

EXPERIMENTAL INVESTIGATION OF NOISE GENERATED BY PNEUMATIC BRAKE SYSTEM COMPONENTS ON A TEST BENCH

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Abstract - This paper presents the results of an experimental investigation of noise generated by pneumatic brake system components under test bench conditions and proposes a generalized mathematical model for predicting their sound power spectra. Experimental measurements were performed using pneumatic brake system components of a Category N3 vehicle. Sound power levels were determined in one-third-octave frequency bands over the range from 16 Hz to 8 kHz. The measured spectra revealed similar spectral characteristics for different pneumatic components, with pronounced maxima in the frequency range of 3–8 kHz and local peaks at low frequencies associated with pressure pulsations and structural vibrations. Based on the experimental results, a generalized mathematical model of the sound power spectrum is proposed. The model represents the spectral distribution as the ratio of two third-order polynomial functions supplemented by two scaling parameters that describe the sound power level at the characteristic low-frequency peak and at the maximum frequency. The proposed approach reproduces the measured spectra with an error not exceeding 1% while preserving the characteristic spectral shape for different pneumatic components. The developed model provides an efficient tool for predicting the acoustic characteristics of pneumatic brake system components and for evaluating the effectiveness of noise reduction technologies during the design and optimization of low-noise pneumatic systems for commercial vehicles.

Keywords: Pneumatic brake system; Sound power level; Spectral analysis; Aerodynamic noise; Mathematical modelling; Noise reduction; Commercial vehicles.

1. Introduction

Noise generated by vehicles operating on urban streets and roads is one of the key factors of negative impact on the environment and human health. Elevated levels of acoustic pollution lead to deterioration of living conditions, reduced work capacity, and increased stress levels in people [1, 2]. It is well known that the primary sources of vehicle noise include the engine, transmission units, and the interaction between tires and the road surface [3, 4, 5, 6]. At the same time, a significant contribution to the overall noise level is made by aerodynamic noise generated during vehicle motion, as well as noise produced by the pneumatic drive of the braking system, transmission control systems, and the vehicle suspension system [7, 8, 9, 10].

The pneumatic braking system is characterized by complex non-stationary processes associated with rapid compressed air flow, operation of valve devices, and periodic impulsive air release into the atmosphere through exhaust valves of pneumatic components [11, 12, 13, 14]. As a result, irregular impulsive noise over a wide frequency range is generated, particularly in the high-frequency domain [15, 16, 17, 18]. Such noise has an adverse effect on the human psycho-emotional state [19, 20, 21, 22], causing changes in the functioning of the central nervous system, decreased concentration, and increased fatigue [23, 24, 25].

In this regard, experimental studies of the acoustic characteristics of pneumatic components of vehicle braking systems are of significant relevance in order to identify the patterns of noise formation and determine ways to reduce it [26, 27, 28]. Modern

acoustic measurement methods are used to evaluate noise characteristics, including sound pressure level measurements using sound level meters and measurement microphones, spectral analysis of signals using the Fast Fourier Transform, and time-domain analysis of impulsive processes [28, 29, 30].

Studies of acoustic characteristics can be carried out both under laboratory conditions using specialized test benches and under real operating conditions of vehicles. However, measurements performed under actual driving conditions are significantly complicated by the influence of external noise sources (engine, tires, aerodynamic disturbances), as well as by the instability of pneumatic braking system operating modes, which limits the ability to accurately isolate and analyze the intrinsic noise of pneumatic components.

Therefore, the use of bench-based experimental studies is more appropriate, as they ensure controlled testing conditions and reproducibility of operating modes [31, 32]. Specialized test benches allow isolation of the investigated object from external acoustic influences, precise control of compressed air parameters (pressure, flow rate, pulse duration), and simultaneous recording of both pneumatic and acoustic characteristics of the process. In addition, bench conditions enable repeated experiments with variation of individual parameters, which is necessary for identifying noise formation patterns and performing accurate spectral and time-frequency analysis.

2. Objective and Problem Statement

Pneumatic brake system components are among the principal sources of noise in heavy-duty vehicles because their operation involves the periodic release of compressed air into the atmosphere. Despite the considerable number of studies devoted to the measurement of the acoustic characteristics of pneumatic devices and the development of noise reduction technologies, existing research has primarily focused on individual component types or specific design solutions. At the same time, generalized mathematical models capable of accurately describing the spectral distribution of the sound power level of various pneumatic brake system components and supporting the evaluation of noise reduction technologies are practically unavailable. This significantly limits the ability to predict the acoustic characteristics of pneumatic systems during the design and optimization stages.

The aim of this paper is to experimentally investigate the spectral characteristics of noise generated by pneumatic brake system components of Category N_3 vehicles and to develop a generalized mathematical model of the sound power spectrum suitable for predicting the acoustic characteristics of

various types of pneumatic brake system components and evaluating the effectiveness of noise reduction technologies.

To achieve this aim, the following objectives were accomplished:

- experimental investigations of the noise generated by the main pneumatic brake system components were carried out under test bench conditions;
- the sound power spectra were determined over the frequency range from 16 Hz to 8 kHz;
- a generalized mathematical model of the sound power spectrum was developed;
- the accuracy of the proposed model and its applicability for predicting acoustic characteristics and evaluating the effectiveness of noise reduction technologies were assessed.

3. Experimental Test Bench for Noise Investigation of Vehicle Pneumatic Brake System Components

To investigate the noise characteristics of pneumatic devices used in the pneumatic braking systems of heavy-duty vehicles, the experimental test bench of the Department of Automobiles at the Kharkiv National Automobile and Highway University was employed (Figure 1). The facility consists of an isolated chamber *I* and a control room for operating personnel *II*. The walls and ceiling of the isolated chamber are lined with sound-absorbing wedges 1, while the floor is constructed of concrete. A pit 2, covered by a lid 3, is located at the center of the chamber floor. The underside of the lid is coated with sound-absorbing material, whereas its upper surface is faced with sheet metal. The pit 2 is connected to the personnel room *II* by means of a tunnel 4.

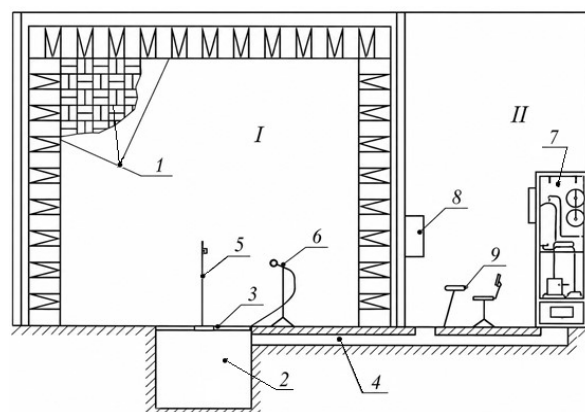


Figure 1: Experimental Test Bench for Noise Investigation of Heavy-Duty Vehicle Pneumatic Brake System Components

In the isolated chamber *I*, a mounting stand 5 is installed to accommodate the pneumatic brake system component under investigation. Noise

measurement equipment 6 is arranged around the stand 5.

It should be noted that the measurement equipment 6 is positioned in accordance with the arrangement shown in Figure 2, at specified heights and at designated measurement locations within the isolated chamber I.

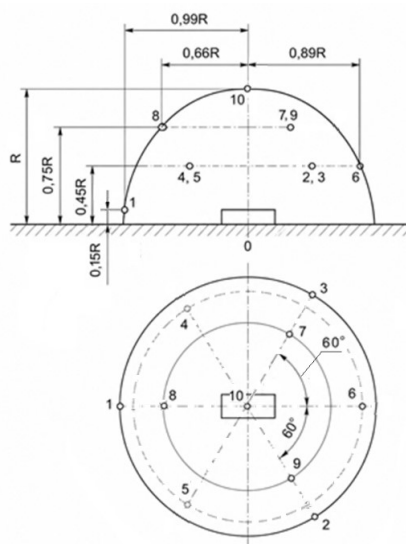


Figure 2: Layout of the measurement locations within the isolated chamber for noise measurements of heavy-duty vehicle brake system components

The control room II is equipped with a display screen 8 for presenting the measurement results, a control console 9 for operating the pneumatic braking system, and a cabinet 7 containing equipment for the accumulation, purification, and storage of compressed air used in the vehicle pneumatic system installed in the pit 2 of the isolated chamber I.

The isolated chamber has a volume 33 m³ (total volume of the room 70 m³) and provides acoustic conditions close to a free field. The chamber allows noise sources with dimensions up to 1 m to be tested using the engineering method and sources up to 0.5 m in size using the precision method.

The following acoustic characteristics can be measured in the chamber: the sound power level in frequency bands (L_W), the A-weighted sound power level (L_{WA}), and the directivity index, G , of the investigated pneumatic brake system component.

It should be noted that the A-weighted sound power level is a measure of the total acoustic power emitted by a noise source, corrected according to the frequency-dependent sensitivity of human hearing.

4. Experimental Methodology

To perform tests on the component of interest, the section of the vehicle braking system incorporating the component under investigation is installed in the pit of the isolated chamber.

Through tunnel 4, the brake system circuit is connected to the air supply unit located in cabinet 7 in the control room.

The component under investigation is mounted on stand 5 at the center of the isolated chamber and connected to the pneumatic brake circuit located in pit 2. The installation conditions of the noise source during testing are arranged to be as close as possible to the actual operating conditions of the source on a vehicle. Particular attention is paid to the distance between the component and sound-reflecting surfaces, while the lengths and cross-sectional dimensions of the pipelines are selected in strict accordance with those used in the specific vehicle.

After installation, the pneumatic brake system is filled with compressed air, and its leak-tightness and operability are verified. Following the leak-tightness check, pit 2 is tightly sealed with cover 3 in order to prevent the brake system components located in the pit from generating acoustic interference during noise measurements of the brake system component mounted on stand 5.

Microphone 6 is connected to the measuring equipment located in the control room by means of a cable routed through tunnel 4. The microphone is positioned at the measurement location in accordance with the arrangement shown in Figure 2 and is oriented toward the brake system component under investigation.

The measurement locations are uniformly distributed over the measurement surface. The measurement surface is an imaginary surface surrounding the component on all sides and terminating at the sound-reflecting floor of the chamber. A hemispherical surface is adopted as the measurement surface, with its center coinciding with the projection of the center of the noise-source parallelepiped onto the sound-reflecting floor plane of the chamber. The radius of the hemispherical measurement surface is equal to 1 m. Ten measurement locations are used when measurements are performed on the hemispherical measurement surface. The relative coordinates of the measurement locations are given in Table 1.

Table 1: Coordinates of the measurement locations on the hemispherical measurement surface

The measurement locations (Figure 2)	X, m	Y, m	Z, m
1	- 0.99	0.00	0.15
2	0.50	- 0.86	0.15
3	0.50	0.86	0.15
4	- 0.45	0.77	0.45
5	- 0.45	- 0.77	0.45
6	0.89	0.00	0.45
7	0.33	0.57	0.75
8	- 0.66	0.00	0.75
9	0.33	- 0.57	0.75
10	0.00	0.00	1.00

During measurements, the sound level meter is set to the appropriate time weighting depending on the type of noise: the slow characteristic is used for steady noise, whereas the impulse characteristic is employed for the measurement of impulsive noise. For non-steady noise, the equivalent *A*-weighted sound pressure level, (L_{Aeq}), dB(A), is determined.

The noise characteristics of the pneumatic brake system component may be presented in the form of test reports, tables, or graphs.

For uniformly distributed measurement locations on the measurement surface, the average sound pressure level, L_{pm} , in frequency bands (equation (1)), expressed in dB, or the average *A*-weighted sound level, L_{pAm} , expressed in dB(A), is determined using equation (2) in accordance with ISO 3744 [15]:

$$L_{pm} = 10 \log_{10} \left(\frac{1}{n} \sum_{i=1}^n 10^{0.1 L_{pi}} \right), \quad (1)$$

or

$$L_{pAm} = 10 \log_{10} \left(\frac{1}{n} \sum_{i=1}^n 10^{0.1 L_{pAi}} \right), \quad (2)$$

where L_{pi} – sound level measured at the *i*-th measurement location, dB; L_{pAi} – *A*-weighted sound level measured at the *i*-th measurement location, dB(A); *n* – number of measurement locations.

The sound power level in frequency bands L_w (equation (3)) or the *A*-weighted sound power level L_{wA} of the investigated component is determined using equation (4):

$$L_w = L_{pm} + 10 \log_{10} \left(\frac{S}{S_0} \right) + K, \quad (3)$$

or

$$L_{wA} = L_{wAm} + 10 \log_{10} \left(\frac{S}{S_0} \right) + K, \quad (4)$$

where: $S = 2\pi r^2 = 6.28 \text{ m}^2$ – the area of the measurement surface; $S_0 = 1 \text{ m}^2$ – the reference surface area; *K* – the environmental correction accounting for deviations of the acoustic field from ideal free-field conditions.

The environmental correction *K* is determined according to equation (5):

$$K \cong K_1 - K_2 - K_3, \quad (5)$$

where: K_1 – the background noise correction; K_2 – the environmental correction; K_3 – correction for temperature and atmospheric pressure of air in an isolated chamber.

The formulation of equation (5) provides a clear physical interpretation of the correction terms:

K_1 – increases the estimated sound power level by accounting for the influence of background noise;

K_2 – decreases the estimated sound power level by compensating for the increase caused by sound reflections within the test environment;

K_3 – adjusts the measured result to standard atmospheric conditions and may either increase or decrease the estimated value, depending on the air temperature and atmospheric pressure.

We will determine the corrections using the following equations:

$$K_1 = 10 \log_{10} \left(\frac{1}{1 - 10^{-0.1(L_{pi} - L_{bg})}} \right), \quad (6)$$

where: L_{bg} – background noise level, dB.

$$K_2 = 10 \log_{10} \left(1 + \frac{4(1 - \alpha_m)S}{\alpha_m S_v} \right), \quad (7)$$

where: $S_v = 62 \text{ m}^2$ – total area of the internal surfaces of the isolated chamber; $\alpha_m = 0.91$ – average sound absorption coefficient of the chamber was determined by the equation:

$$\alpha_m = 1 - \exp \left(- \frac{0.161 \cdot V}{S_v \cdot T_{60}} \right),$$

where: $V = 33 \text{ m}^3$ – volume of isolated chamber; $T_{60} = 0.035 \text{ s}$ – reverberation time;

$$K_3 = 10 \log_{10} \left(\frac{p}{p_a} \sqrt{\frac{273.15 + T_0}{273.15 + T}} \right), \quad (8)$$

where: $p_a = 101325 \text{ Pa}$ – the average atmospheric pressure (adopted according to ISO 3744 and ISO 3745.); *p* – atmospheric pressure during research, Pa; $T_0 = 20 \text{ }^\circ\text{C}$ – air temperature under standard conditions (adopted in accordance with acoustic measurements of sound power levels according to ISO 3744 and ISO 3745); *T* – air temperature during research, $^\circ\text{C}$.

Unlike the ISO 3744 standard methodology, the proposed methodology takes into account ambient temperatures and atmospheric pressure.

For measurements performed on a hemispherical measurement surface, the directivity index is determined as the difference between the sound pressure level at the considered measurement location and the average sound pressure level over the measurement surface, corrected by 3 dB to account for sound radiation into a half-space. That is, the directivity index of the noise source at the *i*-th measurement location on the hemispherical measurement surface can be determined using equation (9):

$$G = L_{pi} - L_{pm} + 10 \log_{10} (2). \quad (9)$$

The measurement results are subsequently presented as graphs using the following scales: 10

dB per 20 mm on the vertical axis and one octave per 15 mm on the horizontal axis.

The proposed methodology is applicable to the investigation of noise emissions from pneumatic brake system components of heavy-duty vehicles. The results obtained in this study indicate that, for components equipped with an atmospheric exhaust port, it is unnecessary to perform measurements at all positions on the measurement surface shown in *Figure 2* and subsequently recalculate the results using equations (1) – (9).

Instead, the noise characteristics can be determined from measurements conducted at a single position located 0.5 m from the atmospheric exhaust port outlet (located at a height of 1.5 m from the floor) and oriented at an angle of 45° with respect to the axis of the atmospheric exhaust port of the investigated pneumatic brake system component.

5. Results of the Experimental Investigation of Some Components of the Vehicle Pneumatic Brake System

The overall noise generated by a vehicle pneumatic brake system results from the combined acoustic emissions of its individual components. Each pneumatic device exhibits distinct acoustic characteristics, including its frequency spectrum, sound power level, and noise duration. Therefore, effective noise reduction of pneumatic brake systems requires the identification of the dominant noise

sources and a detailed investigation of their acoustic characteristics.

It is well known that the primary noise sources in pneumatic brake systems are the components equipped with atmospheric exhaust ports, through which compressed air is discharged into the atmosphere. Experimental measurements of the noise generated by various pneumatic brake system components installed on a Category N_3 vehicle were carried out at the Department of Automobiles of Kharkiv National Automobile and Highway University (KhNAHU). The spectral analysis of the sound power level (L_W) of the pneumatic brake system components (*Figure 3*) revealed that the emitted noise is broadband in nature, with a predominance of high-frequency components. The maximum sound power levels were observed in the frequency range from 3 to 8 kHz, reaching 90–130 dB, depending on the air pressure in the pneumatic brake system. This spectral characteristic is attributed to the intense turbulent discharge of compressed air through the exhaust port of the tested pneumatic brake component.

In the low-frequency region, a local maximum of the sound power level (L_W) was observed within the range of 45–90 dB at frequencies of 63–125 Hz. This phenomenon is most likely associated with pressure pulsations and vibrations of the pneumatic system components.

It should also be noted that the overall A-weighted sound power level (L_{WA}) of components of this type ranges from 95 to 125 dB(A).

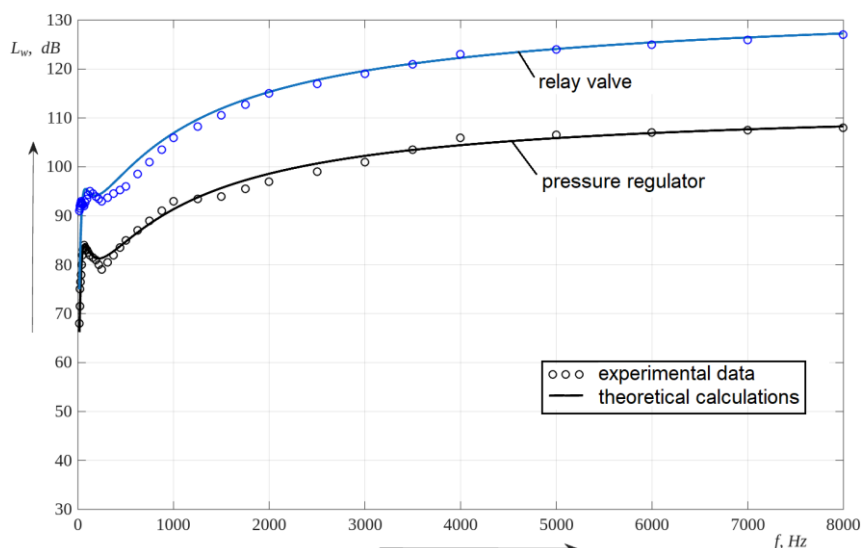


Figure 3: Sound power level (L_W) spectra of pneumatic brake system components of a Category N_3 vehicle measured at a pneumatic system pressure of 0.6 MPa

6. Results

The conducted study showed that the sound power level (L_W) of various pneumatic brake system components can be predicted with an error of less

than 1%, depending on frequency, using Equation (10). The proposed equation is formulated as the ratio of two third-order polynomial functions supplemented by a scaling term, ΔL_W , expressed on the sound power level scale.

$$L_w = \frac{113f^3 + 5.4 \cdot 10^4 f^2 + 2 \cdot 10^6 f + 10^8}{f^3 + 850f^2 - 870f + 2 \cdot 10^6} + \Delta L_w, \quad (10)$$

The value of the scaling factor, ΔL_w , can be determined using Equation (11).

$$\Delta L_w = A + (B - A) \frac{\log_{10}(f) - \log_{10}(f_p)}{\log_{10}(f_{\max}) - \log_{10}(f_p)}, \quad (11)$$

In Equation (11), the ratio of the two logarithmic functions, together with the coefficients A and B , defines the sound power level offset at the characteristic peak frequency, f_p , and at the maximum frequency, $f_{\max}$, respectively. The coefficients A and B are readily obtained from Equations (12) and (13), respectively.

$$A = L_{w(at f_p)}^{unit} - 84, \quad (12)$$

$$B = L_{w(at f_{\max})}^{unit} - 108, \quad (13)$$

For example, a pressure regulator with $L_{w(at f_p)}^{unit} = 84$ dB and $L_{w(at f_{\max})}^{unit} = 108$ dB is described by the scaling factors $A = 0$ and $B = 0$. For a relay valve with $L_{w(at f_p)}^{unit} = 95$ dB and $L_{w(at f_{\max})}^{unit} = 127$ dB, the corresponding scaling factors are $A = 6$ and $B = 19$. Despite these changes, the characteristic shape of the sound power level (L_w) spectrum remains unchanged throughout the frequency range from 16 Hz to 8 kHz (see Figure 3).

7. Discussion

The experimental investigation confirmed that the sound generated by pneumatic brake system components is characterized by a broadband spectrum with dominant high-frequency components caused by the turbulent discharge of compressed air through atmospheric exhaust ports. The obtained results are consistent with the findings reported in previous studies on aerodynamic noise generated by pneumatic valves and compressed air exhaust systems, where turbulent jet flow has been identified as the primary mechanism of noise generation [7, 8, 9, 12].

The measured sound power spectra of different pneumatic brake system components exhibited similar spectral shapes despite considerable differences in their absolute sound power levels. A pronounced maximum was observed within the frequency range of 3–8 kHz, while a local low-frequency peak occurred between 63 and 125 Hz. The latter is most likely associated with pressure pulsations and structural vibrations of the pneumatic components during the transient exhaust

process. Similar low-frequency phenomena have been reported in studies investigating the unsteady flow characteristics of pneumatic valves [10, 11, 13].

Unlike previous investigations, which mainly focused on the acoustic performance of individual pneumatic devices or on evaluating the effectiveness of specific noise reduction measures, the present study proposes a generalized mathematical model capable of describing the sound power spectra of different pneumatic brake system components using a common analytical expression. The introduction of two scaling parameters allows the model to preserve the characteristic spectral shape while accurately reproducing the measured sound power levels for different component types.

The proposed model demonstrated an approximation error not exceeding 1% over the frequency range from 16 Hz to 8 kHz. Such accuracy is sufficient for engineering applications, including preliminary acoustic assessment of pneumatic brake system components, comparison of alternative design solutions, and evaluation of the effectiveness of noise reduction technologies without the need for extensive experimental testing.

The proposed approach may also serve as a basis for further research aimed at developing predictive acoustic models that account for the influence of operating pressure, flow rate, valve geometry, and transient flow processes on the sound power spectrum. In addition, combining the developed analytical model with numerical flow simulations could improve the prediction accuracy and facilitate the optimization of low-noise pneumatic brake system components during the design stage.

8. Conclusions

The conducted experimental and analytical investigations demonstrated that the sound power spectra of pneumatic brake system components exhibit common characteristic features regardless of the component type. This made it possible to develop a generalized mathematical model capable of accurately describing the spectral characteristics of different pneumatic devices using a common analytical expression with only two scaling parameters.

The main findings of the study are summarized as follows:

1. Experimental investigations of pneumatic brake system components of a Category N3 vehicle demonstrated that the dominant noise sources are components with atmospheric exhaust ports, generating broadband aerodynamic noise with maximum sound power levels in the frequency range of 3–8 kHz.

2. The measured sound power spectra exhibit a similar characteristic shape for different pneumatic components despite differences in their absolute sound power levels. This similarity provides the

basis for the development of a unified mathematical model describing their spectral characteristics.

3. A generalized mathematical model describing the sound power spectrum as the ratio of two third-order polynomial functions supplemented by frequency-dependent scaling factors was developed. The proposed model reproduces the measured spectra with an error not exceeding 1% over the frequency range from 16 Hz to 8 kHz.

4. The proposed scaling parameters enable the model to be adapted to various pneumatic brake system components while preserving the characteristic spectral shape, thereby simplifying the prediction of their acoustic characteristics.

5. The developed model provides an effective tool for evaluating the effectiveness of noise reduction technologies and supports the design and optimization of low-noise pneumatic brake system components for commercial vehicles.

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