Sound Generation under Rotary-Screw Propulsion Unit Base Cylinder Bending

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Abstract: Theoretical results of sound generation parameters under rotary-screw propulsion unit base cylinder bending are dealt with in this article. Base cylinder parameters impact such as diameter, wall thickness, length as well as such vehicle movement parameters impact as speed and mass on noise level appearing at rotary-screw propulsion unit passing a single obstacle are analyzed in the paper. Sound vibrations onset conditions are shown. Research results allow one to conclude that a considerable sound level generated by rotary-screw propulsion units can appear only when the vehicle mass to rotor diameter ratio is greater than 25. The model produced and the results obtained might be useful when designing cross-country vehicles.

Key words: rotary-screw propulsion unit, bending, noise level

Introduction
Cross-country vehicles providing transport accessibility for all Russia’s regions are the only possible way of underpopulated regions development where road building and maintenance is economically impractical. Herewith, one of the basic problems of cross-country vehicles is a high-level noise impact on the operator and passengers which appears when vehicles and road machinery are moving over uneven or rough terrain. Thus, modelling noise level generated by different types of propulsion units in general and especially by rotary-screw propulsion unit, as well as finding ways to lower the noise level are acute research tasks.

Previous papers review
Nowadays only rotary-screw propulsion unit is considered relevant in transporting people and cargoes over water surface and frozen-over rivers [1,2,3,4,5,6,7]. However, due to the fact that not a single vehicle of this type has been produced in series up to now, a number of issues arise when making such vehicles, one of them being noise generation of this or that type.

Methods
The existing models of rotary-screw vehicles movement suggest no rotor base cylinder distortion [1,2,3,4,5,6]. Meanwhile, in operation and forcing an obstacle, the object is impacted by considerable forces (before the vehicle total mass) due to which the rotor will be bent and the vehicle forward movement will turn this bend into a wave process. Beam bending energy is conventionally described by equation [5,6,7,8,9]:

\[ W = \frac{m_p \omega^2 A_m^2}{2} \]  

where \( m_p \) – rotary mass, \( A_m \) – bending amplitude, \( \omega \) - oscillation frequency equal to the vehicle travel speed \( V \) (m/s) to deflection length ratio \( l \) (m). At this the maximum deflection length is limited by the base
cylinder length. As soon as work is equal to the difference of energy conversion, base cylinder bending capacity can be presented as follows:

\[
N = \frac{A}{r} = \frac{W}{r} = \frac{m_p V^2 (A_m^2 - A_0^2)}{2l^2} = \frac{m_p V^3 (A_m^3 - A_0^3)}{2l^3}
\]

(2)

Thus, to determine rotor bending capacity it is necessary to specify the deflection amplitude under a given loading pattern. In this case transverse load causes rotor shaft bending in the plane of load with rotor cross-sections displacement. The bent beam axis is called defected axis or deflection curve. New positions of cross-sections are characterized by linear and angular displacements shown in Fig. 1.

Figure 1. Calculation model to determine rotor bowing value

Deflection curve approximate differential equation has the form of [5,6,9,16,17,18,19]:

\[
EI \frac{d^2 A_m}{d l^2} = M
\]

(3)

where \( EI \) – rotor cross-section rigidity under bending, \( A_m \) - the deflection value second derivative, \( M \) – bending moment value, equal to:

\[
M = F (1 - \frac{l}{L}) \cdot l
\]

(4)

Substituting \( l \) as part of \( L \), that is \( l = \frac{mL}{12} \) we obtain:

\[
A_m^\prime = \frac{F L (\sigma - \sigma_0^2)}{EI}
\]

(5)

where \( L \)- total rotor length, \( l \) – a span between force application point and front rotor support. Integrating equation 5 rotor shaft rotation angle equation is obtained first time:

\[
\theta = \int \frac{F L (\sigma - \sigma_0^2) \cdot d\sigma}{EI} = \frac{F L}{EI} \left( \int \sigma d\sigma - \int \sigma^2 d\sigma \right) = \frac{F L}{EI} \left( \frac{\sigma^2}{2} - \frac{\sigma^3}{3} + C \right)
\]

(6)

taking into account, that in case the load is applied to point \( l = L/2 \) exactly in the middle of the rotor, angle \( \theta \) must be equal to 0 we obtain:

\[
\theta = A_m^\prime = \frac{F L}{12 EI} (6\sigma^2 - 4\sigma^3 - 1)
\]

(7)

integrating equation 7 for the second time, considering that at the rotor fixturing point (at points \( \sigma = 1 \) and \( \sigma = 0 \)) axle travel \( A_m \) must be equal to 0, the required rotor shaft deflection equation is obtained:

\[
A_m = \frac{F L}{12 EI} (2\sigma^3 - \sigma^4 - \sigma^2)
\]

(8)

Simultaneously solving equations 2 and 8, considering that a single rotor maximum load is equal to a half of total vehicle mass we obtain:
\[ N = \frac{m_p V^3 (FL \alpha (2\sigma^2 - \sigma^3 - 1))^2}{12 E I \frac{2L^3}{\sigma^3}} = \frac{m_p V^3 G^2 (2\sigma^2 - \sigma^3 - 1)^2}{1152 E I L \alpha} \] (9)

Taking into account that rotor resisting moment is conventionally estimated by value [5,6,9]:

\[ I = \frac{D^4(1 - C^4)}{2}, \quad \delta = \frac{d - D}{D} \] (10)

where \( D \) – base cylinder diameter, \( \delta \) - wall thickness. Taking into account that rotary mass is equal to its volume multiplied by material density \( m_p = \rho L \pi D \delta m \) the maximum radiated sound power appearing at the base cylinder bending is determined by equation:

\[ L_p = 10 \lg \left( \frac{\rho L V^3 G^2}{2350 ED^3(1 - C^4)} \right) \] (11)

Solution of equation 14 is shown in Fig. 2

Results

The results obtained indicate that for actual vehicles with the mass of over 2 tons and the base cylinder diameter 0.3 m, the generated noise level under rotor bending will not exceed 5 dB. For light cross-country vehicles with the mass of up to 1 ton, base cylinders with diameter of over 0.2 m completely exclude noise generation under bending. Thus, the necessary dependences have been obtained to design rotary-screw propulsion units of cross-country vehicles completely meeting normative noise requirements [11,12,13,14,15].

![Figure 2. Generated noise level change, appearing under base cylinder bending, of its diameter; 1- For 2.5 ton vehicle; 2 – for 750 kg vehicle](image)

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