study was focused on experimentally investigating pump noise at design and off-design operations and its relations with pressure fluctuations. Two small size pumps are placed in a semi-anechoic chamber and operated at various system conditions and speeds. Pump operational data, noise data and time dependent pressure data are recorded. Coherence spectrum between sound pressure level and hydraulic pressures at inlet and outlet are obtained. During study specific software Soundbook SAMURAI was used. The experiments have indicated that system characteristics or pump size do not have any influence on the noise of pump investigated in the study.

On the other hand, pump characteristics are found to be distinguishable by means of peak frequencies on the sound spectra which are proportional to blade passing frequency. Results of cross correlations also show that, pump outlet pressure is a more significant source of noise than pump inlet pressure. [24]

2.3 Flow Induced Noise

Frequently in commercial work in the oil and gas sector, one can come across situations in which the flow of gas through a given component generates extremely high levels of noise. This noise can be an occupational hazard, environmental or safety issue and in certain cases can lead to equipment damage.

Tractors have hydraulic systems in which pipes and hoses are used mostly to transfer hydraulic oil. Short-term management of the problem involves expensive loss of productivity and long-term solutions can be extremely costly, not least owing to the downtime necessary to install them.

In order for a treatment solution to be cost effective, the problem must first be accurately diagnosed and second the effectiveness of the treatment must be accurately and reliably evaluated. Both these aspects require prediction methods that are accurate and reliable, but at the same time fast enough to be employed in design iterations.

Direct flow simulation has not yet advanced sufficiently to be able to predict noise generation and analytical techniques, while playing an extremely important role in establishing engineering practice, are insufficient on their own. Naturally, empirical methods have played an essential role in the development of current methodologies, but for trouble-shooting new or modified geometries, they are extremely time-consuming and expensive.

Flow induced noise and its effects can be seen in other industries as well as in automotive industry. Faruk Emre Güngör [25] studied the some of the noise source like air-flow noise, fan noise in HVAC systems and prepared noise prediction software for HVAC systems. Aerodynamic noise from all types of fans can be broadly divided into a rotational component and vortex component associated with turbulent mixing process.

The rotational component is associated with the impulse given to the air each time a blade passes a given point and is hence a series of discrete tones at the fundamental blade passing frequency and harmonics thereof. In addition to aerodynamic noise, there are usually several non-aerodynamic sources of noise in equipment involving fans as well as other types of rotating machinery. [25]

Such mechanical sources include noise resulting from unbalance and misalignment, bearing noise, brush noise, magnetic noise, and belt noise. The number of blades of a centrifugal fan generally is governed by optimum airflow design. [25]

The noise generation decreases but slightly for more than the optimum number of blades. A shroud around a propeller fan may serve to reduce noise considerably if it is working properly. Such a reduction is generally most effective at the higher harmonics. However, if the flow breaks down over part of the shroud, the noise may become considerably worse than for an un-shrouded case. As the operating pressure across axial fans is increased, the maximum sound intensity is shifted from the fundamental to higher harmonics. This effect is not observed for centrifugal fans. [25]

Although fans are a major source of sound in HVAC systems, they are not the only sound source. Aerodynamic sound is generated at duct elbows, dampers, branch takeoffs, air modulation units, sound attenuators, and other duct elements. [25]

Produced by the interaction of moving air with the structure, the sound power levels in each octave frequency band depend on the duct element geometry and the turbulence of the air flow and the air flow velocity in the vicinity of the duct element. During the study, noise prediction software focused mainly on the noise sources generated by flow induced noise. [25]

The previous study was about the air-flow noise sources in HVAC systems. This type of study was done for axial fans in railcar systems by Cleon and Williaime [26].

The study illustrated the operation of an axial fan and then the main sources of noise generated by this type of fan. The interactions between acoustic emissions and mass output are then described to illustrate the advantages of an acoustic and pneumatic predictive device. Finally, a newly designed of axial fan on the railcar reduced the noise emission by 10 dB (A) without decreasing the original cooling performances. Besides, this application helped to reduce energy consumption. [26]

CHAPTER 3

IDENTIFICATION OF NOISE SOURCES

3.1 Methodology

This thesis is aimed at identifying the possible noise sources of a wheeled tractor and reducing the noise levels below the necessary legislative limits by controlling these noise sources using the proper methodologies.

These methodologies consist of sound intensity measurements and spectral analysis of exterior noise and comparison of data acquired by these tests. In order to understand the current situation of original prototype tractor's noise levels, the operator's ear and pass-by noise levels of original prototype tractor were measured and compared to the legislative limits shown in Table 3.1.

Table 3.1 Legislative Noise Limits

Wheeled Tractors		Noise Level Limit
Legislation	Description	dB (A)
74/151 [2]	Pass-By	89
77/311 [1]	Operator's Ear	86

Then, sound intensities at three sides of the tractor were measured. Sound levels were measured on the test tractor to generate a basis for thesis research at the early stage.

During these measurements and tests, as a first impression, at the engine idle speed and the test tractor is at stationary position, the engine noise is noticed clearly. However, considering the measurements when the tractor moves shows that there is noticeable transmission noise or drive-train noise.

Spectral analysis of the noise radiated by the test tractor is conducted to describe the current situation. Estimates for frequency spectra of noise at front, right and left sides were obtained. Possible noise sources of the machine are examined through inspection of these estimated spectra. That is, frequencies at which peaks are observed in the noise spectra are compared with the calculated operational frequencies of the possible noise sources. The classification of the sources helps to clarify the sources.

There are several dominant and minor possible noise sources on tractors. It is important to classify them according to their contribution into overall sound power emitted by tractor. However, it is not easy to identify and control all the possible noise sources. Usually, it is a better engineering approach in terms of cost management to know which source has the biggest contribution and which source has the smallest. Therefore, sound pressure spectra are inspected to understand the significance of each frequency. Due to this fact, noise source ranking is used in noise source identification studies. The necessary terms and formulae et al [27] are described for noise source ranking. The physical acoustic pressure based on sound pressure level can be expressed as;

$$Lp = 20 \log \left[\frac{P_{rms}}{P_{ref}} \right] \quad \text{[dB]} \qquad (3.1)$$

where the reference pressure is taken as,

$$P_{ref} = 20\,x10^{-6} \qquad \text{[Pa]} \qquad (3.2)$$

By using this equation, with known A-weighted sound levels at each frequency, one can calculate A-weighted acoustic pressure values for each frequency. Also, the overall A-weighted acoustic pressure value can be calculated. Therefore, the ratio of an acoustic pressure of a chosen frequency over the overall pressure will be acquired.

$$P_{rms} = 10^{\frac{L_p}{20}}\,x\left(20\,x10^{-6}\right) \qquad \text{[Pa]} \qquad (3.3)$$

$$P_{i} = 10^{\frac{L_{pi}}{20}}\,x\left(20\,x10^{-6}\right) \qquad \text{[Pa]} \qquad (3.4)$$

Finally the ratio is,

$$RATIO = \left(\frac{P_i}{P_{rms}}\right)x100 \qquad [\%] \qquad (3.5)$$

Where,

P_i : A-weighted acoustic pressure value for i^{th} frequency

P_{rms} : A-weighted acoustic pressure value for whole frequency range

L_{pA} : A-weighted sound level value for whole frequency range

L_{pA_i} : A-weighted sound level value for i^{th} band frequency

Noise source ranking should be done by using A-weighted sound levels, because A-weighted values are more correlated with hearing loss incurred for comparison concerning on humans due to exposure to noise. The A-weighting is the standard weighting for outdoor community noise measurements and is commonly used for noise measurements within architectural spaces and within vehicles. It accounts for the sensitivity of human ear to low intensity sounds. The A-weighting reduces the sensitivity of the measuring instrument to both low and very high frequency sounds. In the study, each frequency is considered as a noise source, and the corresponding amplitude at that frequency can be considered as the contribution by that noise source. If the ratio (Equation 3.5) of source level (Equation 3.4) to the overall level (Equation 3.3) is calculated, then the amplitude at that frequency, i.e. the contribution of that noise source can be acquired.

Dominant peak center frequencies where noise energy peaks out in the spectral analysis are obtained. These observed frequencies from spectral analysis are compared with the operational frequencies on the tractor. The peaks are measured, ranked and finally classified. After this comparison, some of the noise sources can be identified.

Nevertheless, this method cannot be used for the sources with high ranking and not related with operational frequencies. These types of unidentified sources are accepted as either hydrodynamic or vibro-acoustic originated sources. So, it would be better to reveal the relation between the bandwidth of the mentioned peaks in the spectra between vibrating parts of tractor and the measured noise. The peaks with wider bandwidth could be from hydrodynamic origin and on the other hand the peaks with narrower bandwidth could be from vibro-acoustic origin.

It is known that anything moving or spinning in a direction at any speed will generate vibration at some level. Most systems will experience wear in time, increasing vibration levels. It needs only a small amount of vibration energy to generate noticeable, audible noise. In other words, a small vibration force can create a significantly audible noise problem. The unidentified vibro-acoustic originated sources should be investigated by means of tractor plates called fenders covering tires, front hood covering engine and steps or platform on which seat and operator were positioned.

On the other hand, hydro-acoustics or hydrodynamic originated noise term developed from the need to understand the causes and origin of flow-induced noise. The major way to the knowledge has been the study of sound generated by particular flow processes. The source of sound, rather than the sound itself, became the center of attention, and real problems of identifying the source had their origin in the fact that the source is ambiguous. That is, the source determines the sound but the sound cannot prescribe the source.

Hydrodynamic sources are always associated with unsteady flow, most noisy flows being unsteady. Small perturbations about the steady state grow into turbulent sources of sound. Sound generated by turbulence can be the triggering disturbance of instabilities that grow into turbulence.

Another detailed analysis to study these vibro-acoustic and hydrodynamic noise sources is the measurement of sound intensities. Sound intensity is the measurement of flow of sound power through a unit area. In addition to this description, sound intensity can be defined as sound energy flux at a certain point or surface.

$$\vec{I} = P\vec{U} \tag{3.6}$$

$$W = \int_S I_n dS \tag{3.7}$$

In discrete sense, Equation (3.7) can be thought as,

$$W = \sum_{i=1}^{N} I_{ni} S_i \qquad (3.8)$$

where,

P: Instantaneous Pressure at a Point (Pa)

$\vec{U}$: Particle Velocity Vector at a Point (m/s)

$\vec{I}$: Sound Intensity Vector (W/ m²)

W : Sound Power (W)

S : Surface Area (m²)

Sound intensity can be calculated by Equation (3.6). It is seen that sound intensity is a vector quantity. Moreover, sound power of a source could also be calculated using sound intensity measurements on the hypothetical surfaces such as, hemispheres, box shapes, etc. The surface can be divided into equal sized grids and over each grid center sound intensity is measured. Then intensity values and the total area are used to calculate sound power emitted by that side of the tractor. As a more general explanation, sound intensity can be used for the determination of sound power, noise source location and determination of the transmission loss of structures. The sound power can be determined from intensity measurements on a hypothetical enveloping surface enclosing the noise source. The sound power calculation using sound intensity measurements is covered in relation to the ISO standards numbered as 9614-1 [8], 9614-2 [9] and ISO 9614-3 [10].

This surface can be a hemisphere or a cube. Considering the fact that sound intensity is a vector quantity, steady background noise does not affect the measured data in sound intensity measurements. Consider a steady background noise source in an environment where sound intensity test was conducted. Sound power emitted by background noise source for the measurement surface is calculated as zero, because the net sound energy propagating into hypothetical surface is zero. As a result, background noise does not affect the intensity measurement for the source of interest.

Therefore, sound power levels can be calculated both from sound pressure measurements and sound intensity measurements. Advantages of calculating sound power levels from sound pressure measurements might be summarized as;

- The instrumentation is more available and is quite cheaper
- Sound pressure is quite easy to measure and do not require expertise
- Numerous international standards are available for converting sound pressure measurements to sound power levels

On the other hand, advantages of calculating sound power levels from sound intensity measurements can be summarized as;

- The measurements can be carried out in an open site eliminating the requirement of special chambers or rooms
- The effect of continuous background noise need not be considered due to the fact that sound intensity is a vector quantity
- Intensity measurements help to determine the direction from which the noise tends to propagate.

3.2 Sound Intensity Mapping

As it was mentioned earlier, sound intensity is the measurement of flow of sound power through a unit area. This measurement quantity, i.e. sound intensity is a vector quantity and hence it has both magnitude and direction. This measurement tends to help the designers to determine the direction from which the noise tends to propagate.

Sound intensity measurement tests form a basis for the determination of sound power. Moreover, it provides an edge over sound pressure measurements because of its capability to phase out steady background noises. This also has the advantage of measuring the noise intensity in the near field of the machines.

Sound intensity values can be negative or positive. If it is negative, then it means that there is sound energy absorption instead of propagation. Whereas, if it is a positive value, then this time it is called as the source behavior because it means that there is energy flow away from the source.

Measuring sound intensity and calculating sound power is a necessary process to be done during noise source identification applications. One of the other advantages of sound intensity measurements includes the ability to determine the directionality of sound, and to quickly localize noise "hot spots". In order to show these “hot spots”, it is important to generate intensity maps. Proper mapping software will provide the noise engineer with a contour map of noise levels for use in noise control studies on tractors or any other industrial products.

The pictures shown below are from different industries related to intensity mapping applications. In Figure 3.1 an intensity map for a washing machine is shown. The “hot spots” appeared at the bottom of the washing machine.

In Figure 3.2, an intensity map for an earth moving machine and in Figure 3.3, an intensity map for an internal combustion engine is shown.

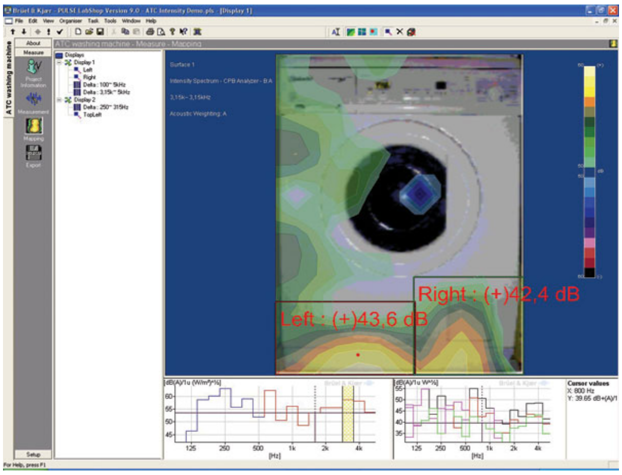


Figure 3.1 The intensity map of a washing machine [33]

Figure 3.2 The intensity map of an earth moving machine [34]

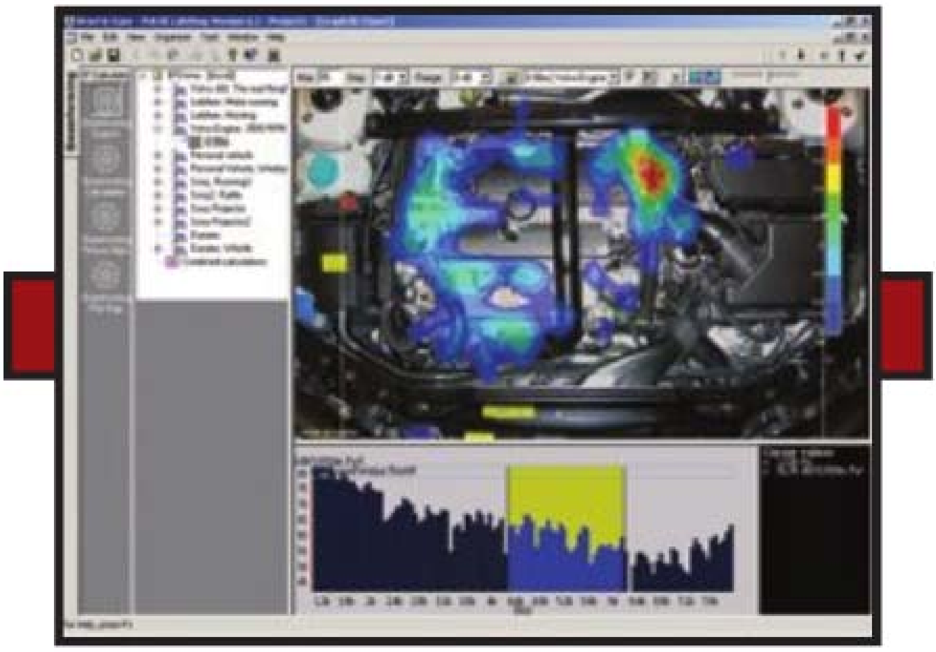


Figure 3.3 The intensity map of an internal combustion engine [35]

As it can be seen in above figures, the so called “hot spots” can be clearly identified as in red color having the highest intensity values. By analyzing and studying these intensity maps carefully, design engineers can focus on the “hot spots” and apply necessary noise control applications to these “hot spots”. Otherwise, it will be very difficult and time-consuming using trial-and-error type efforts.

3.3 The Test Tractor T480

The study of this thesis work has been done on the test tractor named as T480 for domestic market. The pictures of T480 tractor can be seen in Figures 3.4 and 3.5. The tractor is a 2-wheel drive less cab tractor having a 48 Hp, naturally aspirated, diesel engine and a very economical small duty driveline for basic agricultural work especially for vineyards and orchard applications.

The developed noise source identification methodologies and noise control strategies have been applied to T480 tractor.

Figure 3.4 The Side View of the Test Tractor

Figure 3.5 The Front View of the Test Tractor

The engine of T480 tractor is a 4-cylinder diesel engine. The engine is water cooled vertical in-line and 4-cycle naturally aspirated. Its rated speed is 2800 rpm with 48 BHP and the maximum speed is nearly 3000 rpm. The exhaust emission of the engine satisfies Tier III norms which will be an obligation to tractor manufacturers for domestic market in January 2011.

The fan speed ratio is 1.25÷1. The hydraulic lift gear pump has a capacity of 16.8 cc per revolution and the steering system gear pump has a capacity of 8.8 cc per revolution.

The picture of the engine and the wireframe 3D model of the engine can be seen.

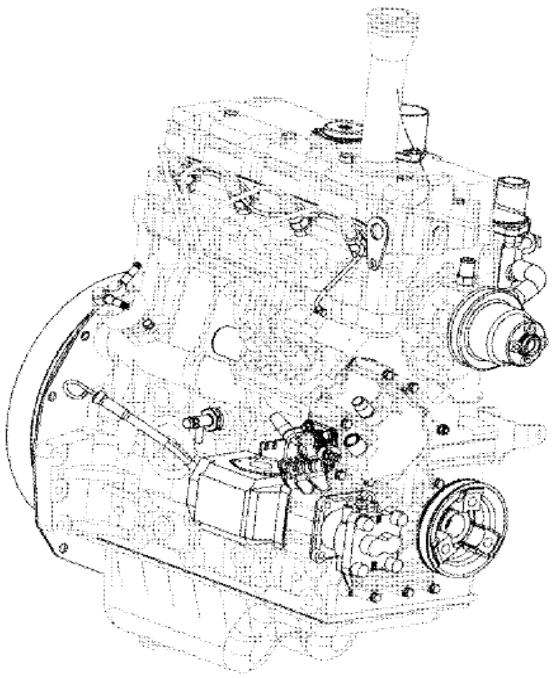


Figure 3.6 The Engine Wireframe 3D Model

Figure 3.7 The Front View of the Engine

In order to be able to start the study, the original tractor's noise levels are measured. Obtained sound pressure level is compared with the permissible maximum level stated in the legislations.

Operator's ear noise level was measured according to the instructions stated in the legislation. After the comparison, it was seen that the operator's ear noise level of the original prototype tractor was higher than the legislative limit, 86 dB (A). Potential noise sources of the tractor can be seen in Figure 3.8.

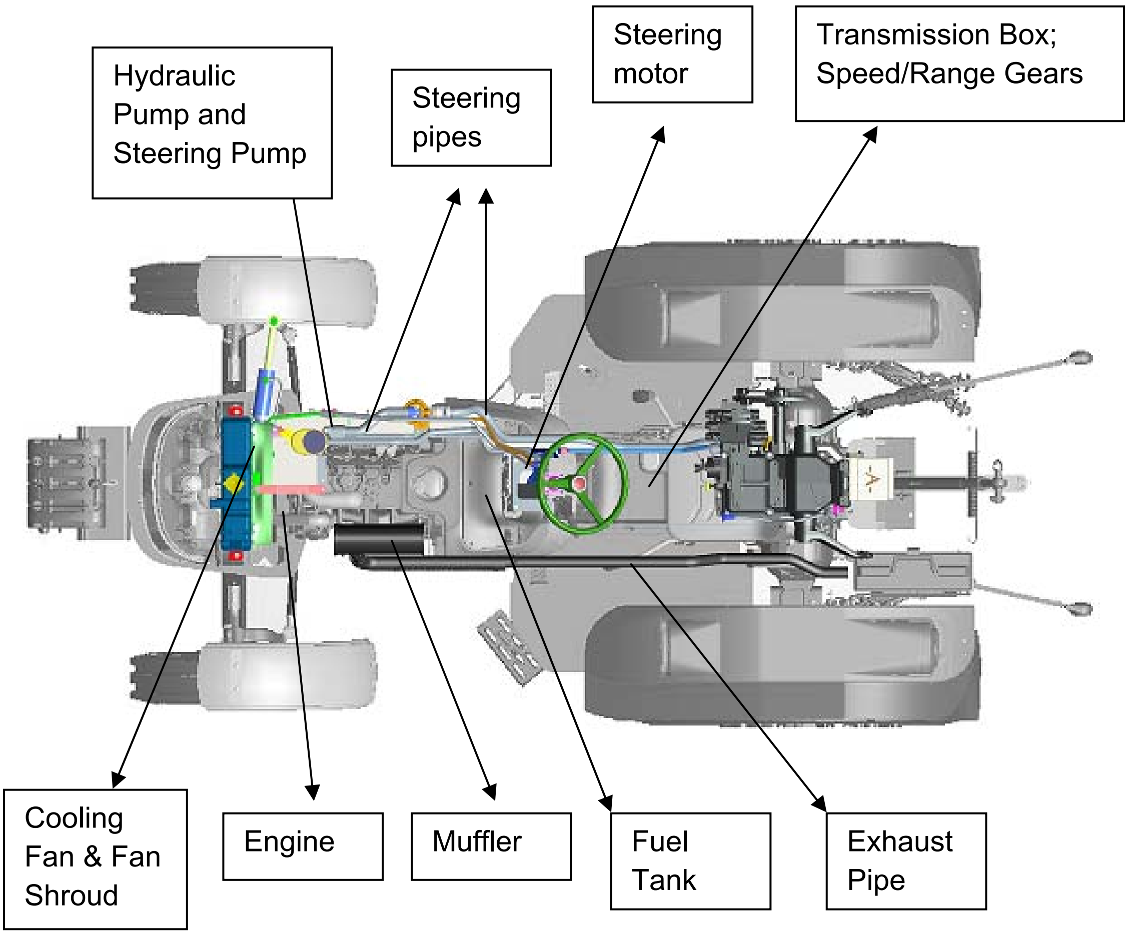


Figure 3.8 Potential Noise Sources on the Tractor

According to the legislation 77/311/EEC, the noise level for T480 tractor must not exceed 86 dB (A). The first set of measurements taken from the original prototype T480 tractor completed in September 2009. Measurements showed that the original prototype tractor's noise level is between 88.5 dB (A) and 89 dB (A). Therefore, 3 dB (A) reduction in noise levels at the operator's ear should be achieved to meet the requirement stated in the directives and to complete the processes necessary for a complete tractor homologation.

Noise sources on the tractor should be determined. Spectral analyses of the exterior noise from three sides of the machine are performed. Operational frequencies on the machine are first calculated. Then, these frequencies are compared with the estimated frequencies in the spectral analyses. Generally, noise sources can be mainly classified in three groups:

- Sources related to the engine rotational speed
- Sources not related to the engine rotational speed (vibro-acoustic, hydrodynamic or hydro-acoustic, aero-acoustic)
- Other sources.

For example, cooling fan rotational frequency, fan blade passage frequency and engine firing frequency can be classified into the first group. However, vibration due to fender could be covered under third group.

Noise source ranking is the next step in thesis methodology. Flow generation devices such as cooling fan or engine and firing are the most potential sources which can be identified with the operational speed. In addition to this, the uncorrelated sources with operational speeds, classified into the second group, are also found to be that they are relatively of higher importance.

Sound intensity measurements are carried out to examine the vibro-acoustic noise level radiated from mainly large surfaces in vibration like front hood, fenders or hydraulic and steering pumps. Moreover, intensity maps were generated to describe the noise emission from these areas.

Acoustic intensities on front hood, fenders, exhaust pipe and hydraulic pump should be measured; because the area of the front hood is larger than the other parts of the machine. The front hood due to its size might act as a possible vibro-acoustic noise source. The operational frequency of the hydraulic pump is also investigated by analyzing results of intensity measurements.

During original prototype tractor’s measurement tests in which the test tractor is moving at the required speed, it was realized that there is significant drive-train noise. Tests have been conducted to verify this statement. Hence, this drive-train noise could cause structure-borne noise as well.

CHAPTER 4

EXPERIMENTAL STUDIES

The original prototype tractor used during the experiments is manufactured by Türk Traktör and Ziraat Makineleri A.Ş. These experiments were accomplished on an engine having a 4-cylinder diesel engine. The engine is water cooled vertical in-line and 4-cycle naturally aspirated. Its rated speed is 2800 rpm with 48 BHP and the maximum speed is nearly 3000 rpm. There are two separate hydraulic pumps on the test tractor. One of them is the hydraulic pump a fixed displacement external gear pump. This pump serves the necessary flow 16.8 cc/rev to the main hydraulic system including the tractor's hydraulic lift. The other pump is the steering pump serving flow to the steering system, i.e. steering motor of the tractor. This pump has 8.8 cc/rev flow rate. Both pumps can be seen in Figure 4.1.

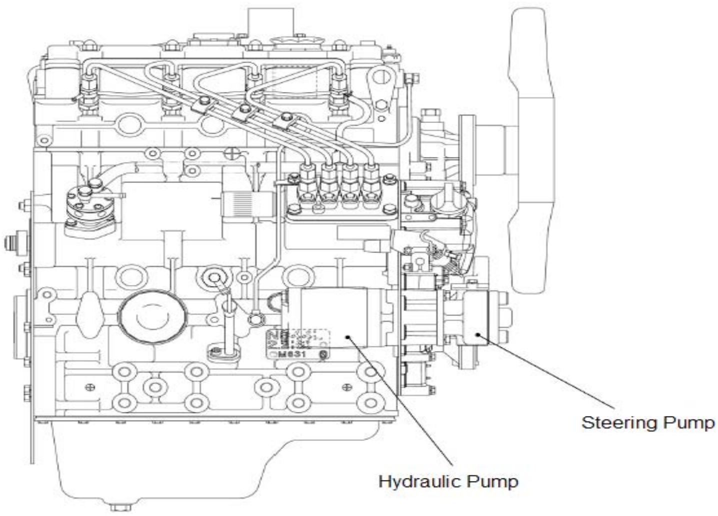


Figure 4.1 The Left View of the Engine Wireframe

The specifications for both hydraulic and steering pumps are given in Tables 4.1 and 4.2.

Table 4.1 Specifications of the Steering Pump on the Engine

Specifications of Steering Pump		
Volume	[cc/rev]	8,8
Maximum Pressure	[MPa]	13,7 (140 kgf/cm2)
Instantaneous Max. Pressure	[MPa]	16,2 (165 kgf/cm2)
Pump Speed	[rpm]	600 ~ 3000

Table 4.2 Specifications of the Hydraulic Pump on the Engine

Specifications of Hydraulic Pump		
Volume	[cc/rev]	16,8
Maximum Pressure	[MPa]	17,2 (175 kgf/cm2)
Instantaneous Max. Pressure	[MPa]	20,6 (210 kgf/cm2)
Pump Speed	[rpm]	600 ~ 3000

Noise level experiments will be classified into three groups:

1. Sound Power Level Determination
2. Spectral Analysis of Tractor Noise
3. Sound Intensity Measurements

Tests related to tractor noise source identification studies will be explained in detail. Firstly, original sound power level of the machine is determined. Secondly, frequency spectrum of the exterior noise is determined. This spectrum is evaluated with the expected dominant noise frequencies in terms of tractor operational frequencies. Third step is a rather more detailed noise source identification technique called sound intensity technique.

Fourth step is the comparison of the results and finally fifth step is the final measurements on the tractor having final modifications and explaining the changes, applications and their comparisons.

4.1 Sound Power Level Determination Using Sound Pressure Level Measurements

Effective noise source identification studies require measurement of sound pressures and intensities. When a noise source generates noise, the sound pressure from that source is generally dependent on the surroundings. Because, the sound pressure level will be different for the same source whether the source is positioned indoors or outdoors.

On the contrary, the sound power is generally independent of surroundings. Due to this reason, for design engineers who are interested in noise control applications, sound power calculations and comparisons are essential. Also, European Union requires the declaration of noise emissions in terms of sound power levels otherwise stated.

However, the sound power cannot be measured by using equipments. It can only be calculated and there are two ways to calculate it. One way is to calculate sound power from sound pressure levels and the other way is to calculate it from sound intensity levels together with necessary surface area and time values.

In the literature, there are 7 ISO standards explaining how to calculate sound power from sound pressures. These ISO standards can be seen below list.

- ISO3740:2001 Acoustics: Determination of sound power levels of noise sources. Guidelines for the use of basic standards [40]

- ISO3741:2000 Acoustics: Determination of sound power levels of noise sources using sound pressure. Precision methods for reverberation rooms [41]

- ISO3743-1:1995 Acoustics: Determination of sound power levels of noise sources. Engineering methods for small, movable sources in reverberant fields. Comparison for hard-walled test rooms [42]

- ISO3743-2:1997 Acoustics: Determination of sound power levels of noise sources. Engineering methods for small, movable sources in reverberant fields. Methods for special reverberation test rooms [43]

- ISO3744:1995 Acoustics: Determination of sound power levels of noise sources using sound pressure. Engineering method in an essentially free field over a reflecting plane [44]

- ISO3746:1999 Acoustics: Determination of sound power levels of noise sources using sound pressure. Survey method using an enveloping measurement surface over a reflecting plane [29]

- ISO3747:2000 Acoustics: Determination of sound power levels of noise sources using sound pressure. Comparison method in situ [45]

Among these standards, ISO3746:1999 [29] was taken as a reference for the calculation method due to its application which can be done over a reflecting surface, such as a tractor parking fleet area.

As it was explained in ISO3746:1999 [29], the sound power generated by a source can be measured in an open field free from nearby reflective surfaces while the engine was running at about maximum speed of 3000 rpm. Three microphone locations were selected and necessary measurements were recorded. Figure 4.2 shows an example of test area and microphone locations. The radius of hypothetical sphere is 4m, which is nearly twice the wheel base of tractor. According to ISO3746:1999 [29], the radius must be at least 3m.

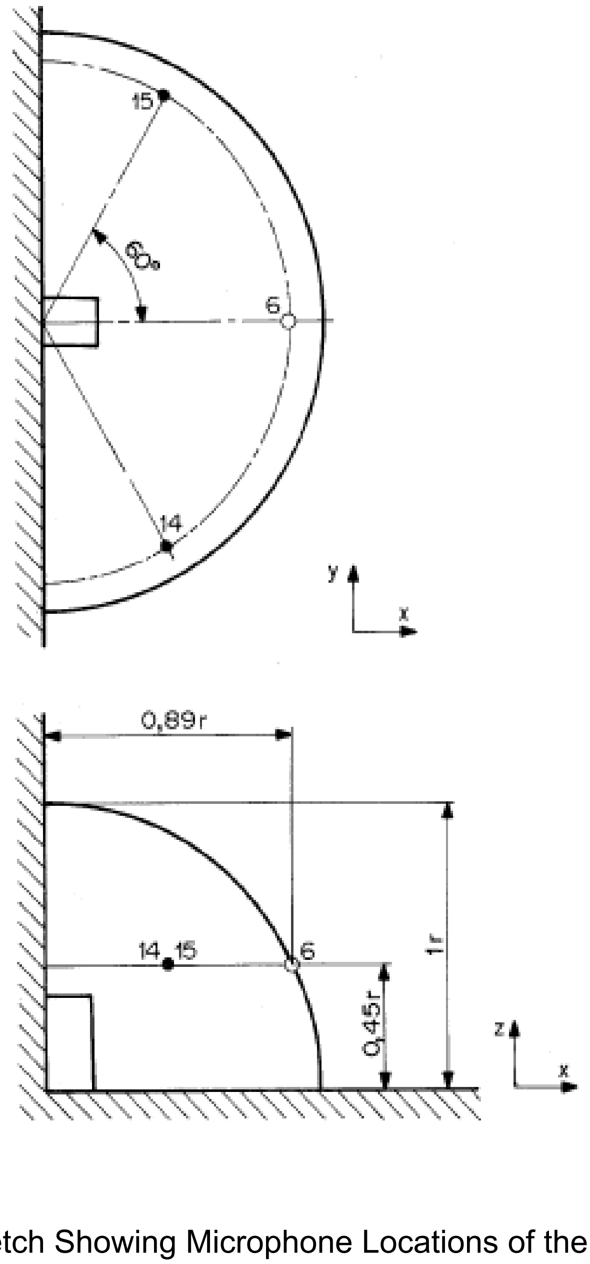


Figure 4.2 The Sketch Showing Microphone Locations of the Test Area [29]

Figure 4.3 Microphone Locations on the Test Site

According to ISO3746:1999 [29], the sound power calculation can be done by using the formulae;

$$\overline{L'_{pA}} = 10 \log \left[\left(\frac{1}{N} \right) \sum_{i=1}^{N} 10^{0.1 L'_{pAi}} \right] \qquad \text{[dB (A)]} \qquad (4.1)$$

Where;

$\overline{L'_{pA}}$: Average A-weighted sound pressure level of sound source for the reference surface area

L'_{pAi} : A-weighted sound pressure level as dB for i^{th} microphone location

N : Number of microphone positions

Normally, depending upon test environment and process, some correction factors should be considered during these calculations. However, since the difference between the background noise measured during tests and the sound pressure levels is above 10 dB, there is no need to have a correction factor for these calculations as it was explained in ISO standard.

Since there are no correction factors,

$$\overline{L_{pfA}} = \overline{L'_{pA}} \qquad (4.2)$$

Finally, the sound power equation becomes,

$$L_{WA} = \overline{L_{pfA}} + 10 \log \left(\frac{s}{s_0} \right) \qquad \text{[dB (A)]} \qquad (4.3)$$

Where,

S : Measurement Surface Area (m²)

And the reference area is,

s_0 : 1 m²

The measurement surface area is then,

$$s = \pi r^2 \quad , \quad r = 4 \ m$$

4.2 Spectral Analysis of Exterior Noise on the Test Tractor

In order to identify noise sources, spectral analysis of noise emitted by tractor which is the second step in the methodology was performed.

Measurements were carried out in free field where test site is tractor fleet area having asphalt surface in compliance with the standards. The test tractor is located at the center of the enveloping sphere as shown in Figure 4.4. Transmission of the tractor is in neutral position and the engine is kept running at nearly 3000 rpm.

Both tests were conducted in an open free field. Due to size of the tractor, there should be no sound-reflecting surface near the test area. Meteorological conditions such as weather, wind speed, relative humidity and temperature data are all recorded. Measurements are not performed during precipitation. In case of exceeding wind speed from 1 m/s, windscreen over the microphone is used. If wind speed exceeds 5 m/s, the sound level measurement tests should not be done.

In order to obtain steady state conditions, before measurements, the engine and the hydraulic system had run for some time to bring tractor up to normal operating condition temperatures.

In Figure 4.4, three sides used on spectral analysis can be seen. These are front, left and right sides. Rear side was not used during measurements. And in Figure 4.5, the measurement area can be viewed.

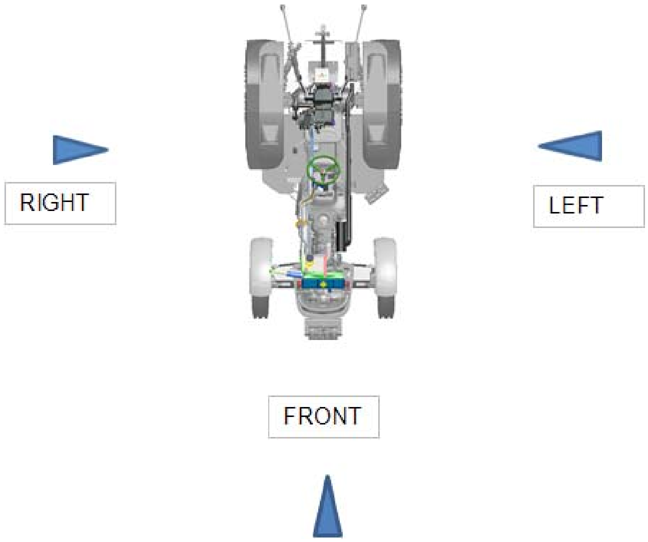


Figure 4.4 Microphone Locations on the Test Area of Spectral Analysis

Figure 4.5 Measurement Area

4.3 Sound Intensity Measurements

Sound intensity measurements providing the location and the frequency information of sound energy flux from the tractor is the third step in the noise source identification methodology. Sound intensity radiated by the test tractor is measured over suspected vibro-acoustic noise sources such as engine front hood and hydraulic pump. The grid application was done by using the grid instrument designed and produced by R&D Department of Türk Traktör and Ziraat Makineleri A.Ş. The grid frame has a flat surface divided into grids with edge lengths of 200mm and 200mm. This frame can be seen in Figure 4.6.

Figure 4.6 The Grid Frame used for Sound Intensity Measurements

The grid numbering was started from left bottom to left top and then from left to right in order to identify the measurement points. Sound intensity is measured from the center of squares constituting the grid.

During measurements Brüel&Kjaer Type 3595 Sound Intensity Probe Kit was used. Type 3595 is a two-microphone probe kit for measuring sound intensity and it can be seen in Figure 4.7. The probe set includes the 1/2" Sound Intensity Microphone Pair Type 4197 serving for 1/3-octave centre frequency measurements between 20 Hz and 6.3 kHz. [39]

Besides, UA-0781 type ellipsoidal windscreen on microphones is also used to minimize the wind effects during measurements.

Figure 4.7 The Sound Intensity Probe

In this experiment, the distance between the measurement surface and the microphones was chosen not to be closer than 300mm. In addition to that, 12mm microphone spacing was chosen due to its frequency range from 20 Hz and 6.3 kHz.

Calibration is performed by Brüel&Kjaer Type 4231 with its adapter type DP-0888. Calibrator can be seen in Figure 4.8 below.

Figure 4.8 The Sound Calibrator

The Sound Level Meter Type 2260 Brüel&Kjaer was used during all sound pressure level and intensity measurements and it can be seen in Figure 4.9.

Figure 4.9 The Sound Level Meter

The engine was running at 3000 rpm, the maximum engine speed, during sound intensity measurements. Engine revolution speed was measured with the digital laser remote tachometer Monarch PLT200 in Figure 4.10.

Figure 4.10 Digital Laser Remote Tachometer

The grid frame was set near tractor as it can be seen in Figures 4.6 and 4.11. Sound intensity measurements were started from the left bottom grid to top left. By using this method, left, front and right side of tractor were measured at the center of square grids.

Figure 4.11 The Left Side View of the Test Tractor during Sound Intensity Measurements

The grid numberings of right, front and left sides of tractor were shown in Figures 4.12, 4.13 and 4.14.

4	5	12	13	20	21	28	29	36
3	6	11	14	19	22	27	30	35
2	7	10	15	18	23	26	31	34
1	8	9	16	17	24	25	32	33

Figure 4.12 Grid Numberings for the Right Side of the Tractor

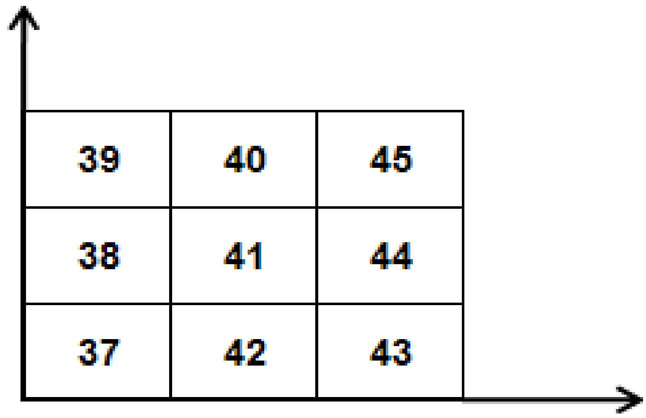


Figure 4.13 Grid Numberings for the Front Side of the Tractor

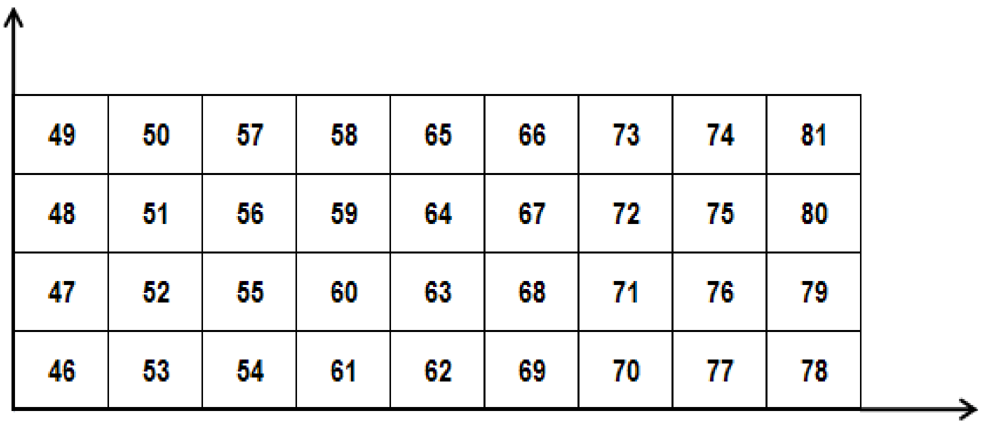


Figure 4.14 Grid Numberings for the Left Side of the Tractor

There are 78 grids on cumulative. The measurement time for each grid is 5 seconds already tuned on sound level meter. After every 5 seconds measurement grid was changed automatically by sound level meter, therefore all 78 points or grids were scanned and intensities were measured.

4.4 Sound Power Level Determination Using Sound Intensity Measurements

Another way of calculating the sound power level for a noise source is calculating from sound intensities over measurement surface areas. In the literature, there are three standards explaining sound power calculation from sound intensities:

- ISO 9614-1: 1994 [8] Acoustics – Determination of sound power levels of noise sources using sound intensity – Part 1: Measurement at discrete points

- ISO 9614-2: 1994 [9] Acoustics – Determination of sound power levels of noise sources using sound intensity – Part 2: Measurement by scanning

- ISO 9614-3: 1994 [10] Acoustics – Determination of sound power levels of noise sources using sound intensity – Part 3: Precision method for measurement by scanning

From intensity measurements, sound intensity levels were recorded. According to ISO 9614-2 and 9614-3 standards, the calculation can be done as followed;

$$\overline{I_{ni}} = I_{ref} \; 10^{\frac{\overline{L_{In}}}{10}} \qquad (4.4)$$

where, $I_{ref} = 10^{-12}$ (W/ m²)

Then, partial sound power values can be found by using formula,

$$W_i = \overline{I_{ni}}.S_i \tag{4.5}$$

Where, $S_i = 0,04$ m² for a square having sides of 20cm.

From here,

$$W = \sum_{i=1}^{N} W_i \tag{4.6}$$

Finally, to find sound power level, equation (4.7) should be used.

$$L_W = 10 \log \left(\frac{|W|}{W_{ref}} \right) \tag{4.7}$$

where, $W_{ref} = 10^{-12}$ (W)

CHAPTER 5

RESULTS OF EXPERIMENTS

5.1 Results of Noise Measurements

As it was explained in the methodology section, original prototype tractor's noise levels were measured at operator's ear position. Originally measured data is tabulated in Table 5.1. There are a number of requirements in order to do sound measurements. During the measurements, the weather was suitable, nearly 10°C and the wind speed was below 5 m/s. Moreover, the measurement field had no reflective parts or surfaces closer than 50m. The measurements were according to ISO 5131 [28].

Table 5.1 Noise Levels of the Original Prototype Tractor

Operator' Ear Level Measurements	Measurement Locations	
Noise Level dB (A)	Left Side	Right Side
T480 Tractor Original Prototype	88,5	89

5.2 Sound Power Level Calculation Using Sound Pressure Level Measurements

By using the techniques described in ISO3746:1999 [29] Standard and formulae given in Chapter 4, sound power calculation of the original prototype tractor was completed. The overall sound power generated by tractor was calculated as 107,2 dB (A).

Table 5.2 Sound Power Level Calculation of the Original Prototype Tractor

Microphone Position Point	Background Noise dB (A)	L'_{pA1}	L'_{pA2}	L'_{pA3}	$\overline{L'_{pA,T}}$	$\overline{L'_{pA}}$	L_{wA}
14	56,7	89,2	88,8	88,9	88,97	90,2	107,2
15	56,5	90,5	89,9	90,0	90,13		
6	56,6	91,1	90,8	91,0	90,97		

After analyzing the results shown in Table 5.2, it can be concluded that the measurements at point 6 yield the highest values. Point 6 was in front of tractor. This is simply due to the engine noise can propagate more easily through the grill in front of the tractor than from sides where engine front hood forms an obstacle to this propagation.

5.3 Spectral Analysis of Exterior Noise on the Test Tractor

The first two sections of this chapter showed that the original prototype tractor has an unacceptable noise level according to legislative limits and also the original tractor has sound power levels relatively high compared to other homologated tractors.

Having realized at the situation, the first step mentioned in this study is to find the critical frequencies from spectral analysis. In this study, 1/3 octave frequency bands were used. In Figure 5.1, plots of peak center frequencies with sound levels taken from measurements can be seen.

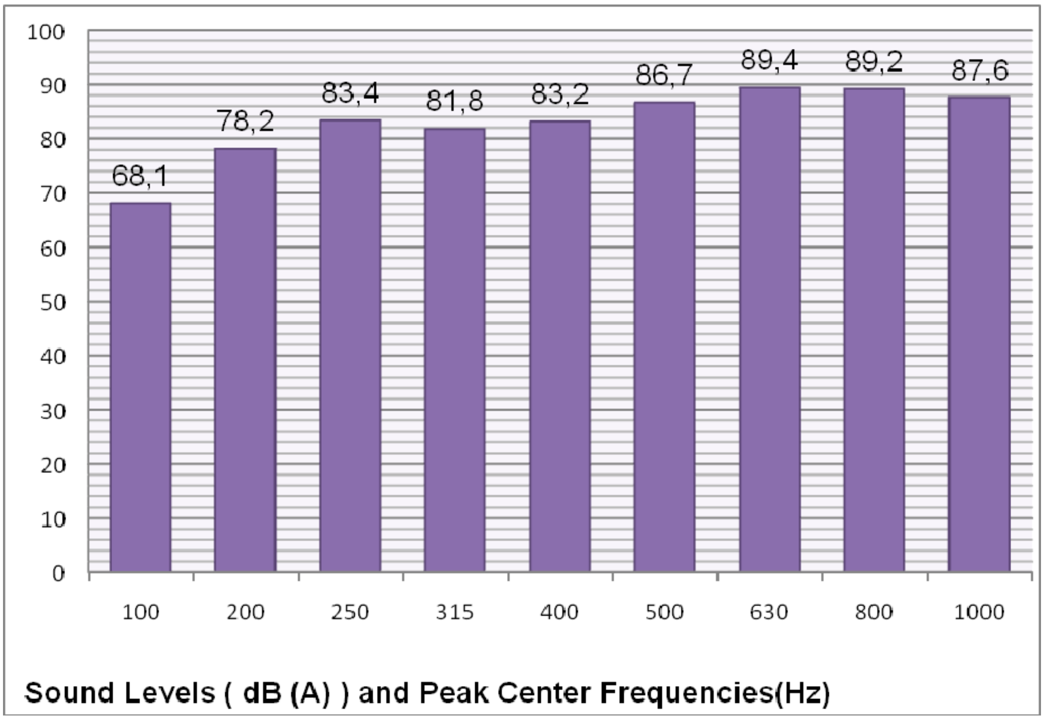


Figure 5.1 Sound Levels and Peak Center Frequencies on the Front Side

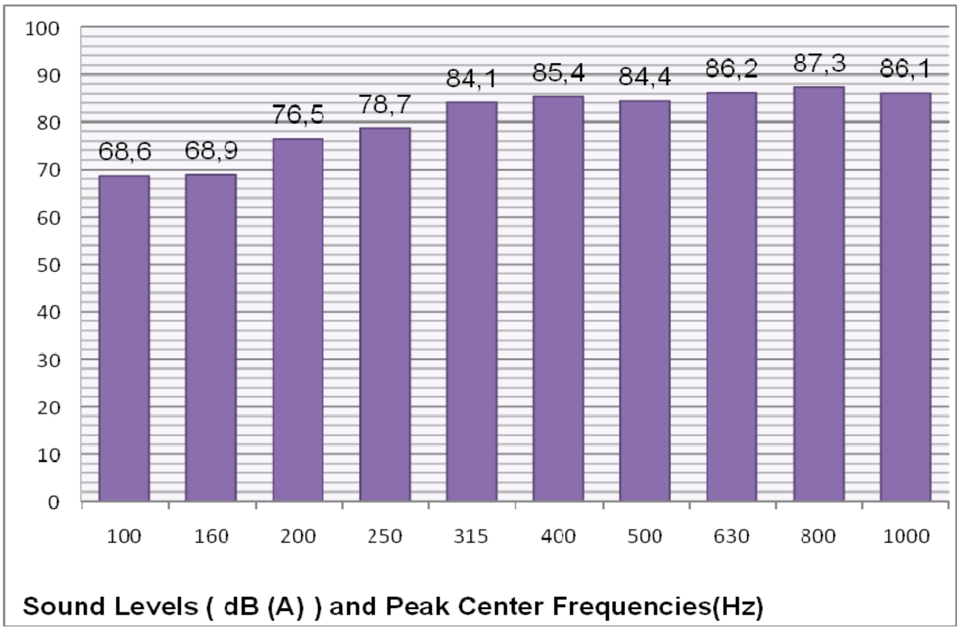


Figure 5.2 Sound Level and Peak Center Frequencies on the Left Side

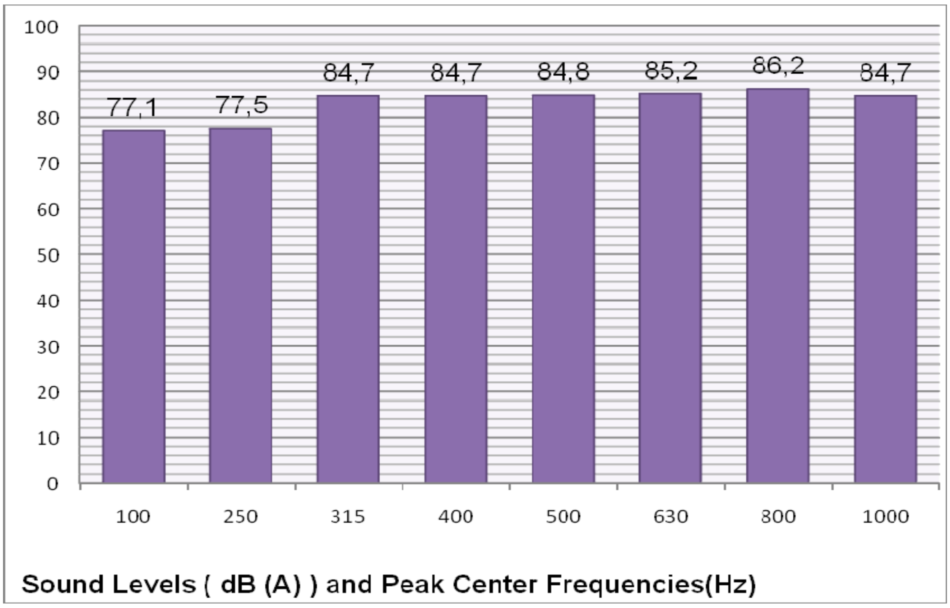


Figure 5.3 Sound Level and Peak Center Frequencies on the Right Side

5.4 Ranking Noise Sources of the Test Tractor

By using the equations given in Chapter 3, with known A-weighted sound levels at each frequency, the calculation was completed with known A-weighted acoustic pressure values for each frequency. Besides, the overall A-weighted acoustic pressure value was calculated. Thus, the ratios of acoustic pressure of chosen frequencies over the overall pressures can be ranked.

After the ratio calculations giving the contribution of each noise source, it was seen that 630 Hz and 800 Hz frequencies have the highest ratios for all three sides.

Table 5.3 Highest Ratio Peak Center Frequencies and Sound Levels on the Front Side

Peak Center Frequency (Hz)	Sound Level dB (A)
630	89,4
800	89,2

Table 5.4 Highest Ratio Peak Center Frequencies and Sound Levels on the Left Side

Peak Center Frequency (Hz)	Sound Level dB (A)
630	86,2
800	87,3

Table 5.5 Highest Ratio Peak Center Frequencies and Sound Levels on the Right Side

Peak Center Frequency (Hz)	Sound Level dB (A)
630	85,2
800	86,2

After analyzing these tables and peak center frequencies, it can be concluded that the ratios at these frequencies are the highest ratios and therefore, the study must focus on the noise sources corresponding to these frequencies as a first strategy.

Table 5.6 Peak Center Frequencies and Noise Sources

Peak Center Frequency (Hz)	**Noise Source(s)**
100	Engine Firing Frequency
200	Engine Firing Frequency
400	Engine Firing Frequency + Cooling Fan
800	Cooling Fan 2nd harmonic

5.5 Results of Sound Intensity Measurements of the Original Prototype Tractor

Sound intensity measurements are completed as described in previous chapters in order to identify vibro-acoustic originated noise sources of tractor. During the sound intensity measurements, the weather was 3°C and the wind speed was below 5 m/s. Moreover, the measurement field had no reflective parts or surfaces closer than 20m. The engine was at full throttle and running at 3022 rpm. During sound intensity measurements, 1/3 octave center frequencies were used having the frequency range from 20 Hz and 6.3 kHz for a 12mm spacer on sound intensity probe kit. [39]

The cooling fan on test tractor has 6 wings and the ratio is 1.25/1. Considering the engine maximum speed is 3000 rpm, the cooling fan blade passage frequency can be found as 360 Hz. According to 1/3 octave band center frequencies, this refers to 400 Hz. For 1/3 octave bands, 400 Hz center frequency has a range between 356,4 Hz and 449,0 Hz. Similarly, the second harmonic of 360 Hz will be 720 Hz referring to the center frequency of 800 Hz according to 1/3 octave band. For 1/3 octave bands, 800 Hz center frequency has a range between 712,7 Hz and 898,0 Hz.

Significant frequencies obtained from the results of sound intensity measurements are tabulated in Table 5.7.

Table 5.7 Peak Center Frequencies and Intensity Values from Sound Intensity Measurements

Frequency (Hz)	Tractor Right Side	Tractor Front Side	Tractor Left Side
	Sound Intensity (dB)	Sound Intensity (dB)	Sound Intensity (dB)
100	75,7	70,9	62,4
200	72,8	77,0	77,6
250	79,7	84,4	79,0
315	86,6	81,5	84,4
400	86,1	82,4	86,7
500	85,6	88,5	85,8
630	85,6	89,9	88,5
800	87,6	89,7	87,8
1000	85,2	87,4	86,6
1250	84,5	86,5	85,1

By analyzing the results, it can be inferred that the tractor from all three sides radiates high sound energy at 200 Hz, at 400 Hz and at 800 Hz. Besides, high sound energy can be seen at 500 Hz and at 1000 Hz. It is clear that at 400 Hz and 800 Hz cooling fan dominates noise level. Moreover engine firing frequency and its second harmonic appeared at 200 Hz and 400 Hz. Finally, at 315 Hz and at 630 Hz there is high sound energy radiated by the tractor.

The intensity contour maps of the test tractor for the important frequencies can be seen in below figures. The coloring is ordered such that red colored contours show the highest intensity values and the most important noise sources. However, yellow colored contours show the least important sources or the lowest intensity values. Finally, blue colored contours are used to differentiate these highest and lowest intensity values.

Figure 5.4 Left Side View for Sound Intensity Measurements

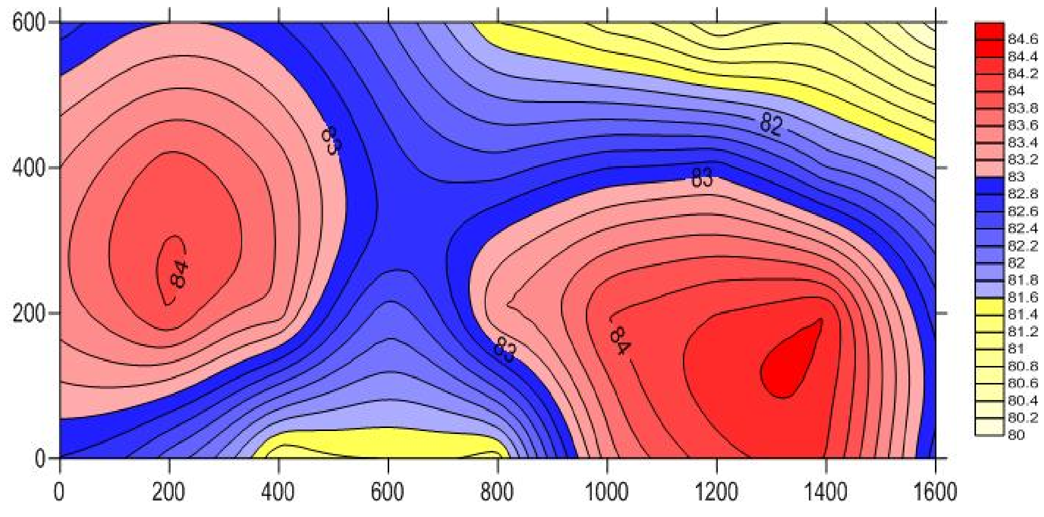


Figure 5.5 The Original Intensity Map of the Left Side of the Tractor at 315 Hz

Figure 5.6 Left Side View for Sound Intensity Measurements

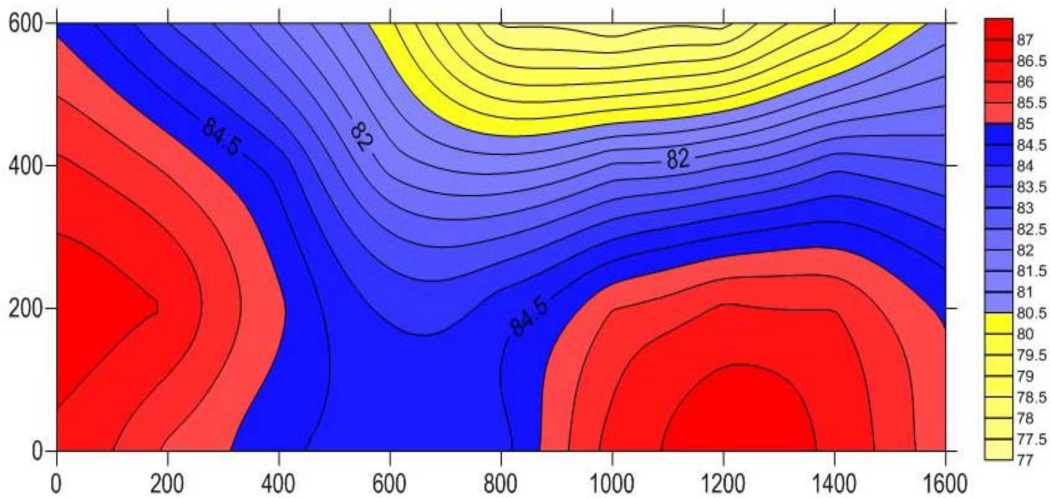


Figure 5.7 The Original Intensity Map of the Left Side of the Tractor at 400 Hz

Figure 5.8 Left Side View for Sound Intensity Measurements

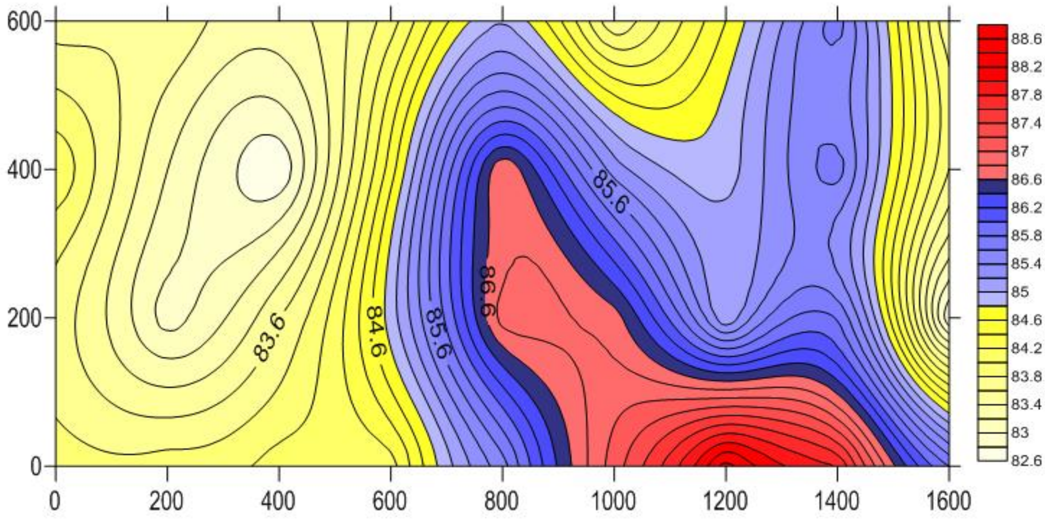


Figure 5.9 The Original Intensity Map of the Left Side of the Tractor at 630 Hz

Figure 5.10 Left Side View for Sound Intensity Measurements

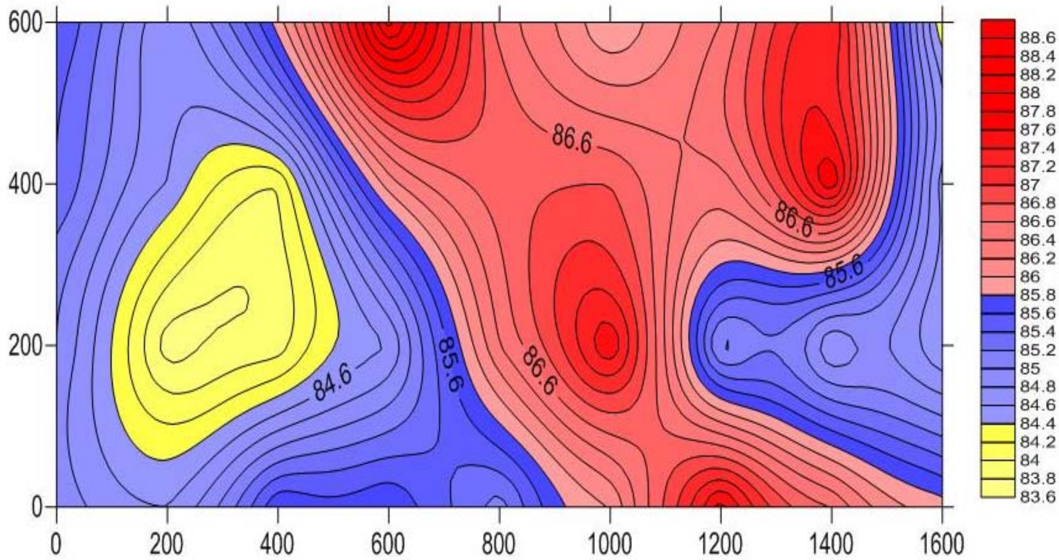


Figure 5.11 The Original Intensity Map of the Left Side of the Tractor at 800 Hz

Figure 5.12 Front Side View for Sound Intensity Measurements

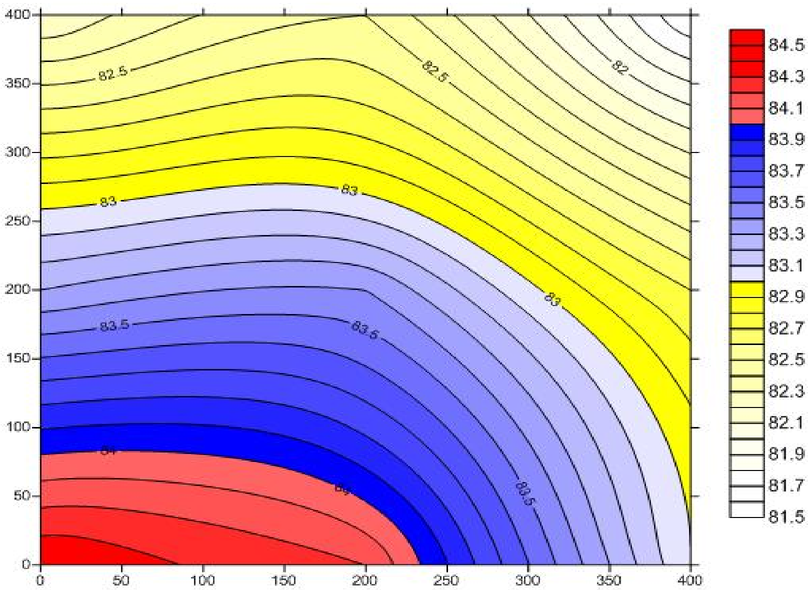


Figure 5.13 The Original Intensity Map of the Front Side of the Tractor at 250 Hz

Figure 5.14 Front Side View for Sound Intensity Measurements

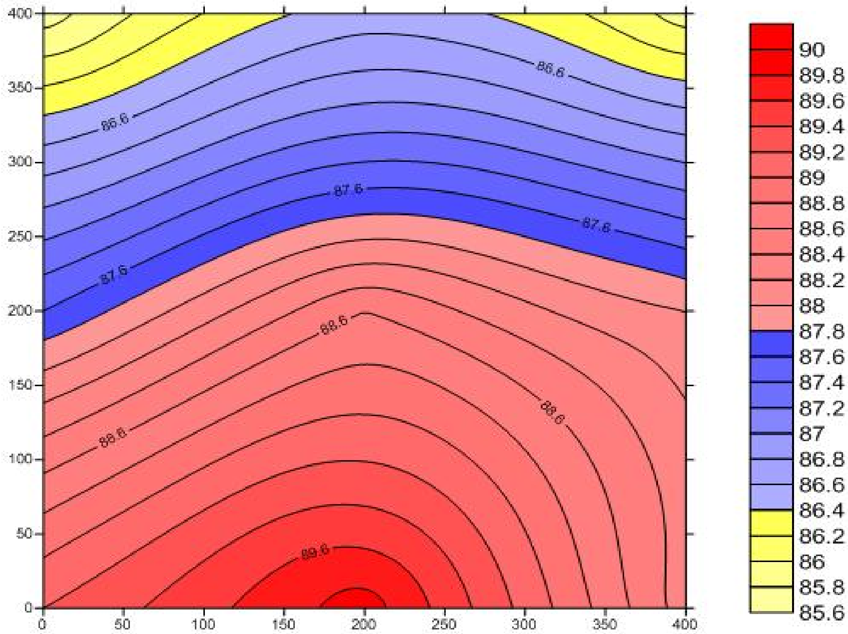


Figure 5.15 The Original Intensity Map of the Front Side of the Tractor at 630 Hz

Figure 5.16 Front Side View for Sound Intensity Measurements

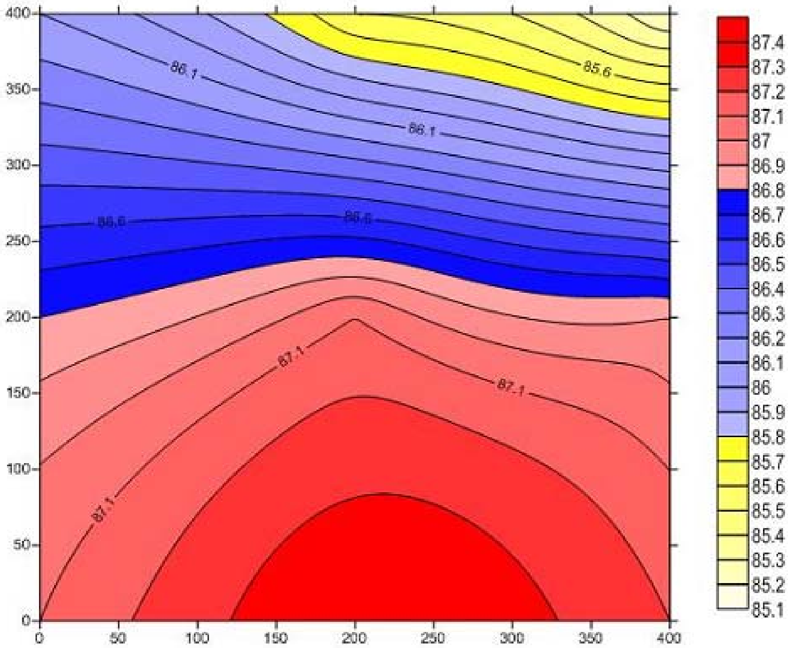


Figure 5.17 The Original Intensity Map of the Front Side of the Tractor at 1000 Hz

Figure 5.18 Right Side View for Sound Intensity Measurements

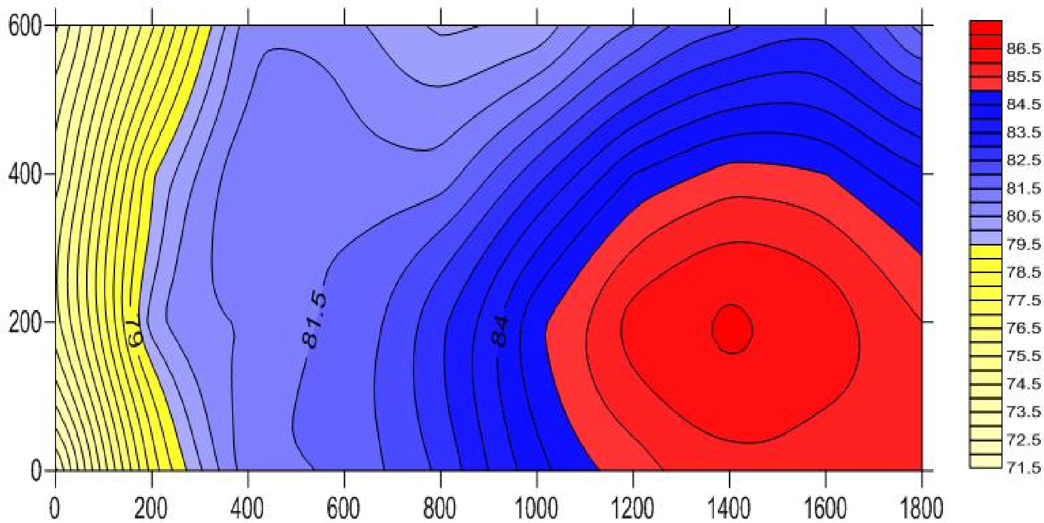


Figure 5.19 The Original Intensity Map of the Right Side of the Tractor at 315 Hz

Figure 5.20 Right Side View for Sound Intensity Measurements

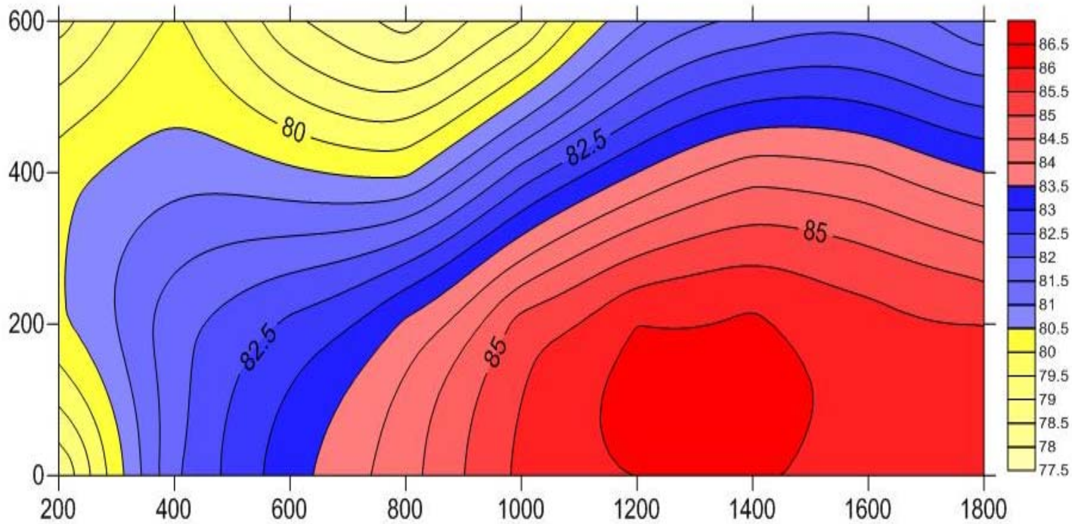


Figure 5.21 The Original Intensity Map of the Right Side of the Tractor at 400 Hz

Figure 5.22 Right Side View for Sound Intensity Measurements

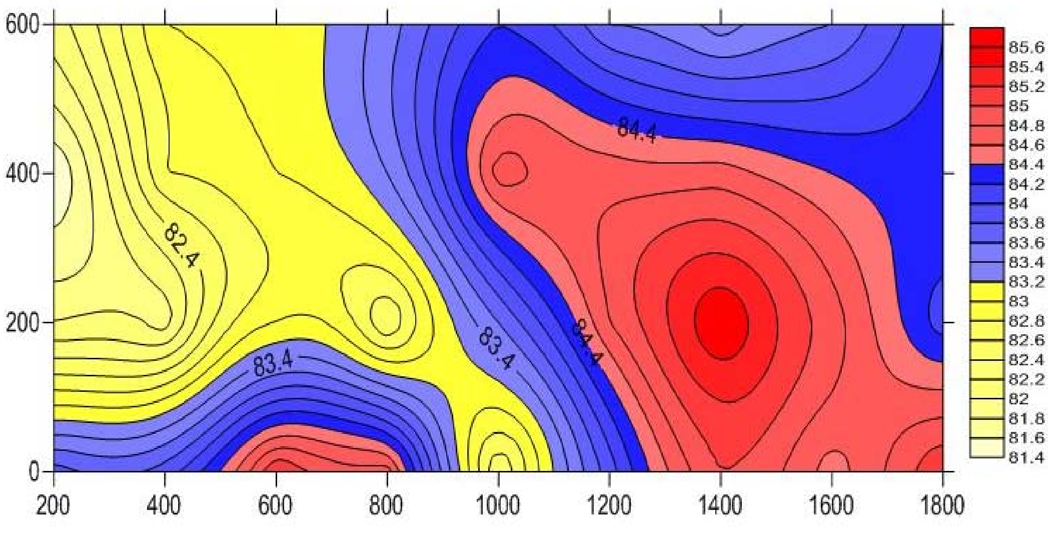


Figure 5.23 The Original Intensity Map of the Right Side of the Tractor at 630 Hz

Figure 5.24 Right Side View for Sound Intensity Measurements

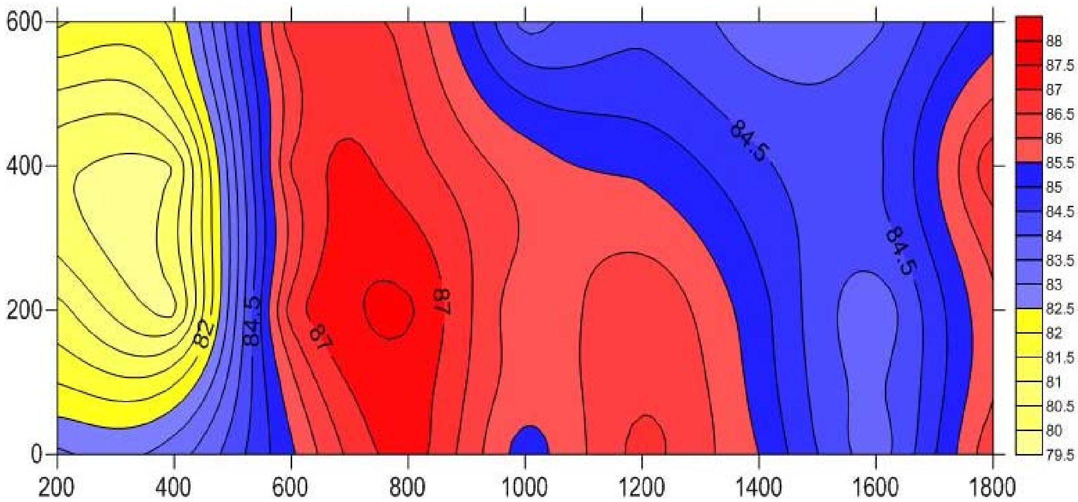


Figure 5.25 The Original Intensity Map of the Right Side of the Tractor at 800 Hz

5.6 Sound Power Level Calculation Using Sound Intensity Measurements of the Original Prototype Tractor

By using the techniques described in ISO 9614-2 [9] and 9614-3 [10] standards and formulae given in previous chapter, sound power calculation of the original prototype tractor is completed. The overall sound power level produced by the tractor can be calculated as 99,2 dB (A).

Table 5.8 Sound Power Level Calculation Results from Sound Intensities of the Original Prototype Tractor

Sound Power (W)			Sound Power (W)	Sound Power Level
Left	Right	Front	Sum of All Surfaces	A-Weighted dB (A)
3397,2E-6	3546,1E-6	1443,4E-6	8386,7E-6	99,2

The overall sound power level is lower than the value found from sound pressure measurements. The difference is due to the reason that sound intensity measurement surfaces were not covering the whole tractor. Only the front, right and left sides of the tractor were scanned during measurements.

5.7 Results of Sound Intensity Measurements after Initial Modifications

By analyzing spectral analysis results, it was obtained that the highest contributions were discovered at frequencies, 630 Hz and 800 Hz. The noise sources referring to these frequencies are muffler or exhaust pipe and cooling fan, respectively. In addition to that, the domination of cooling fan and exhaust pipe can be seen on intensity maps at these frequencies.

Sound intensities were measured on original prototype tractor after first modifications consisting of changing cooling fan blade angle from 36° to 43° and adding a vibration isolator on the exhaust pipe connection to the platform. During the sound intensity measurements, the weather was 13°C and the wind speed was below 5 m/s. Moreover, the measurement field had no reflective parts or surfaces closer than 20m.

Table 5.9 Peak Center Frequencies and Associated Noise Sources

Peak Center Frequency (Hz)	Noise Source(s)
100	Engine Firing Frequency
200	Engine Firing Frequency
315	Muffler/silencer + Exhaust Pipe
400	Engine Firing Frequency + Cooling Fan
500	Engine Firing Frequency
630	Muffler/silencer + Exhaust Pipe 2nd harmonic
800	Cooling Fan 2nd harmonic

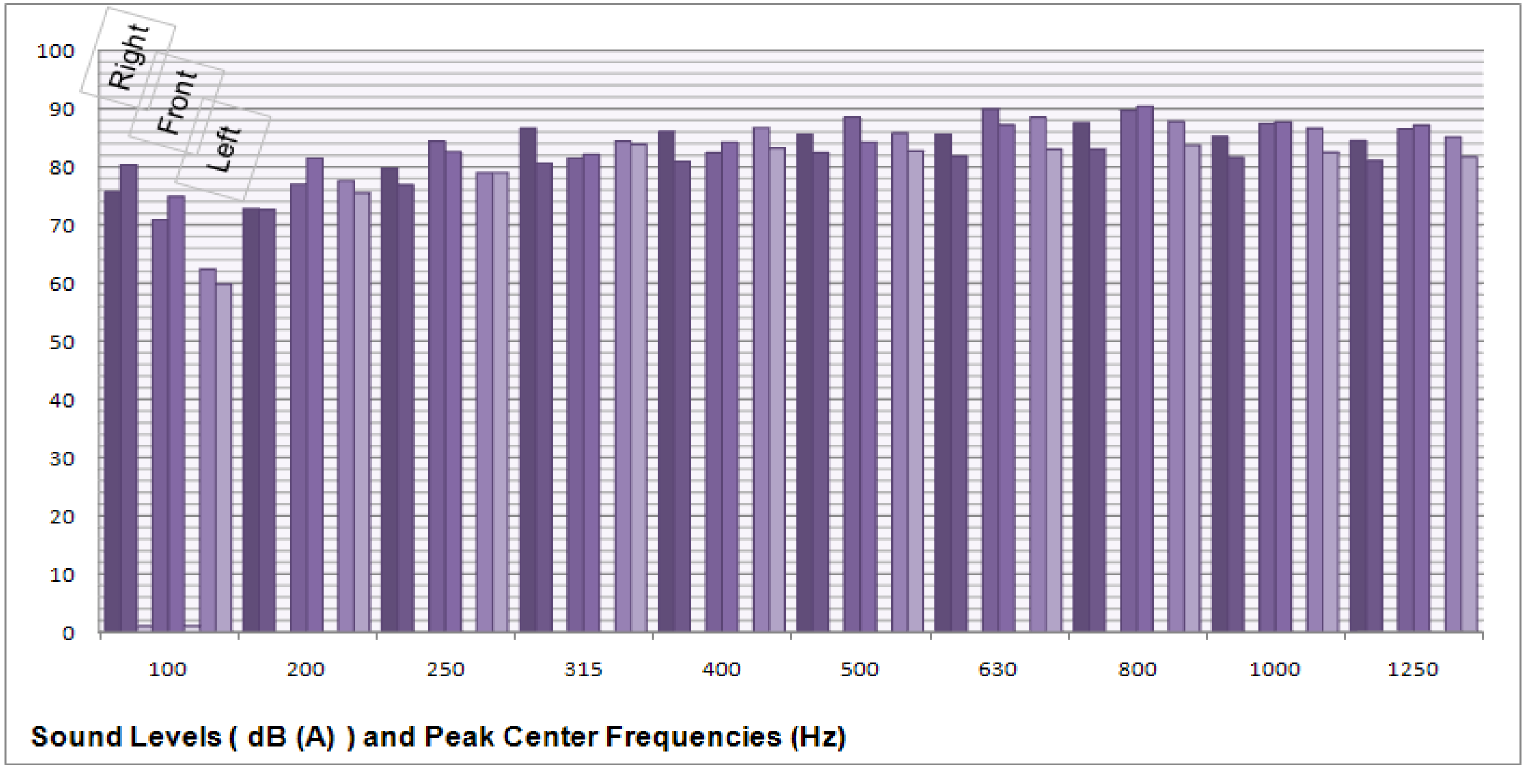
Initial modified tractor's significant frequencies obtained from the results of sound intensity measurements and comparison to the first measurements are tabulated in Table 5.10.

Table 5.10 Comparison of Sound Intensity Measurements Peak Center Frequencies and Intensity Values after Initial Modifications

Freq. (Hz)	Right Side	*Right Side (Initial Modified)*	Front Side	*Front Side (Initial Modified)*	Left Side	*Left Side (Initial Modified)*
	Intensity (dB)	*Intensity (dB)*	Intensity (dB)	*Intensity (dB)*	Intensity (dB)	*Intensity (dB)*
100	75,7	*80,3*	70,9	*74,9*	62,4	*59,8*
200	72,8	*72,6*	77,0	*81,5*	77,6	*75,5*
250	79,7	*76,9*	84,4	*82,6*	79,0	*79,0*
315	86,6	*80,6*	81,5	*82,1*	84,4	*83,9*
400	86,1	*80,9*	82,4	*84,2*	86,7	*83,2*
500	85,6	*82,4*	88,5	*84,2*	85,8	*82,7*
630	85,6	*81,8*	89,9	*87,2*	88,5	*83,0*
800	87,6	*83,0*	89,7	*90,4*	87,8	*83,7*
1000	85,2	*81,6*	87,4	*87,7*	86,6	*82,5*
1250	84,5	*81,1*	86,5	*87,1*	85,1	*81,7*

In addition to this table, plots from these values were generated and shown in Figure 5.26 below.

Figure 5.26 Octave Band Analysis of Noise after Initial Modifications



5.8 Results of Sound Intensity Measurements after Final Modifications

After the initial modifications, it was seen from intensity measurements that intensity values were decreased. However, in order to comply with the legislation, more modifications were required for the noise sources at other peak frequencies, like 400 Hz, 315 Hz, etc.

Sound intensity measurements were completed with the final modified tractor with all necessary modifications. During the sound intensity measurements, the weather was 9°C and the wind speed was below 5 m/s. Although the weather was suitable, some measurements were refreshed due to unwanted noises. Moreover, the measurement field had no reflective parts or surfaces closer than 20m.

All the modifications including the initial modifications on final prototype tractor can be listed as;

- Steering Column Vibration Isolation from Transmission Box Casting
- Exhaust Pipe Vibration Isolation from the Platform
- The Cooling Fan
- Serial Production Silencer/Muffler
- The Engine Side Sound Absorbers/Isolators
- Platform Vibration Isolators

After all noise control applications and improvements were conducted on the final modified tractor, the comparison of all intensity values can be seen in Table 5.11 and final sound intensity contour maps can be seen below figures.

Table 5.11 Comparison of Peak Center Frequencies and Associated Intensity Values after Final Modifications

Freq. (Hz)	Right Side	Right Side (Initial Modified)	*Right Side (Final Modified)*	Front Side	Front Side (Initial Modified)	*Front Side (Final Modified)*	Left Side	Left Side (Initial Modified)	*Left Side (Final Modified)*
	Intensity (dB)	Intensity (dB)	*Intensity (dB)*	Intensity (dB)	Intensity (dB)	*Intensity (dB)*	Intensity (dB)	Intensity (dB)	*Intensity (dB)*
100	75,7	80,3	*72,8*	70,9	74,9	*75,4*	62,4	59,8	*66,2*
200	72,8	72,6	*71,2*	77	81,5	*80,8*	77,6	75,5	*72,5*
250	79,7	76,9	*74,3*	84,4	82,6	*78,7*	79	79	*74,9*
315	86,6	80,6	*80,7*	81,5	82,1	*80,3*	84,4	83,9	*79,6*
400	86,1	80,9	*82,4*	82,4	84,2	*81,4*	86,7	83,2	*80,9*
500	85,6	82,4	*80,3*	88,5	84,2	*85,1*	85,8	82,7	*80,7*
630	*85,6*	*81,8*	*79,8*	*89,9*	*87,2*	*85,6*	*88,5*	*83*	*80,4*
800	*87,6*	*83*	*80,7*	*89,7*	*90,4*	*89,3*	*87,8*	*83,7*	*81,4*
1000	85,2	81,6	*79,8*	87,4	87,7	*87,1*	86,6	82,5	*80,5*
1250	84,5	81,1	*79,7*	86,5	87,1	*84,9*	85,1	81,7	*79,9*

Figure 5.27 Left Side View for Sound Intensity Measurements

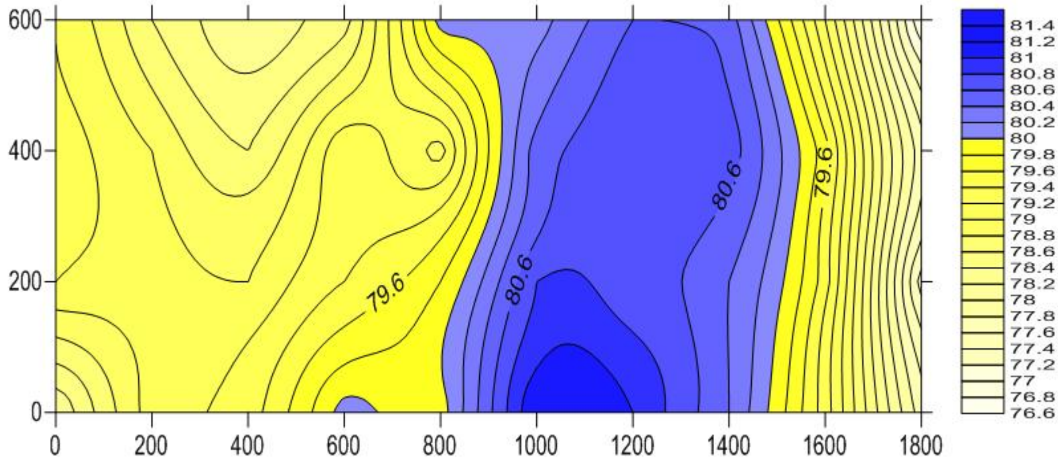


Figure 5.28 The Final Intensity Map of the Left Side of the Tractor at 315 Hz

Figure 5.29 Left Side View for Sound Intensity Measurements

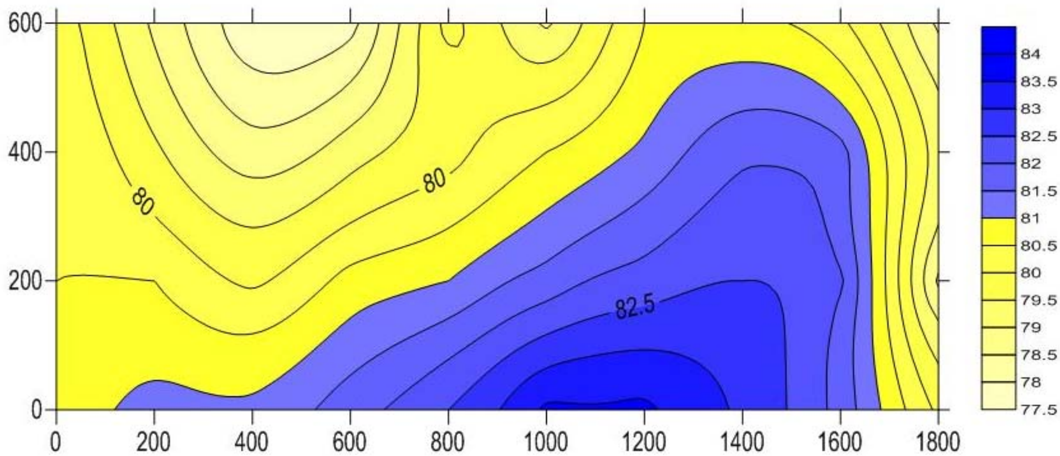


Figure 5.30 The Final Intensity Map of the Left Side of the Tractor at 400 Hz

Figure 5.31 Left Side View for Sound Intensity Measurements

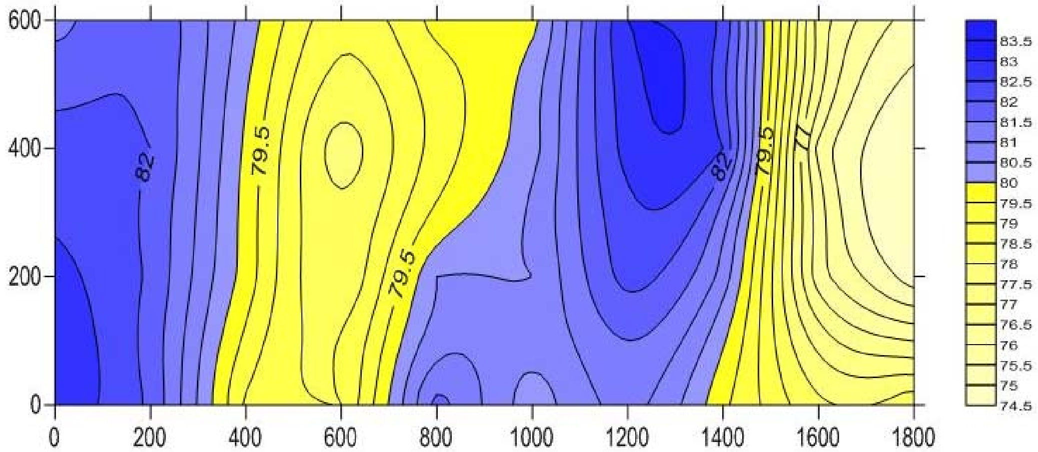


Figure 5.32 The Final Intensity Map of the Left Side of the Tractor at 630 Hz

Figure 5.33 Left Side View for Sound Intensity Measurements

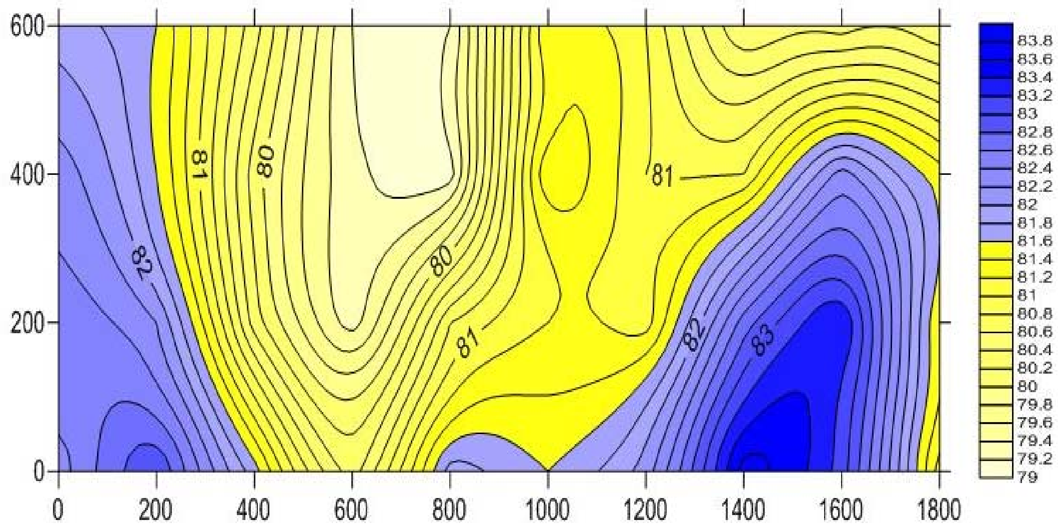


Figure 5.34 The Final Intensity Map of the Left Side of the Tractor at 800 Hz

Figure 5.35 Front Side View for Sound Intensity Measurements

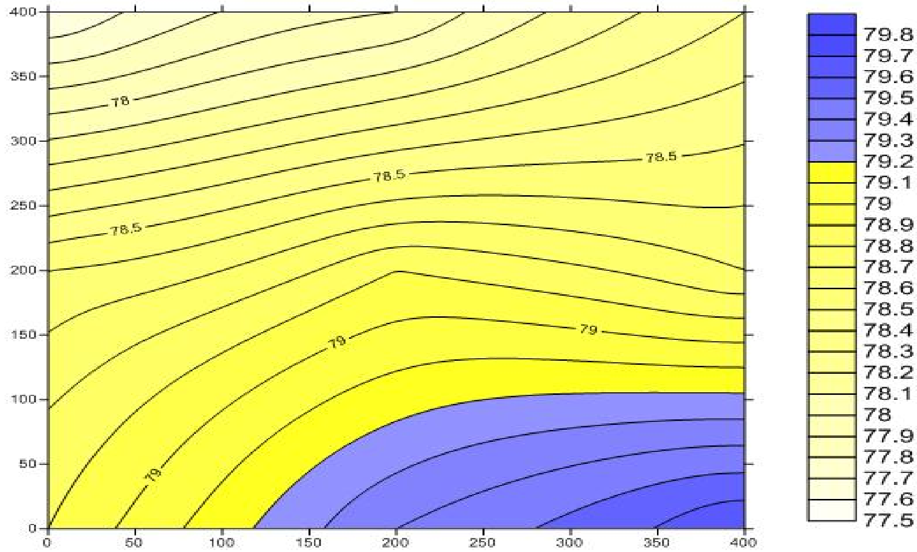


Figure 5.36 The Final Intensity Map of the Front Side of the Tractor at 250 Hz

Figure 5.37 Front Side View for Sound Intensity Measurements

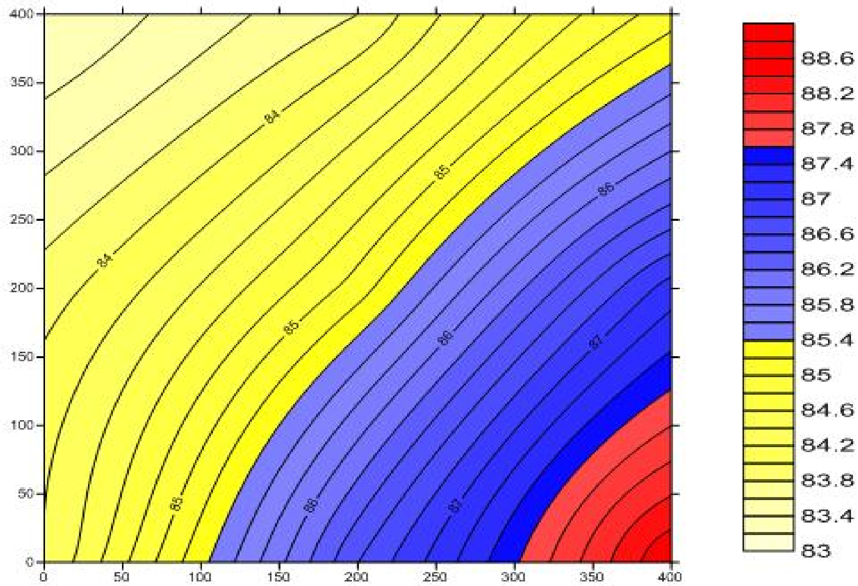


Figure 5.38 The Final Intensity Map of the Front Side of the Tractor at 630 Hz

Figure 5.39 Right Side View for Sound Intensity Measurements

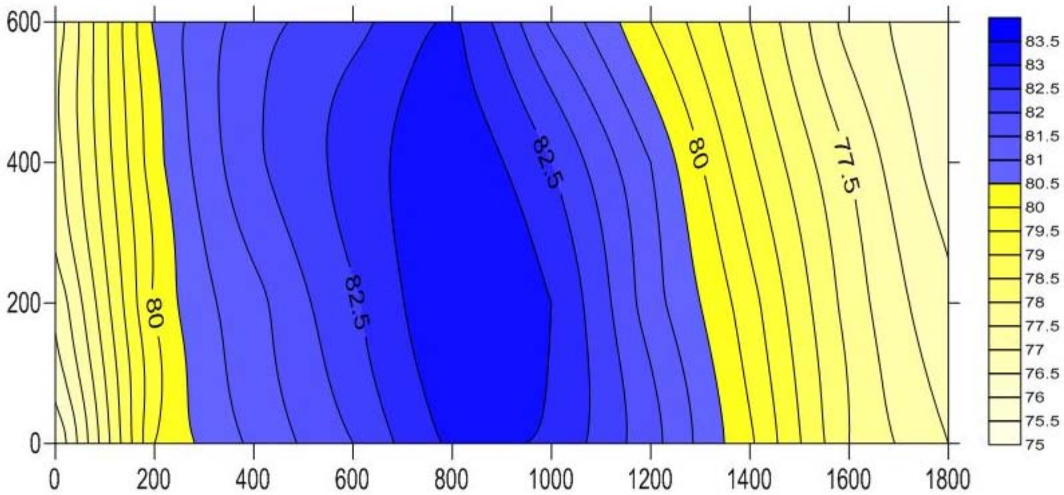


Figure 5.40 The Final Intensity Map of the Right Side of the Tractor at 315 Hz

Figure 5.41 Right Side View for Sound Intensity Measurements

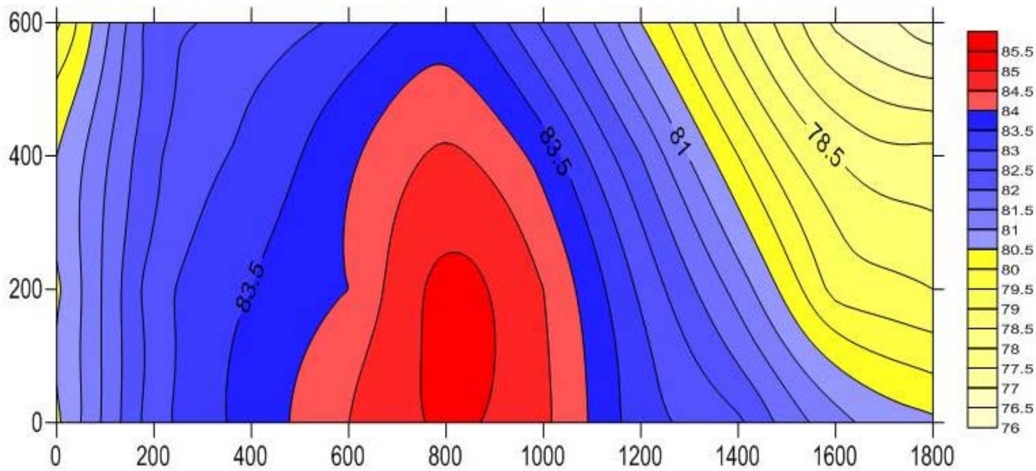


Figure 5.42 The Final Intensity Map of the Right Side of the Tractor at 400 Hz

Figure 5.43 Right Side View for Sound Intensity Measurements

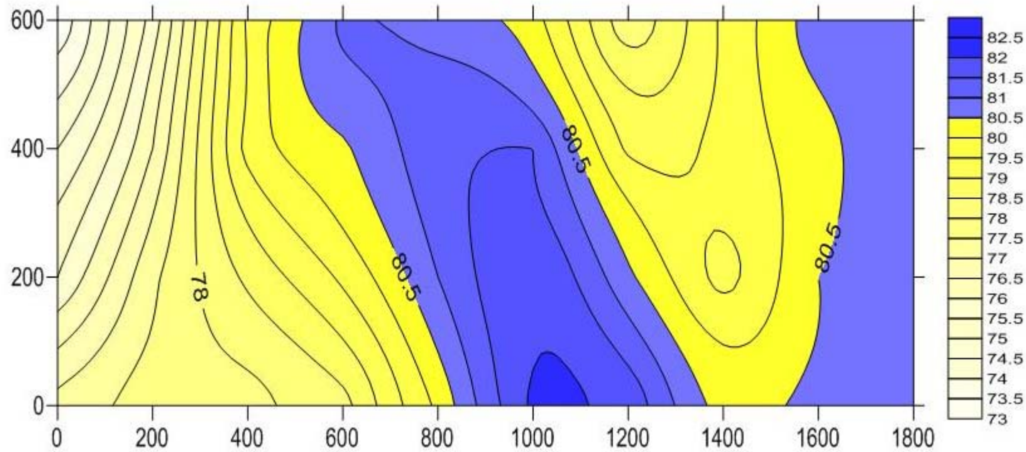


Figure 5.44 The Final Intensity Map of the Right Side of the Tractor at 630 Hz

Figure 5.45 Right Side View for Sound Intensity Measurements

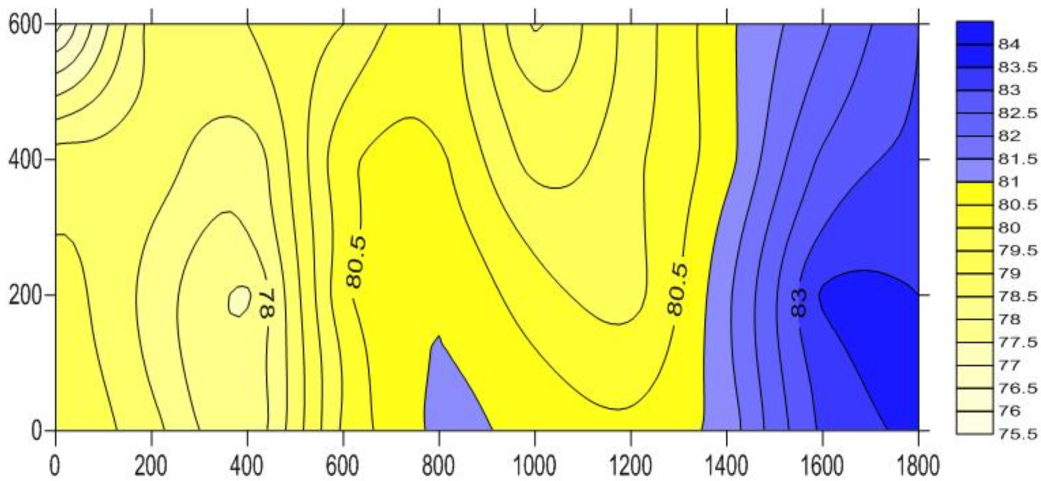


Figure 5.46 The Final Intensity Map of the Right Side of the Tractor at 800 Hz

5.9 Dominant Noise Sources from Test Results

Concerning both frequency analysis results and sound intensity maps, it can be seen that all three sides of the tractor radiate high sound energy at 100 Hz, 200 Hz, 400 Hz and at 800 Hz frequency bands. In addition, high sound energy can be noticed at 500 Hz and at 1000 Hz frequency bands. It is clear that at 400 Hz and 800 Hz cooling fan dominates noise level. Also engine firing frequency and its second harmonic appeared at 200 Hz and 400 Hz. Lastly, at 315 Hz and its second harmonic 630 Hz there is a high sound energy emitted by tractor due to exhaust pipe.

For 400 Hz and 800 Hz cooling fan dominates noise level. For the original prototype tractor, the metal cooling fan had 6 wings and 36° blade angle. After the analysis, new cooling fan fitted to the tractor and new results were satisfactory. The new plastic cooling fan has 6 wings but 43° blade angle.

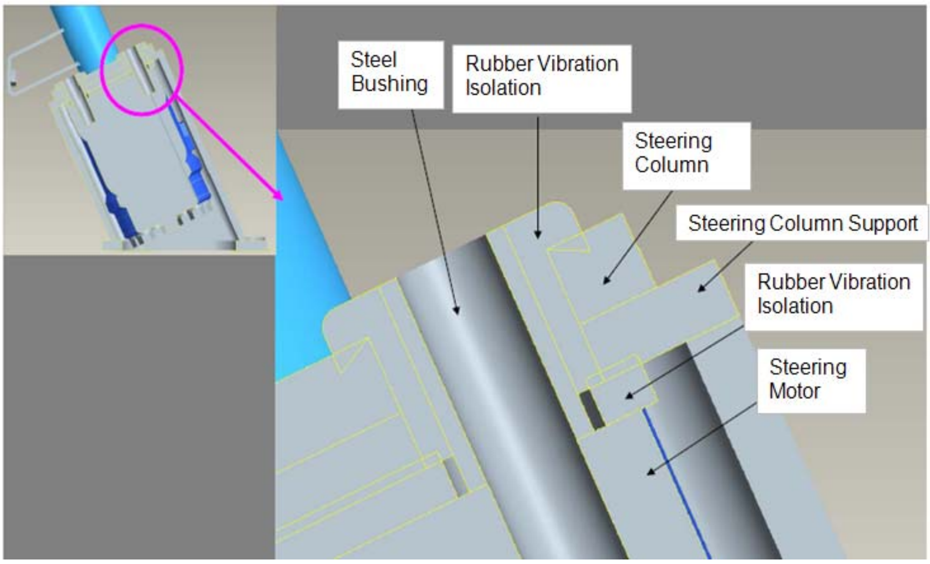


Figure 5.47 A Sectional View of Steering Motor Column and its Support

In addition to the cooling fan improvement, the steering motor and its support's connection were studied and isolated. The modifications can be seen in Figure 5.47.

After analyzing intensity maps, it was seen and decided that rubber vibration isolation materials could be assembled to eliminate metal to metal contact and structure-borne noise. This situation was eliminated from steering support bottom connection as well. This can be seen in Figure 5.48

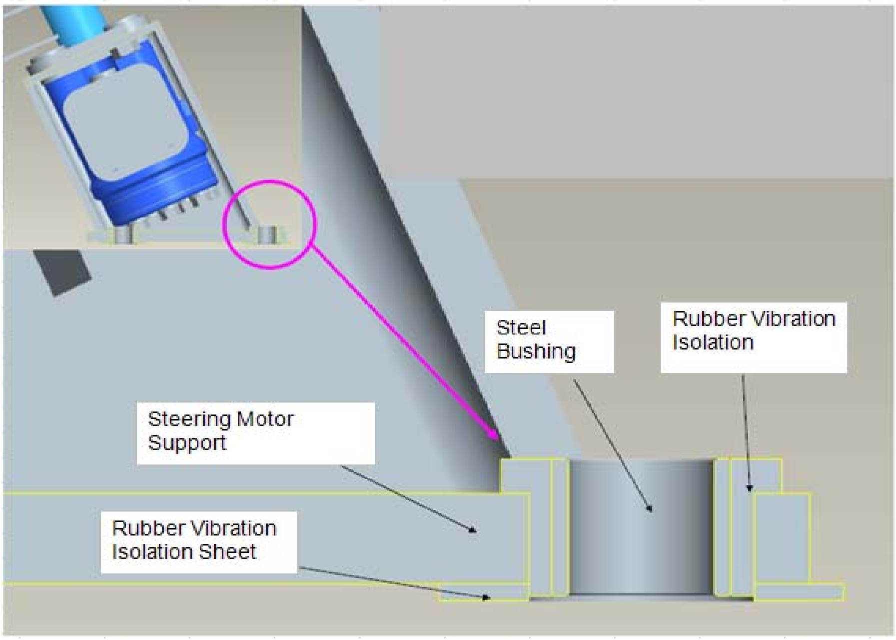


Figure 5.48 A Sectional View of Steering Motor Column and its Support

Moreover, exhaust pipe on this tractor is horizontal and it was connected to the platform on the tractor. Therefore, rubber vibration isolators were devised to this connection to get rid of metal to metal contact. A snapshot taken from 3D master model for this part can be seen in Figure 5.49.

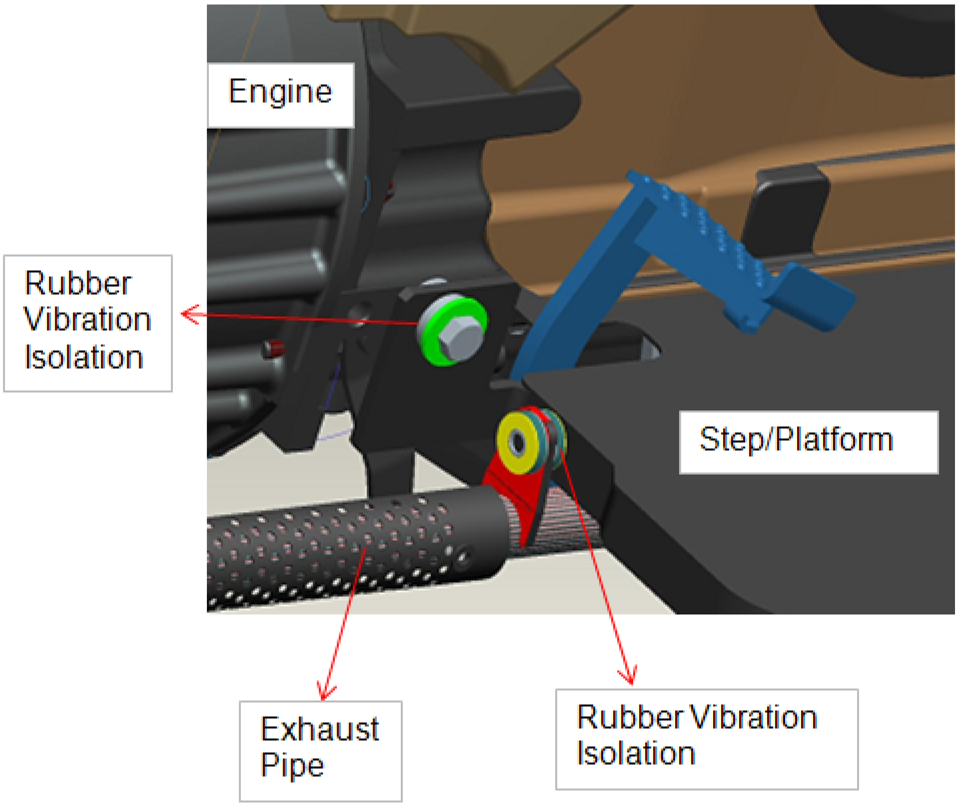


Figure 5.49 The Exhaust Pipe Isolation view taken from 3D Master Model of the Tractor

5.10 Sound Power Level Calculation Using Sound Intensity Measurements after Final Modifications

Sound power level of the test tractor having final modifications was calculated. The overall sound power generated by this tractor was calculated as 96,9 dB (A) and shown Table 5.12.

Table 5.12 Sound Power Level Calculation Results from Sound Intensities after Final Modifications

T480 Tractor	Sound Power (W)			Sound Power (W)	Sound Power Level
	Left	Right	Front	Sum of All Surfaces	A-Weighted dB (A)
Original	3397,2E-6	3546,1E-6	1443,4E-6	8386,7E-6	99,2
FINAL	1773,0E-6	1785,3E-6	1388,2E-6	4946,5E-6	96,9
The Decrease in Sound Power Level [dB]					2,3

Finally, it can be concluded that a 2,3 dB decrease in sound power level of the tractor having final modifications was achieved from 99,2 dB (A) to 96,9 dB (A) after exercising final noise control applications.

5.11 Sound Power Level Calculation Using Sound Pressure Measurements after Final Modifications

Sound power level of the tractor after final modifications was calculated according to ISO3746:1999 [29] Standard and formulae given in Chapter 4. The overall sound power generated by the tractor was calculated as 105,2 dB (A) and shown below Table 5.13.

Table 5.13 Sound Power Level Calculation Results after Final Modifications

Microphone Position	Background Noise dB (A)	L'_{pA1}	L'_{pA2}	L'_{pA3}	$\overline{L'_{pA,T}}$	$\overline{L'_{pA}}$	L_{wA}
14	52,7	86,0	86,5	85,8	86,1	88,2	105,2
15	50,0	88,4	88,3	88,6	88,4		
6	49,7	89,4	89,6	89,5	89,5		

From the results tabulated in Table 5.13, it can be concluded that the noise measurements at point 6 took the highest values again. However, decrease in level was observed in the latest. The decrease on overall sound power level of the tractor is totaled as 2 dB, from 107,2 dB (A) to 105,2 dB (A).

Consequently, both sound power level calculations showed that the overall sound power of the test tractor was decreased by 2 dB. Through proper noise control measures yielded comparably similar results.

CHAPTER 6

SUMMARY AND CONCLUSIONS

This study mainly focused on noise source identification techniques and applications on the new project T480 tractor and development of proper noise control strategies and applications on this tractor. In order to achieve these, necessary measurements were completed on the T480 tractor and related noise mapping studies were generated using relevant software. Several aspects of the study were summarized and explained in the following sections. The first section is devoted to the strategies applied on the tractor and the second section describes future works not only for this project tractor but also for the other type of tractors.

6.1 Noise Control Strategies on the Test Tractor

Noise source identification methods were explained and applied on the original prototype tractor. These methods were composed of sound power level determination, spectral analysis of noise data and sound intensity measurements on three sides of the tractor.

1/3 octave band frequency analysis is used to obtain spectral analysis results. Sound intensity measurements were accomplished for in depth analysis of noise sources of the tractor which could be identified by only a narrower spectral analysis.. The intensity maps of sound intensity values from intensity measurements were carried in 1/3 octave bands.

Significant frequencies obtained from the results of sound intensity measurements showed that the cooling fan, exhaust pipe, muffler and the engine itself due to its high speed, i.e. 3000 rpm, are potential noise sources affecting the overall noise level more than the others.

Noise control applications for reducing the flow induced noise by the cooling fan were exercised by replacing the existing fan with a new one. The original metal fan had 6 wings and 36° blade angle. The new plastic cooling fan has 6 wings and 43° blade angle. Finally, the new cooling fan helped reduce the noise level nearly by 1 dB. However, the noise contribution is not the only design constraint for a cooling fan. As the name implies, in order to avoid any overheating problems, the new cooling fan has also been checked on air-to-boil test.

During the operator's ear noise level test of the original prototype tractor, it was realized that there is noticeable drive-train noise. This noise could only be felt while tractor is moving. Because of the fact that both sound intensity tests and spectral analysis tests are conducted at stationary positions, this drive-train noise is detected only when the tractor is moving.

According to the legislation, the required speed of the test tractor should be nearly 7.5 km/h. The related range-gear combination referring to that speed is 1st range and 3rd speed gear. The existing gear sets are spur gear sets owing to the fact that this project is cost effective. Therefore, it is decided that replacement of the 3rd and 4th speed gears and related shafts with helical ones would decrease the drive-train noise levels. New helical gear sets are used in order to reduce the noise level for final measurements on final modified tractor. Using helical gears for 3rd and 4th speed gears helps decrease in overall noise level 1 dB. The noise level measurements on the modified tractor having helical gears can be seen in Table 6.1.

Considering the original prototype tractor's noise levels in mind, all noise control actions resulted in noise reduction of 3 dB. The measured data from the final modified tractor is tabulated in table 6.1. As it can be seen from the table the results are under the legislative limits. Finally, T480 tractor is homologated according to European Noise Directives 77 / 311 / EEC [1] and 74 / 151 / EEC [2] and certified.

Table 6.1 The Homologation Tractor Noise Levels

Sound Level dB (A)	Left Side	Right Side
Operator' Ear Noise Level	85.1	85.5
Pass-By Noise Level	83	77.6

Moreover, absorptive materials are assembled to isolate the engine noise which peaks out at engine firing frequencies on the interior part of the left side engine panel and the right side engine panel. These can be viewed in Figures 6.1 and 6.2.

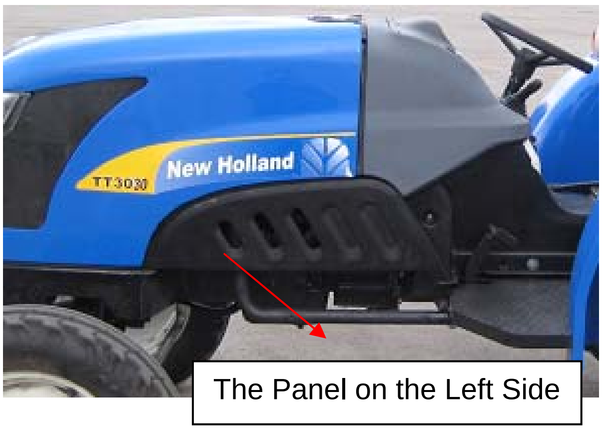


Figure 6.1 The Engine Panel on the Left Side

Figure 6.2 The Engine Panel on the Right Side

Sound intensity measurements on the original tractor show that at 400 Hz, the cooling fan is dominant. In Figure 6.3, it can be seen that the “hot spot“ vertical axis line is at the same line with cooling fan.

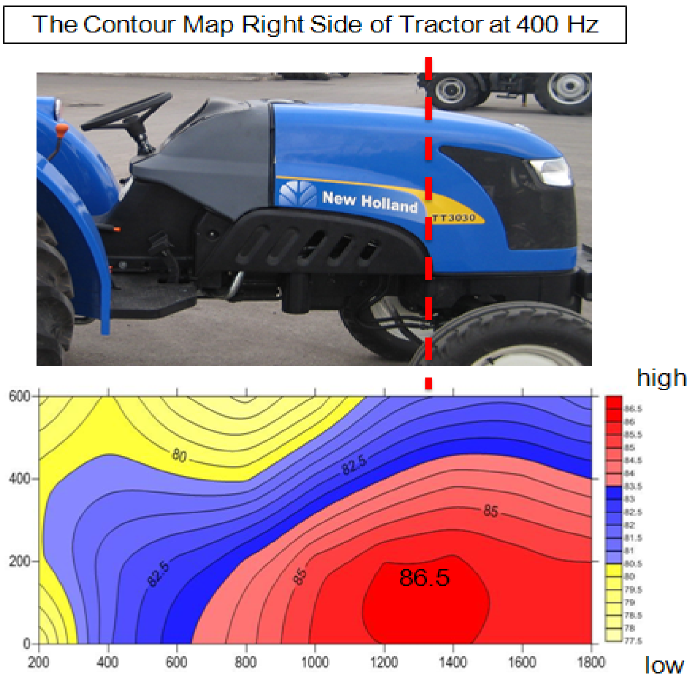


Figure 6.3 The Original Intensity Map of the Right Side at 400 Hz

After the installation of the new cooling fan, it is seen in sound intensity maps that at 400 Hz, the “hot spot“ vertical axis line is not at the same line with cooling fan. The line shifts rearwards of tractor and the maximum value drops from 86,5 dB (A) to 85,5 dB (A). After final modifications, the tractor’s intensity map at the same frequency showing the shifted line can be seen in Figure 6.4.

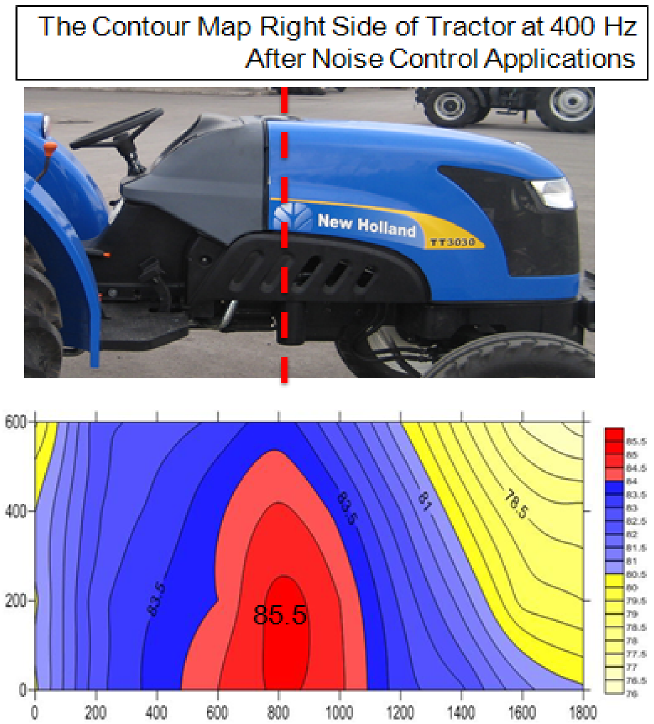


Figure 6.4 The Final Intensity Map of the Right Side at 400 Hz

By analyzing the spectral analysis and the noise source ranking calculations, it is found that the contribution of noise sources at 630 Hz is the second highest source. Considering this in mind, the intensity map has been generated after original tractor’s sound intensity measurements. It is seen that at 630 Hz, exhaust pipe is the dominant noise source. In Figure 6.5, red contour lines on intensity map mark the exhaust pipe.

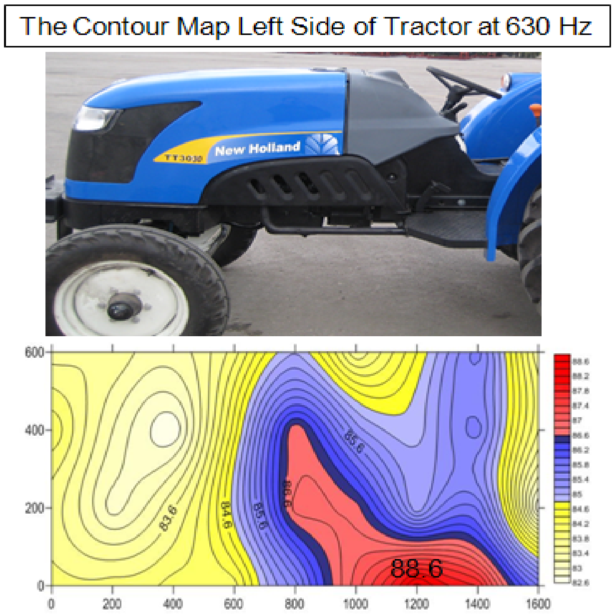


Figure 6.5 The Original Intensity Map of the Left Side at 630 Hz

After the addition of vibration isolation element between the exhaust pipe and the platform, it is seen that, the red contour lines in sound intensity measurements at 630 Hz are disappeared.

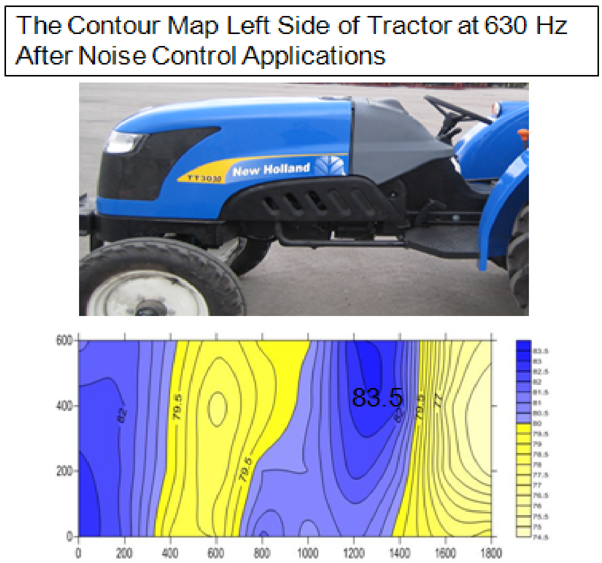


Figure 6.6 The Final Intensity Map of the Left Side at 630 Hz

6.2 Potential Future Studies on Tractors

Even though the fact that this thesis planned and realized necessary noise control application studies not to exceed legislative limits; there remain some other techniques and methods to be used for further improvements.

It is anticipated that future studies can include exhaust pipes and the muffler. The engine muffler could be analyzed acoustically with the help of suitable software. By this way, a noise attenuation of 1-2 dB could be achieved in reference to sound intensity measurements and intensity maps.

Moreover, due to the fact that this tractor's engine has a relatively high maximum speed, i.e. 3000 rpm, the vibrations on this tractor are significant. Coherence techniques can be used to obtain cause-effect relation for vibration and noise levels for engine hood plate, side fenders and platforms. A more detailed spectral analysis of noise and vibration of engine parts need to be planned using FFT analysis techniques.

The study could be enlarged by using the methods covered in ISO 16902-1:2003 [31] and ISO 15086-1:2001 [32]. Intensity measurements can be conducted to undercover the hydraulic and steering pumps. Silently operating gear pumps can be sought and similar analysis for resulting noise level drops can be investigated.

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