Control and Mechatronics 322

Active Control of Vibration in Tall Buildings

Paul Livanos,
Nick Raffan,
Matthew Coltrona,

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A/Prof Jie Pan

INTRODUCITON

The control of vibration by vibration absorbers (or neutralisers) is of considerable interest in practical engineering. The dynamic vibration absorber or simply dynamic absorber was invented at the beginning of the 19th century by Frahm (1909), and since then, it has proven to be an indispensable device to reduce the undesirable vibrations in many applications such as gas turbines and engines, ship rolling, helicopters, electrical transmission lines, tall buildings etc. The discrete dynamic absorber was first analysed by Ormondroyd and Den Hartog (1928), and optimum damping was later derived by Brock (1946). Their studies covered a main system consisting of a mass and spring and a dynamic absorber with a mass, spring and viscous damper. Later, Thompson (1981) extended the study to a viscously damped main system.

More recently the past 20 years has seen further advances in control and mechatronics engineering and in particular the application of this engineering. There are many practical applications of vibration absorbers seen throughout the industry and one such example is seen in the control of vibrations in tall buildings. These systems aim to reduce building sway caused by either wind or seismic excitation. The main idea employed in these systems is the use of a tuned mass damper, in which a large mass, placed near the top of the building, is connected to the building with springs and dampers and is 'tuned' to the building's natural frequency. When the building begins to vibrate, the tuned mass damper resonates out of phase with the building, reducing the overall vibration of the building to acceptable levels. The *active* control systems employed, sense the motion of the building and control the motion of the vibration absorber, typically by feedback control, so as to increase the vibration attenuation.

THEORY OF PASSIVE VIBRATION ABSORBERS

Essentially, a vibration absorber is a mass-spring-damper system which, when attached to another mass-spring-damper system, will, to a limited extent, absorb the vibrational energy (usually of a sinusoidal nature) of that system.

Vibration absorbers are tuned spring-mass mechanical oscillators that produce high mechanical impedance at the design resonance frequency at the point of attachment to a vibrating surface, thus reducing the response of the vibrating system at the absorber's resonance frequency. If the absorber is tuned to match the resonance frequency of the vibrating system, the combined resonance is split into two closely spaced resonances. Damping material is included in the absorber to absorb energy at the tuned resonant frequency of the absorber. The damping material broadens the frequency response at resonance while reducing the amplitude of the response.

For ease of understanding let us consider the example of an undamped vibration absorber.

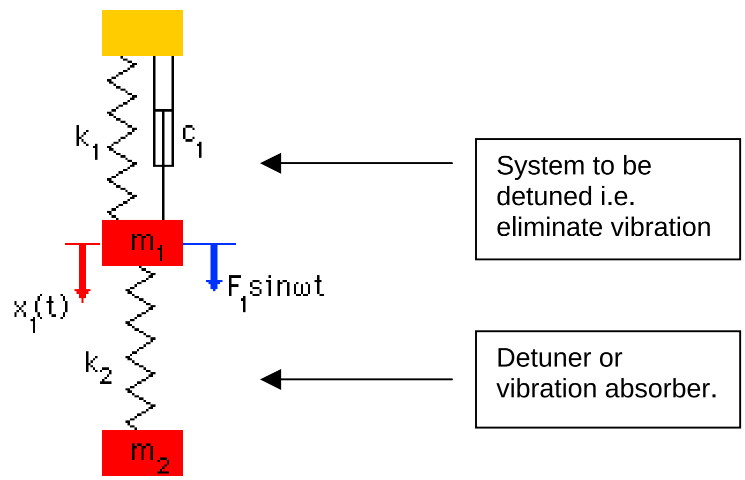


Figure 1: Undamped absorber (Stone 1999).

It can be shown that the steady state solution, $\frac{X_1}{F_1} = 0$ when $\omega = \sqrt{\frac{k_2}{m_2}}$. That is, the detuned frequency is the undamped natural frequency of the lower spring-mass system (Stone 1999), i.e. the vibration absorber.

However as described earlier, damping material is included in the absorber to absorb energy at the tuned resonance frequency of the absorber. The damping material broadens the frequency response at resonance while reducing the amplitude of the response. This can be effectively illustrated by considering the response of a single degree-of-freedom system and how it is modified by both undamped and damped absorbers (Stone 1999).

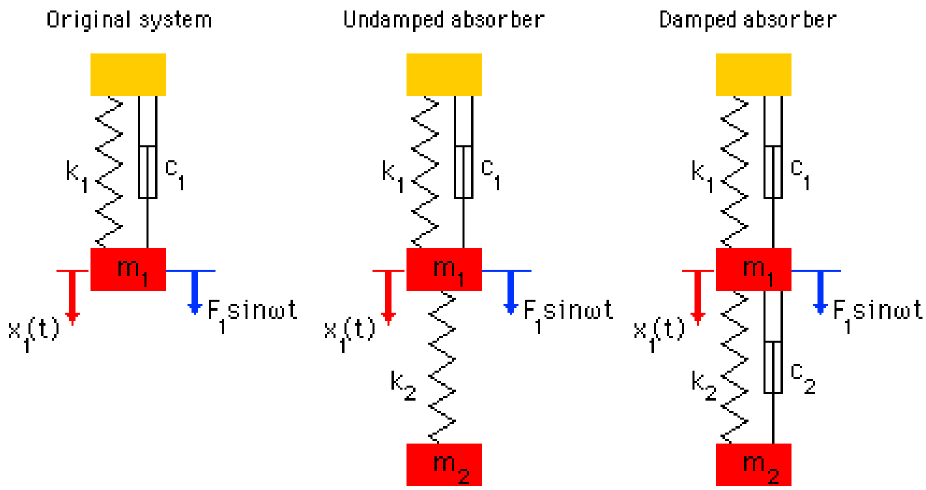


Figure 2: Damped and undamped absorbers (Stone 1999).

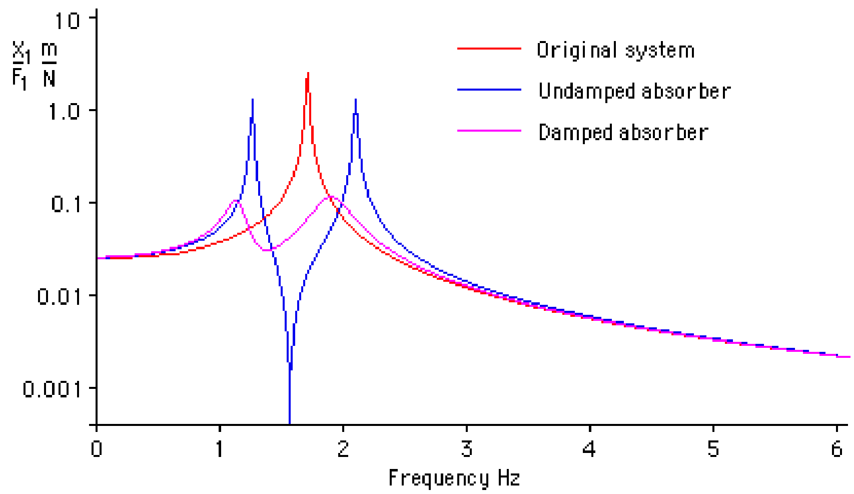


Figure 3: System response to damped and undamped absorbers (Stone 1999).

Here we see that the detuner (undamped absorber) will stop the vibration of the mass at its natural frequency, however either side of this frequency it produces two more resonance peaks. Therefore, the reasoning behind a dynamic absorber (optimised damped absorber) is that it greatly reduces these two resonance peaks while still reducing the original systems resonance.

DESCRIPTION AND PURPOSE OF THE VIBRATION ABSORBER

For ease of understanding, a real world example of a tall building, which utilises a vibration absorber, will be illustrated. The Citicorp Building is one such example. The Citicorp Building is 279 metres high, and comprises 63 storeys (Irvine 2002). The building is made of steel, with aluminium and glass cladding. As a result, it is quite lightweight and has very low inherent structural damping. This means that the building is particularly susceptible to wind induced vibration. William LeMessurier designed a tuned mass damper (TMD) to reduce the effects of wind-excited vibration of the building (Irvine 2002). The Citicorp Building was the first ever skyscraper to make use of a tuned mass damper to reduce its dynamic response to external excitation. It is strong enough to withstand deflections of over two metres at the top floor. However, it is unlikely that office workers would happily work in an environment that shifted this much every few seconds. In addition, if the frequency of the wind excitation should happen to match the natural frequency of the building, resonance could occur and the building could become vulnerable to collapse. Thus, it is necessary to reduce the response of the structure from wind-induced excitation.

The 366 000 kg translational tuned mass damper in the Citicorp Building sits in the crown of the building, on the 63rd floor. It consists of a 9.1 x 9.1 x 2.6 m^3 concrete block resting on a series of twelve 0.6 m diameter hydraulic pressure bearings. The mass is raised to its operating position by hydraulic actuators if the acceleration of the top floor exceeds 0.003 *g* for two consecutive cycles. (Connor 1999). Springs and dampers connect the inertial mass to supports, which are rigidly connected to the building. The tuned mass damper has two sets of springs and dampers at right angles, to absorb vibrations in both directions. It is designed to be biaxially resonant, i.e. it will resonate out of phase with the building in two orthogonal directions. The vibration absorber is capable of reducing building sway by up to 40 %. (Irvine 2002)

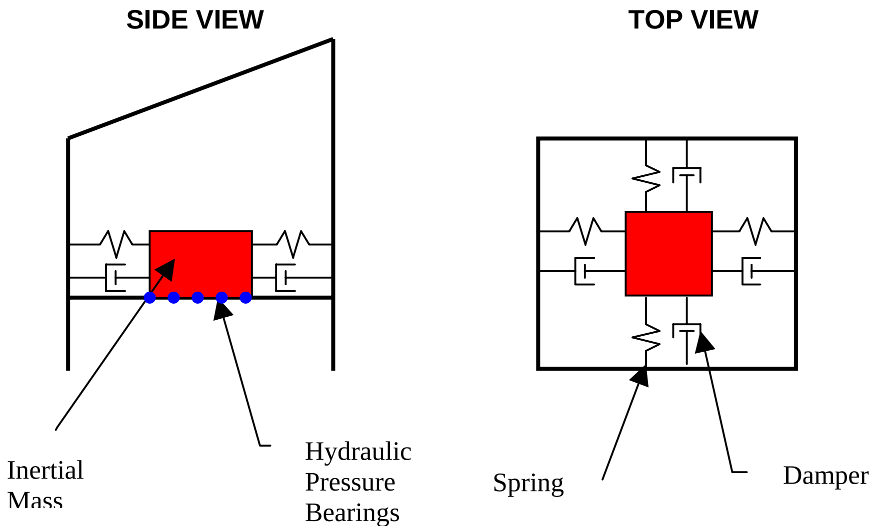


Figure 4. The translational tuned mass damper.

In essence, the translational tuned mass damper works as follows; when the building moves, the inertial mass tends to slide over the hydraulic bearings and remain in the same position relative to earth. This stretches the springs, which then pull the building back towards the inertial mass as they try to revert to their initial position. This has the effect of pulling the building back towards its mean position. Whilst this happens, the dampers are also absorbing the vibration energy.

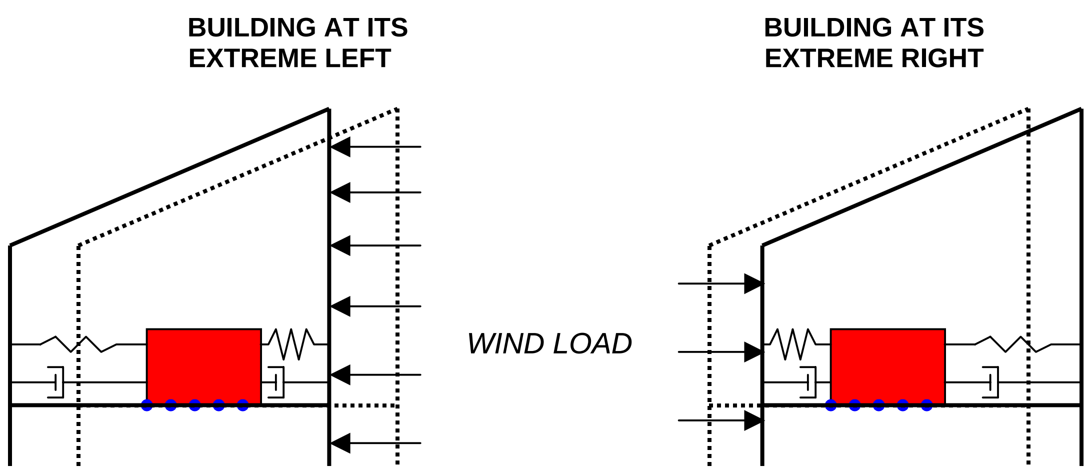


Figure 5. Function of the Tuned Mass Damper.

CONCEPT OF ACTIVE VIBRATION ABSORBERS

With passive absorbers, the tuned condition stipulates that the resonance of the absorber corresponds to that of the structure. The mathematical interpretation of the cancellation is that the absorber and the structure vibrate 180° out of phase, thus leading to zero magnitude of vibration. In more technical terms the addition of the absorber leads to an increase in impedance for the structure at its problem resonance leading to a far smaller magnitude of vibration. Furthermore, by referring back to Figure 3, one is able to see that the tuned passive vibration absorbers are only effective over a specific frequency range. In the case of active vibration absorbers the aim is to produce a system that can vary its frequency of vibration to match that of the structure across a range of frequencies by using feedback and feedforward control systems.

Passive control typically involves a form of structural augmentation or redesign, often including the use of springs and dampers that leads to a reduction in the vibration. Active control aims to reduce the measured vibration levels through the use of sensors, actuators and a control system.

Active vibration absorbers are characterized by a secondary mass, mounted to a vibratory primary system that is directly connected to an actuator and electronic control system. Generally a closed-loop feedback control system is used to control the active vibration absorber although some research has looked at a combined feedforward and feedback control (Yoshida K. et al. 1988). Several forms of active control have been developed including:

- Active Mass Damper (AMD).
- Active Tendon Systems (ATS).
- Active Bracings Systems ABS).

The AMD has been installed in various buildings to control wind-induced vibration and its effectiveness in combating seismic excitation has been studied. ATS and ABS are less widely used in practice.

Numerous applications involving active control of dynamic absorbers have dealt with actuating the absorber mass directly. This study is different in the way that the "active" component is implemented. Here, the actuator will be controlling the stiffness of the spring, thus controlling the natural frequency of the absorber system (the secondary system). Ryan et al (1994) designed an adaptive-passive vibration absorber using a variable spring as the adaptive component. The stiffness was controlled using a spring inserted through a sliding plate, which could then be moved to alter the effective number of coils in the spring, which in turn affected the overall stiffness of the spring. The plate was moved according to the feedback signal linked accelerometer, mounted on the primary vibratory system. This method of control has the potential to be less energy consuming, as the power required to adjust the spring variable is expected to be less than the power required to actuate the mass directly. This method is best described as an adaptive-passive approach to vibration control. Adaptive-passive methods involve the use of passive elements that can be optimally tuned to perform over a certain frequency range. These vibration absorbers have the advantage that the controller is able to guarantee closed-loop stability, even with minimal knowledge of the controlled system (Lee-Glauser, 1995). Unlike a fully active system, this form of control cannot add "negative" damping to an existing system (i.e. the poles of the system cannot migrate across the imaginary axis). Thus a system that is already stable cannot be made unstable through the use of this.

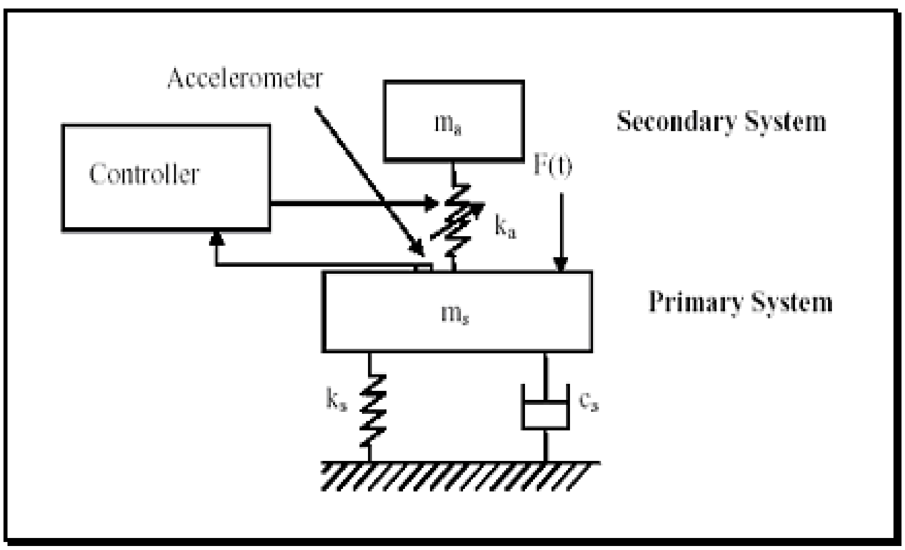


Figure 6: Representation of an Adaptive Vibration Absorber (Ting-Kong 1999)

A feedback controller is designed to generate an output that causes some corrective effort to be applied to a process so as to drive a measurable process variable towards a desired value known as the set point. Shown is a typical feedback control loop with blocks representing the dynamic elements of the system and arrows representing the flow of information, generally in the form of electrical signals. Virtually all feedback controllers determine their output by observing the error between the set point and the actual process variable measurement

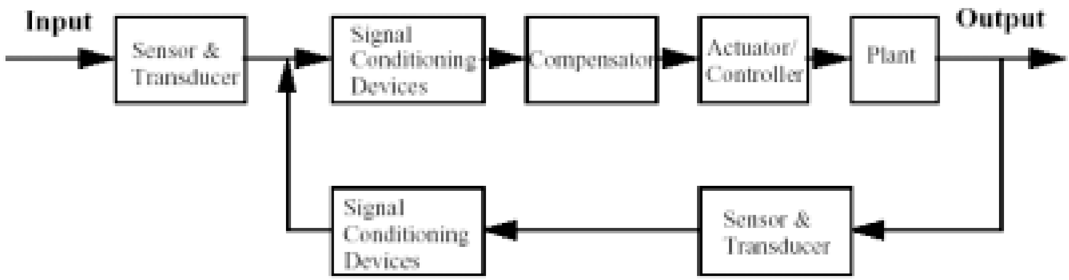


Figure 7: Typical Feedback Control System.

Although active control of structures has its benefits it is not without its problems. These include:

- Robustness of the control device.
- Time delay effects.
- Cost-effectiveness.
- Optimum placement of the control device.

To overcome or limit these problems more research is needed on the subject.

ANALYSIS OF THE CONTROL SYSTEM

Instrumentation of the Control System

The system designed to control vibration in tall buildings comprises several devices. Although there are some variations in control systems for different individual cases we will look at a common form of the control system used. Many of the devices used are typical of what is contained in any control system. The following contains a description of each instrument and its effect on the system.

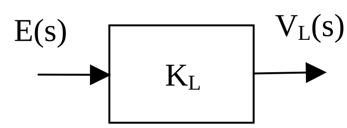
<u>Sensors</u>

The role of the sensor is to measure the input and output signal. For this case we want to measure the frequency of vibration for the entire building and for the secondary mass.

An accelerometer is used to measure the excitation frequency for the entire building. A linear transducer is used to measure the frequency of the secondary mass.

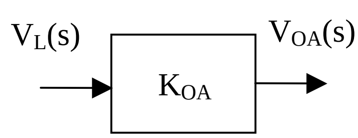
Linear Transducer

The position of the absorber mass at any point in time is measured by incorporating a linear transducer into the system. A linear transducer consists of a rotational pot that gives out a voltage variation depending on the number of turns it has completed. The number of turns completed will affect the number of coils in the spring. This alters the stiffness, k and in turn the frequency of vibration. It can be shown that there is a linear relationship between the voltage output from the transducer and the natural frequency of the dynamic absorber.



Comparator

Once the input and output has been measured and the values have been converted to voltages the two values need to be compared. To do this a differencing Operational Amplifier is used. A diagrammatic representation of the amplifier and its block diagram can be show,



Compensation

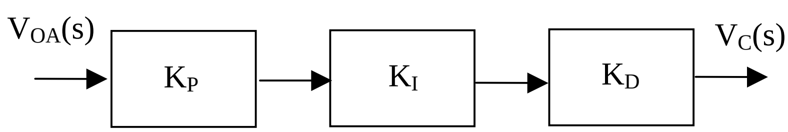
Once the signal is passed through the operational amplifier it is now time to add compensation to the system. Critical to any control system is its performance. Performance is measured in terms of stability, accuracy and speed of response using values such as the steady state error, subsidence ratio and settling time. To improve the performance of the system compensation is included. For this system it is common for PID controllers to be utilized. Other forms of compensation such as fuzzy logic and least square predictive have been used but are less common. The PID controller is a combination of three separate devices.

Proportional Controller: The role of this controller is to affect the speed of response. As the gain is increased the response time of the system will increase. However the gain cannot be increased too far as this will cause system instability and a high overshoot,

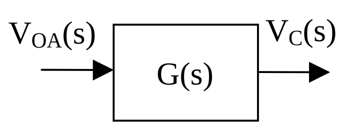
Integral Controller: This is attached to the system to eliminate the steady state error however it has the negative effect of slowing down the speed of response. Therefore a trade off between the two measures of performance is required.

Derivative Controller: This controller affects Damping of the system. If appropriately designed, this controller will cause the system to be critically damped.

The PID controllers can be represented as,



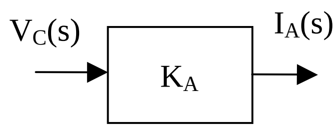
This can be simplified to,



Where, $$G_c(s) = K_p + K_D s + \frac{K_i}{s}$$

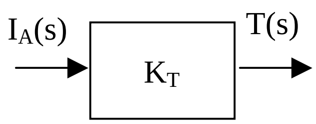
<u>Bi-directional Servo amplifier</u>

A motor can be controlled by current or voltage. If an output torque is required then the motor requires a current input. Therefore for this system a servo amplifier is added to convert the voltage output from the PID controllers to a current. This can be represented as,



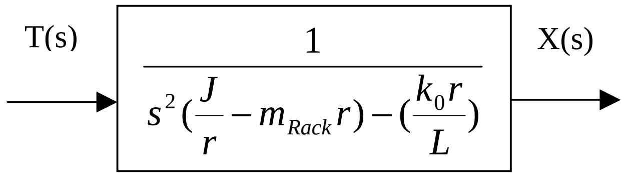
<u>Actuator</u>

The role of the actuator is to transfer energy from electrical energy to a mechanical form of energy. In this case we a use a DC motor to transform the electric current to a torque. This can be represented by,



<u>Plant</u>

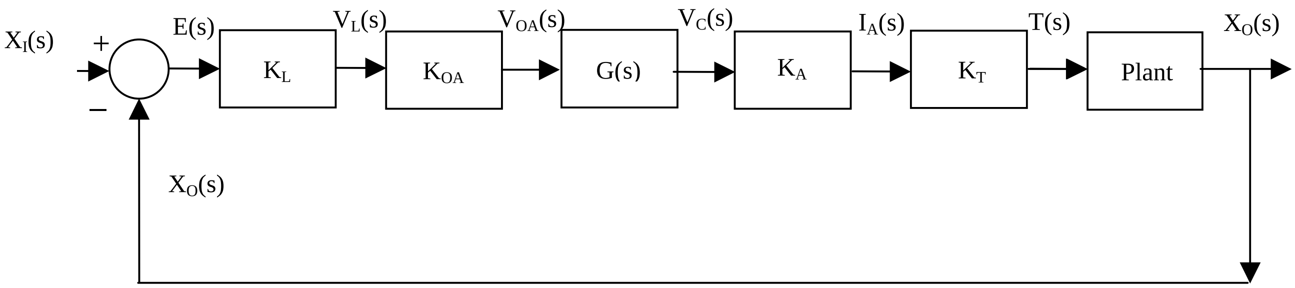
The mechanics of how the control system converts the Torque supplied by the motor into the linear motion required to alter the length of spring, in turn altering the stiffness of the spring, in turn altering the frequency of vibration of the vibration absorber is described below.



Calculation of this transfer function is detailed below.

<u>Complete Control System</u>

Combining all the elements that have been mentioned produces the complete diagram of a typical system for controlling the vibration absorbers in large structures.

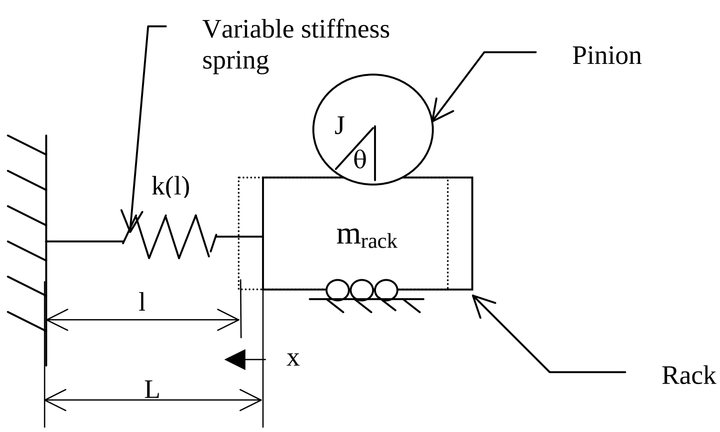


The circuit is a closed loop feedback control system with compensation. Using this block diagram the system can now be analysed.

Calculation of transfer function for the motor and rack system used to alter the spring stiffness.

Essentially, the system works as follows;
Firstly the frequency of vibration of the building is measured. A computer package will then be utilised to calculate the required length for the vibration absorber in order for it to alter the effective length of its spring, which will in turn alter the spring's stiffness and therefore alter the natural frequency of the vibration absorber. The length calculated by the computer package will then be the input into the control system. The control system will process this information and without going into the detailed workings of how the spring's length is altered, a motor will move a rack by the required distance.

The system may be modelled as follows:



Where:

J is the moment of inertia of the motor and pinion

k(l) is the stiffness of the spring which varies with its length l

L is the initial length of the spring

k_0 is the initial stiffness of the spring

x is the linear displacement of the spring (output)

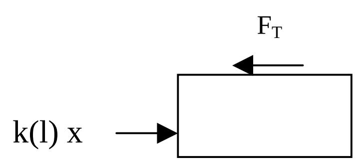
θ is the rotational displacement of the pinion and motor arm

For brevity, it is assumed that the spring stiffness varies linearly with its length l (i.e. the motion of the rack reduces the number of coils in the spring).

Thus, k(l) may be written as:

$$k(l) = k_0 \frac{l}{L} = k_0 (\frac{L-x}{L})$$

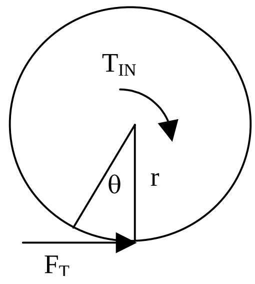
Summing forces horizontally for the rack yields the following equation:



$$k(l)x - F_T = m_{Rack}\ddot{x}$$

Where F_T is the tangential force that the pinion exerts on the rack.

Similarly, a relationship may be found between the input torque (T_{IN}) and angular displacement of the pinion and motor shaft:



$$T_{IN} - F_T r = \frac{J\ddot{x}}{r}$$

Note also that:

$$x = r\theta$$
$$\dot{x} = r\dot{\theta}$$
$$\ddot{x} = r\ddot{\theta}$$

Combining these equations together yields:

$$T_{IN} - [k_0(\frac{L-x}{L}) - m_{Rack}\ddot{x}]r = \frac{J\ddot{x}}{r}$$

This equation may eventually be written as:

$$\ddot{x}(\frac{J}{r} - m_{Rack}r) + x(\frac{-k_0 r}{L}) = T_{IN}$$

Taking the laplace transform of this function, we have:

$$(\frac{J}{r} - m_{Rack}r)s^2 X(s) + (\frac{-k_0 r}{L})X(s) = T_{IN}(s)$$

Rearranging this gives the transfer function in the s domain:

$$H_{Rack} = \frac{1}{s^2(\frac{J}{r} - m_{Rack}r) - (\frac{k_0 r}{L})}$$

Thus, the overall compensated open loop transfer function may be written as:

$$H_{CO} = \frac{K_L K_{OA}(K_P + K_D s + \frac{K_i}{s})K_A K_T}{s^2(\frac{J}{r} - m_{Rack} r) - (\frac{k_0 r}{L})}$$

Then the compensated closed loop transfer function for the overall system will be given by:

$$Hc = \frac{H_{CO}}{1 + H_{CO}} = \frac{K_L K_{OA}(K_P + K_D s + \frac{K_i}{s})K_A K_T}{s^2(\frac{J}{r} - m_{Rack} r) - (\frac{k_0 r}{L}) + K_L K_{OA}(K_P + K_D s + \frac{K_i}{s})K_A K_T}$$

This transfer function may now be analysed to optimise the response of the system, whilst allowing for stability considerations.

CONCLUSION

Active vibration control has become an integral division in the field of vibration analysis. Through the use of control systems the once commonly used passive method for controlling unwanted vibration in tall buildings has been superseded by the newer active methods. Vibration caused by wind and seismic excitation can be controlled by the use of active mass dampers in conjunction with an appropriately designed control system.

This project has outlined the history and theory behind vibration control of tall buildings. Initially the dynamics of the vibration was described, then the system used to control the vibration absorber was outlined and analysed. This included a description of the individual devices that make up the control system. Whilst further work is required to refine and improve the field of active vibration control, it has shown to be an adequate method for cheap, reliable control of vibration in tall buildings.

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