DEVICES FOR INSITU TUNING OF MACHINERY STRUCTURES TO MINIGATE RESONANCE

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Abstract –

For on-shore and grout less design applications, the support structure for a rotating machine consists of a base-frame (base plate) normally known as skid. Most of the time these structures show up with high vibration on machine bearing cap due to structural resonance, unidentified cracks. Detection and mitigation of such anomaly is time consuming job. The paper proposes a blueprint of active and passive resonance control at location without removal of base frame from foundation / main support and any structural modification which requires any hot work.

Introduction -

Structural resonance is commonly identified as symptom of excessive vibrations of non-rotating components, usually machinery parts or supporting structures. This is the more common resonant condition usually occurs at or near the rotating speed of the machine. Even slight vibratory forces due to residual unbalance and misalignment effects of the machine can excite to resonate the base structure, resulting in severe vibration.

During resonant condition, the support structure normally demonstrates vibrations in rigid body modes and aggravating the shaft vibration conditions. This issue occurs normally with rigid rotors with antifriction bearings or with machines where ratio of pedestal dynamic stiffness and bearing stiffness are less than 3 such as electric motors, generators, steam and gas turbine casing structures.

Sometimes, the evaluation of resonant frequencies for equipment base frame is not given its due importance by the OEM particularly for medium sized rotating machines. To identify such condition at site and rectification is cumbersome due to the complexity of the components in structural base frame.

Descriptions of structural resonance and methods to detect -

Sometimes in spite of good quality of balance, correct alignment, rotor exhibits excessive vibration at 1x harmonics. Often it is assumed that simply balancing the rotor with a stricter grade will make the vibration problem go away. However, upon further investigation it is not uncommon to find that a structural resonance is the root cause of excessive vibration, and balancing is not a viable or long-term solution to the problem.

The large amplitudes of vibration are seen when a machine's running speed is at or near a natural frequency even if the vibration amplitude due to unbalance or misalignment or other higher harmonics such as vane pass frequency are low This problem is particularly common in variable speed equipment or for those machines having specially fabricated base plate with non- grouted design option.

Various field tests are performed to verify that "resonance is really in fact the cause" of excessive vibration in a system. Two of the most common tests are a modal impact test and a start-up or coast-down data collection with a cascade plot. The cascade plot below shows an example of typical vibration exhibited during start-up of a machine with a structural natural frequency in its operating speed range as shown in fig 1.

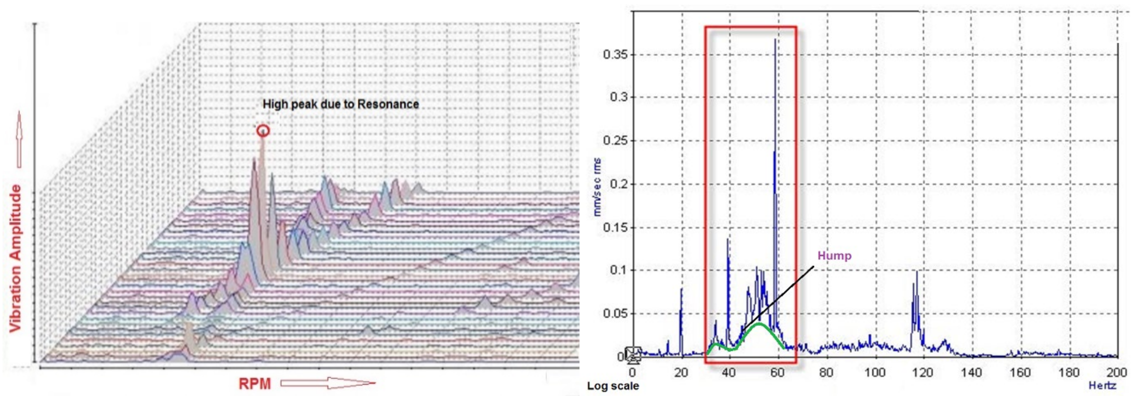

Fig 1 – Occurrence of resonance as detected in cascade plot and FFT spectra

Resonance is characterized by a large increase in vibration at a particular frequency with humps but generally lower amplitudes at all other operating speeds / frequencies as shown in below FFT spectra of a machine.

Sometimes, bump test combined with ODS (operational deflection shape) analysis also used as verification of existence of structural resonance. It is easy to detect pure horizontal or vertical movement of rotor / machine but without visual aid such as wmv or animation file, it is not possible to find location of deflection on structure.

The normal modes of structural and machinery movement due a resonant condition is shown in simplified manner.

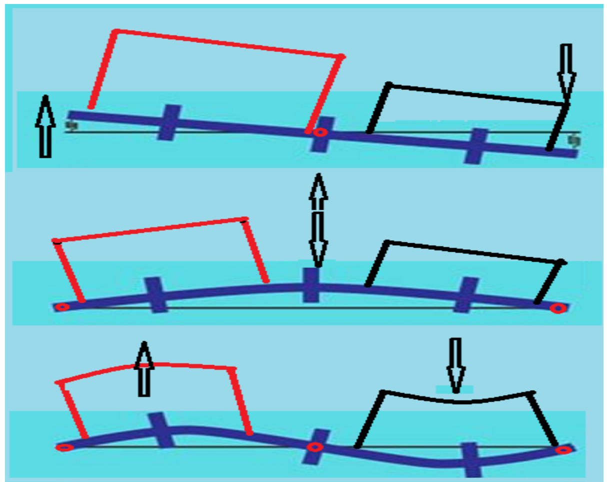

Fig 2 various modes of structural vibration

Any rigid-body displacement of a structural element can be resolved into these six independent displacements as shown in fig. Hence, the rigid block has six degrees of freedom and six natural frequencies. Of six types of motion, translation along the Z axis and rotation about the Z axis can occur independently of any other motion. However, translation about the X axis (or Y axis) and rotation about the Y axis (or X axis) are coupled motions. Therefore, in the analysis of a block, we have to concern ourselves with four types of motions. Two motions are independent and two are coupled. For determination of the natural frequencies, in coupled modes, the natural frequencies of the system in pure translation and pure rocking need to be determined

The results with wmv or animation file of modal analysis or ODS can provide a clear picture of coupled rigid body modes as example below.

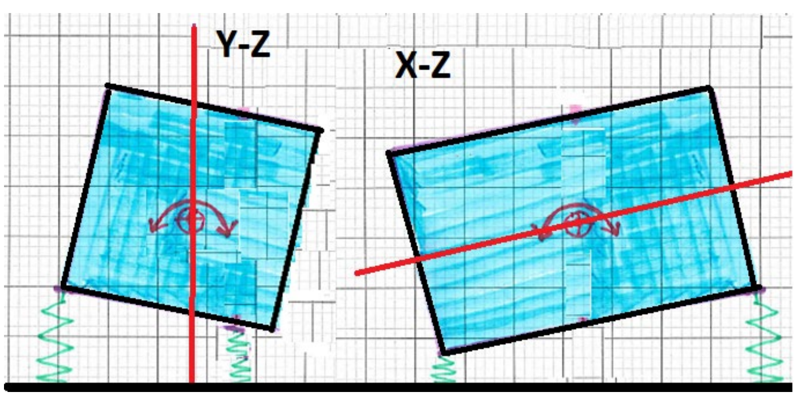

Fig3 - Coupled rigid body modes - most common due to weak structural members

Practiced Solutions -

As we've seen, resonance occurs when the natural frequency of a system coincides with any expected forced vibration frequencies (such as unbalance) which can lead to severe levels of vibration. If it is determined that resonance is in fact the cause of excessive vibration, what can be done to stop or minimize the effect of a resonant condition?

The natural frequency of a system is dependent upon two main factors; stiffness, and mass. If the natural frequency is ω, $\omega = \sqrt{(k/m)}$.

Where k is the stiffness and m are the mass of base frame. Therefore, in order to change the natural frequency, we need to change either k or m or both.

•Increasing mass or decreasing stiffness will reduce the natural frequency

•Decreasing mass or increasing stiffness will increase the natural frequency

Typically, the objective is to increase the natural frequency such that it is above any expected vibration frequencies. If the natural frequency is above or significantly far away from any expected vibration frequencies the resonance will likely no be excited. This theory forms the basis for any structural redesigns implemented to avoid resonance.

In practice, the following rules also can be used to shift a natural frequency and minimize the vibration response of a system;

- Increasing damping reduces the peak response but widens the response range
- Decreasing damping increases the peak response but narrows the response range
- Reducing forcing amplitudes reduces response at resonance

Tuning of the base frame are mostly done by increasing stiffness k by stiffing the base frame to shift its resonant frequency. This is done by adding structural members or by changing the shape / size of structural member to decrease the deflection / twist in particular direction.

Complications with present day analytical method -

Detection of resonance need special equipment and multichannel vibration analyser with trained personnel to conduct such tests.

The natural frequencies of a structure are related to its design and are greatly affected by the mass and stiffness of the structure. If we made the tines of a tuning fork stiffer, the note made by the tuning fork would increase in frequency. And the same is true for the structure supporting our machine; which is normally what is resonating. If we know that we have a resonance problem we can consider either adding mass to reduce the natural frequency or increasing stiffness to increase the natural frequency. The aim is to change the natural frequency so that it is no longer excited by the machine.

This is a very sophisticated process and there are a number of issues that must be considered. In essence this is what we are trying to do – modify the structure so that the vibration generated by the machine is no longer amplified and therefore harming the machine or the structure

If stiffener / mass is arbitrarily added to the structure, it could easily shift the natural frequency into another range of operation thus making the vibration significantly worse. Calculation of adding mass to the system and shift the natural frequency down to a frequency where the forces are lower is a cumbersome exercise as addition of mass increases the stiffness of that local area.

Hence a validated finite element analysis (FEA) model of a structure is constructed to determine the optimal design changes in order to fix and avoid resonant conditions. Validation of model is done first by free -free resonant test of structure, putting the data to FE model and then correcting the FE model which is a complicated work. This allows to test various different potential design changes in a computer simulation first before recommending any structural changes.

Based on criticality of machine, combination of frequency sweep test, bump test and then creating an animation file of operating deflection shape the resonant frequency and nodal points of deflection can be found. Based on location of nodal and anti-nodal points structural modification can be carried out.

To carry out structural modification, the machine needs to be removed from base frame and the base frame need to be sent to work shop / fabrication shop to undertake controlled welding for adding structural elements to increase stiffness. All these activities may render the unit out of operation for 2-3 months.

Proposed solutions -

If the particular structural member is subjected to lateral compression or tension, its bending stiffness and torsional stiffness increases and this is the basis of proposed solution detailed in forthcoming paragraphs. As we know, structural member undergoes a temporary deflection during a coupled rigid body mode and if that defection is arrested then we increase the directional stiffness of subject structural element.

Two types of controls to restrain the temporary defection are being proposed based on location, criticality and sensitivity of machinery installation.

For **passive control,** some devices shop fabricated devices can used which can mitigate the structural resonance problem. The proposed methods shall try to cover the possible discrepancy on-site grouting issues where a support structure may undergo push pull action. The push-pull phenomena aggravate machinery vibration. The device proposed can provide additional directional stiffness.

The proposal is to design base frame for a horizontal centrifugal machine where the effective stiffness can be increased in a particular direction based on readings taken at installation site itself. The design incorporates devices on the peripheral structural parts through which adequately pre-sized structural element is to be introduced, thus increasing the stiffness and restricting coupled modes of concern such as rolling or pitching which is mainly caused by structural resonance modes of base frame.

To facilitate a base frame tuneable at site, an overall drawing of a proposed base frame is shown in figure 10. To increase the stiffness of cross (traverse) members, additional devise (hollow cylindrical elements) are inserted across the two longitudinal main members.

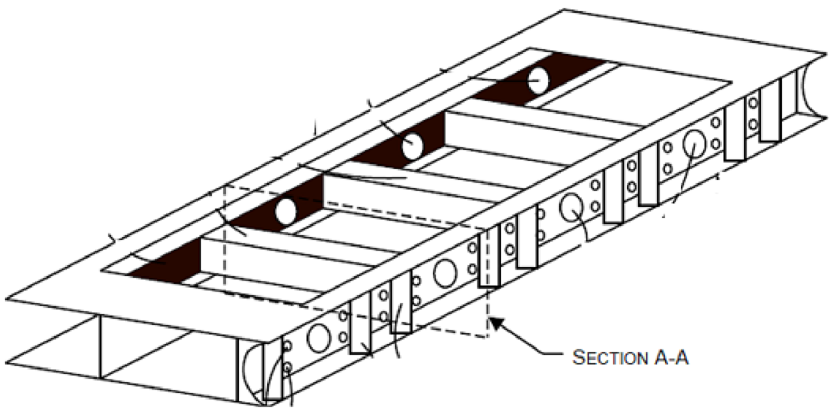

Fig 4 - Base frame with open channel type design

To overcome the rolling mode of base frame and machines during operation, directional stiffness shall be increased by inserting hollow stiff cylinders and then tightening with spring washers as shown in picture. To prevent the cylinder locking device getting loose, tab type washers or locknut can be used.

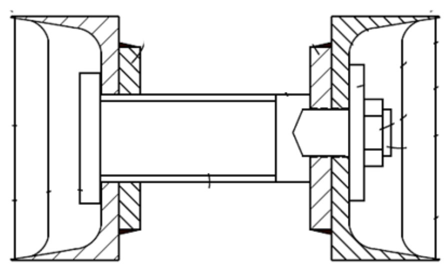

Fig 5 cylindrical Stiffening device attached across channels section A-A

To overcome the pitching motion of base frame and machines during operation, directional bending stiffness shall be increased by a fastening a plate along with the web portion of open sections. This is shown in figure 13.

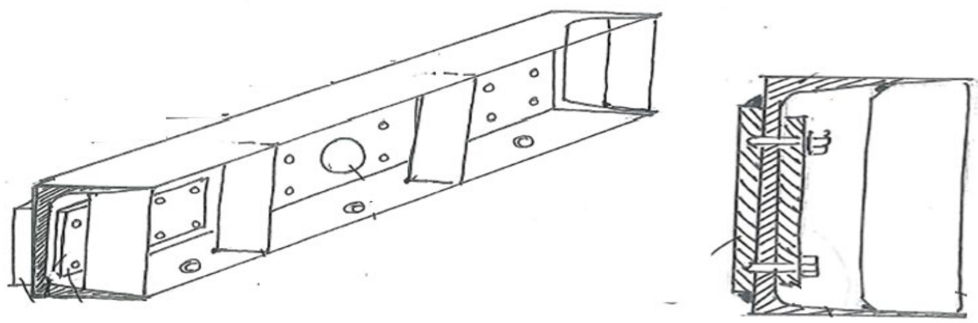

Fig 6 - Stiffening of longitudinal beams

To prevent the rolling and pitching movement of stool mounted motor, a turn buckle type arrangement can be used with circular slot having angular freedom of movement as shown in figure 14.

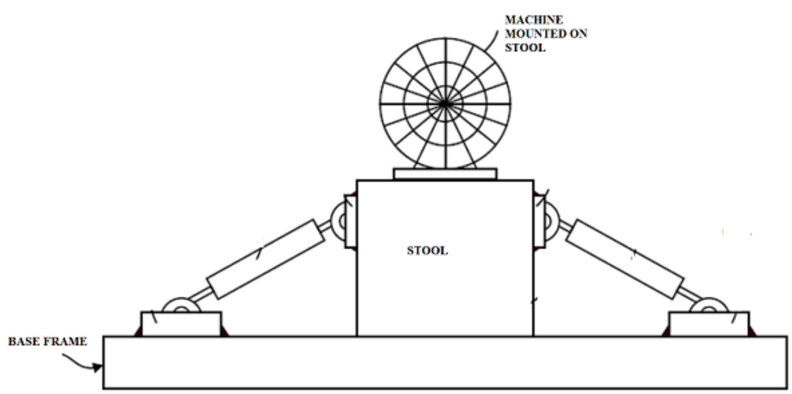

Fig 7- Stool holding device using turn buckle arrangement to prevent swaying

With some fine tuning such as adjusting torque values of bolt, spring washers changing the thickness to higher and lower range as part of supplied device, the machine vibrations resulting due to structural resonance can be brought down to acceptable values.

Active control -

An active control of coupled modes of vibration (rocking and pitching) can be achieved with instrumentation using strain sensors, servo valves and hydraulics by increasing the directional stiffness.

For proposed active control shown in figure, miniature opposed single acting hydraulic cylinders are inserted into hollow part of base frame area just besides the welded cross member.

As we know, structural member undergoes a temporary deflection during a coupled rigid body mode. This deflection shall cause strain and change in force acting on particular direction.

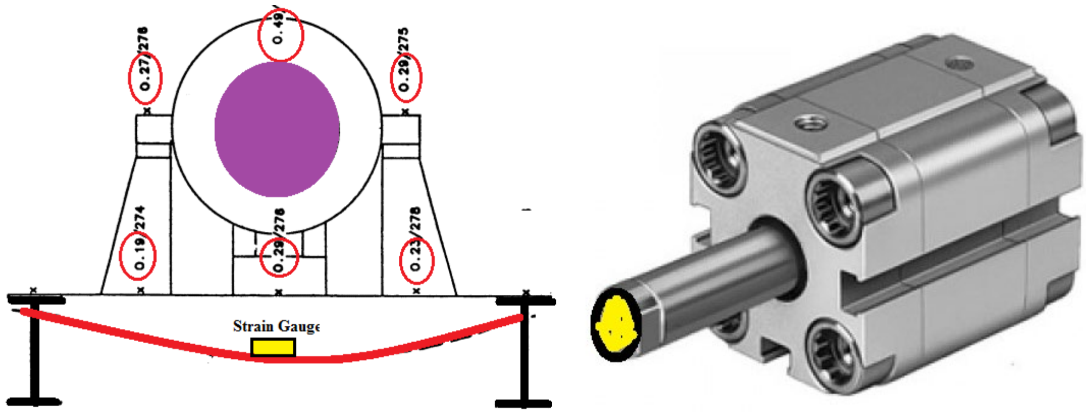

Fig 8 -Left- location of a strain gauge, Right - Miniature hydraulic cylinder

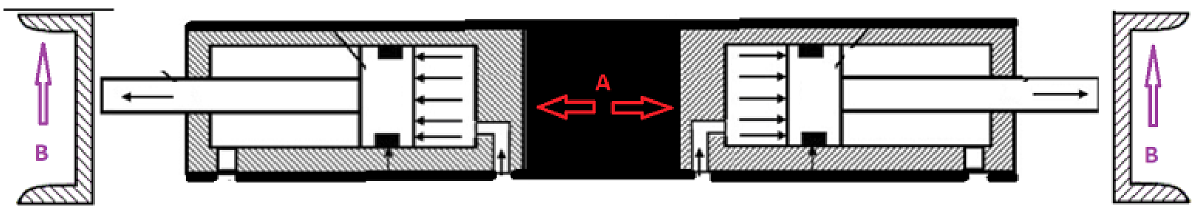

Fig 9 Hydraulic device arrangement with two miniature opposed hydraulic cylinder

So, if the bonded strain gauge tape (deflection / twist sensor) or strain rosette is used on subject structural element then they can give exact position of equipment in terms of deflection and coupled modes of vibration. If the signal output is in form of x-y milliamps, it can control any device based on closed loop feedback control.

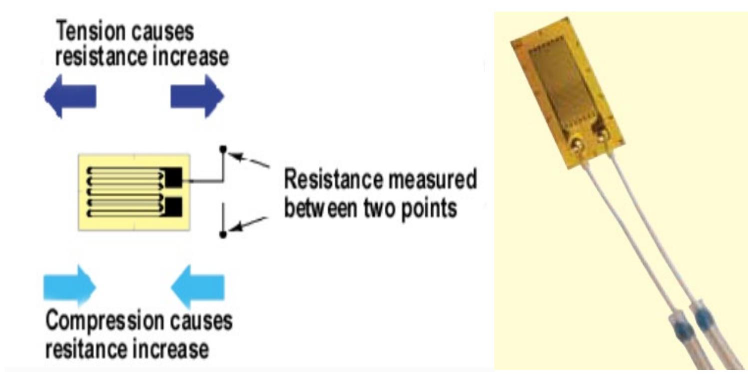

Fig 10 Pre-wired strain gauge

The proposed device uses strain gauge / strain rosette pasted on structural member. A structural member bends by an applied moment or load exerted perpendicular to that member. One side of beam shall be in tension and other side at compression (shape attained by member as convex - concave). A feedback device wired with a signal conditioner shall control directional control valve to create required push action by opposed single cylinder with extended pistons which shall work as additional traverse member. In this proposed design, multiple miniature opposed single acting hydraulic cylinder shall be used. Adjustable force ranging from "negligible" to maximum cylinder capacity, by adjusting the input pressure can be achieved.

If the deflection / strain is found in longitudinal structural elements such miniature single direction hydraulic device can create required vertical stiffness. The strain gauge tape and location of hydraulic device should be in middle of foundation bolt location to have maximum effect.

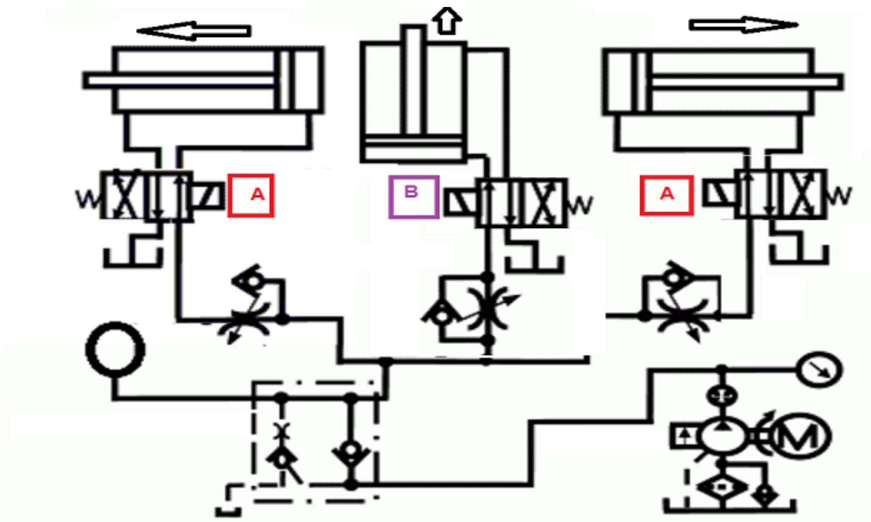

Fig 11 - Hydraulic circuit

Solenoids A shall be connected to banded strain gauges on traverse structural elements to pick command to extend pistons in horizontal directions. Solenoid B shall be connected to force sensor or banded strain gauges on vertical direction of structural elements to pick command to extend pistons in vertical directions to prevent spring action.

Three traverse structural elements and two longitudinal structural elements with hydraulic actuated devices can control any unwanted coupled modes by shifting local stiffness to higher values than operating frequency of machine. Hence when the state of no change in resistance in bonded strain gauge is registered after action by hydraulic cylinders, then system stays put and no action is required. Thus, system becomes completely adaptive to coupled mode due to possible structural resonance.

For controlling rocking mode of a motor or slow operating equipment stool, two hydraulic actuators with hinge arrangement can control the mode as shown below -

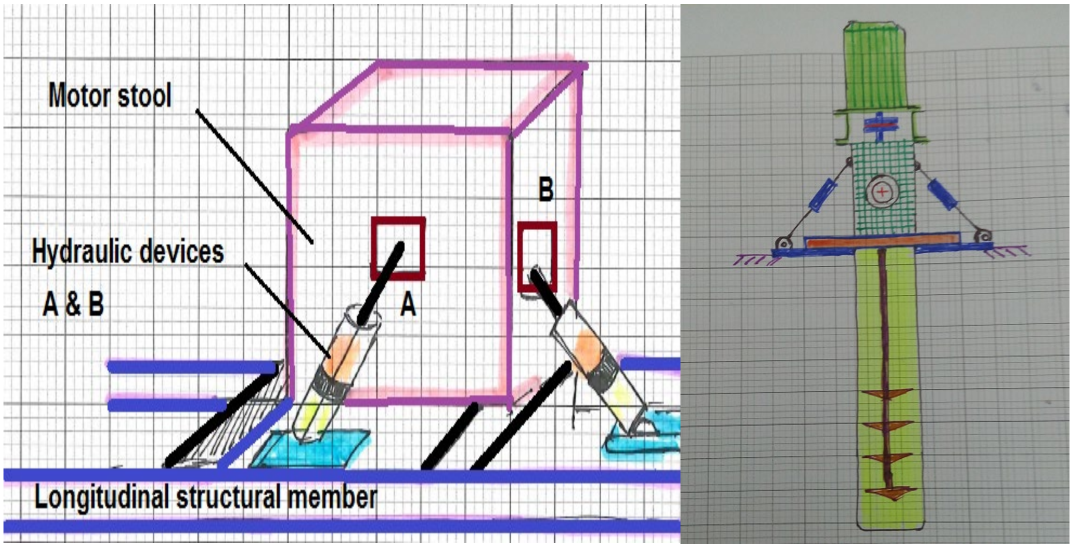

Fig 12 Motor Stool (left), vertical pump (right) rocking and rolling mode control

Conclusion –

The proposed devices can be very helpful on offshore installations or a live gas plant where obtaining with hot work permit is a long exercise. With passive tuneable device, the structural related vibration issues can be greatly mitigated. Even some the cases it is not even necessary to undertake modal analysis.

Active structural resonance control can be a very useful tool to find the location reinforcement required to mitigate structural resonance by adding stiffness to that particular section. For variable speed driven machines, the concept and blue print of active vibration control can be very useful.

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