Innovative structural solution for heavy loaded vibrating screens

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Abstract
In this paper an innovative design solution, that allows to enhance the structural resistance and the dynamic performances of a vibrating screen for inert materials, is presented. The new design does not significantly affect the geometry of the traditional screens, keeping the same global dimensions and almost the same mass value. In fact the aim of this study was to design a new vibrating screen having almost the same dimensions but that could give a much higher dynamic structural resistance at frequencies and load amplitudes much higher than the nominal ones. Numerical finite element models were generated to investigate the structural and dynamic behavior of a standard vibrating screen. These analyses allowed to modify the geometrical parameters of the traditional screen and to design the new one. Accurate three-dimensional FE models were so generated in order to evaluate the best design solution, in terms of dynamic structural resistance, able to reduce the stress values at the most stressed area. The fatigue resistance of all the components of the new screen, with particular attention to the welding joints, was checked. Experimental full scale tests on a prototype of the new screen were carried out in order to validate the numerical models and mostly to verify the structural integrity of the vibrating screen during the working conditions. Strains at the surface of the most stressed areas of the screen were measured in dynamic working conditions, at different frequencies and load amplitudes; these strain values were compared with the numerical ones in order to validate the numerical results. The new screen was patented [1].

Keywords: Vibrating screens, new structure, resistance, dynamic optimization

Nomenclature

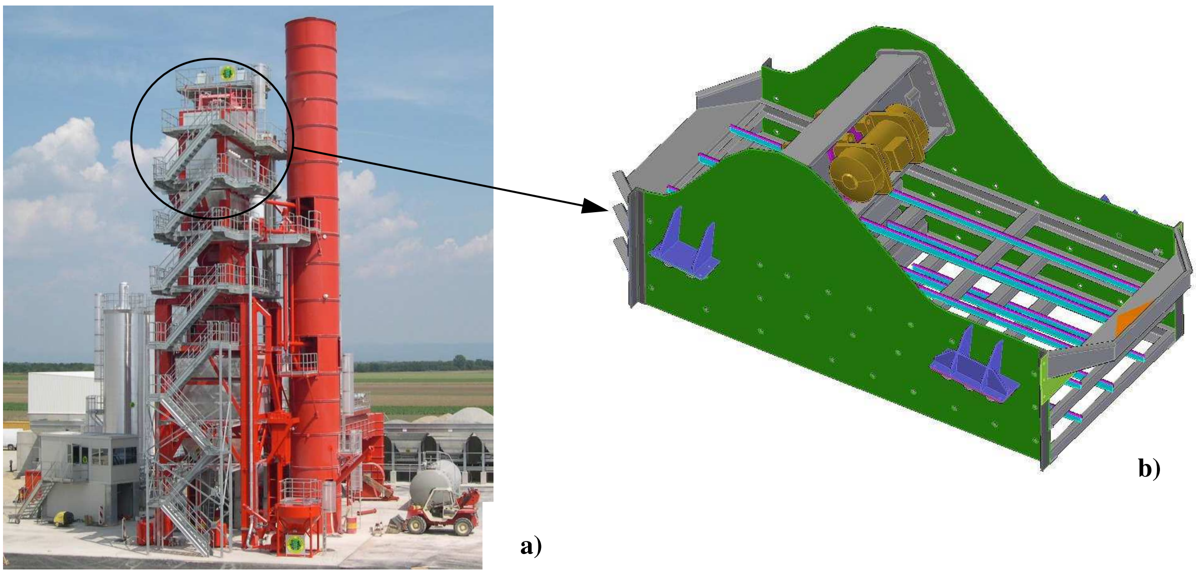
1. Introduction

Heavy load vibrating screens, horizontal or inclined, are mounted at the top of batch type asphalt plants or continuous asphalt plants. The screen allows the selection and segregation of prescribed components of a powder or of an aggregation of pebbles and stones according to specific particle size. The accuracy of the selection is very important in a lot of application for which reaching the correct recipe of the final mixture is mandatory. Among such applications surely the asphalt plants, that allow the production and installation of the asphalt for roads and highways, require a perfect dosage of all the components. A prescribed percentage of inert materials with designed particle size has to be achieved for the asphalt mixture and a good vibrating screen has to guarantee its performances for 24 hours a day [1, 2]. The temperature of the inert materials that flow through the vibrating screen is in the range 150-200 °C. Pebbles and stones have to be accurately heated and dehydrated before the selection into the screen in order to be ready to be put in the mixture and bond to the other constituents of the asphalt. The asphalt production plant is usually a vertical tower which can be high up to 30 m. The inert materials (pebbles and stones) have to be pre-dosed, heated and dehydrated before entering the vibrating screen at the top of the tower (they reach the top of the tower by means of a cups elevator that feeds a drying heater. After the screening process the pebbles having different particle size are gathered into steel hoppers put under the screen. The material stored in the hoppers feeds the mixer according to prescribed percentage ratios; during the mixing process the bitumen is added with specified additives that allow the production of a compact texture asphalt.
The dynamic analysis of vibrating screens was carried out in [3-5]. In [3] the Authors propose a theoretical model for the dynamic behavior evaluation of vibrating screen and a new screen with elliptical trace is presented. The multi degree of freedom vibration theory was used and no experimental validation of the model is available. In [4] the aim of the paper is to improve the reliability of large vibrating screens. The Authors present a new design of a hyperstatic net-beam structure and an optimal dynamic analysis of the vibrating screen. Finite element models of the screen were developed and the new screen proves to have much higher structural strength and the dynamic behavior is enhanced. More recently the author of this paper proposed a general theoretical model to study the dynamic behavior of high loaded vibrating screens[5]. A general dynamic theoretical model is proposed and a procedure for dynamic optimization of the mail parameters of a screen is described. Dynamic and static finite element models of a high loaded screen were developed and the theoretical procedure was implemented in Matlab®. Full scale experimental tests were carried out on a real screen in order to confirm the theoretical and numerical models. A complete strain gages campaign was prepared and several dynamic tests were carried out confirming the values of the dynamic stresses calculated through the FEM models. DEM (Discrete Element Method) was used to simulate the particle flow in vibrating screens in [6-11]. In [6]

the Authors consider a banana screens of large dimensions (three decks or five decks). Numerical experiments (3D DEM models) were used to set and control the performances of the screens, useful to optimize and control the screening process. Screens performances were improved by reducing the operating amplitude and frequency of vibrations. In [7, 8] a double deck big banana screen for high accelerations was studied and the DEM method was implemented to study and improve the performances. Different peak accelerations were simulated. In [11] the DEM numerical method was used to simulate the particle motion. The particle kinetics and motion is treated in also in some other references [12-16]. In [12] the Authors gives guidelines to check the screening performance and efficiency and checks the effects of many process parameters. In [13] the Authors report an accurate study of the screening dynamic motion of cylindrical particles. Full scale tests were conducted by using a CCD-camera and an image analyzing software. The measurements give a good prediction of the grade of separation and the particle distribution is like the one of a closed surface. In [14-16] the Author reports a study on the transport velocity of the material on the screen, the analysis of the mechanisms of stratification and passage of the material through the screen and the study of the effect of particle size on the screening performance. In [17] the fatigue failure analysis of the support beam of a vibrating screen is reported. A double deck screen is considered and the fatigue assessment is done according to BS 5400 standard. [17] is one of the few references, as far as the author of this paper knows, in which the structural assessment of a vibrating screen in studied. The failure occurred at the beam area of the screen, were the exciter mechanism is installed. The operating frequency of the screen is 15 Hz and millions of cycles can be reached in some hundreds of hours. The screen is expected to work for 30,000 hours and special attention has to be paid to the stress-strain and fatigue design of all the structural component of the screen. The screen studied in [17] is a large screen of dimensions 6.5 m by 3.0 m and the screen in comparable to the one presented in this paper.
The aim of this work is to propose a new design solution for vibrating screens that does not significantly modify the geometry of the traditional screens, keeping the same global dimensions and almost the same mass value and at the same time exert a much higher dynamic structural resistance at frequencies and load amplitudes much higher than the nominal ones. Accurate three-dimensional FE models were so generated in order to evaluate the best design solution, in terms of dynamic structural resistance, able to reduce the stress values at the most stressed area. The fatigue resistance of all the components of the new screen, with particular attention to the welding joints, was checked. Experimental full scale tests on a prototype of the new screen were carried out in order to validate the numerical models and mostly to verify the structural integrity of the vibrating screen during the working conditions. Strains at the surface of the most stressed areas of the screen were measured in dynamic working conditions, at different frequencies and load amplitudes; these strain values were compared with the numerical ones in order to validate the numerical results. The new design of the vibrating screen was patented [18]. The vibrating screen is usually made of low carbon steel and the components of the screen can be assembled both by means of bolted connections or by using welded joints. Bolted and welded connections are currently used to build structures that can rarely be used under dynamic loads on where an important vibrating load is expected to act on the equipment. For this reason bot the reliability and the performances of the vibrating screens have to accurately verified. The utilization of higher quality materials and the execution and non-destructive control processes of higher quality welded connections would put the vibrating screen out of the market. On the other hand the asphalt producers require always better performances (quantity of screened material per hour) and absolute absence of fatigue induced damages during the 24 h per day screening process. Bearing in mind these consideration it was decided not to change the materials and the welding technological process to connect the components of the screen but to modify the structure of the vibrating screen in order to improve its performances.

2. The vibrating screen

A sample plant for the production of the road asphalt is shown in figure 1.

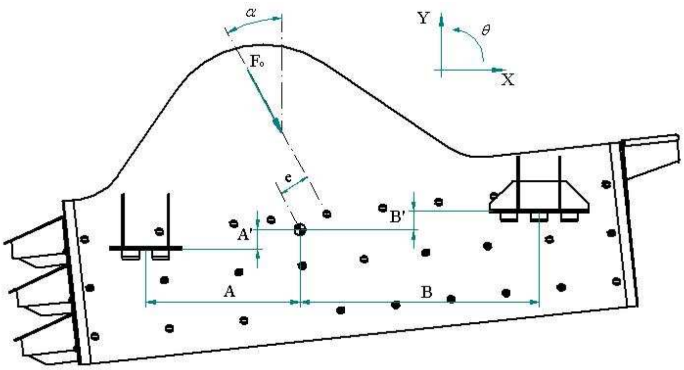


a)

b)

figure 1: a) Example of a batch type asphalt plant: the vibrating screen is at the top of the plant and b) Heavy loaded vibrating screen

The vibrating screen studied in this paper is a low carbon steel made one. Its main dimensions are shows in figure 2.



a) **b)**

figure 2: a) picture of the vibrating screen and b) sketch of the vibrating screen

The vibrating screen of the type shown in figure 2 in composed by a case-shaped structure, or body, provided with suitable screening surfaces and supported by a base through springs, to be set in vibration by a suitable vibrating device. Screening is an operation done for the most different purposes. By way of example, the following purposes are mentioned: separating the biggest fragments (or refuse) contained in a mixture, either to eliminate them or to reduce them by crushing or grinding; separating the smallest (fine) fragments both to agglomerate them in the granulation workshops and to eliminate them as waste or recycle them for other uses; classifying the crushed goods according to their commercial sizes (materials for building roads, abrasives, coals, etc.); classifying the goods for operations of mechanical or physicochemical treatment that shall take place with materials of homogeneous sizes (ore concentration, coal washing, selection of cube-shaped materials, etc.); material drying, etc. In the vibration screens, transport of the material on the screening surface (grids, perforated plates, wire nets or meshes, for example) is obtained by impulse, while vibration of the mass gives rise to a stratification bringing the greatest sizes to the top surface and the smallest (the finest) ones to the layer base, which is very useful for screening. The vibration-generating device can be a merely mechanical, electromechanical (motor and eccentric masses) or also electromagnetic device. To make a screening machine capable of putting into practice the processes for selection or treatment of the materials at its best, it is necessary to carry out a dedicated planning of the machine itself. Planning of a vibrating machine is complicated because both criteria of static resistance and planning criteria for fatigue strength are required to be applied; in addition, determination of the loads acting on the machines appears to be a very hard task due to the difficulty of fully understanding the operating dynamics of the vibrating machines. Since in the vibration screen of the above mentioned type the whole case-shaped body is submitted to vibrations, this body must be made strong enough to withstand the dynamic stresses due to vibration. On the other hand, the structure in known machines cannot be strengthened beyond a certain degree because the stresses that are harmful to the machine become more numerous and stronger as the weight of the structure set in vibration increases. In addition too big a mass brings to the necessity for vibrating devices having an unacceptable power and bulkiness. Usually, the combination of the operating parameters makes these known machines reach accelerations that are of 4-5 g at most. However, the availability of vibrating screens with much greater accelerations would be of the greatest interest because the greater the acceleration is, the greater the screening yield. In addition, greater accelerations would also enable an optimal screening of big sizes to be obtained as well as an efficient unclogging of the screening surfaces. However, for the above reasons, a person skilled in the art deems it practically impossible to obtain such greater accelerations to acceptable costs. It is a general aim of the present invention to obviate the above mentioned drawbacks by providing a vibration screening machine capable of withstanding high accelerations, even in the order of 18-20 g, with relatively low costs and reduced complexity. In view of the above aim, the manufacture of a vibration screen has been conceived which comprises a case-shaped body with parallel side walls between which perforated screening surfaces are disposed and which is suspended over a base through spring means and is provided with vibrating devices, characterized in that the side walls are each made up at least partly of two plate sheets closely spaced and parallel so that each of them forms one of the two faces of the wall and both delimit a hollow space there between, the sheets being interconnected with each other by means of interconnecting elements disposed in the hollow space. The principal dimensions and characteristics of the screen are reported in Table 1.

Table 1: principal dimensions and characteristics of the screen

Maximum vertical dimension of the screen	**1400**
Maximum horizontal dimension of the screen	**3000 mm**
Number of selections of the screen	**6**
Mass of the screen	**1962 kg**
Horizontal distance between the springs (loading side-discharge side	**2100 mm**
Number of springs at the loading side	**6**
Number of springs at the discharge side	**4**
Diameter of one spring	**110 mm**
Diameter of the wire of the springs	**17 mm**
Number of active coils of the springs	**8**
Length of one spring	**248 mm**
Pitch of the springs	**29 mm**
Axial stiffness of one spring	**77,5 N/mm**
Horizontal distance of the centre of gravity of the screen from the loading side springs (B in Fig. 2b)	**1240 mm**
Horizontal distance of the centre of gravity of the screen from the discharge side springs (A in Fig. 2b)	**860 mm**
Vertical distance of the centre of gravity of the screen from the loading side springs (B' in Fig. 2b)	**47 mm**
Vertical distance of the centre of gravity of the screen from the discharge side springs (A' in Fig. 2b)	**247 mm**
Type of electrical vibrators	**Italvibras® MSVI S90**
Number of electrical vibrators mounted on the beam	**2**
Inclination angle of the beam of the screen on which the vibrators are mounted (α in Fig. 2b)	**27°**
Frequency of load application of the electrical vibrators	**1200 rpm**
Maximum load of each vibrator (F_0 in Fig. 2b)	**100000 N**
Eccentricity of the applied vibrating load (e in Fig. 2b) for the optimum screening performance	**0 mm**

3. The dynamic behavior of the screen: literature analytical models

The dynamic behavior of the vibrating screens was accurately studied in [5]. It is assumed that temperature and the inert material flowing through the vibrating screen have no effects on the dynamic behavior of the mechanical system. The screen was schematized as rigid body having the dimensions of the real screen and the same mass and inertia properties. This hypothesis is supported by the fact that the stiffness of the springs on which the screen is mounted is much lower than the stiffness of the steel structure of the screen itself. The equilibrium equations were written first and the analysis of the harmonic behavior of the screen was performed; the natural frequencies of the vibrating screens were calculated afterword and the model gave all the cinematic and dynamic parameters useful to calculate all the forces acting on the structure (displacement, velocity and acceleration fields). The whole screen has six degrees of freedom, three rotational and three translational degrees of freedom but the three dimensional problem can be easily studied as a planar problem, reducing the number of degrees of freedom to three, two translational and one rotational degree of freedom as shown in figure 2b. For a better selection of the material that flows through the screen the angle θ (Fig. 2b), the pitching angle, should be kept constant and equal to zero. In fact, only during the start-up phase of the screening process the pitching angle varies significantly but these variations tend to reduce to zero as soon as the vibrating screen reaches the normal working condition (the start-up phase usually takes non longer than some tens of seconds). The inertial forces along the coordinate axis are assumed to be concentrated in the centre of gravity of the structure. The weight of the whole structure will not be taken into account in the dynamic equilibrium equations because it only has an influence on the static equilibrium position of the screen. Not only the axial behaviour of the springs that support the screen was taken into account but also the shear behaviour of the springs and their hysteretic damping behaviour.

The amplitude of the force given by the electric vibrators mounted at the top of the screen is:

$$F_0(t) = m_\varepsilon \omega^2 \cos(\omega t) \tag{1}$$

The hysteretic damping forces given by the springs are the following ones (2):

$$\begin{cases} F_{c1} = (\dot{x} + A\dot{\theta})c_1 \\ F_{c2} = (\dot{x} - B\dot{\theta})c_2 \\ F_{tc1} = (\dot{y} + A\dot{\theta})c_{t1} \\ F_{tc2} = (\dot{y} - B\dot{\theta})c_{t2} \end{cases} \tag{2}$$

The differential dynamic equilibrium equations of the mechanical system are reported in the system of equations (3):

$$\begin{cases} m\ddot{x} + c_1\dot{x} + c_2\dot{x} + A'c_1\dot{\theta} - B'c_2\dot{\theta} + k_{t1}x + k_{t2}x + k_{t1}A'\theta - k_{t2}B'\theta = F_{0x}\cos(\omega t) \\ m\ddot{y} + c_{t1}\dot{y} + c_{t2}\dot{y} + Ac_{t1}\dot{\theta} - Bc_{t2}\dot{\theta} + k_1 y + k_2 y + k_2 B'\theta - k_1 A'\theta = F_{0y}\cos(\omega t) \\ k_1 Ay + k_{t1}A'^2\theta + c_1 A\dot{y} + c_1 A^2\dot{\theta} + c_{t1}A'\dot{x} - c_2 B\dot{y} + c_2 B^2\dot{\theta} + c_{t2}B'^2\dot{\theta} - c_{t2}B'\dot{x} + \\ + k_{t2}B'^2\theta - k_2 By + c_{t2}A'^2\dot{\theta} + k_1 A^2\theta + k_2 B^2\theta + k_{t1}A'x - k_{t2}B'x + J\ddot{\theta} = -e \cdot F_0\cos(\omega t) \end{cases} \tag{3}$$

It is possible to simplify system (3) by introducing the following terms (4):

$$\begin{aligned} &K_{A-eq} = k_1 + k_2 \quad - \quad K_{T-eq} = k_{t1} + k_{t2} \quad - \quad C = c_1 + c_2 \quad - \quad D = A'c_1 - B'c_2 \\ &E = k_{t1}A' - k_{t2}B' \quad - \quad G = c_{t1} + c_{t2} \quad - \quad H = Ac_{t1} - Bc_{t2} \quad - \quad I = k_2 B' - k_1 A' \\ &L = k_1 A - k_2 B \quad - \quad M = k_{t1}A' - k_{t2}B' \quad - \quad N = c_1 A - c_2 B \quad - \quad O = c_{t1}A' - c_{t2}B' \\ &P = k_{t1}A'^2 + k_{t2}B'^2 + k_1 A^2 + k_2 B^2 \quad - \quad Q = c_1 A^2 + c_2 B^2 + c_{t2}B'^2 + c_{t2}A'^2 \\ &F_X = F_{0x}\cos(\omega t) \quad - \quad F_Y = F_{0y}\cos(\omega t) \quad - \quad F = F_0\cos(\omega t) \end{aligned} \tag{4}$$

By using the terms listed in (4), the system of differential equilibrium equations (3) changes into system (5):

$$\begin{cases} m\ddot{x} + C\dot{x} + K_{T-eq}x + D\dot{\theta} + E\theta = F_X \\ m\ddot{y} + G\dot{y} + H\dot{\theta} + K_{A-eq}y + I\theta = F_Y \\ J\ddot{\theta} + Q\dot{\theta} + O\dot{x} + N\dot{y} + Mx + Ly + P\theta = -eF \end{cases} \tag{5}$$

The vibrating screen gives the best performances, in terms of selected inert material, if the conditions reported in (6) are satisfied:

$$\begin{cases} \theta = 0 \\ \dot{\theta} = 0 \\ \ddot{\theta} = 0 \end{cases} \tag{6}$$

The equations governing the screening dynamic behaviour are reported in (7).

$$\begin{cases} m\ddot{x} + C\dot{x} + K_{T-eq}x = F_X \\ m\ddot{y} + G\dot{y} + K_{A-eq}y = F_Y \\ O\dot{x} + N\dot{y} + Mx + Ly = -eF \end{cases} \tag{7}$$

In equations (7) the optimum parameters for the screening performance are considered: thus $\dot{\theta} = 0$ and the load eccentricity e=0. The problem has only two degrees of freedom, one in the x duration and one in the y direction. Equations (7) describe only the stationary regime behaviour of the screen. With these hypotheses linear system (7) reduces into linear system (8).

$$\begin{cases} m\ddot{x} + C\dot{x} + K_{T-eq}x = F_X \\ m\ddot{y} + G\dot{y} + K_{A-eq}y = F_Y \end{cases} \tag{8}$$

Equations in (8) are two second order ordinary differential equations that give the following solution for the dynamic behavior along *x* and *y* respectively (eq. (9)) [19-21]:

$$\begin{cases} x(t) = e^{-\frac{c_x}{c_{cx}}\omega_{nx}t}(C_1 \sin \omega_{Dx}t + C_2 \cos \omega_{Dx}t) + \\ + \dfrac{F_{0x}}{K_{T-eq}} \dfrac{1}{\sqrt{\left(1-\left(\dfrac{\omega_F}{\omega_{nx}}\right)^2\right)^2 + \left(2\dfrac{c_x}{c_{cx}}\dfrac{\omega_F}{\omega_{nx}}\right)^2}} \cdot \cos(\omega_F t + \varphi_x) \\ \\ y(t) = e^{-\frac{c_y}{c_{cy}}\omega_{ny}t}(C_3 \sin \omega_{Dy}t + C_4 \cos \omega_{Dy}t) + \\ + \dfrac{F_{0y}}{K_{A-eq}} \dfrac{1}{\sqrt{\left(1-\left(\dfrac{\omega_F}{\omega_{ny}}\right)^2\right)^2 + \left(2\dfrac{c_y}{c_{cy}}\dfrac{\omega_F}{\omega_{ny}}\right)^2}} \cdot \cos(\omega_F t + \varphi_y) \end{cases} \tag{9}$$

The values of the phase displacements are reported in (10) and (11):

$$\varphi_x = \arctan\left(\frac{2\xi_x\beta_x}{1-\beta_x^2}\right) \tag{10}$$

$$\varphi_y = \arctan\left(\frac{2\xi_y\beta_y}{1-\beta_y^2}\right) \tag{11}$$

The first term of the first equation in linear system (9) represents the transient response (the constants C_1 and C_2 depend on initial conditions). On the other hand this response tends to fade with time, while the second term of the first differential equation in (9) represents the steady-state response, which has the pulsation out of the phase φ_x. The same considerations also apply to the second differential equation in linear system (9).

Si introducono ora i seguenti parametri:

- ***Rapporto di smorzamento***

$$\xi = \frac{c}{c_c} \tag{12}$$

c = costante di smorzamento viscoso.

$c_c = 2\sqrt{km}$ = costante di smorzamento critico.

- ***Rapporto di frequenza***

$$\beta = \frac{\omega_F}{\omega_N} \tag{13}$$

ω_F = frequenza della forzante.

$\omega_N = \sqrt{K_{eq}/M}$ = frequenza naturale del sistema.

- ***Delta statico***

$$\delta_{ST} = \frac{F}{K_{eq}} \tag{14}$$

δ_{ST} = spostamento statico delle molle sottoposte ad una forza *F*.

- ***Pulsazione smorzata***

$$\omega_D = \omega_N \sqrt{1-\xi^2} \tag{15}$$

ω_D = rappresenta la frequenza delle pulsazioni smorzate.

Sostituendo i parametri appena introdotti nella (8) e nella (9) si ottengono:

$$\begin{cases} x(t) = e^{-\xi_x \omega_{nx} t}[C_1 \sin(\omega_{Dx} t) + C_2 \cos(\omega_{Dx} t)] + \\ + \dfrac{\delta_{ST-X}}{\sqrt{(1-\beta_x{}^2)^2 + (2\xi_x \beta_x)^2}} \cdot \cos(\omega_F t + \varphi_x) \\ \\ y(t) = e^{-\xi_y \omega_{ny} t}[C_3 \sin(\omega_{Dy} t) + C_4 \cos(\omega_{Dy} t)] + \\ + \dfrac{\delta_{ST-Y}}{\sqrt{(1-\beta_y{}^2)^2 + (2\xi_y \beta_y)^2}} \cdot \cos(\omega_F t + \varphi_y) \end{cases}$$

(16)

(17)

2.4.1 Risultati vaglio attuale

Di seguito verranno esposti i risultati del vaglio proposto dalla Bernardi Impianti per quanto riguarda le grandezze cinematiche e tutti i parametri funzionali.

In questa prima tabella (continua alla pagina seguente) sono riportati i dati principali riguardanti la dinamica del sistema. Nelle tabelle successive vengono esposti i risultati relativi alla dinamica orizzontale e verticale. Nell'ultima tabella sono esposti i risultati riguardanti il moto globale del vaglio ottenuti componendo i due moti orizzontale e verticale.

DATI PRINCIPALI		
Modello motovibratore:	MVSI 10/7000-S90	
Momento statico:	6.250	kg·mm
N° di giri motovibratori	1.000	g/1'
Forza totale:	130.720	N
Inclinazione della forzante:	27	gradi
Massa totale del vaglio "M":	1.962	kg
N° molle lato carico:	6	
N° molle lato scarico:	4	

Rigidezza assiale molla:	201.978	N/m
Rigidezza trasversale molla:	79.909	N/m
Loss Factor:	0,10	

Pulsazione della forzante:	104,7	rad/s
Forza totale verticale “Fy”:	122.137	N
Forza totale orizzontale “Fx”:	62.232	N
Rigidezza assiale totale "Ka-eq":	799.089	N/m
Rigidezza trasversale totale "Kt-eq":	2.019.784	N/m
Coefficiente di smorzamento "Cx-eq":	1.929	N·s/m
Coefficiente di smorzamento "Cy-eq":	763	N·s/m
“δst” vaglio spento:	24,0	mm

Tabella 1. *Dati principali del vaglio attuale.*

DINAMICA ORIZZONTALE		
Spostamento statico "δst-x":	0,031	m
Pulsazione naturale "ωx":	32,09	rad/s
Rapporto di frequenza "βx":	3,26	
Rapporto di smorzamento "ξx":	0,0153	
Spostamento max "X":	***0,0032***	*m*

Velocità max "Vx":	0,334	m/s
Accelerazione max "Ax":	***3,57***	***"g"***

Tabella 2. *Risultati della dinamica orizzontale.*

DINAMICA VERTICALE		
Spostamento statico "δst-y":	0,153	m
Pulsazione naturale "ωy":	20,18	rad/s
Rapporto di frequenza "βy":	5,19	
Rapporto di smorzamento "ξy":	0,0096	
Spostamento max "Y":	***0,0059***	*m*
Velocità max "Vy":	0,617	m/s
Accelerazione max "Ay":	***6,59***	***"g"***

Tabella 3. *Risultati della dinamica verticale.*

PARAMETRI FONDAMENTALI		
Accelerazione max totale "A":	***7,49***	***"g"***
Spostamento picco-picco "D":	***13,41***	***mm***
Forza totale trasmessa in dir. "x":	*6.447*	*N*
Forza totale trasmessa in dir. "y":	*4.711*	*N*

Forza totale trasmessa a terra:	*7.985*	*N*
Rapporto di "sollevamento":	***0,245***	

Tabella 4. *Risultati dinamica globale*

DATI PRINCIPALI		
Modello motovibratore:	MVSI 10/10000-S90	
Momento statico:	8.673	kg·mm
N° di giri motovibratori:	1.000	g/1’
Forza totale:	190.220	N
Inclinazione della forzante:	27	gradi
Massa totale del vaglio "M":	2.050	kg
N° molle lato carico:	6	
N° molle lato scarico:	4	
Rigidezza assiale molla:	201.978	N/m
Rigidezza trasversale molla:	79.909	N/m
Loss Factor:	0,10	

Pulsazione della forzante:	104,7	rad/s
Forza totale verticale “Fy”:	169.487	N
Forza totale orizzontale “Fx”:	86.358	N

Rigidezza assiale totale "Ka-eq":	799.089	N/m
Rigidezza trasversale totale "Kt-eq":	2.019.784	N/m
Coefficiente di smorzamento "Cx-eq":	1.929	N·s/m
Coefficiente di smorzamento "Cy-eq":	763	N·s/m
"δst" vaglio spento:	25,2	mm

Tabella 5. *Dati principali del vaglio modificato.*

DINAMICA ORIZZONTALE		
Spostamento statico "δst-x":	0,043	m
Pulsazione naturale "ωx":	31,39	rad/s
Rapporto di frequenza "βx":	3,34	
Rapporto di smorzamento "ξx":	0,0150	
Spostamento max "X":	***0,0042***	*m*
Velocità max "Vx":	0,442	m/s
Accelerazione max "Ax":	***4,72***	***"g"***

Tabella 6. *Risultati della dinamica orizzontale.*

DINAMICA VERTICALE		
Spostamento statico "δst-y":	0,212	m
Pulsazione naturale "ωy":	19,74	rad/s

Rapporto di frequenza "βy":	5,30	
Rapporto di smorzamento "ξy":	0,0094	
Spostamento max "Y":	***0,0078***	*m*
Velocità max "Vy":	0,819	m/s
Accelerazione max "Ay":	***8,74***	***"g"***

Tabella 7. *Risultati della dinamica verticale.*

PARAMETRI FONDAMENTALI		
Accelerazione max totale "A":	***9,90***	***"g"***
Spostamento picco-picco "D":	***17,72***	***mm***
Forza totale trasmessa in dir. "x":	*8.524*	*N*
Forza totale trasmessa in dir. "y":	*6.247*	*N*
Forza totale trasmessa a terra:	*10.568*	*N*
Rapporto di "sollevamento":	***0,311***	

Tabella 8. *Risultati dinamica globale*

4. The structural behavior of the screen

4.1 Traditional screen

4.2 Modified and improved screen

Claims

1. A vibration screen comprising a case-_shaped body (11) with parallel side walls (16) between which perforated screening surfaces (17) are disposed and which is suspended over a base (13) through spring means (12) and is provided with vibrating devices (14), **characterized in that** the side walls (16) are each made up at least partly of two plate sheets (19, 20) closely spaced and parallel to each other so that each of them forms one of the two faces of the wall and both delimit a hollow space (21) therebetween, the sheets being interconnected with each other by means of interconnecting elements (22, 23) disposed in the hollow space.

2. A vibration screen as claimed in claim 1, **characterized in that** the two sheets (19, 20) of a wall have different thickness values from each other.

3. A vibration screen as claimed in claim 2, **characterized in that** one sheet has a thickness included between 1.5 and 2.5 times the thickness of the other sheet and, in particular, about twice the thickness of the other sheet.

4. A vibration screen as claimed in claim 2, **characterized in that** the sheet of greater thickness is the outer one.

5. A vibration screen as claimed in claim 4, **characterized in that** the outer sheet has a thickness of approximately 6 mm and the other sheet of approximately 3 mm.

6. A vibration screen as claimed in claim 1, **characterized in that** the sheets are 2 and 8 mm thick.

7. A vibration screen as claimed in claim 1, **characterized in that** a connecting beam (15) supporting the vibrating devices (14) is disposed on the upper part between the two walls.

8. A vibration screen as claimed in claim 7, **characterized in that** the side walls have an upper intermediate bulge between the ends, close to which th

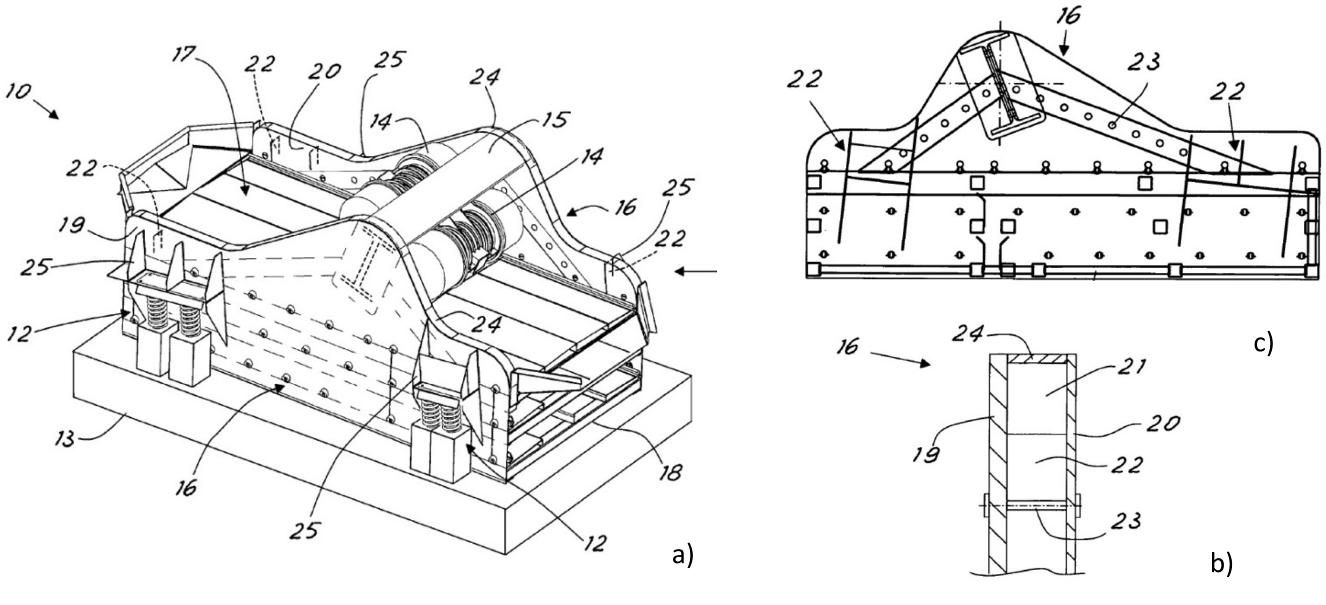


figure : a) the new modified vibrating screen, b) geometrical configuration of a side wall and c) detail of the reinforced side-wall made up at least of two plate sheets each of them forms one of the two faces of the wall and both delimit a hollow space

5. Experimental procedures

6. Analysis of the results

7. Conclusions

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