DELIBERATIONS ON FOUNDATION DESIGN METHODS FOR ROTATING MACHINERY.

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Abstract –

For on-shore applications, the support structure for a rotating machine consists of a block foundation and base-frame (base plate) normally known as skid. Adequate dynamic analysis of support structures for rotating equipment is necessary to safeguard other machinery in the vicinity of the subject rotating machine, as well as ensure good conditions for the operation of the supported machine.

The present method tries to achieve an under-tuned support structure which considers the machine mass and block foundation mass as one entity. The design procedure assumes that the assume that all vibrations are transferred to machinery foundation irrespective to the type of machine. This paper endeavours to highlight that how and why current design procedures for support structures of rotating machinery should consider the stiffness ratio of the bearing support (casing) and the rotor - bearing system, With the advent of foundation damping methods, the size of foundation can be lowered with a correct deliberation on subject.

Introduction

Heavy duty rotating machinery train in on shore application with high speed rotating masses requires a support system that can tackle dynamic forces minimize vibrations and transfer minimum vibration to surrounding soil. Excessive vibrations are detrimental to machinery, its support system, and pose hazard for operating personnel. The present practice of design of rotating machines includes gas turbines, steam turbines, turbo-expanders, compressors, fans, electric motors, and high energy centrifugal pumps etc. Normally block type foundations are used to support this type of machinery for on shore applications. In block-type foundation, dynamic machines are preferably located close to grade to minimize the elevation difference between the machine dynamic forces and the centre of gravity of the machine-foundation system. Block foundations are generally designed as rigid structures with pedestal and thick footing or mat. The loads on machine foundations are static and dynamic. Static loads are principally a function of the weights of the machine and all its auxiliary equipment along the weight of foundation itself. Dynamic loads, which occur during the operation of the machine, result from forces generated by unbalance, inertia of moving parts, or both, and by the flow of fluid and gases for some machines. The magnitude of these dynamic loads primarily depends upon the machine’s operating speed and the type, size, weight, and arrangement (position) of moving parts within the casing. The basic goal in the design of a machine foundation is to limit its motion to amplitudes so subject machine ensures satisfactory operation of the machine and other machines in immediate vicinity the dynamic response of a rigid block foundation depends on the transmitted dynamic load from rotating machinery, foundation's mass, it's dimensions, and soil characteristics.

Present practice of foundation design: civil & structural engineering perspective

Machinery Foundations are subjected to free damped vibrations and forced vibrations. Forced vibrations occur with continuous external forces on machine foundation. When a damped system is subjected to external dynamic force arising from the operation of machine F (t), the equation of motion can be written as-

$$M(d^2 z/ dt^2) + C (dz/dt) + Kz = F(t)m$$

Where M is mass, C is damping and k is stiffness of foundation. z is the displacement of foundation at particular direction. Normally machine foundation has 6 degree of freedom (known as rigid body modes) and above equations need to be solved separately for all rigid body modes.

Accordingly, we can say that for successful operation two aspects become critical for foundation design: 1) The machine itself should run smoothly (round the clock); 2) The foundation supporting the equipment is capable of sustaining the various loads coming from the machine under operation (dynamic loads) as well as those that could develop during transient operations thermal bows, electrical faults, short circuits etc. The analysis is hence considered a very complex problem because of the interaction of the structure, the subsurface soil, and the vibrating machine. Two different models for machine mass location are considered. First one by applying the seismic force at the machine anchorage locations, and the second model by applying the seismic force at the centre of gravity of the machine.

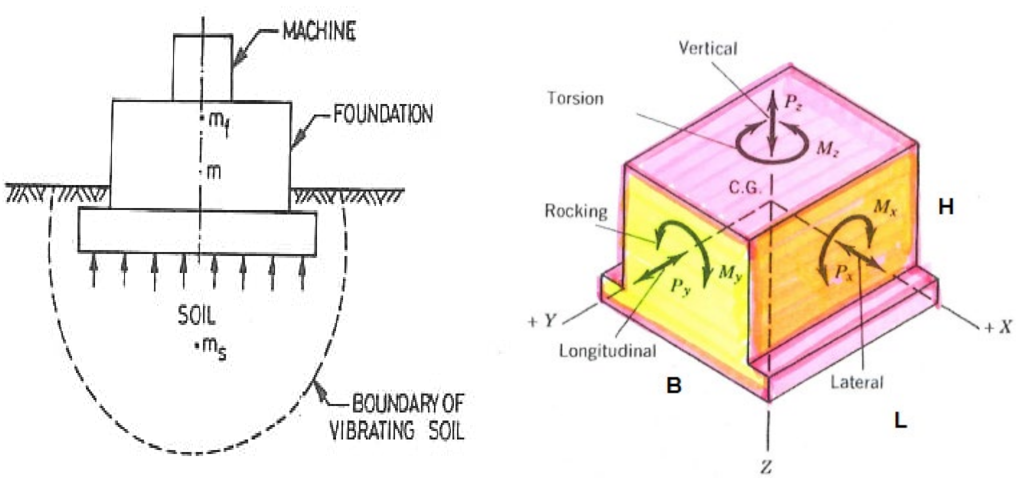

Fig 1 Block type Foundation layout and rigid body modes

The dynamic forces those are produced by the operating machine are transferred to the machine foundation. The major contributor to the dynamic forces is unbalance forces during machinery operation caused by residual unbalance (eccentricity of mass distribution) of rotor. Other limiting dynamic criterion is to avoid event of resonance and excessive transmissibility to the supporting soil or structure. If there is excessive vibration transmitted to adjoining soil by machinery foundation, then soil also vibrates randomly and continuous compacting takes place. This can cause differential settlement of foundation and eventual machinery failure.

To avoid resonant conditions, tuning of foundation based on operating frequencies is required. The value of r (frequency ratio) is considered based on required separation margin i.e. resonant frequency of foundation and affected soil as shown in figure2. Figure 2 below shows the effect of amplification of high tuned and low tuned foundation. It is clear from figure 2 that low tuned foundation should be preferred for high speed machines for better attenuation of resonance.

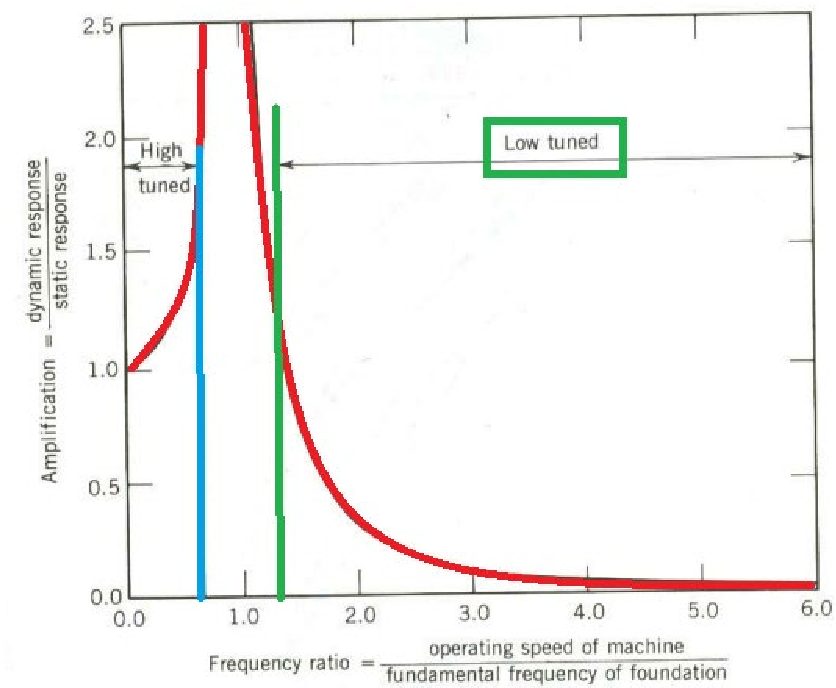

Fig 2 tuning of foundation

Above evaluation becomes more complicated of equipment operating with variable speed drives. For that reason, the natural frequency of foundation should be kept away from complete operating frequency band.

Another key ingredient to a successful design is the careful engineering analysis of the soil-foundation response to dynamic loads from the machine operation. The foundation's response to dynamic loads can be significantly influenced by the soil on which it is constructed.

Effect of Embedment of Foundation

All machine foundations are invariably embedded partly in to the ground. There are two different school of thoughts - a. effect of embedment causes increase in natural frequency and b. it causes reduction in amplitudes. By and large, it has been generally agreed that embedment tends to reduce the dynamic amplitudes. The reduction in the amplitudes could either be on account of change in stiffness, change in damping, change in soil mass participation or their combination.

For the soil to behave as an elastic material, it is necessary that the total pressure (static + dynamic) exerted by the foundation on the soil remains within elastic limits. A reasonable margin therefore should be kept while assigning bearing capacity to the soil intended to be used for machine foundation application. The dynamic pressure produced by machines not only' affects the foundation directly under the machine but to other foundations too, which are away from machine as the energy gets transmitted through soil in all directions. It is therefore desirable to keep intended margins for even static foundations i.e. foundations for static equipment including foundations of the building housing machines etc.PIP recommendation states bearing pressure should be 75% for combined static and dynamic loads which means 50% for static case only.

Static bearing capacity: proportion of footing area for 50% of allowable soil pressure, which means that the actual soil pressure should be less than 50% of static bearing capacity. The actual soil pressure equals to the weight of machine and foundation divided by the base area of footing as shown -

Actual soil pressure = $W_{m/c} + W_{found.}$ / Area of foundation

However, practised guidelines for permissible soil pressures for machine foundations and buildings/structures housing machine foundations are:

~ For low rpm machines, no reduction of soil stress is needed i.e. one can go up to 100% of the bearing capacity.

~ For medium rpm machines, reduction factor should be 10% i.e. permissible bearing pressure should be limited to 90% of the allowable bearing capacity.

~ For high rpm machineries, reduction factor should be 20% i.e. permissible bearing pressure should be limited to 80% of the allowable bearing capacity.

To determine soil property, for a construction site a site, dynamic soil investigation is carried out using wave propagation test or by vertical vibration resonance test, cyclic plate load test. The details of these tests are not the base scope of this paper.

The amplitude of vibration of a machine at its operating frequency is the most important parameter to be determined in designing a machine foundation, in addition to the natural frequency of a machine foundation soil system.

During design of block type foundation following loads and parameters are considered -

Gravity (dead) loads due to the weight of the machine, auxiliary equipment, pipe, valves, and deadweight of the foundation structure are to be considered in design. However, it is common to assume the machine frame is rigid and that its weight is uniformly distributed between support points

Static operating loads include the weight of gas or liquid in the machinery equipment during normal operation and forces, such as the drive torque developed by machines at the connection between the drive mechanism and driven machinery. The torque load is generally resolved into a vertical force couple by dividing it by the centre-to-centre distance between longitudinal soleplates or anchor point. A wider sole plate on a levelled foundation gives a better stability. Start-up torque and emergency drive torque such as short circuit torque, phase fault, auto-start obtained from the motor manufacturer.

Dynamic loads due to unbalanced masses-Dynamic forces transmitted to the bearing pedestals and then to casing under the following conditions— a) Under design unbalance levels over operating speed range; b) At highest vibration when passing through critical speeds; at or near a critical speed, the force from rotating unbalance is highly amplified. c) At a vibration level where the machine is just short of tripping on high vibration; and d) Under the maximum level of upset condition / transient condition such as passing through critical speed the machine is designed to survive. The forcing function F(t) equation shown below is a generic function to determine the dynamic force applied on the foundation at any operating frequency **Ω** corresponding to each of the rotating machine.

$$F(t) = M_i .(G.\Omega^2 / \omega) \times \sin (\Omega t+a_i)$$

Where M_i is mass of rating part i , G is balance quality, G = e.ω , ω is designed rotation speed in radian / sec , Ω is rotational speed of machine for which unbalance force is calculated.

Most of the rotors are balanced to an initial balance quality either in accordance with the. ISO 1940 - balance quality of rigid rotors in terms of a constant emωo. The normal balance quality Q for parts of process-plant machinery is 6.3 mm/s-(Balance quality 6.3). The rotor intended for higher speeds are balanced to balance quality grade G2.5 or less as per ISO 1940 table.

Machine unbalance equivalent to trip vibration level and effective bearing stiffness should be considered. Since a rotor is often set to trip on high vibration, it can be expected to operate continuously at any vibration level just below the trip limit and hence foundation must withstand this type of abnormal situation which may occur.

Transients - emergency torque due to entire short – circuit faults which can occur. A line to line short circuit at the driver terminals causes the most severe loading of foundation. Load due to bowed rotor-A bowed rotor can impose large dynamic forces on the machinery foundation. This is very difficult to quantify during foundation design stage.

Loads from multiple rotating machines—if a foundation supports multiple rotating machines, such as speed increaser or reducer gear drives, the foundation designer computes unbalanced force based on the mass, unbalance, and operating speed of each rotating component. The response to each rotating mass is then combined to determine the total response-While we state foundation as rigid, that means elastic deformation of thick foundation shall be negligible. But the soil on which machinery block foundation is placed with some embedment is not so rigid. Hence elastic properties of soil -foundation interface must be addressed.

During dynamic analysis of machine foundation, following design parameters are taken for soil and machinery operating conditions -

Soil – These include soil density, Poisson ratio, shear wave velocity / dynamic shear modulus, coefficient of friction for sliding, damping ratio, allowable soil bearing capacity.

Machinery operating and design parameters- These include Machine and Foundation Dimensions, Operating Frequency of Rotating Parts, Coordinates of CGs, Coordinates for Point of Application of Forces, Eccentricity of Unbalanced Forces for Rotating Parts, Critical Speeds, Rotor Weights, complete machinery weight

Foundation- These include grout type and extent in plan, levels, base plate and anchor bolt details / setting details with uprooting force.

As a rule of thumb, dynamic analysis of foundation is carried out for equipment heavier than 22kN with frequency checks, amplitude and velocity checks are carried out with time-based plots.

Torsional and rocking modes are controlled by adequately sizing length, breadth and height of foundation. Typically, stability checks are done with static loads

Methods of Shear, compression waves absorption / dampening

A foundation on absorbers is usually made of two parts: a lower slab or a sole plate on which the absorbers are placed and an upper foundation block resting on the absorbers. The machine is anchored to the upper foundation block. A schematic sketch of a machine foundation on absorbers is shown in Fig 8. V_c is compression wave velocity and V_s is shear wave velocity.

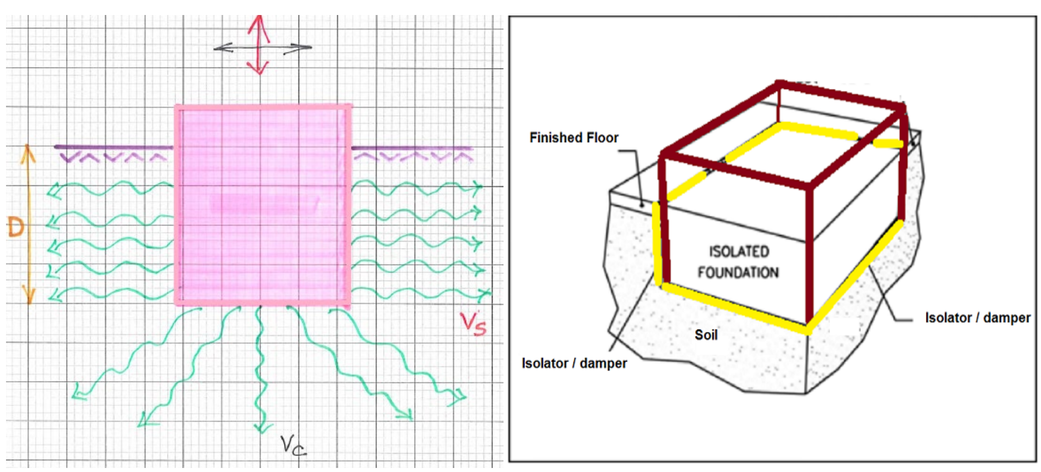

Fig 3 -Shear & compression waves and dampening devices (shown in yellow)

Materials capable of undergoing elastic deformation can be used as vibration absorbers. Commonly used vibration absorbers are; steel or metal springs, cork pads, rubber pads, timber pads, neoprene pads and pneumatic absorbers. However, it is important to consider that waves should not get reflected back to origin i.e. machinery foundation.

A correct isolation between machine foundation and the surrounding area will result in trouble free operation to machine and neighbouring machines, negligible possible compacting during operation.

The isolation / damping material absorbs machine-induced energy, limits the transmission of higher frequency disturbances and provides isolation from ambient and induced shock and vibration

A foundation block or Vibration isolation mountings for a high dynamic machine is required in order to reduce the transmission of vibration and shock to nearby machines / building structures. To control the source of vibration disturbance through the use of resilient insulating materials is known as active vibration isolation.

When it is not possible to prevent or sufficiently lower the transmission of shock and vibration from the source, a resiliently supported vibration insulating foundation block can be used for the passive vibration isolation of critical equipment in vicinity.

Open trenches and (gas cushions type panels) are the only wave barriers that have been extensively studied and used since long time. However, many designers have successfully tested elastomer-based isolation / damping devices.

A seismic isolation pad produced by utilizing the scrap tire rubber which contains interleaved steel reinforcing cords has been proposed. The steel cords are expected to function similar to the steel plates used in conventional laminated rubber bearings.

A new type of seismic isolation system has been introduced using the scrap tire rubber as easily available material at low or negligible cost. It can be found that the research on scrap tire pad has been initiated by Turer et al. In the previous studies, the researchers produced the specimen samples and tested in vertical compression and in horizontal shear. These bonded samples would be produced by stacking one on top of another with the application of adhesive. The car tire usually contents synthetic fibre as reinforcing cords and are very stable in such kind of forces.

Observations on current methodology of design -Machinery dynamics perspective

Following are major observations in present theory of foundation design –

The theory does not identify the type of rotors and bearings being used in rotor bearing system of subject rotating machine. It assumes that all vibratory forces generated during machine operation are completely transferred to machine foundation. This can only happen when rolling element bearings are used where the structural damping due to metal to metal contact is very less. Machinery having rigid rotors uses vibration measurement at bearing cap which is a part of casing because vibration magnitudes are indicative of excitation forces generated when rolling element bearings are used. Medium energy pumps, Fans and blowers, LV electric motors and aero-derivative gas turbines measure vibration on bearing cap / housing in terms of mm/ sec RMS as per ISO and API guidelines.

Machinery having flexible rotors (such as high energy pumps, centrifugal compressors), vibration measured only on bearing cap is not be good indication of machinery health as flexible rotor may experience high displacement while due to fluid film damping and stiffness of bearing pedestal, vibration measured at bearing cap is quite low. As experimental verification, shaft displacement measured (overall vibration up with FFT range of 1000 Hz) on an operating centrifugal compressor as 25 micron pk pk (meeting API 617 criteria) showed 0.3 mm/sec rms as bearing housing vibration. For easy understanding, we considered this phenomenon occurring at a single frequency (1 X only) and converted displacement into velocity mm/sec rms using Sinusoidal Waveform Vibration Calculator. This showed the vibration at bearing cap as 9.2 mm/ sec rms, which may be termed as faulty machine as per table A.1 of ISO 10816-3.

Frequency					166.666 Hz 10000.0 cpm
Displacement	0.348	mil rms	8.839	µm rms	
	0.492	mil pk	12.500	µm pk	@ 0.0 °
	0.984	mil pk-pk	25.000	µm pk-pk	
Velocity	0.364	ips rms	9.256	mm/s rms	
	0.515	ips pk	13.090	mm/s pk	@ 270.0 °
	1.031	ips pk-pk	26.180	mm/s pk-pk	

Fig 4 a – snap shot of vibration conversion calculator. Displacement to velocity

When we converted 0.3 mm/sec RMS to displacement assuming same "one" frequency of interest, we found equivalent displacement is quite low as shown below –

Frequency					166.666 Hz 10000.0 cpm
Displacement	0.011	mil rms	0.286	µm rms	
	0.016	mil pk	0.405	µm pk	@ 0.0 °
	0.032	mil pk-pk	0.810	µm pk-pk	
Velocity	0.012	ips rms	0.300	mm/s rms	
	0.017	ips pk	0.424	mm/s pk	@ 270.0 °
	0.033	ips pk-pk	0.849	mm/s pk-pk	

Fig 4 b – snap shot of vibration conversion calculator. velocity to Displacement

This observation clearly proves that amount of vibration occurring at rotor is not fully transmitted to casing for every rotating machine even though the readings may close to each other as per graph below –

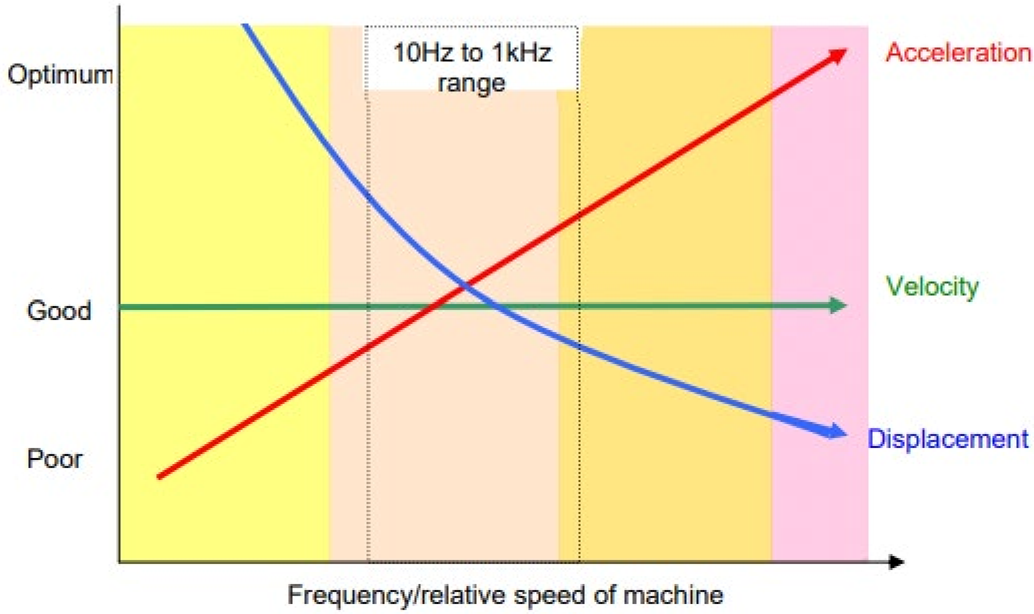

Fig 5 – Range of various measurement accuracy of vibration and trade-off

In case of rotors using hydrodynamic journal fluid film bearings, a substantial amount of vibration is attenuated due to dynamic stiffness of rotor bearing system as shown -

K Dynamic Stiffness= $(K-M\Omega 2) + j(D\omega - \lambda D\ \Omega)$

Where K is the spring stiffness, M is the rotor modal mass, D is the rotor/bearing viscous modal damping, and ω is a term that represents the frequency of the vibration precession (or orbiting), which may be different from the rotation speed, Ω. λ is the Fluid Circumferential Average Velocity Ratio of the fluid in the bearing or seal.

As known, Vibration Displacement =

Summation of Forces Acting on the Rotor System divided by the System Dynamic Stiffness

In the case of rotor with fluid film bearing, vibration energy is absorbed and dissipated via the direct damping term, $D\omega$, which is the rotor pushing on the fluid, as a shock absorber effect. In high speed turbo machines, adequate damping occurs within the bearings as they are rigidly mounted to the casing and foundation and the foundation spring rigidly connected to ground.

Due to this viscous damping by lube oil and viscous shear, energy is dissipated per cycle in form of heat and hence oil temperature is controlled by air cooler / heat exchanger in a rotating machinery employing hydrodynamic bearings.

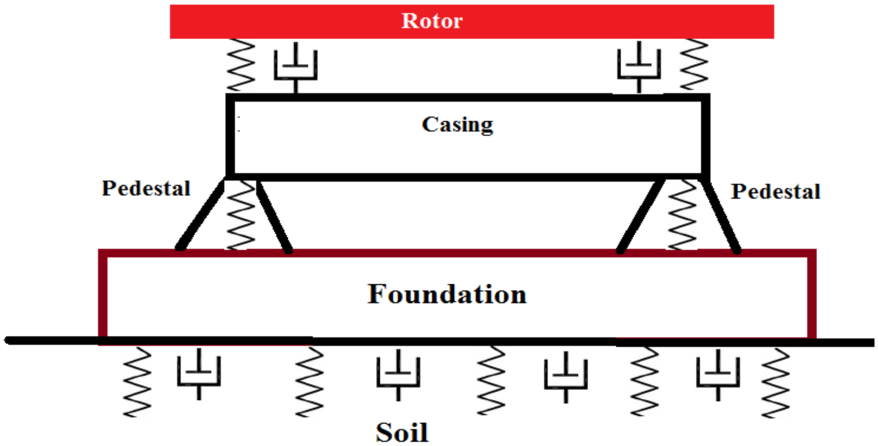

Fig 6 -Transmission of vibration to foundation -rotor with hydrodynamic bearings

The design methodology does not consider machines according to support flexibility Two conditions are used to classify the support assembly flexibility in specified directions: – rigid supports; – flexible supports. These support conditions are determined by the relationship between the machine and support / pedestal flexibilities/ stiffness.

With current method of foundation design, it can be seen below that maximum amplitude are found much below than as required or prescribed by OEM / text book requirement (20 am vs 1.6 am) which means foundation is overdesigned in terms of dynamics.

Type	**Permissible amplitudes** (m)
Low-speed machinery (500 rpm)	0.0002 to 0.00025
Hammer foundations	0.001 to 0.0012
High -speed machinery: a. (3000 rpm) b. (1500 rpm)	 0.00002 to 0.00003 0.00004 to 0.00006

Fig 7 - Generic permissible amplitudes of foundation vibration

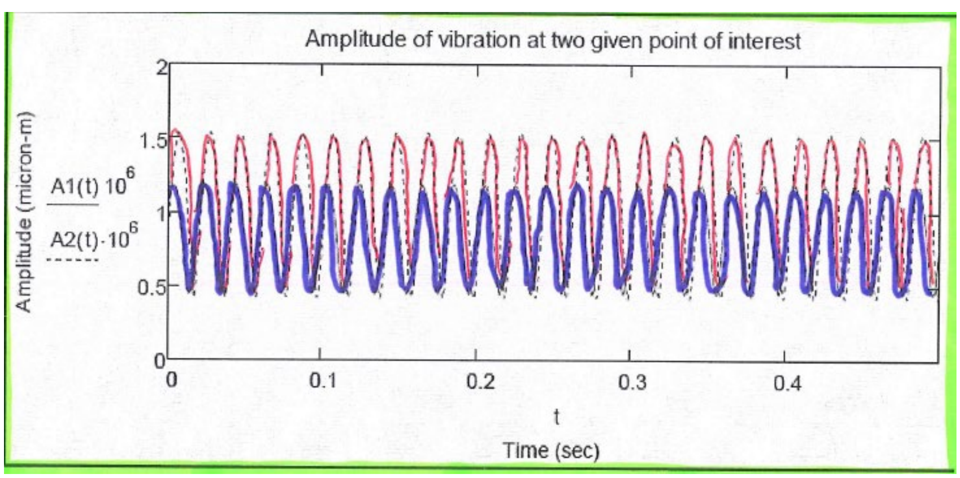

Fig 8 Foundation analysis Output file for actual amplitude

Proposed approach for foundation design - machinery dynamics -

In rotor-dynamic point of view the foundation design should consider classification of machine in terms of type of bearings, actual balance grade, dynamic stiffness ratio of pedestal and bearings and type of device used to measure vibration.

It is seen that fluid film bearings dampen the rotor vibration and a low fraction of vibration is transmitted to foundation which is based on stiffness ratio of bearing pedestal and rotor bearing. With high thickness of casing of heavy-duty rotating equipment, the stiffness is quite high compared to rotor and hence negligible amount of vibration is transmitted to foundation. Fig 8 illustrates the visual comparison of bearing stiffness vs casing stiffness.

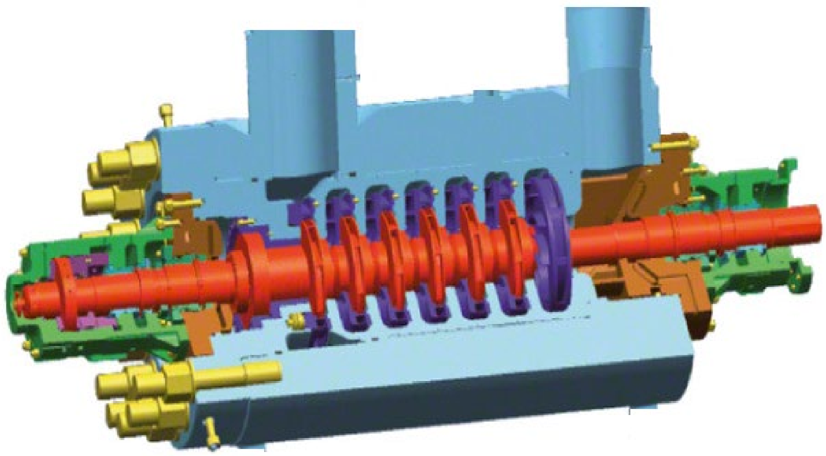

Fig 9 - High speed turbo-machine Rotor (red) with highly stiff casing (in blue)

If the support stiffness is 3.5 times higher than bearing stiffness at particular operating frequency, support is considered adequately rigid as suggested by API. With conservative value of estimated amplification factor, unbalance response of rotor at first critical speed and vibration values at centre span can be calculated. This value can be used as eccentricity to calculate total force exerted on bearing pedestal / casing even in case of machine having rotor sag or temporary rotor bow. When machine and foundation is considered as one entity, vibration measured and transmitted shall be quite lower than required.

According to API standards the permitted initial mass eccentricity is tighter than ISO balance quality grade G2.5, which is normally applied to high speed equipment under ISO 1940. With API consideration, actual grade is ISO grade G0.67 equivalent. Rotating machine manufacturers often do report the unbalance that remains after balancing to demonstrate that their internal Quality plan and API criteria are met. For machines with high casing stiffness and using active magnetic bearings it can be considered that very low vibration is transmitted due to magnetic levitation of rotor.

Another point is to consider the method of measuring vibration of various types of machines based on the pedestal / casing stiffness and bearing stiffness. Some of high-speed machines use relative shaft displacement methods since long time to measure onerous data for machinery protection and monitoring. As per ISO TR 19201:2013(Titled

-Mechanical vibrations – methodology for selecting appropriate machinery vibration standards), this choice is based on ratio of dynamic stiffness of pedestal (bearing housing connected with casing) and dynamic stiffness of bearing. The dynamic stiffness related to damping and mass effect of components.

For massive casings rigidly connected to foundation designed for static loads, the overall vibration amplitude is miniscule and can be considered as negligible as they normally use relative shaft vibration measurement (zone in yellow) based on following chart from ISO TR 19201:2013.

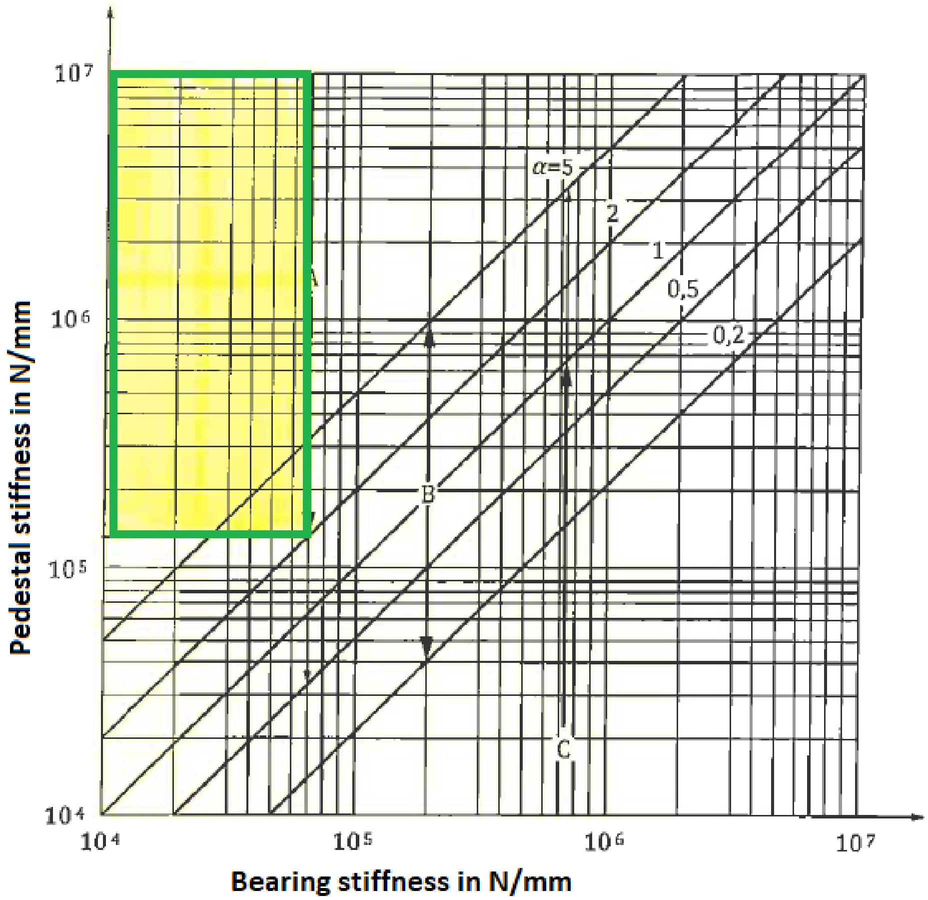

Fig 10 -Zone of measuring relative shaft vibration based on stiffness ratio

It is worth to note that machines equipped with active magnetic bearing create adequate stiffness and damping by magnetic flux and enable rotor to levitate. Hence very low vibration transmission occurs from rotor to stator. This means the foundation also is subjected to very low vibration amplitude.

Effect of transients – Foundation Design against short circuit - Now a days the protection system such as IGBT type transistors, rotor flux monitoring system can dictate and isolate the short circuit from source and in motor itself. The instrument can monitor and evaluate up to four machines at the same time with stored the base line measurements, so that when a rotor shorted turn is detected, an alert relay is activated.

Conclusion with proposed methodology

The rotating machinery should be classified as machine with antifriction bearings and / or machines with lower ratio (<3.5) of dynamic stiffness as listed in ISO TR 19201:2013 to conduct dynamic analysis of foundations. They include - OH2 pumps, low energy pumps as defined in API 610, centrifugal fans, electric motors, screw compressors, steam and industrial gas turbines turbine. For high speed turbomachines with higher ratio of dynamic stiffness, relevant sections of ISO 7919 with a conjunction of ISO 19201 to be followed. Below table Table A.1 of ISO 10816-3. shows shaft displacement and bearing cap vibration velocity both are to be considered together as acceptance criteria as shown

Support class	Zone boundary	R.m.s. displacement µm	R.m.s. velocity mm/s
Rigid	A/B	29	2,3
	B/C	57	4,5
	C/D	90	7,1
Flexible	A/B	45	3,5
	B/C	90	7,1
	C/D	140	11,0

However , in both type of machines, it is imperative to ensure that bearing housing natural frequencies do not coincides with any operating frequency and its harmonics .

For turbo-machines, dynamic analysis of foundation is not needed to be carried out if –

- Machine incorporates a "between bearing design" with operating frequency equal or higher than 100 Hz or machines with active magnetic bearings or hydrodynamic fluid film and
- Rotor dynamics analysis has considered absolute vibration and relative vibration as close or same and
- Machinery protection probes are installed on casing and rotor meets ISO 1940 balance quality as 1 or lower (API residual unbalance standard) and
- No vibration trip multiplier is incorparted in set up.

In that case, only static analysis of foundation can be done with following guidelines –

1. The bottom of block foundation should be above water table. It should not be resting on back filled soil or soil prone to excessive settlement.

2. The mass of rigid foundation shall be 2- 3 times the mass of supported machine (for centrifugal machines with between bearing, fluid film design. In addition, foundation mass shall not be less than 10 times mass of rotor.

3. The top of block is to be kept (0.3 m) above finished floor or high point of paving to prevent damage from surface water run-off.

4. The vertical thickness of block should not be less than .600 mm. However, if the installation is under the shade, same can be optimized to 300 mm. Also, thickness of block should be enough to accommodate anchor bolt embedment

5. The foundation should be wide enough to increase damping in the rocking mode. The width should be at least 1.5 to 2 times height of machine centreline from machine base

6. The combined centre of gravity of machine and foundation should coincide with the centre of gravity of the foundation.

7. Static bearing capacity is proportion of footing area for 50% of allowable soil pressure, which means that the actual soil pressure should be less than 50% of static bearing capacity.

8. Static settlement must be uniform; centre of gravity of footing and machine load should be within 5% of each linear dimension from the foundation centre.

9. End user can insist machinery OEM that rotor should have lower amplification factor i.e. less than 4. Lower amplification factors mean high damping and that means lower force on bearing during resonance condition.

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