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Design Sensitivity Analysis of Raft Foundation for Marine Engines and Machinery in Warships

A. L. Tappu[1], A. K. Sen[2], M. M. Lele[3]
[1](Sr. Associate Professor, Tolani Maritime Institute, Talegaon, Pune, India)
[2](Principal, St. Francis Institute of Technology, Mount Poinsur, Borivali (West), Mumbai, India)
[3](Professor, Maharashtra Institute of Technology, Pune, India)

ABSTRACT

It is necessary to design warships so as to acoustically insulate them from enemy vessels. The conventional designs of various systems have to be modified for achieving this purpose. The aim is to keep all the types of sound transmission to a low level viz. air-borne, water-borne or structure borne. As such the structure borne noise i.e. engine or machinery noise transmitted through the foundation and hull of the vessel has to be minimized. The design of the machinery mounting system hence becomes very important. One of the techniques used is to mount all vital machinery on double stage vibration isolation systems. A two-stage mounting system (also called as raft mounting) is employed where there is a demand for high structure-borne noise attenuation.

The present paper deals with the dynamic analysis and design sensitivity analysis of such a marine engine foundation system considering it as a two degree freedom system. An algorithm has been developed for the same. The vertical vibrations of the system are assumed to be most predominant. The other types of vibrations like rocking or transverse type of vibrations are assumed to negligible and to be taken by the mounts and limiters.

Keywords*:* Double-stage vibration isolation, raft foundation for warships, marine engine foundation, computational methods, dynamic analysis.

1.1 INTRODUCTION:

Marine diesel engines are supported by anti-vibration mounts that are designed to provide both structural rigidity and vibration isolation. Structural rigidity is required in order to maintain alignment of connecting shafts, piping and other connecting elements, whereas vibration isolation is required to minimize the vibrations generated from the engine from being transmitted to the rest of vessel and beyond. A mounting system for marine engine should also be able to react to the strong dynamic force caused by wave slap, cornering loads and docking impact. Improvement of the mounting systems can be attempted by a special seating design or by changing from a conventional single stage design to a double stage mounting system. [1]

In this paper development of a mathematical model and an algorithm for a predictive design of a double stage foundation system has been done. The sensitivity analysis of the system for the change in various parameters like stiffness and damping of various mounts, mass of the raft and so on, has been done so as to manipulate and improve the system response. This can further serve the purpose of a basis for optimizing certain parameters, for example to keep force transmissibility and engine bounce to a minimum value and also the static deflection within limits so as to finally reduce the structure borne sound transmission.

1.2 FEATURES OF THE DOUBLE STAGE FOUNDATION SYSTEM:

In a double stage foundation system, the marine engine is placed on an upper layer of mounts supported by a raft (a simple steel plate or any rigid plate like structure, e.g. a concrete block) which is further supported on the hull girder through a set of lower level of mounts.

The special features of the double stage system are:

It improves vibration isolation and that too over a broad frequency range.

(a) Appreciable reduction in under water noise.

(b) Limiters are provided for lateral stability.

The critical design aspects of the system are:

1) Optimum mass of raft (intermediate mass).

2) Design of suitable mounts for both stages for desired dynamic performance.

3) Minimum height of the system for ensuring stability.

Recently NSTL (Naval Science & Technology Laboratory, Vishakhapatnam, India) had developed and installed a two-stage mounting system for shipboard HP air compressor used on a warship. A significant vibration reduction (25 dB) had been achieved, which in turn lead to reduced radiated noise levels. [2]

1.3 REQUIREMENTS OF DESIGN

The mounting systems should be designed to carry out the following functions:

1) To provide structural rigidity

2) To isolate base from engine vibration (force transmission caused due to perturbation force)

3) To avoid excessive vibration of engine i.e. bounce due to shock excitation at base (motion transmissibility).

The challenge for the design engineer is how to select the vibration isolators and raft size and how to properly install them so as to (a) keep the static deflection of isolators within a limit for keeping the balance of engine under the strong impact, (b) keep the foundation stiffness low in order to minimize the structure-borne noise and vibration level in the cabins or noise breakout into the water, (c) keep sufficient frequency distance to avoid resonance, (d) keep the dynamic amplitude of steady state vibrations of the engine and raft within limits.

In the case of a warship the objective is to minimize the transmitted force. This can be done either by moving the system natural frequencies away from an undesired frequency or by directly optimizing the force transmission by adjusting the design parameters. It should also be confirmed that engine bounce does not go beyond limits. It is found that the structure-borne noise or vibration level transmitted to other spaces significantly depends on the vertical force (normal to the floor) which generates the bending wave in the structure. Thus the objective is to minimize the transmitted force normal to the base. This is because only the force normal to the base can excite the bending wave which contributes most of energy of structure-borne noise. [3]

The different parameters which affect the dynamic response of an engine/machine and its foundation system are the nature and magnitude of exciting forces, the excitation frequency, the masses of the engine and foundation, the stiffness and damping properties of the vibration isolators and so on.

The designer should be able to first make a prediction of the dynamic performance of the foundation and then optimize the parameters if necessary. This paper tries to address the issue of predictive dynamic analysis and further parametric analysis of the same.

2.1 DESIGN PROCEDURE:

The techniques of analysis of a machine or an engine foundation system can be applied with varying degrees of difficulty. In this paper we have assumed the engine-foundation system as a two degree freedom system. The engine and the raft have been assumed to be rigid. The vertical vibrations of the system are assumed to be most predominant. The other types of vibrations like rocking or transverse type of vibrations are assumed to negligible and to be taken by the mounts and limiters which are placed at the sides to avoid lateral movement.

Fig 1(a) A two stage foundation system with mounts

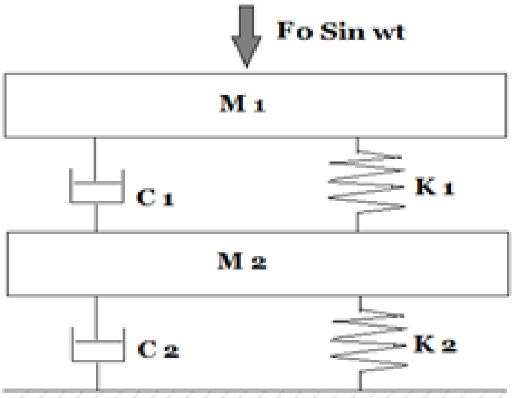


Fig 1(b) Model for the system

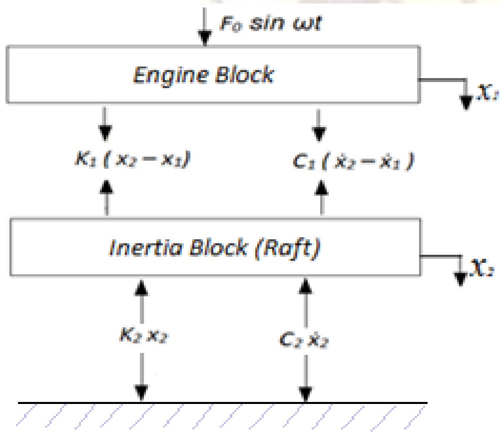


Fig 2(a) FBD (Force transmissibility)

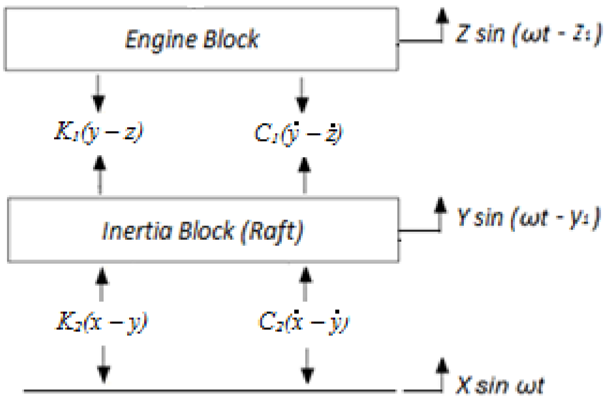


Fig 2(b) FBD (Motion transmissibility)

2.2 MATHEMATICAL MODELING AND ANALYSIS:

Modeling of the engine - raft - hull girder system has been done as shown in the Fig no. 1(b). Upper and lower level of mounts and be considered to be made of a number of springs and dashpots in parallel. Mathematical analysis has been done as follows:

2.3 STATIC DEFLECTION:

Let M_1 and M_2 be the masses of the engine and raft respectively, K_1 and K_2 the equivalent stiffnesses of the upper and lower level mounts, C_1 and C_2 be the equivalent damping coefficients of the upper and lower level mounts respectively. Static displacement amplitudes will be given by:

Static Deflection for the raft =
: $Xst(raft), = (M_1 + M_2)\, g / K_2$ …(1)
Additional Static Deflection for the engine =
: $Xst(engine) = M_1\, g / K_1$ …(2)

2.4 FREE DAMPED VIBRATIONS

The equations of motion for the *free damped vibrations* of the two degree freedom system shown in Figure no. 2(a), (neglecting exciting force) are as follows:
: $M_1\, \ddot{x}_1 - C_1\, (\dot{x}_2 - \dot{x}_1) - K_1\, (x_2 - x_1) = 0$ …(3)
: $M_2\, \ddot{x}_2 + C_1\, (\dot{x}_2 - \dot{x}_1) + K_1\, (x_2 - x_1) + C_2\, \dot{x}_2 + K_2\, x_2 = 0$ …(4)

Where x_1 & x_2 are the displacements of the engine and raft respectively. These are two coupled homogenous second order linear differential equations of motion.
The complimentary solutions can be written in the form
: $x_1 = e^{-rt}\, \{ P_1\, Sin(dt + \phi_1)\} + e^{-ut}\, \{ Q_1\, Sin(vt + \phi_2)\}$ …(5)
: $x_2 = e^{-rt}\, \{ P_2\, Sin(dt + \phi_1)\} + e^{-ut}\, \{ Q_2\, Sin(vt + \phi_2)\}$ …(6)
where values of constants P_1, P_2, Q_1, Q_2, ϕ_1, ϕ_2 can be derived from initial conditions.

The equations represent a transient type of vibration which dies out in a very short time. Hence the steady state vibrations which are more important are analyzed.

2.5 MODAL ANALYSIS: Eigen values (natural frequencies) and Eigen vectors (mode shapes) for the two degree freedom system are found out by programming in ANSYS. The values of natural frequencies of the system are required so as to keep proper frequency distance between the perturbation frequencies and natural frequencies as per recommendations of classification societies.

2.6 FORCED VIBRATIONS: The analysis is done for forced vibrations assuming a harmonic type of excitation force of $F_0 \sin \omega t$ i.e. the imaginary part of $F_0\, e^{i\omega t}$. This leads to a steady state vibration. The analysis is important as we are interested in getting the values of the dynamic amplitudes of the engine bed X_1 and raft X_2 at various frequencies. These have to be in an allowable range as per the norms set by. The free body diagram of the two masses is as shown in Fig no. 2(a). The equations of motion are:

: $M_1\, \ddot{x}_1 - C_1\, (\dot{x}_2 - \dot{x}_1) - K_1\, (x_2 - x_1) = Im\, \{ F_0\, e^{i\omega t}\}$ …(7)
: $M_2\, \ddot{x}_2 + C_1\, (\dot{x}_2 - \dot{x}_1) + K_1\, (x_2 - x_1) + C_2\, \dot{x}_2 + K_2\, x_2 = 0$ …(8)

Assuming the solutions as $x_1 = X_1 e^{i\omega t}$ and $x_2 = X_2\, e^{i\omega t}$ and substituting in the above equations

: $\{(-M_1\, \omega^2 + K_1) + i\, (C_1\, \omega)\}\, X_1 - \{ K_1 + i\, (C_1\, \omega)\}\, X_2 = F_0$ …(9)
: $-\{ K_1 + i\, (C_1\, \omega)\}\, X_1 + \{ (-M_2\, \omega^2 + [K_1 + K_2]) + i\, ([C_1 + C_2]\, \omega)\}\, X_2 = 0$ …(10)

Solving the above two equations we get complex values of x_1 and x_2. By mathematical treatment we get the values of X_1 and X_2 i.e. we get the dynamic amplitudes of vibration engine and inertia block respectively as:

: $X_1 = NUM_1 / DEN$ ... (11)
: $X_2 = NUM_2 / DEN$ where ... (12)
: $NUM_1 = F_0\, \{ [(K_1 + K_2) - M_2\, \omega^2]^2 + [(C_1 + C_2)\omega]^2 \}^{1/2}$ ... (11a)
: $NUM_2 = F_0\, \{ K_1^2 + (C_1\omega)^2 \}^{1/2}$ ... (12a)
: $DEN_1 = \{ M_1\, M_2\, \omega^4 - [K_1 M_2 + M_1\, (K_1 + K_2) + C_1 C_2]\omega^2 + K_1 K_2 \}^2$ ...(11b)
: $DEN_2 = \{ -[M_2 C_1 + M_1 (C_1 + C_2)]\omega^3 + [C_1 K_2 + K_1 C_2]\omega \}^2$ ... (12b)
: $DEN = (DEN_1 + DEN_2)^{1/2}$ ... (13)
If we neglect damping (considering it to be low) then we have,
: $X_1 = F_0\, \{ (K_1 + K_2) - M_2\, \omega^2 \} / \{ M_1\, M_2\, \omega^4 - [K_1 M_2 + M_1\, (K_1 + K_2)]\omega^2 + K_1 K_2 \}$ …(14)
: $X_2 = F_0\, K_1\, / \{ M_1\, M_2\, \omega^4 - [K_1 M_2 + M_1\, (K_1 + K_2)]\omega^2 + K_1 K_2 \}$ …(15)

The general solution consists of the complimentary function and particular integral and is of nature:

: $x = x_{cf} + x_{pi}$ …(16)

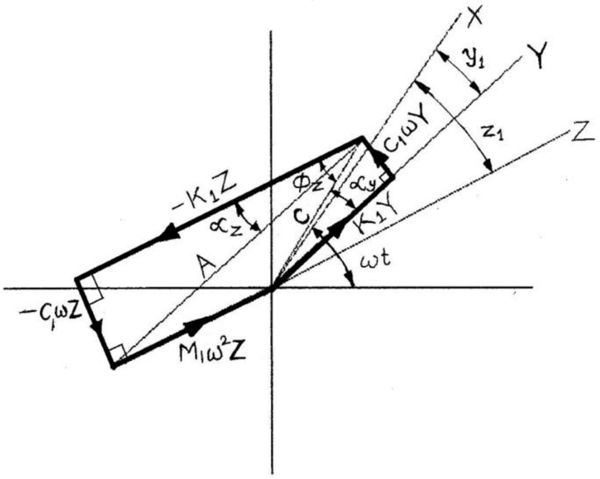


Fig 3(a) Equilibrium of forces at mass M_1

Out of this x_{cf} (which has been dealt with in the previous section) dies out in a short time and x_{pi} represents the steady state vibration. The steady state motion of the two masses are given by the following equations,

: $x_1 = X_1 \, Sin(\omega \, t - \psi_1)$ and : $x_2 = X_2 \, Sin(\omega \, t - \psi_2)$ …(17 a & b)

The solution shows that the response is a sinusoidal motion with the amplitudes X_1 for the engine and X_2 for the raft foundation block. The constants ψ_1 and ψ_2 are to be found out from initial conditions.

2.7 FORCE TRANSMISSIBILITY

Generally force transmitted to foundation due to a single stage would be given by the relation

: $Ftr_1 = [(K_1 X_1)^2 + (C_1 \, \omega \, X_1)^2]^{1/2}$ …(18)

Whereas appropriate addition of the second stage foundation would lead to a transmitted force of

: $Ftr_2 = [(K_2 X_2)^2 + (C_2 \, \omega \, X_2)^2]^{1/2}$ …(19)

: Force transmissibility for double stage foundation systems = Ftr_2 / F_0 …(20)

2.8 DISPLACEMENT TRANSMISSIBILITY (ENGINE BOUNCE AND RAFT BOUNCE)

Let the hull girder be excited by a sinusoidal displacement of $x = X \, Sin \, \omega t$. The response of the raft would be $y = Y \, Sin \, (\omega t - y_1)$ and that of the engine would be $z = Z \, Sin \, (\omega t - z_1)$, where Y and Z are the amplitudes of the bounce of the raft and engine respectively. (Refer Fig. no. 2(b)). These amplitudes have to be in limits or else it would affect the functioning of the engine adversely.

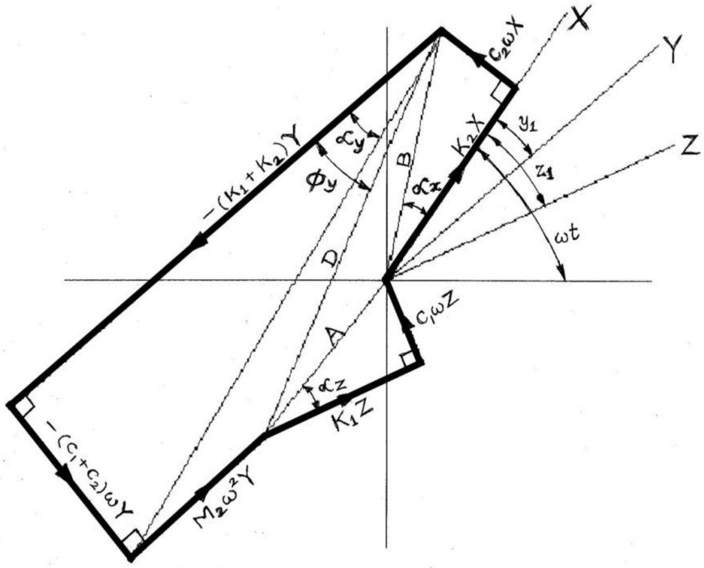


Fig 3(b) Equilibrium of forces at mass M_2

The calculations for motion transmissibility are as follows:
We define the following terms (referring to force polygons in Fig no. 3(a & b))

: $\Phi_z = \tan^{-1}(C_1\omega / (K_1 - M_1 \, \omega^2))$
: $\Phi_y = \tan^{-1}((C_1 + C_2)\omega / (K_1 + K_2 - M_2 \, \omega^2))$
: $\alpha_x = \tan^{-1}(C_2\omega / K_2)$
: $\alpha_y = \alpha_z = \tan^{-1}(C_1\omega / K_1)$
: $A = \{K_1^2 + (C_1\omega)^2\}^{(1/2)} Z$
: $B = \{K_2^{\,2} + (C_2\omega)^2\}^{(1/2)} X$
: $C = \{(K_1 - M_1 \, \omega^2)^2 + (C_1\omega)^2\}^{(1/2)} Z$
: $D = \{(K_1 + K_2 - M_2 \, \omega^2)^2 + [(C_1 + C_2) \, \omega]^2\}^{(1/2)} Y$
: $E = Cos \, (\Phi_y + \Phi_z - 2 \, \alpha_y)$

We get the following results:

: $Raft \; bounce = Y/X = BC / \{(CD)^2 + A^4 - 2A^2 CDE\}^{(1/2)}$
where Y is Raft Bounce amplitude …(21)

: $Engine \; bounce = Z/X = AB / \{(CD)^2 + A^4 - 2A^2 CDE\}^{(1/2)}$
where Z is Engine bounce amplitude …(22)

Motion transmissibility neglecting damping:

: $Engine \; bounce = Z/X = K_1 \, K_2 / \{K_1 \, K_2 - (K_1 \, M_1 + M_1 \, [K_1 + K_2]) \, \omega^2 + M_1 \, M_2 \, \omega^4\}$ …(23)

: $Raft \; bounce = Y/X = K_2 \, (K_1 - M_1\omega^2) / \{K_1 \, K_2 - (K_1 \, M_1 + M_1 \, [K_1 + K_2]) \, \omega^2 + M_1 \, M_2 \, \omega^4\}$ …(24)

3.1 SENSITIVITY ANALYSIS

The above mathematical modeling and analysis has been used to develop an algorithm and a computer program which can be used to get a predictive design of the double stage foundation system. A sensitivity analysis has been carried out to understand in what way and how much each parameter contributes to the functioning of the foundation system as a vibrating body and what effect can be brought out by changing these different parameters.

3.2 A CASE STUDY

As an example, a problem of sensitivity analysis and design of a foundation system has been solved by using the computer program developed.

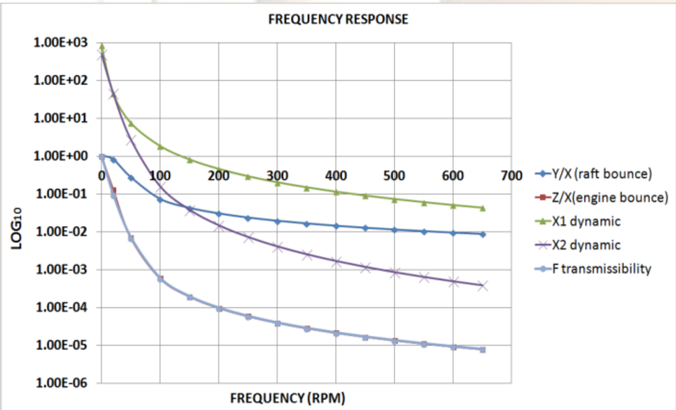
Problem Statement: A foundation system is to be designed for an engine with mass 5000 kg running at a constant speed of 500 rpm.

Solution: A set of trial values are chosen arbitrarily as follows:

M_1=5000 kg, M_2=500 kg, K_1=3000N/m, K_2=2000 N/m, C_1=300 N-sec/m, C_2=300 N-sec/m.

The frequency response would be as shown in chart – 1. The static deflection however (not shown in chart) is too high.

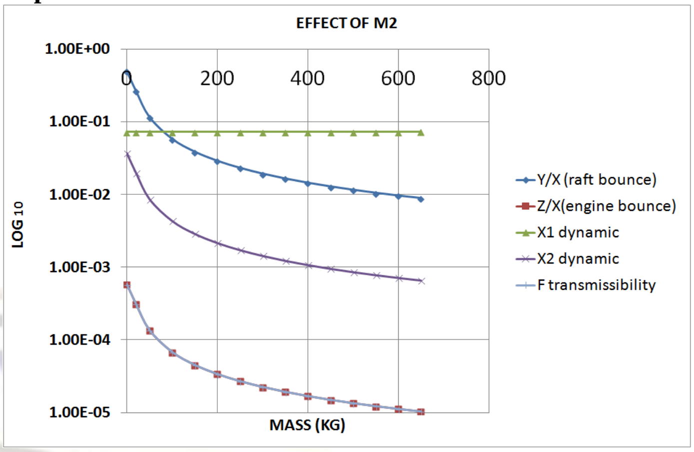
Chart -1: Response of system with variation in frequency (first trial)



Setting the speed of 500 rpm the effect of mass of raft M_2 is analyzed (as shown in chart – 2).

M_1 = 5000 kg, N = 500 rpm, K_1 =3000 N/m, K_2=2000 N/m, C_1=300 N-sec/m, C_2=300 N-sec/m

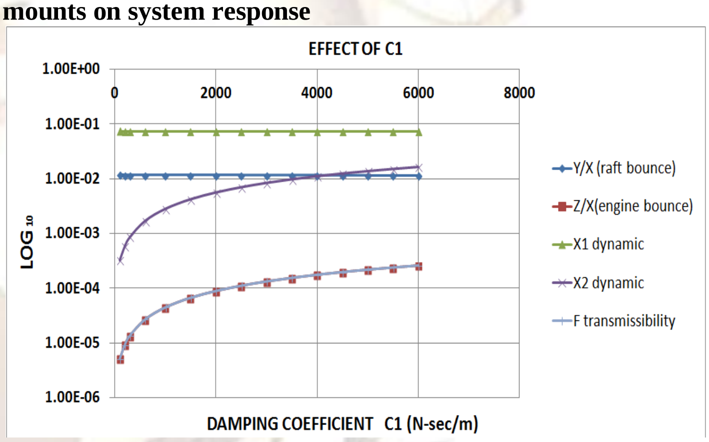
Chart – 2 : Effect of change in mass on system response



Setting the mass M_2 as 500 kg keeping in mind that raft should not be a huge structure and should be a plate like structure with minimal height, the effect of C_1 is analyzed (as shown in chart – 3).

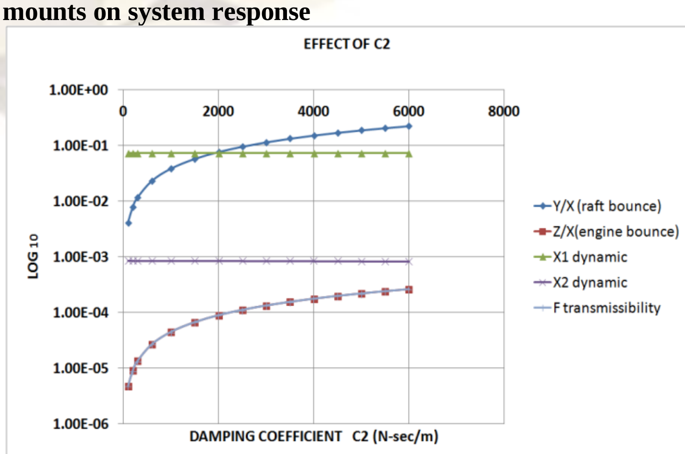
M_1 = 5000 kg, M_2 = 500 kg, N = 500 rpm, K_1 = 3000 N/m, K_2 = 2000 N/m, C_2 = 300 N-sec/m

Chart - 3: Effect of change in damping in upper mounts on system response



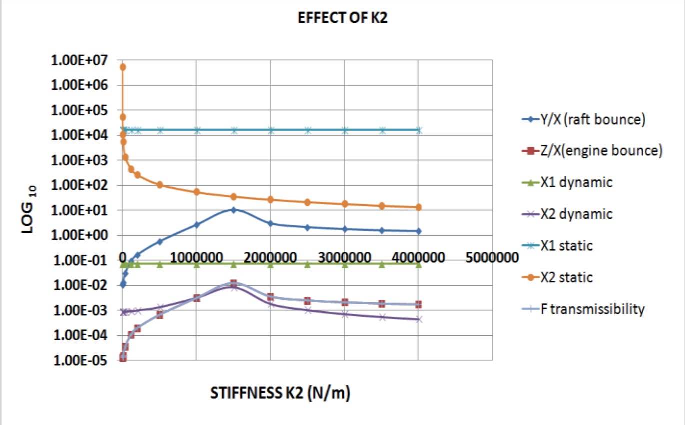
Setting a value of C_1 as 300 N-sec/m, the effect of C_2 is analyzed (as shown in chart no – 4) M_1=10000 kg, M_2=400 kg, N=200 rpm, K_1=30000 N/m, K_2=20000 N/m, C_1=200 N-sec/m

Chart - 4: Effect of change in damping in lower mounts on system response



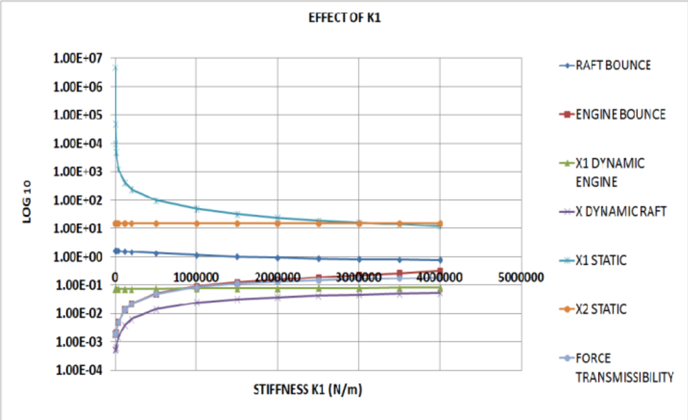
The value of C_2 is selected as 300 N-sec/m tentatively. By now M_2, C_1 and C_2 have been finalized. Now the effect of K_2 is analyzed. Note that the static displacements of the engine and raft are also taken into consideration.
(as shown in chart – 5)
M_1=5000 kg, M_2=500 kg, N=500 rpm, K_1=3000 N/m, C_1=300 N-sec/m, C_2=300 N-sec/m

Chart – 5: Effect of change in stiffness in lower mounts on system response



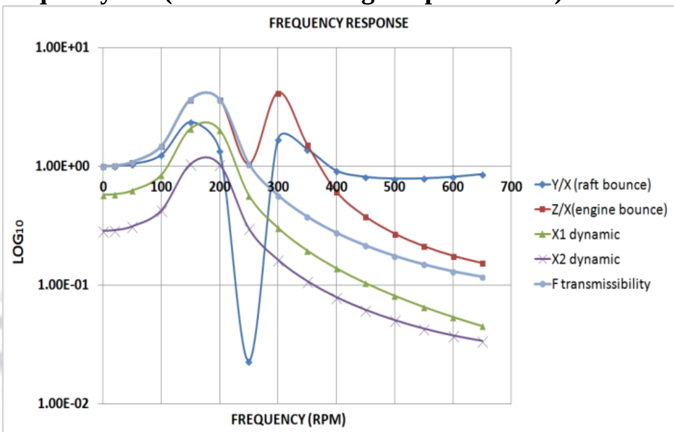
Deciding a higher value of stiffness, so as to curtail static deflection, value of K_2 has been set as 35,00,000 N/m. Finally the effect of stiffness of upper spring i.e. K_1 is analyzed (as shown in chart - 6)
M_1 = 5000 kg, M_2 = 500 kg, N = 500 rpm, K_2 = 35,00,000 N/m, C_1 = 300 N-sec/m, C_2 = 300 N-sec/m

Chart – 6: Effect of change in stiffness in upper mounts on system response



Again taking into consideration the static deflection and other parameters like force transmissibility, engine and raft bounce steady state amplitude of engine and raft the value of K_1 is finalized as 3500000 N/m. Thus the parameters of the system have been finalized as: M_1 = 5000 kg and N = 500 rpm (given). M_2= 500 kg, C_1= C_2 =300 N-sec/m, K_1 = K_2 = 3.5 x 10^6 N/m. (selected)
The system response achieved due to this design is as follows: (refer chart – 7)

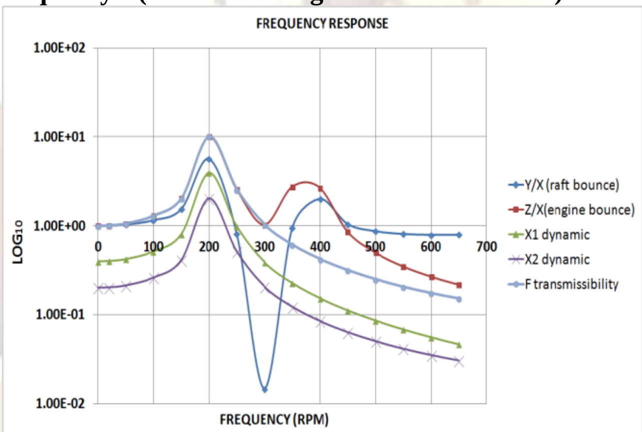
Chart – 7: Response of system with variation in frequency (after selecting parameters)



Engine Bounce = Z/X = 0.2713, Raft Bounce = Y/X = 0.7902, Steady state amplitude of engine = X_1 = 0.0808 mm, Steady state amplitude of raft = X_2 = 0.05025 mm, Force transmissibility = 0.175, Static deflection for the raft = *Xst(raft)*, = 15.415 mm, Additional static deflection for the engine = *Xst(engine)*= 14.014 mm. Total static deflection of engine = 29.429 mm.

Further to enhance the response, the values of K_1 and K_2 are increased to 5 x 10^6 N/m. We get a better response (as shown in chart - 8)

Chart – 8: Response of system with variation in frequency (after selecting better stiffnesses)



Engine Bounce = Z/X = 0.4996, Raft Bounce = Y/X = 0.8687, Steady state amplitude of engine = X_1 = 0.08627 mm, Steady state amplitude of raft = X_2 = 0.04998 mm, Force transmissibility = 0.2499, static deflection for the raft = *Xst(raft)*, = 10.79 mm, additional static deflection for the engine = *Xst(engine)*= 9.81 mm. Total static deflection of engine = 20.6 mm Thus we can instantly compare the change in the system response by changing the stiffness values of the system. All the constraints for design are satisfied. Hence the design is suitable.

5.1 Conclusion

It is observed that the values of static deflection and force transmissibility seem to be more decisive factors and are very sensitive to stiffness of the springs and to excitation frequency. Mathematical modeling and analysis has been done for a double stage foundation system so as to replace the general single stage foundation system. An algorithm has been developed which can be used to have a predictive design of the same. The system response can be instantly predicted by use of charts developed by the computer program. It is further useful in parametric analysis of the system so as to get the values of static and dynamic deflection of engine and raft i.e. *Xst(engine), Xst(raft),* X_1 , X_2, force and displacement transmissibility of the system at a time and manipulate the same. These computations would make it possible for a designer to select appropriate mounts at higher and lower level and mass of raft so as to conform to the norms set by the classification societies. This would further help to achieve minimal force transmissibility and bounce and thus attenuate structure borne noise.

6.1 Future Scope

In the present paper we have assumed the engine-foundation system as a two degree freedom system assuming the engine and raft both to be rigid. We have also concentrated on vertical vibrations, neglecting the rocking type and transverse type of vibrations (considering the limiters). These assumptions also have been justified previously. However if we consider flexibility of raft and the rocking type and transverse type of vibrations, the analysis would further become complex as it would involve additional degrees of freedom. Also a criterion for deciding to go in for the selection of a double stage foundation instead of a single stage one has to be explicitly addressed in terms of vibration parameters. In other words, the advantage in going in for a two stage system has to be verified. The program developed can serve to be a base in further optimizing any parameter (say reduction in force transmissibility) of the system response.

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