SPRING FOUNDATIONS OF MACHINES WITH ROTATING MASSES – AS AN APPLICATION THE TURBINE-GENERATOR SET

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SUMMARY

Design principles are presented for foundations mounted on springs. The main properties of steel and concrete foundations are compared. A short description about isolation elements are also given. Simple formulas are derived for the needs of a preliminary design. Three-dimensional analyses pointed out the complexity of a dynamic behaviour of low-tuned systems. The results obtained proved the effectiveness of artificial damping elements in controlling the amplitudes of vibration.

INTRODUCTION

Traditional machine foundations are bulky and rigid reinforced concrete foundations which provide high-tuned supports for the machine. The total weight of this kind of the foundation is typically 2 or 3 times greater than that of the machine. The massiveness of the foundation affect clearly the design of the structures and equipments of the building where the machine is. The operating frequencies of machines have increased so much that, in some cases, it may be difficult to construct a high-tuned massive foundation. Nowadays, a principal tendency is the use of a low-tuned foundation instead of a high-tuned one.

Low-tuning can be achieved by structural solutions like slender columns or piles. The mass of the foundation is clearly decreased compared to the high-tuned system. However, the foundation is a separate structure which is not connected to the other parts of the building. In addition, it may be difficult to realize a low-tuned system through structural modifications if the operating frequency is low or the foundation is directly resting on the soil. In designing of a low-tuned foundation an alternative is to mount the foundation on special spring and damping units. Recently, these dynamically isolated foundations have also been used for large machines.

The dynamic behaviour of an isolated light foundation is studied in this occasion. The purpose is to give background to a detailed design. The aim is to investigate the use of viscous dampers as a way to reduce the mass of the foundation, since a light steel foundation is an interesting alternative to affect the costs of the foundation and the whole building.

Static and temperature loads are not considered here, since they are typically not so problematic as the dynamic loads. However, the static stiffness of the foundation may be a problem affecting directly the alignment of the machine. This effect is not included in the study.

STEEL OR REINFORCED CONCRETE FOUNDATION

The main target is an economic and reliable foundation. So the price of the material is an important factor. A weight unit of constructed steel is about 15 times more expensive than constructed concrete. From the structural point of view an important difference between steel and concrete is that the stiffness of steel is about 7 times greater than that of compressed concrete. The difference is much greater in bent structures if it is assumed that only reinforcement bars resist tensile stresses. However, in practice vibration calculations are made assuming an uncracked concrete cross-section. Usually the dimensions of concrete foundations are so selected that this assumption is a realistic one. The mass density of steel is over three times larger than that of concrete. It is obvious that the steel structure has to be much lighter than the concrete one to be economical.

Steel has favourable properties. At the range of the linear theory its behaviour is practically time-independent and its Young's modulus is known exactly, in advance. So, the dynamic behaviour of the welded steel structure can be calculated accurately. The structural steel is also a though material the behaviour of which under a fatigue load is well-known. Steel is, however, sensitive to temperature changes, requires corrosion protection and vibrating steel structures are also noisy compared to concrete ones.

The properties of reinforced concrete are more random than those of steel. The calculations of concrete structures are usually made by using so-called design values, which can differ quite clearly from the actual values of the foundation. The actual values are determined afterwards from test samples. Concrete has also clear time-dependent properties like shrinkage and creep. To improve the dynamic properties of concrete special mixtures as fiber concrete are developed. These are more expensive than a traditional concrete. For example, the adding of steel fibers to concrete increases price per a weight unit about 20 %.

The material damping of concrete is larger than that of steel, but in turbine foundations the material damping is small in any case. For concrete the design value of the damping ratio is about 2 % and for steel less than 1 %. The values reflect the fact that the amplitudes of vibration are so small that the foundation behaves almost as an elastic material.

From the practical point of view, the steel foundation requires not so much space as the concrete one does. The space saved makes it easier to place equipments and pipes. The steel foundation acts also as an assembling frame of the turbine. In case of the concrete foundation a separate steel frame for assembling the machine is usually needed. As advantages of the steel foundation compared to the concrete one are usually mentioned shorter construction time, easiness to align a machine and the scrap value of a disassembled foundation /4/. On the other hand, concrete is the main building material in Finland and the use of steel decreases the number of available building contractors.

SPRING FOUNDATION

The idea of the spring foundation is to separate the machine and the foundation from the other parts of the building in a dynamic sense. The isolated system does not spread vibrations to the surroundings and does not receive them from the surroundings. In the case of the turbines, the dynamic analysis is restricted to the isolated system and the structures under the isolating elements are analysed for static loads. This simplifies the design since same structures are used to support the isolators and other parts of the building. It is more important, however, that the isolated foundation is not so sensitive for the movements or depressions of the support points. The changes in the vertical position of support points can be corrected during the normal operation by adding metal sheets between the foundation and isolators.

The spring foundation is particularly suitable for large turbines the axis of which is normally quite long, even dozens of meters. In such cases, it is practically impossible to guarantee that the long-time displacements of supports stay within an allowable range, but it must be prepared to correct the position of the supports during the life-time of the turbine. It is said that turbines, the power of which is greater than 200 MW, should always be implemented on isolators.

The isolated turbine-foundation system is a low-tuned one. Movements are normally kept within limits compatible with the proper operation of the machine by the mass of the foundation block below the machine. Isolators are placed under the block. Just below the machine, the structures of an isolated foundation are quite similar to the structures of a conventional foundation.

An interesting alternative to minimize vibrations is the use of artificial dampers. These are viscous type elements connected to the foundation. They produce frequency dependent damping forces to reduce the amplification of vibrations near the resonances. A disadvantage of damping is that it increases amplitudes of forces transmitted by isolation elements compared to the undamped vibration at frequencies higher than $\sqrt{2}$ times the resonance frequency. In the case of a low-tuned foundation, it means that the support reactions caused by the eigenmodes clearly below the excitation frequency increase due to the damping added. However, the contribution of the eigenmodes near the excitation frequencies is usually most important to total vibrations and the artificial damping is a valid method in decreasing a total effect of these eigenmodes.

Structures under isolators can often be slender and lighter than the corresponding structures of a conventional foundation. The costs of the isolators can often be paid by the save in these structures. This is especially true for frame foundations where the base mat is avoided. Construction costs should not be the only factor to consider if the isolators are used. Most economical benefits appear during the operating time of the machine.

Isolators have to have certain stiffness and damping properties. The physical difference between these two properties is so clear that separate elements are used in producing damping and stiffness to the system. Dampers are usually viscous type. The damping force is produced by some viscous fluid. With small displacements the dampers can behave hysteretic but they are always viscous if movements are large. There are various alternatives to make a stiffness element which is commonly called a spring. Rubber, air and steel springs are all well-known and widely used. Rubber is a suitable material if the amplitudes of dynamic forces are small and static loads are almost constant. The spring constant of the rubber spring depends on the magnitude of the static loads, and the rubber spring owes a clear non-linear behaviour if the amplitude of the dynamic load is large /6/. The temperature changes may also be a problem for rubber springs. Thus, the rubber spring is not a very attractive alternative for turbine foundations.

Air springs are usually rubber or fabric bellows which contain a column of compressed air. The bellow itself does not provide or support load. This is done by the column of air. The eigenfrequency of the air spring can be very low (0.5 hz) and the frequency is easily controlled by the pressure of air in the bellow. The height of the air spring is smaller than the height of the corresponding steel spring. A special feature of air springs is the need of the compressed air system where pressure is regulated by valves. The system itself is relatively easy to construct. In the factory building, compressed air is usually not a problem but anyway the pressurized system is an extra feature demanding maintenance. As far as the writer knows, air springs are not used in turbine foundations. Since 1930's they are, however, widely used in other applications /7/ and there should not be any restriction in using them with turbine foundations.

The springs of turbine foundations are normally out of helical steel. Its main advantage is that the behaviour of the spring is linear over a wide range of displacements and the properties of steel stay practically constant as a function of time. The most important matter is to use steel whose properties and quality are good enough for demanding conditions where springs are used. The lowest practical eigenfrequency of steel springs is usually 2-3 Hz. The frequency can be even smaller but then the height of the spring is quite large. In very low frequencies, air springs are more suitable than steel springs.

Isolators affect mainly the rigid body eigenmodes of the system on the isolators. The foundation itself must be analysed as carefully as the unisolated foundation, since higher bending modes may be problematic ones. It has to be checked that structures below isolators are rigid enough to guarantee a proper behaviour of isolators. In some cases, it is necessary to make a dynamic analysis for the whole system including the structures below isolators.

DYNAMIC ANALYSIS OF A SPRING FOUNDATION

The analysis of isolated foundations is quite similar to that of unisolated ones. There are few matters which might be problematic. What is a proper ratio between the stiffness of springs and the bending stiffness of the structure on the springs? The designer has to determine the location of springs and also the span between them. In what kind of situations are dampers necessary? The method to analyse the system is also a decision which affect both results and costs. These are problems which arise during the design of an isolated turbine foundation.

The proper stiffness of springs is simple to determine as a function of excitation frequencies. A general advice for the structure on the springs is that it is stiff enough, but the scale to measure the stiffness is not given. Same feature is typical for all the rules concerning the design of machine foundations.

The different parts of the vibrating system are coupled together and the problem is to select an appropriate relative stiffnesses for these parts. Another basic question is the role of mass. Mass is used in two purposes. On the one hand, the adding of mass decreases eigenfrequencies and affect the tuning of the system. On the other hand, mass decreases the amplitudes of vibrations in a low-tuned system. In the case of turbine foundations mass is not used in tuning but it is still used to control the amplitudes of vibrations.

In the study, these problems are approached by investigating a possible frame type foundation of a standard gas turbine-generator. The frame is made of structural steel and it is supported by springs. The frame has two longitudinal beams with equal mass distribution. The total mass of the machine is 208.8 tons. The foundation is 21 m long and 4 m wide. The distribution of the machine mass as a function of the length is represented in Figure 1.

The foundation was first modelled as a beam on springs. This model was used to investigate the influence of the difference between the bending stiffness of the beam and the stiffness of the springs. Finally, the frame was described by a complete model. Using this model, the effect of dampers on the amplitudes of vibrations was especially studied. The frame is a quite light one. Its mass is about 25 % of the mass of the machine.

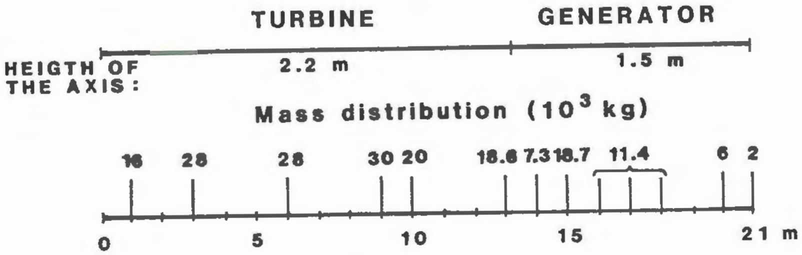

Figure 1. The mass distribution of the turbine-generator studied.

The machine was not modelled in the analysis, but its effects were added to the calculation model. The mass of the machine was described by discrete masses and discrete mass moments of inertia. The dynamic loads were given at the supporting points of the machine. The rigidity of the machine itself and its casing was not included in the analysis. These may have a stiffening effect on the behaviour of the foundation.

In the preliminary design, it is useful to know the effect of springs to the eigenfrequencies of the beam. This can be evaluated by the energy method. Let us assume that the beam is continuously supported by springs, and the support reaction, R(x), in every point is described by an equation:

$$R(x)=k(x)w(x) \tag{1}$$

where k is a spring constant, w the deflection of the beam and x the coordinate of the beam axis. The maximum potential energy, U_{max}, of the beam-foundation system is

$$U_{max}=\tfrac{1}{2}\int_0^l EI(x)(w''(x))^2dx+\tfrac{1}{2}\int_0^l k(x)w(x)^2dx \tag{2}$$

where EI is bending stiffness and l the length of the beam. The maximum kinetic energy of the system, V_{max}, can be written as follows:

$$V_{max}=\tfrac{1}{2}\omega^2\int_0^l m(x)w(x)^2dx \qquad (3)$$

The angular velocity of the vibration is marked by ω and m is the mass of the beam per unit length.

In a conservative system the sum of potential and kinetic energies is constant. Thus, we may write

$$V_{max}=T_{max} \qquad (4)$$

and by substituting Equations (2) and (3) into Equation (4) the following expression for the natural frequency of the system is obtained

$$\omega^2=\frac{\int_0^l EI(x)(w''(x))^2dx+\int_0^l k(x)w^2(x)dx}{\int_0^l m(x)w^2(x)dx} \qquad (5)$$

Assuming that the mass and the both of the stiffnesses are constants it may be written

$$\omega^2=\frac{EI\int_0^l (w''(x))^2dx}{m\int_0^l w^2(x)dx}+\frac{k}{m} \qquad (6)$$

In Equation (6) the last term presents the contribution of the spring foundation to the eigenfrequency. The first term gives the bending frequency of the beam itself. In respect of bending vibrations, the beam and the foundation form a system with a parallel coupling. The foundation adds the eigenfrequencies by a constant value, which does not depend on the serial number of the analysed eigenfrequency. This is true if we have a continuous foundation. In practice, the foundation consists of discrete springs and it is possible to have eigenmodes in which the springs do not obtain displacements at all.

Based on Equation (6) some general conclusions can be drawn for the needs of a practical design. The lowest eigenfrequency of the system should be controlled by the stiffness of the foundation. If the beam is considerable stiffer than the foundation, the bending frequencies of the system are determined by the stiffness of the beam and the lowest eigen-frequencies are pure rigid-body modes. This is the situation typical for the spring isolated machine foundations. If the first term is smaller than the second one in Equation (6) the lowest eigenmodes are not rigid-body modes. In this case the stiffness of the beam has a clear effect to the lowest eigenmodes. This is a situation which should be avoided.

The location of the springs is optimal if the springs carry equal parts of the static load. In addition to the alignment of the machine, the load distribution affects also the dynamic behaviour of the system. The properties of springs may vary as a function of loading. This is particularly true for rubber springs. The load distribution affects the rigid-body modes of the system. If the load distribution is even then the translation and the rotation are uncoupled. Otherwise, these two movements occur simultaneously in the rigid-body modes. During the operation of the machine, the behaviour of the system is easier to interpret and improve if the lowest modes are as simple as possible.

To illustrate the use of the equations derived, the described gas-turbine foundation was analysed by a simple beam model. The distribution of machine mass was as given in Figure 1. The model used was a beam on 8 springs and the length of one element was 1 m. So the model consisted of 21 elements altogether. The rotational inertia of the machine mass was included. Otherwise the analyses were made by standard routines of the program MSC/NASTRAN. Eigenmodes and eigenfrequencies were calculated by varying the stiffness ratio between the foundation and the springs. The number of springs were also altered without changing the total stiffness of the spring system.

The bending stiffness of the beam were evaluated by an equation

$$k_b = 4.73^4 EI/l^3 \tag{7}$$

and the stiffness of the spring foundation was the number of springs times the spring constant of a spring. All the springs had the same spring constant. Equation (7) corresponds to the stiffness of a free beam in its first bending mode /1/. The stiffness ratio was defined as a ratio of the bending stiffness of the beam to the stiffness of the spring foundation.

Calculations were made by four stiffness ratios, which were approximately 50, 5, 0.05 and 0.0005. The three first modes obtained in each case are given in Figure 2. The results prove that if the springs are soft compared to the beam stiffness the first two modes consist of pure rigid-body movements. Slight bending can already be seen with a stiffness ratio 5 in the first two modes. If the ratio is 0.05 the rigid-body modes vanish and all the springs act more or less as supports for the beam. The phenomenon is very clear with the smallest ratio in which case the span between the springs determine the lowest eigenfrequency.

In practical design, it may arise a question: should a spring system with a certain stiffness be constructed by using a small number of stiff springs or large number of soft springs. The

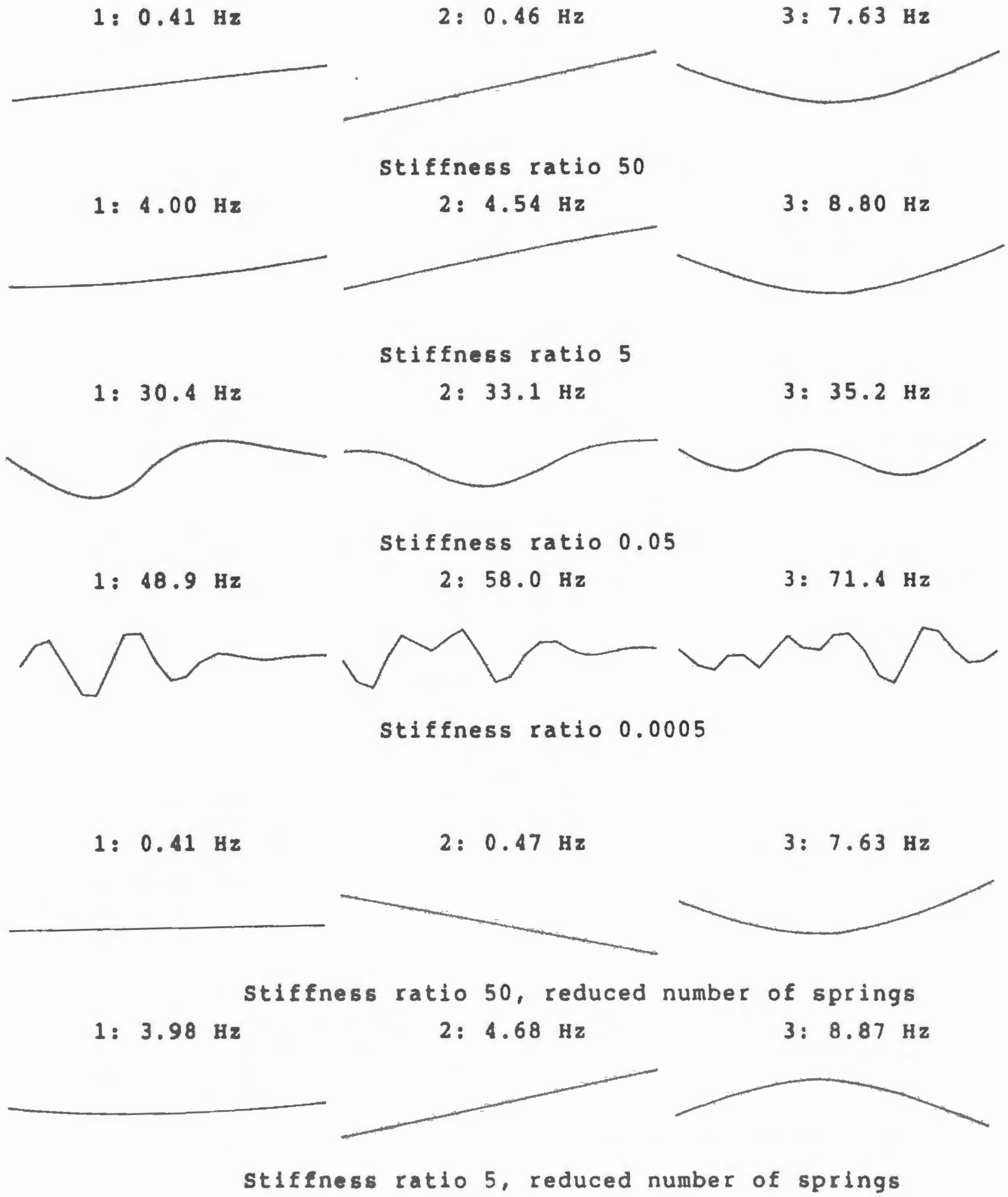

Figure 2. Eigenmodes of the beam model with different ratios of the bending stiffness of the beam to the vertical stiffness of the spring system.

number of the springs does not significantly affect eigenmodes if the spring system is softer than the upper structure. For stiffness ratios of 50 and 5 this is also illustrated in Figure 2. From the economical point of view, it is useful to minimize the number of springs, since it reduces assembling and maintenance costs. The reduced number of the springs was 5 in Figure 2.

The first eigenmodes of a well designed spring foundation can be determined by simple methods during a preliminary design. The frequency of the first bending mode of the beam model is a little difficult to determine exactly by simple methods if the system has eccentric masses producing a large mass moments of inertia. The effect of eccentricity is studied more detailed in Reference /2/. Applying Equation (7) together with Eq. (6) the obtained estimate for the frequency of the first bending mode relating to the stiffness ratio 5 was 9.96 Hz which is about 12 % higher than the frequency given in Figure 2. The difference is mainly caused by the mass moment of inertia. In the calculations, the value of EI was 7308 MNm^2 and the total mass 121 tons.

It is often reasonable to make a preliminary design by simple calculation models. However, the foundation is a three dimensional structure and its springs act in all the directions of coordinate axes. The amplitudes of vibration should be determined by using complete models without any artificial boundary conditions. To illustrate a three-dimensional effect, Figures 3 and 4 present the lowest three-dimensional eigenmodes of the frame structure which was analysed as a beam previously. The frame was supported by 10 springs which were equally distributed below the both longitudinal beams. Now, the stiffness ratio defined earlier was 5 in the vertical plane and 0.95 in the xy-plane. The stiffness of the springs was same in all the directions. According to the results the foundation fulfills the condition that the rigid-body modes are the lowest ones. Figures 3 and 4 reveal also the complexity of the dynamic behaviour indicating the necessity of the complete dynamic analysis for a low-tuned system.

In calculating the amplitudes of vibrations the dynamic load was determined by assuming a certain eccentricity for rotating masses. The eccentricity used was

$$e=2.5/\omega \text{ (mm)} \tag{8}$$

where ω is the angular velocity of a rotating mass in rad/s /3/. The load acted on the axis-level of the rotating mass. So, the vertical load produced mainly vertical displacements but the horizontal load, in addition to horizontal displacements, caused also vertical displacements. The frequencies of the turbine and the generator were 85 Hz and 50 Hz, respectively.

The calculation model was similar to that used in computing the results of Figures 3 and 4. To the model, artificial viscous dampers were added to the nodes where spring elements were. So the number of dampers was 10 in general. However, some analyses were made without dampers and with a reduced number of them, namely 6. The dampers had same damping constant in every direction and the ratio between damping and spring constants was 10 s. The amplitudes were also determined for the frame in which the stiffness ratio was 1.1 in both the vertical plane and the xy-plane. This frame is called slender frame in Tables 1,2,3 and 4 in which the results obtained are represented.

The displacements produced by the vertical excitation are acceptable, generally also without the dampers, as showed in Tables 1 and 2. There is a clear resonance near 51 Hz the effect of which is effectively wiped out by the dampers.

Since the horizontal excitation causes both rotation around a

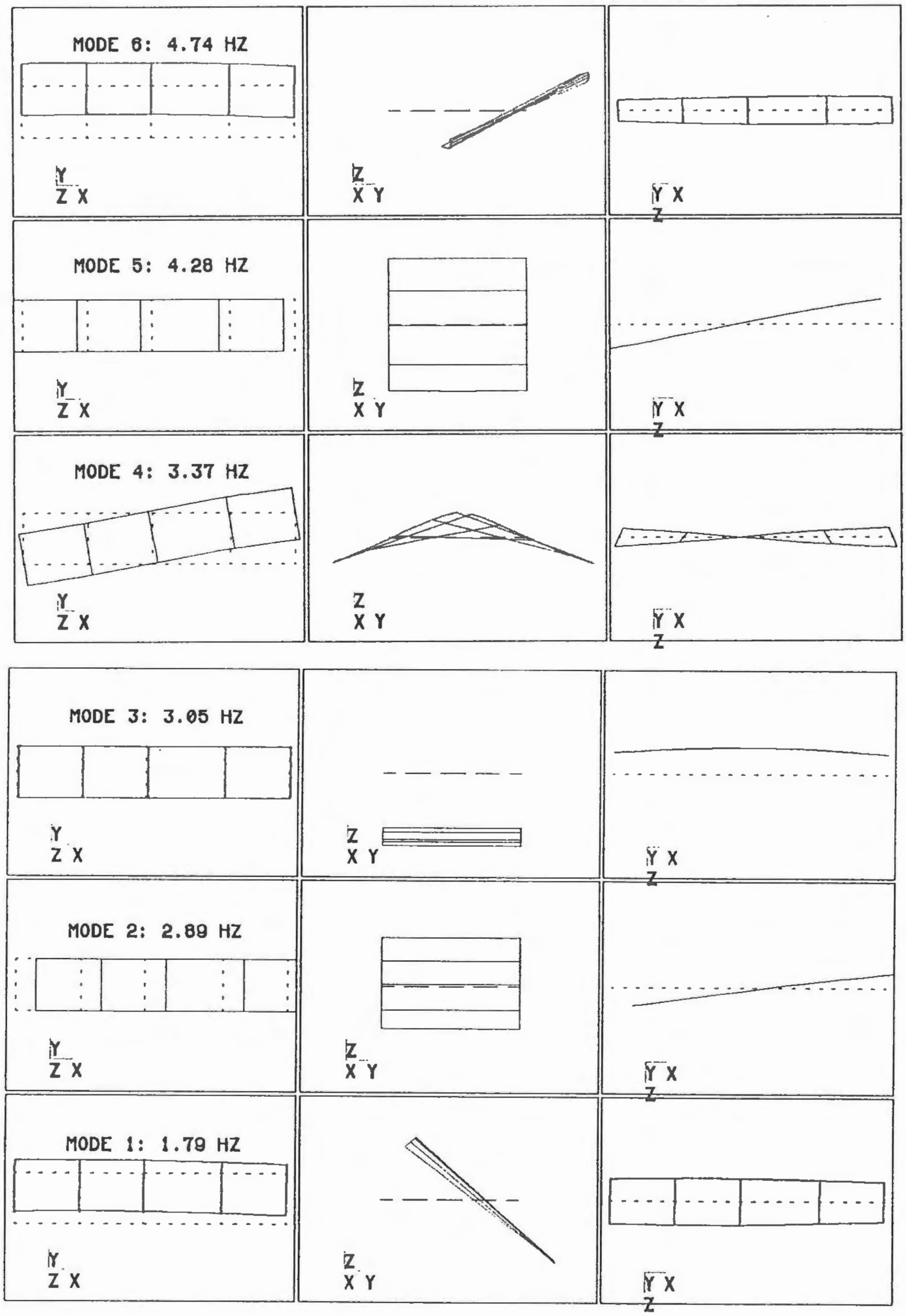

Figure 3. The first six three dimensional eigenmodes of the frame foundation analysed.

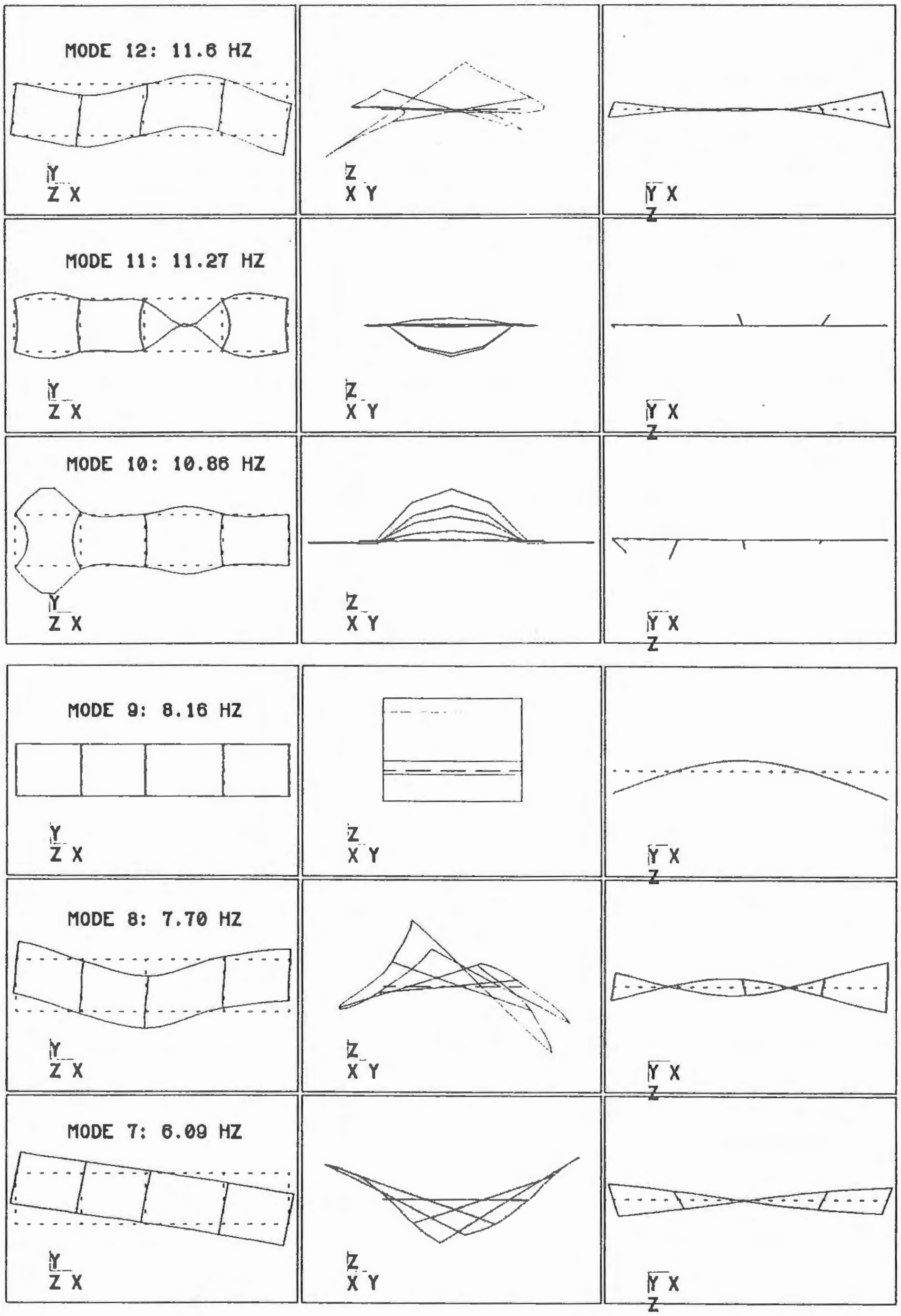

Figure 4. The three dimensional eigenmodes with ordinal numbers 7-12 of the frame foundation analysed.

Table 1. The maximum amplitudes of vertical displacements caused by the vertical excitation in the generator.

VERTICAL LOADING CAUSED BY THE GENERATOR			
	MAXIMUM VERTICAL DISPLACEMENT AMPLITUDE		
EXCITATION FREQUENCY Hz	WITHOUT DAMPERS 0.001 mm	WITH DAMPERS 0.001 mm	
		only vertical dampers	slender frame
49	6.69	2.03	4.14
49.5	5.50	2.04	3.98
50	5.58	2.80	3.84
50.5	7.67	5.08	3.72
51	25.1	3.97	3.63

Table 2. The maximum amplitudes of vertical displacements caused by the vertical excitation in the turbine.

VERTICAL LOADING CAUSED BY THE TURBINE			
	MAXIMUM VERTICAL DISPLACEMENT AMPLITUDE		
EXCITATION FREQUENCY Hz	WITHOUT DAMPERS 0.001 mm	WITH DAMPERS 0.001 mm	
		only vertical dampers	slender frame
84	1.03	0.54	3.00
84.5	1.04	0.55	2.82
85	1.05	0.65	2.66
85.5	1.06	0.66	2.51
86	1.05	0.78	2.30

Table 3. The maximum amplitudes of horizontal displacements caused by the horizontal excitation in the generator.

HORIZONTAL LOADING CAUSED BY THE GENERATOR				
EXCITATION FREQUENCY Hz	MAXIMUM HORIZONTAL DISPLACEMENT AMPLITUDE			
	ONLY VERTICAL DAMPERS 0.001 mm	DAMPERS IN EVERY DIRECTION 0.001 mm		
		6 dampers	10 dampers	10 dampers slender frame
49	50.0	69.8	4.21	3.26
49.5	92.9	52.6	4.62	3.26
50	267.	38.0	4.43	3.26
50.5	520.	20.3	4.65	3.26
51	149.	26.7	4.57	3.26

Table 4. The maximum amplitudes of horizontal displacements caused by the horizontal excitation in the turbine.

HORIZONTAL LOADING CAUSED BY THE TURBINE				
EXCITATION FREQUENCY Hz	MAXIMUM HORIZONTAL DISPLACEMENT AMPLITUDE			
	ONLY VERTICAL DAMPERS 0.001 mm	DAMPERS IN EVERY DIRECTION 0.001 mm		
		6 dampers	10 dampers	10 dampers slender frame
84	12.8	8.10	4.62	2.91
84.5	13.5	8.01	4.70	2.92
85	14.3	7.90	4.79	2.94
85.5	15.1	7.78	4.88	2.95
86	16.1	7.66	4.97	2.97

longitudinal axis and translation, it is normally much more problematic than the vertical load. In this case this is also true as can be seen in Tables 3 and 4. The displacement amplitudes are large without dampers. So if the foundation is a light and low-tuned one the dampers are highly recommendable. The amplitudes can be effectively decreased by adding the damping to the system. The damping increases modal amplitudes if the ratio of the excitation frequency to the eigenfrequency of the mode is larger than $\sqrt{2}$. However, the disadvantage is less important compared to the fact that the damping decreases the amplitudes of the modes near excitation frequencies. If the excitation frequency is higher than 20 Hz it is often difficult to avoid near resonance situations with higher eigenmodes and difficulties may arise with the higher harmonics of the excitation frequency. The results prove that the slender foundation behaves well and the stiffness ratio about 1 seems to be large enough for dynamic loads.

5 CONCLUSIONS

A low-tuned machine foundation is usually an economic alternative. The tuning can be made by standard solutions as slender columns or by special isolation elements. An isolated system is clearly restricted and can be analysed more accurately than a monolithic one.

Both steel and concrete are proper materials for the foundation. In general, risks are smaller with the concrete foundation because of the great mass of the foundation. A massive foundation has a greater damping and it is not so sensitive for temperature changes than a light one. Temperature loading, especially fire, demands extra care in steel foundations.

Stiffnesses of a foundation and springs, should be selected so that rigid-body modes and deflection modes are not coupled. In a low-tuned system resonance cannot be avoided. It may occur in operating speeds of the machine or during speeding up or slowin viscous dampers proved to be a valid method.

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