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# Antiresonant Vibration Isolation in Vibratory Mills

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**Paweł PIEKAJ** [1*]

**Author's affiliations and addresses:**

[1] AGH University of Krakow,
al. Adama Mickiewicza 30,
30-059 Krakow, Poland

*** Correspondence:**

e-mail: piekaj@agh.edu.pl

Abstract:

This paper presents a conceptual review of a novel configuration of vibratory mills based on a two-mass anti-resonant dynamic system. The proposed solution introduces an intermediate drive frame between the grinding chamber and the foundation, enabling redistribution and partial compensation of dynamic excitation forces generated by inertial vibrators. The operating principle is based on tuning system parameters to achieve an anti-resonance condition, which significantly reduces vibration transmission to the supporting structure compared to classical single-mass over-resonant vibratory mill designs.

A simplified mechanical model is used to illustrate the dynamic behavior of the system and to highlight key differences relative to conventional over-resonant configurations. The influence of stiffness and damping tuning on vibration and dynamic force reduction is discussed, along with possible chamber motion trajectories. The paper also outlines key engineering challenges associated with scale-up, synchronization of excitation systems, and sensitivity to variations in operating conditions.

The results suggest that anti-resonance-based vibratory mills may represent a promising direction for reducing dynamic loads in large-scale milling systems while maintaining effective grinding performance.

Keywords: vibratory mill, antiresonance, vibrations, dynamic vibration absorber, vibration isolation

## 1. Introduction to Vibratory Mills

Vibratory mills belong to the group of comminution machines used for material size reduction during the grinding process to obtain a finely grained product. The grinding process takes place in the vibrating chamber of the mill. Material comminution results from collisions between freely moving grinding media and their impacts against the chamber walls. The kinetic energy of the grinding media is generated by the vibrations of the chamber, which are most commonly excited by inertial vibrators. The intensity of the interactions between the grinding media depends on the vibration parameters of the chamber, such as amplitude, frequency, and vibration trajectory, which are determined by both the machine designer and the user [1].

Despite the continuous development of vibratory mills, the over-resonant tubular mill with inertial excitation remains one of the most commonly used design solutions [1, 2, 3]. One of its variants is shown on Fig. 1. The mill presented in the figure, equipped with two inertial vibrators mounted on opposite sides of the chamber, is designed to operate with a quasi-circular vibration trajectory of the chamber. In other variants, where one of the vibrators is replaced by an appropriately selected stationary mass, the machine operates with an elliptical vibration trajectory [4, 5, 6, 7].

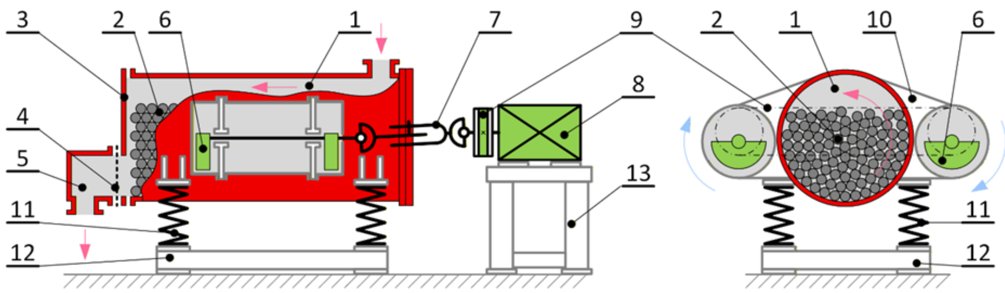


**Fig. 1.** Schematic diagram of a single-chamber vibratory mill with quasi-circular vibration trajectory and two inertial exciters:
1 – grinding chamber, 2 – chamber charge, 3 – chamber cover, 4 – screening partition, 5 – product outlet, 6 – inertia vibrators, 7 – Cardan shaft, 8 – electric motor, 9 – synchronization gearbox, 10 – supporting structure, 11 – spring suspension, 12 – mill foundation, 13 – motor foundation

Mills of this type offer several advantages. They make it possible to achieve a significantly higher degree of comminution while maintaining considerably smaller overall dimensions than conventional ball mills [1, 8]. They are also characterized by substantially lower power consumption [9]. However, one of their main drawbacks is the adverse dynamic impact of the mill on the foundation and the surrounding environment [10, 11, 12]. Consequently, reducing these dynamic effects has become the subject of numerous studies [13, 14, 15, 16]. In industrial practice, the most common solutions include the application of vibration isolation systems, such as spring–rubber isolators, heavy foundations, and vibration-isolated foundation blocks.

For relatively small mills, conventional vibration isolation is generally sufficient. However, the continuously increasing demand for finely ground materials has led to the development of progressively larger mill designs, in which the transmission of vibrations and dynamic forces to the foundation becomes an increasingly significant problem.

These adverse dynamic effects can be effectively reduced by applying anti-resonance vibration isolation based on the operating principle of the dynamic vibration absorber proposed by Frahm [17].

A review of the available literature has not revealed any examples of industrial vibratory mills operating on this principle, apart from prototype laboratory-scale anti-resonant vibratory mills that are

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currently under investigation [18, 19, 20]. One of these design solutions has been granted patent protection [21].

The structurally closest solutions based on the operating principle of a dynamic vibration absorber are vibratory conveyors, which have undergone intensive development in recent years [23, 24, 25, 26, 27, 28]. In these machines, the application of this phenomenon is generally limited to machines operating with a linear vibration trajectory, including designs equipped with a single inertial vibrator [29, 30].

In comparison with other machines of a similar structural configuration [22], the anti-resonant vibratory mill can operate with a linear, circular, or elliptical vibration trajectory of the chamber, while also allowing a higher chamber filling ratio. Furthermore, the anti-resonance effect significantly reduces the vibrations and dynamic forces transmitted to the foundation.

These characteristics indicate that anti-resonant vibratory mills represent a promising direction for the development of large-scale comminution machines, particularly in applications requiring a substantial reduction in the vibrations and dynamic forces transmitted to the foundation.

The aim of this paper is to present the operating principles, design solutions, and development prospects of this new class of machines.

## 2. Vibration Isolation in Anti-Resonant Vibratory Mills

The spring suspension of a vibratory mill is one of its key components, preventing the transmission of excessive vibrations and dynamic forces to the foundation. It is most commonly implemented as a system of steel springs located between the mill body and the supporting structure. Their function is to provide the required compliance of the system while effectively isolating the vibrations generated during machine operation. Depending on the operating requirements, elastomeric elements or hybrid solutions combining the properties of different elastic components may also be used.

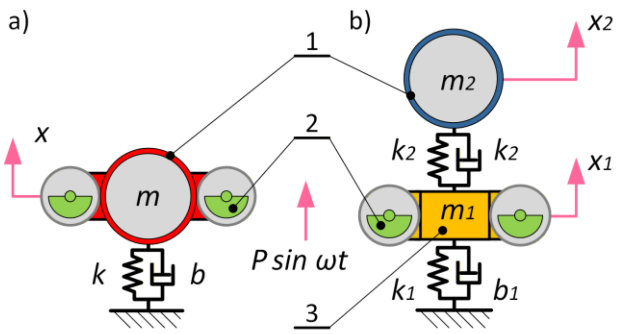


**Fig. 2.** Mechanical model of the mill system with linear (sectional) vertical vibration motion, where:
1 – chamber; 2 – inertial vibrators; 3 – drive frame; m, $m_1$, $m_2$ – equivalent masses; k, $k_1$, $k_2$ – stiffness coefficients of spring suspension;
b, $b_1$, $b_2$ – damping coefficients of spring suspension. (a) classical vibratory mill configuration; (b) anti-resonant vibratory mill configuration

In the conventional mill design discussed in the Introduction, shown in Fig. 1 and represented by the simplified vertical vibration model in Fig. 2(a), the vibration isolation principle is based on the appropriate selection of the spring suspension supporting the chamber, to which the inertial vibrators are attached. The suspension should be designed so that the ratio of the operating frequency of the vibrators, ω, to the natural frequency of the system, $\omega_0$, satisfies the condition $\omega/\omega_0>4$, ensuring over-resonant operation of the mill [31]. A properly selected spring suspension enables the machine to pass rapidly through resonance during start-up and reach its steady-state operating conditions. However, during steady-state operation, the vibrations and dynamic forces transmitted to the foundation are directly related to the vibrating mass of the chamber, its vibration amplitude, and the stiffness of the spring suspension. The steady-state motion of the system can be described by Eq. (1), while the excitation force generated by the inertial vibrators is given by Eq. (2).

$$m\ddot{x} + b\dot{x} + kx = F(t) \tag{1}$$

where:

| | | |
|---|---|---|
| $m$ | – | mass of the chamber and inertial vibrators, kg, |
| $k$ | – | stiffness coefficients of the chamber spring suspension, N/m, |
| $b$ | – | viscous damping coefficient of the chamber spring suspension, Ns/m, |
| $x$ | – | displacement, m, |
| $F(t)$ | – | external excitation force, N(t). |

$$F(t) = m_0\, e\, \omega^2 \cos(\omega t) \tag{2}$$

where:

| | | |
|---|---|---|
| $F(t)$ | – | external excitation force, N(t), |
| $m_0$ | – | unbalanced mass of the exciters, kg, |
| $e$ | – | eccentricity of the unbalanced mass, m, |
| $\omega$ | – | angular frequency, rad/s. |

In anti-resonant vibratory mills, the vibrations and the resulting dynamic forces can be significantly reduced by applying the operating principle of a dynamic vibration absorber and exploiting the anti-resonance phenomenon. A simplified mechanical model of such a mill is shown in Fig. 2(b).

The principal difference compared with the conventional design shown in Fig. 2(a) is the addition of an additional intermediate element, referred to as the drive frame, between the chamber and the foundation, together with the relocation of the inertial vibrators from the chamber to the drive frame.

The spring suspension supporting the drive frame is designed according to the same principles as in the conventional mill. In contrast, the spring suspension located between the drive frame and the chamber must be tuned to the intended steady-state operating frequency of the mill. This is achieved by selecting the stiffness coefficients of spring suspension so that the natural frequency of the chamber subsystem is equal to the excitation frequency, which may be expressed as $\omega/\omega_0 \approx 1$ Such tuning enables the mill to operate under anti-resonant conditions.

The steady-state motion of the system is described by Eq. (3), while the excitation force generated by the inertial vibrators is given by Eq. (2).

$$\begin{gathered} m_1\ddot{x}_1 + b_1\dot{x}_1 + b_2(\dot{x}_1 - \dot{x}_2) + k_1x_1 + k_2(x_1 - x_2) = F(t) \\ m_2\ddot{x}_2 + b_2(\dot{x}_2 - \dot{x}_1) + k_2(x_2 - x_1) = 0 \end{gathered} \tag{3}$$

where:

| | | |
|---|---|---|
| $m_1$ | – | mass of the drive frame and inertial vibrators, kg, |
| $m_2$ | – | mass of the chamber, kg, |
| $k_1$ | – | stiffness coefficients of the drive frame spring suspension, N/m, |

$k_2$ – stiffness coefficients of the chamber spring suspension, N/m,
$b_1$ – viscous damping coefficient of the drive frame spring suspension, Ns/m,
$b_2$ – viscous damping coefficient of the chamber spring suspension, Ns/m,
$x_1$ – displacement of the drive frame, m,
$x_2$ – displacement of the chamber, m,
$F(t)$ – external excitation force, N(t).

Under ideal anti-resonant conditions, when the spring suspension is assumed to be undamped, the dynamic force generated by the vibrating chamber and transmitted through its spring suspension compensates for the excitation force produced by the inertial vibrators mounted on the drive frame. When the system is perfectly tuned to the excitation frequency, the resultant dynamic force acting on the drive frame becomes zero, resulting in the elimination of its vibrations.

When damping is taken into account, the vibrations and dynamic forces transmitted to the foundation cannot be completely eliminated. Nevertheless, in practice their magnitudes are several times lower than those observed in a conventional mill operating with the same vibration parameters of the chamber.

The differences in the amplitude–frequency characteristics obtained for representative parameters of a conventional and an anti-resonant vibratory mill are presented in Fig. 3. For comparison purposes, the simulations were performed assuming that the total unbalanced mass of the inertial vibrators was equal to 20% of the chamber mass including its charge ($0.2m_2$). The mass of the drive frame together with the inertial vibrators ($m_1$) was assumed to be equal to the mass of the chamber ($m_2$). The parameters of the stiffness coefficients of spring suspension ($k$, $k_1$, $k_2$) and damping coefficients of spring suspension ($b$, $b_1$, $b_2$,) were selected according to generally accepted design recommendations [31]. The parameters used in the simulations are summarized in Table 1.

**Table 1.** List of parameters used in the simulation

| Parameter | Value | Unit |
|---|---|---|
| **$m$** | 12000 | [kg] |
| **$m_1$** | 10000 | |
| **$m_2$** | 10000 | |
| **$k$** | 7500000 | [N/m] |
| **$k_1$** | 13750000 | |
| **$k_2$** | 100000000 | |
| **$b$** | 12000 | [Ns/m] |
| **$b_1$** | 22000 | |
| **$b_2$** | 40000 | |
| **$m_0$** | 300 | [kg] |
| **$e$** | 0.3 | [m] |
| **$\omega$** | 100 | [rad/s] |

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As can be seen from the presented characteristics, at the operating point of the mill both the vibration amplitude and the dynamic forces transmitted directly to the foundation are several times lower for the anti-resonant vibratory mill than for the conventional mill.

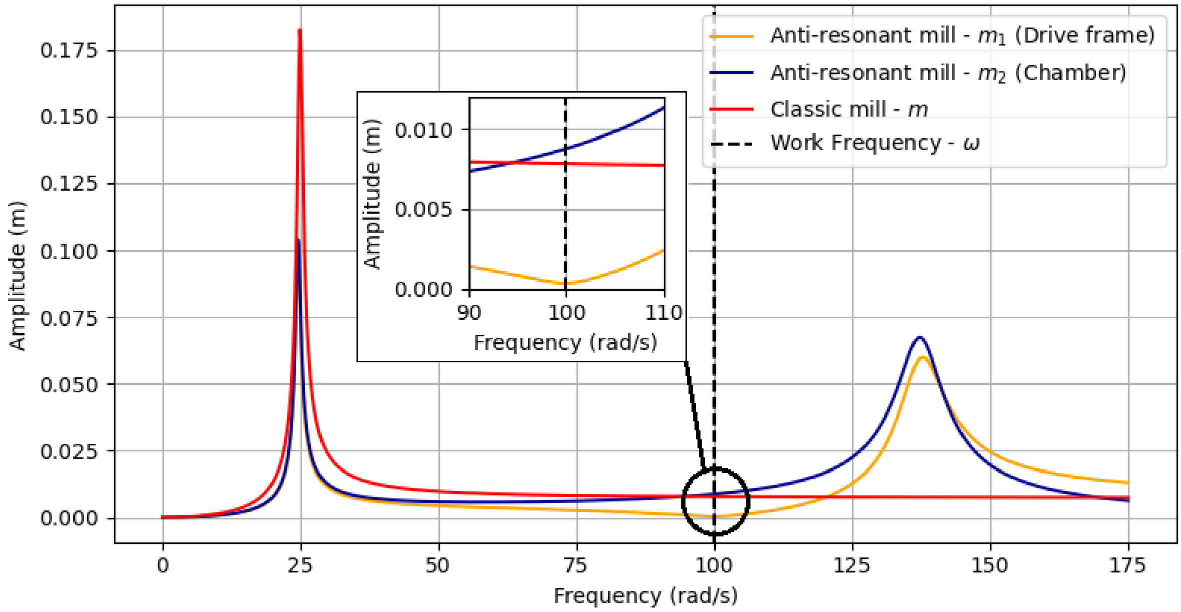


**Fig. 3.** Amplitude–frequency response of a classical and anti-resonant vibratory mill

It should be emphasized that the model presented in this paper considers only vertical motion and is intended solely to illustrate the operating principle of the anti-resonant vibratory mill. The model does not account for horizontal motion or for the behaviour of the chamber charge, both of which significantly influence mill operation. Nevertheless, appropriate selection of the system parameters makes it possible to maintain high effectiveness of anti-resonance vibration isolation for both linear vibration trajectories [19] and circular vibration trajectories [20], as demonstrated by more advanced mathematical models and experimental investigations reported in [18, 19, 20].

## 3. Review of Structural Designs

Anti-resonant vibratory mills are two-mass systems. From the perspective of the operating principle of a dynamic vibration absorber, the chamber assembly functions as the absorber, whereas the drive frame constitutes the primary vibrating subsystem whose vibrations are to be suppressed. Unlike the classical single-direction dynamic vibration absorber illustrated in Fig. 2(b), anti-resonant vibratory mills used combined systems in which the chamber performs more complex motion, typically following a quasi-circular or elliptical vibration trajectory. Such trajectories are obtained by combining two mutually perpendicular vibration absorbers tuned independently in both directions. This design makes it possible to achieve not only a linear vibration trajectory [19], but also quasi-circular [20] and elliptical vibration trajectories.

Since anti-resonant vibratory mills represent a relatively new design concept, the number of technical solutions reported in the literature remains limited and is currently restricted to laboratory-scale machines. Schematics of the existing designs are presented in Figs. 4 and 5.

In all of the presented designs, three main assemblies can be distinguished: the chamber, the drive frame, and the foundation. The chamber assembly is connected to the drive frame by means of its spring suspension (5). The drive frame is supported on the foundation through its own spring suspension (7).

The chamber assembly consists of the chamber (1) mounted inside the chamber frame (2), which facilitates convenient installation and removal of the chamber. The chamber frame incorporates the chamber spring suspension (5), composed of coil springs with appropriately selected stiffness coefficients of spring suspension and damping coefficients of spring suspension in both the vertical and horizontal directions. In addition, the chamber frame allows the attachment of correction masses for fine adjustment of the dynamic properties of the system.

The drive frame assembly consists of the drive frame (6) together with its spring suspension (7), to which two inertial vibrators (3) are attached. The vibrators are synchronized by means of a toothed-belt transmission (4), which ensures identical angular velocities of both vibrators.

The foundation assembly consists of a foundation frame (8) equipped with four adjustable feet, allowing accurate leveling of the machine and, if required, anchoring it to the foundation.

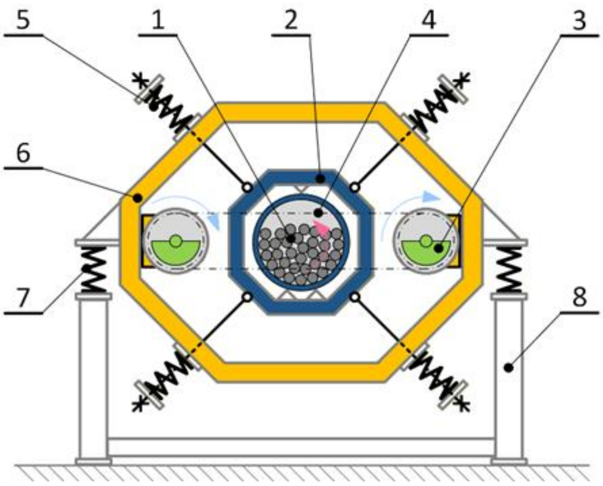


**Fig. 4.** Schematic diagram of a laboratory anti-resonant vibratory mill with circular and linear (sectional) chamber vibration trajectories:
1 – chamber, 2 – chamber frame, 3 – inertial vibrators, 4 – synchronization belt transmission, 5 – chamber spring suspension, 6 – drive frame, 7 – drive frame spring suspension, 8 – foundation frame

The first anti-resonant vibratory mill reported in the literature is a single-chamber design in which the chamber performs vibrations with a quasi-circular trajectory (Fig. 4) [18, 19, 20]. This trajectory represents the standard operating mode of the machine. However, after appropriate design modifications, the mill can also operate with a linear vibration trajectory. This solution employs an X-shaped spring suspension with appropriately selected parameters that provide identical stiffness and damping characteristics in both the horizontal and vertical directions.

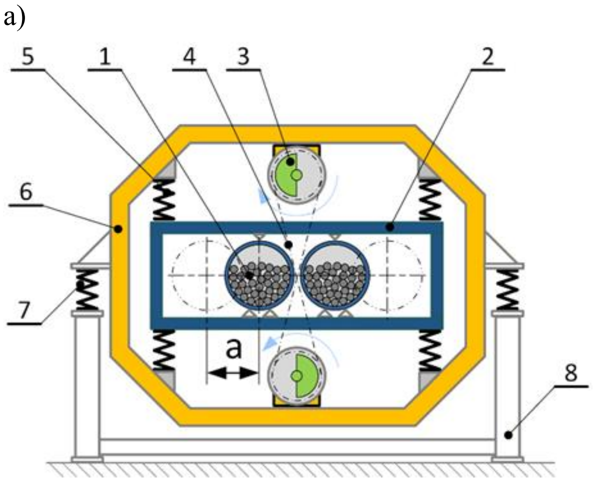


**Fig. 5.** Schematic diagram of a laboratory anti-resonant vibratory mill with elliptical chamber vibration trajectories:
1 – chamber, 2 – chamber frame, 3 – inertial vibrators, 4 – synchronization belt transmission, 5 – chamber spring suspension, 6 – drive frame, 7 – drive frame spring suspension, 8 – foundation frame. (a) anti-resonant vibratory mill with vibrators positioned vertically; (b) anti-resonant vibratory mill with vibrators positioned horizontally

Another design currently under investigation is a two-chamber mill in which the chambers perform vibrations with an elliptical trajectory. After appropriate design modifications, the machine can also operate with linear and quasi-circular vibration trajectories.

In two-chamber mills, the chambers are arranged symmetrically with respect to the center of mass of the system and simultaneously undergo translational and angular motion, resulting in an elliptical vibration trajectory of each chamber. Examples of such designs, differing in the arrangement of the inertial vibrators, are presented in Fig. 5. In these machines, the stiffness coefficients of spring suspension in the longitudinal and transverse directions are generally different and are selected to ensure the occurrence of the anti-resonant condition in both the translational and angular modes of vibration.

In the presented designs, obtaining the required vibration amplitude and trajectory of the chambers requires appropriate synchronization of the inertial vibrators. The only exception is the configuration producing a linear vibration trajectory, in which the phenomenon of self-synchronization of the inertial vibrators occurs, eliminating the need for an additional synchronizing belt transmission.

## 4. Discussion: Advantages and Limitations

The anti-resonant vibratory mills discussed in this paper constitute an alternative to conventional over-resonant vibratory mills. By adapting the operating principle of the dynamic vibration absorber and exploiting the anti-resonance phenomenon, it is possible to significantly reduce the vibrations and dynamic forces transmitted to the foundation. As demonstrated in previous studies [19, 20], these dynamic effects may be several times lower than those observed in conventional designs (Fig. 3), while maintaining the required motion parameters of the chamber.

This improvement is achieved by modifying the structural configuration of the mill through the adding of an additional intermediate mass in the form of a drive frame and relocating the drive assembly, typically consisting of inertial vibrators, from the chamber to the drive frame. In conventional designs, the inertial vibrators are mounted directly on the chamber.

This modification reduces the mass of the chamber assembly compared with the conventional design. Consequently, with the same drive assembly, a greater vibration amplitude of the chamber can be achieved, as shown by the characteristics presented in Fig. 3.

Alternatively, after relocating the inertial vibrators from the chamber to the drive frame, instead of increasing the vibration amplitude of the chamber, its original amplitude may be maintained by appropriately reducing the excitation force generated by the inertial vibrators, thereby reducing the required size and power of the drive system. From the user's perspective, this approach may be particularly advantageous, as it allows a conventional over-resonant vibratory mill to be replaced by an anti-resonant vibratory mill without changing the technological parameters of the grinding process. The motion parameters of the chamber, including vibration frequency, vibration amplitude, and vibration trajectory, can be maintained while simultaneously reducing electrical power consumption and employing smaller drive units. Furthermore, reducing the excitation force leads to an additional decrease in the dynamic forces and vibrations transmitted to the foundation compared with the case in which the excitation force is maintained at the same level as in a conventional mill.

However, the final verification of these assumptions requires further investigation of the influence of the chamber charge. In the model presented in this paper, the charge was simplified as a homogeneous rigid mass attached to the chamber, which represents only an approximation of the actual dynamic conditions occurring during the grinding process [19, 20].

The main disadvantages of the proposed solution are the increased mass and structural complexity resulting from the addition of an additional drive frame and the second stage of the spring suspension. This results in a greater number of structural components, which may increase the manufacturing, assembly, and operating costs of the machine.

The advantages of applying the anti-resonance phenomenon become particularly significant in the case of large-scale industrial mills, where reducing vibrations and dynamic forces transmitted to the foundation is one of the major design challenges. However, the research conducted to date has been limited primarily to laboratory-scale prototypes of anti-resonant vibratory mills [18, 19, 20]. Although the reported results confirm the practical feasibility of exploiting the anti-resonance phenomenon, studies on high-capacity industrial machines are still lacking. Further research is therefore required to evaluate the durability of the elastic elements, the long-term reliability of the system, and the economic feasibility of implementing this technology in industrial applications.

The effectiveness of the anti-resonance system largely depends on accurate tuning of the spring suspension of the chamber to the operating frequency. Changes in the stiffness of the elastic elements caused by wear, manufacturing tolerances, or variations in their material properties may reduce the effectiveness of the anti-resonance phenomenon, resulting in increased vibration amplitudes and dynamic forces transmitted to the foundation.

Another factor that complicates maintaining the optimum tuning of the system is the variation in the mass of the working assembly. In continuously operated mills, these variations result from changes in the chamber filling ratio caused by the continuous inflow of feed material, the discharge of the ground product, and the wear of the grinding media. In batch-operated mills, where the chamber charge remains constant, maintaining the anti-resonant condition becomes more difficult because the degree of material comminution changes during the grinding process [20]. These challenges may be addressed by implementing appropriate drive control strategies and adjusting the correction masses.

## 5. Technical Challenges and Future Work

Despite the promising results obtained for laboratory-scale prototypes, anti-resonant vibratory mills still require the resolution of several technical problems before they can be implemented in industrial applications. One of the most important issues remains the maintenance of the system in an anti-resonant state at the operating point. The anti-resonance effect occurs only within a limited range of dynamic parameters; therefore, the main research challenges to be addressed in future work include, in particular, the analysis of the influence of changes in the chamber filling ratio for quasi-circular and elliptical vibration trajectories during continuous grinding processes, taking into account the effects of both the grinding media and the material being processed.

Preliminary studies have shown that the degree of comminution of the processed material influences the effectiveness of the anti-resonance phenomenon. Consequently, an important direction for further research is a detailed description of the influence of changes in particle size distribution during the grinding process, especially in batch-operated systems, on the effectiveness of anti-resonance vibration isolation.

There is also a lack of studies concerning the operation of anti-resonant vibratory mills in continuous grinding processes. It may be assumed that, under steady-state conditions, the influence of changes in particle size distribution on the dynamic properties of the system will be smaller than in batch processes, since the continuous inflow of feed material and discharge of the product result in a relatively constant granulometric composition of the material inside the chamber. However, this hypothesis requires experimental verification.

Many comminution processes are carried out using wet grinding with appropriate suspensions. However, no research results have yet been reported regarding the possibility of operating anti-resonant vibratory mills under such conditions. Therefore, evaluating the influence of the presence of a liquid phase on the dynamic properties of the system and the efficiency of the grinding process constitutes another important research direction.

Another important step in future research is the development of a model that accounts for the real chamber charge, consisting of both grinding media and processed material. Existing models correctly describe the operation of the system only in the case where the charge consists exclusively of grinding media; therefore, it is necessary to extend them to include the influence of the processed material on the dynamic properties of the system.

## 6. Conclusions and Perspectives

Anti-resonant vibratory mills represent a promising direction in the development of comminution machines, particularly in applications requiring the reduction of vibrations and dynamic forces transmitted to the foundation. The application of the operating principle of the dynamic vibration absorber enables a significant reduction of dynamic interactions while maintaining the required motion parameters of the working chamber. An additional advantage of this solution is the possibility of either increasing the vibration amplitude of the chamber or reducing the required power of the drive system, which may contribute to improved energy efficiency of the grinding process.

Previous studies have confirmed the validity of the proposed concept and demonstrated the possibility of obtaining linear and quasi-circular vibration trajectories of the chamber while maintaining

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the anti-resonant condition. At the same time, it should be emphasized that most available results concern laboratory-scale prototypes and simplified models. Therefore, before this technology can be implemented in industrial practice, several research and engineering issues still need to be resolved.

The main directions for further research include the development of dynamic models that account for the real nature of the working chamber charge, consisting of both grinding media and the processed material; the analysis of the influence of changes in the chamber filling ratio and the degree of comminution of the material on the conditions for the occurrence of anti-resonance; as well as the assessment of mill performance in continuous and wet grinding processes. An important issue also remains the development of methods ensuring proper tuning of the system during long-term operation.

Further development of anti-resonant vibratory mills also requires investigations conducted on pilot-scale and industrial-scale machines. The results of such studies will enable the assessment of structural durability, operational reliability, energy efficiency, and the economic feasibility of this technology. Solving these issues may contribute to the development of a new generation of large-scale vibratory mills characterized by reduced dynamic interaction with the environment while maintaining high comminution efficiency.

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