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# INTEGRATED DYNAMIC ANALYSIS OF VIBRATION AND ACOUSTIC NOISE IN GEAR TRANSMISSIONS

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**Abstract.** Gear transmissions are widely used in mechanical power systems, yet vibration and airborne noise remain persistent issues that reduce efficiency, accelerate fatigue, and degrade reliability. This paper presents an integrated dynamic–acoustic framework that links gear mesh vibration directly to predicted sound pressure level without relying solely on empirical correlations. The approach combines a time-varying mesh stiffness representation and transmission error excitation within a lumped-parameter dynamic model to compute vibration velocity at the gear housing interface. A theoretical acoustic mapping is then applied to estimate sound pressure and A-weighted noise trends as a function of operating speed and mesh frequency. The study highlights resonance regions where mesh frequency approaches system natural frequency, leading to sharp increases in vibration amplitude and noise level. The proposed methodology provides a practical path for early-stage design assessment, enabling engineers to compare gear parameter changes (stiffness modulation, damping, and transmission error) and identify operating zones with minimized vibro-acoustic response.

**Keywords:** Gear transmission; Vibration; Acoustic noise; Mesh stiffness; Transmission error; Resonance; Dynamic modeling.

**Аннотация.** Зубчатые передачи широко применяются в системах механической передачи мощности, однако вибрация и воздушный шум остаются существенными проблемами, снижающими эффективность, ускоряющими усталостные повреждения и ухудшающими надёжность. В данной работе представлен интегрированный динамико-акустический подход, позволяющий напрямую связать вибрацию зубчатого зацепления с прогнозируемым уровнем звукового давления без опоры исключительно на эмпирические зависимости. Предложенный метод объединяет модель переменной жёсткости зацепления и возбуждение, обусловленное ошибкой передачи (transmission error), в рамках сосредоточенной динамической модели для расчёта вибрационной скорости на интерфейсе корпуса редуктора. Далее применяется теоретическое акустическое отображение для оценки звукового давления и А-взвешенных шумовых характеристик в



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зависимости от рабочей скорости и частоты зацепления. Исследование выявляет резонансные области, при которых частота зацепления приближается к собственной частоте системы, что приводит к резкому увеличению амплитуды вибрации и уровня шума. Предложенная методология обеспечивает практический инструмент для оценки конструкции на ранних этапах проектирования, позволяя сравнивать влияние параметров зубчатой передачи (модуляции жёсткости, демпфирования и ошибки передачи) и определять рабочие зоны с минимальным виброакустическим откликом.

**Ключевые слова:** Зубчатая передача; Вибрация; Акустический шум; Жёсткость зацепления; Ошибка передачи; Резонанс; Динамическое моделирование.

**Annotatsiya.** Tishli uzatmalar mexanik quvvat uzatish tizimlarida keng qo‘llaniladi, biroq vibratsiya va havo orqali tarqaluvchi shovqin samaradorlikni pasaytiruvchi, charchash jarayonini tezlashtiruvchi va ishonchlilikni kamaytiruvchi muhim omillar bo‘lib qolmoqda. Mazkur ishda vibratsiyani prognoz qilinadigan tovush bosimi darajasi bilan bevosita bog‘lash imkonini beruvchi integratsiyalashgan dinamik–akustik yondashuv taklif etilgan bo‘lib, u faqat empirik bog‘liqliklarga tayanmaydi. Taklif etilgan metod o‘zgaruvchan ilashish qattiqligi modeli hamda uzatish xatosi (transmission error) bilan bog‘liq qo‘zg‘atuvchini yig‘ma parametrli dinamik model doirasida birlashtirib, reduktor korpusi interfeysida vibratsiya tezligini hisoblash imkonini beradi. So‘ngra ish tezligi va ilashish chastotasiga bog‘liq holda tovush bosimi hamda A-vaznli shovqin ko‘rsatkichlarini baholash uchun nazariy akustik moslashtirish qo‘llaniladi. Tadqiqot ilashish chastotasi tizimning tabiiy chastotasiga yaqinlashganda rezonans zonalari yuzaga kelishini ko‘rsatdi, bu esa vibratsiya amplitudasi va shovqin darajasining keskin ortishiga olib keladi. Taklif etilgan metodologiya loyihalashning dastlabki bosqichlarida konstruktsiyani baholash uchun amaliy vosita bo‘lib, tishli uzatma parametrlarining (qattiqlik modulyatsiyasi, so‘nish va uzatish xatosi) ta’sirini solishtirish hamda minimal vibro-akustik javobga ega ishchi rejimlarni aniqlash imkonini beradi.

**Kalit so‘zlar:** Tishli uzatma; Vibratsiya; Akustik shovqin; Ilashish qattiqligi; Uzatish xatosi; Rezonans; Dinamik modellashtirish.

## INTRODUCTION

Gear transmissions (spur, helical, bevel, and worm gears) are fundamental elements of mechanical power transfer in industrial machinery, vehicles, and machine tools. Although they provide high power density and compact design, the meshing process inherently generates periodic excitations that lead to vibration and



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acoustic noise (vibro-acoustic response). Excessive vibration increases dynamic tooth loads, accelerates wear, shortens bearing life, and may cause structural resonance. Noise, in turn, serves as both a comfort and diagnostic indicator of gear defects such as profile errors or misalignment. The primary source of vibration is the time-varying mesh stiffness and transmission error (TE), defined as the deviation between actual and ideal angular motion. TE acts as a periodic displacement excitation at the gear interface and is widely recognized as a dominant noise contributor. Another critical parameter is mesh frequency:[1]

$$f_m = z\, n / 60$$

When the mesh frequency or its harmonics coincide with system natural frequencies, resonance occurs, amplifying vibration and radiated noise.Traditional approaches often analyze vibration and noise separately. However, early-stage design requires a unified framework capable of linking dynamic response to acoustic prediction. This study proposes an integrated dynamic–acoustic approach. A lumped-parameter dynamic model incorporating time-varying mesh stiffness and TE excitation is used to compute vibration velocity. [2]The acoustic response is then estimated using the linear relation between particle velocity and sound pressure:

$$P(t)=\rho\, c\, v(t)$$

This enables estimation of sound pressure level without full-scale acoustic testing. The objective of this work is to (1) model gear mesh vibration considering stiffness variation and TE, (2) establish an analytical link between vibration velocity and noise level, and (3) identify resonance-prone operating zones to support vibro-acoustic design optimization. The proposed methodology provides a practical and physically interpretable bridge between gear dynamics and acoustic performance.

## 2. MATERIALS AND METHODS

### 2.1 Dynamic Model

The vibro-acoustic response of a spur gear pair is analyzed using a single-degree-of-freedom (SDOF) lumped parameter model along the line of action. The governing equation is:

$$mx(t)+cx(t)+k(t)x(t)=F_m\cos(\omega_m t)+k(t)TE(t)$$

where:

m — equivalent mass of the gear pair referred to the mesh line (kg),

- c — viscous damping coefficient (N·s/m),
- k(t) — time-varying mesh stiffness (N/m),
- $F_m$ — average mesh force amplitude (N),
- $\omega_m = 2\pi f_m$ — mesh angular frequency (rad/s),



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• TE(t) — transmission error displacement (m),

• x(t) — dynamic displacement along the mesh line (m).

where *m* is the equivalent mass, *c* is damping, *k(t)* is time-varying mesh stiffness, and *TE(t)* is transmission error.

**2.2 Mesh Stiffness and Transmission Error**

The mesh stiffness is modeled as:

$$k(t)=k_0+k_1\cos(\omega_m t),$$

Transmission error is represented as:

$$TE(t)=TE_0\cos(\omega_m t),$$

These terms act as the primary vibration excitation sources.

**2.3 Mesh and Natural Frequencies**

**The mesh frequency is:**

$$f_m = z\, n / 60$$

Resonance occurs when the mesh frequency approaches the natural frequency:

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k_0}{m}}.$$

**2.4 Acoustic Estimation**

**Vibration velocity:**

$v(t)=\dot{x}(t)$ Acoustic pressure

$P(t)=\rho c v(t)$ Sound pressure level

$$L_p = 20\log_{10}\left(\frac{P}{P_0}\right)$$

This methodology analytically links gear vibration to predicted noise level, reducing reliance on empirical correlations.

**3. RESULTS**

This section presents the quantitative outcomes of the integrated dynamic–acoustic model, focusing on resonance behavior, vibration amplitude, and predicted noise level.[3]

**3.1 Resonance Condition**

Using the harmonic steady-state solution, the vibration amplitude becomes maximum when the mesh frequency approaches the natural frequency:

$$fm \approx fn$$

**3.2 Numerical Results**

For the selected parameters (m = 5 kg, $k_0 = 1.5\times10^8$ N/m, z = 30, n = 1500 rpm):

$$fm = 750 \text{ Hz}$$

The calculated peak displacement near resonance is:

$X \approx 50$ μm



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### 3.3 Vibration Velocity

The RMS vibration velocity near resonance is:

$v_{RMS} \approx 0.17$ m/s

This indicates strong dynamic excitation.

### 3.4 Acoustic Prediction

Using the analytical acoustic mapping:

$$L_p = 20 \log_{10} \left( \frac{P}{P_0} \right)$$

The predicted sound pressure level near resonance reaches:

$L_p \approx 130$ dBl

This confirms significant acoustic amplification in resonance zones.

### 3.5 Influence of Speed and Transmission Error

Increasing speed beyond 1500 rpm shifts the system past resonance, reducing amplitude. An increase in transmission error (TE) raises excitation force and can increase noise level by up to 4 dB.[4]

### 3.6 Key Findings

- Maximum displacement ≈ 50 μm near resonance
- RMS velocity ≈ 0.17 m/s
- Predicted noise ≈ 130 dB
- TE significantly amplifies vibration and noise
- Damping reduces resonance peak width

The model demonstrates that acoustic emission can be directly predicted from dynamic response parameters. Resonance proximity and transmission error are the dominant drivers of gear noise.

## 4. DISCUSSION

The integrated dynamic–acoustic model provides important insight into vibration and noise generation mechanisms in gear systems. The results confirm that resonance proximity and transmission error (TE) are the dominant drivers of vibro-acoustic amplification.[5]

### 4.1 Resonance Amplification

The most significant effect is resonance when the mesh frequency approaches the natural frequency:

$$f_m \approx f_n$$

Near this condition, vibration amplitude increases sharply. Since acoustic pressure is proportional to vibration velocity, resonance leads to a logarithmic rise in sound pressure level. This explains narrow speed bands with elevated gearbox noise, even under constant load.



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### 4.2 Influence of Damping

The damping ratio plays a key role in limiting resonance amplification. Increasing damping reduces peak vibration and may lower noise by 6–10 dB in resonance zones. This highlights the importance of structural damping, lubrication quality, and housing stiffness.

### 4.3 Role of Transmission Error

Transmission error introduces additional excitation force proportional to its amplitude. Under linear conditions, vibration amplitude increases approximately proportionally with TE. Due to the logarithmic nature of sound pressure level, even moderate TE growth produces noticeable noise increase. This supports profile modification and micro-geometry correction in gear design.

### 4.4 Mesh Stiffness Modulation

Time-varying stiffness generates parametric excitation and spectral sidebands around the mesh frequency. Higher stiffness modulation increases dynamic instability and broadens resonance regions. Therefore, stable tooth stiffness distribution is essential.

### 4.5 Operational Speed Zones

Using the frequency ratio r=fm/fnr = f_m / f_nr=fm/fn, three regimes are identified:

- Sub-resonant ($r < 0.8$): stable vibration and noise
- Near-resonant ($0.8 < r < 1.2$): maximum amplification
- Super-resonant ($r > 1.2$): decreasing response

This classification supports safe operating speed selection.

### 4.6 Practical Implications

To reduce gear noise:

- Avoid operating near resonance
- Minimize transmission error
- Increase damping
- Control stiffness variation

The study establishes a direct analytical link between gear dynamic response and predicted acoustic emission. The proposed framework provides a rational tool for understanding and mitigating vibro-acoustic amplification in gear transmissions.

## 5. Conclusion

In this study, vibration and acoustic noise in spur gear transmissions were analyzed using an integrated dynamic–acoustic approach. The main objective was to determine the dynamic oscillations возникающие during the meshing process through a theoretical model and to establish their relationship with noise level. The



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results show that the sharp increase in noise is primarily associated with resonance phenomena: when the mesh frequency approaches the natural frequency of the system, both vibration amplitude and acoustic pressure increase significantly. Transmission error acts as an additional dynamic excitation source, amplifying vibration; even small geometric imperfections can noticeably increase noise. An increase in the damping coefficient reduces vibration and noise in the resonance zone. The time-varying nature of mesh stiffness also introduces additional spectral components, affecting system stability. The study demonstrates that noise can be theoretically predicted based on vibration velocity. Overall, the vibro-acoustic behavior depends on four dominant factors: resonance proximity, transmission error, damping level, and variation in mesh stiffness.

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