Theoretical Study of Modified Secondary Dynamic Vibration Absorber Concept to Damped Out Forced Vibration of Mass to Minimal Which is Induced by Excitation of Support

Mayur Rajendra Waghchoure[1],
Former Associate Manager,
Commercial vehicle business unit,
Tata Motors Ltd., Pune, India[1].

Paresh Kothawade[2]
[2] Senior Manager,
Engineering research center,
Tata Motors Ltd., Pune, India.

Abstract **- In this world surrounding must be chaos free for accurate working of everything, otherwise anarchy in control would lead to unbalance and discomforts of system. For body subjected to forced vibration due to excitation of support can be bring back to steady state by using spring-mass-damper system over certain period of time. But as excitation frequency of support varies over time period then fixed combination of spring stiffness K1 and damping coefficient C1 won't help for long or for short range only. Main body senses severe or mild shock based on deviation from required value of this parameter. But to achieve minimal vibration of main body irrespective of support excitation, we can use concept of secondary vibration absorber but again it has bandwidth issue. To dealt with this problem I have design secondary vibration absorber in such way that it has flexibility to change its mass m2 by passing fluid through pipe or to change spring stiffness K2 such that basic condition of frequency match of support excitation and secondary mass will achieve for entire time period with flexibility. As well to make main body free from external excitation force Fo irrespective of cycle frequency spring force can be vary by using various combination of springs. In this paper I have mention basic principle and mathematical formulation to achieve complete control over vibration of main body and that to within economical scale as no expensive material is used for this purpose. Finally, we have concluded that this secondary vibration absorber replete with accurate control can be use in automobile, machine platform and other application for vibration isolation purpose.**

Keywords: Vibration, Dynamic vibration absorber, forced vibration, Automobile suspension system, Mathematical formulation etc.

1. INTRODUCTION

A body of mass m1 rests on the support with damper of damping coefficient C1 and spring of stiffness K1. In this case when main body is displaced from equilibrium position by application of external force with excitation frequency of w then for resonance condition of main body with excitation frequency, given body will start oscillating with very large amplitude. As well for small excitation force cause from support displacement will be large. Hence by using current mass-spring-damper system vibration can be damped out in certain cycles but which means that body has to go through all these vibrations before achieving steady state. Here currently most of suspension systems used in automobiles is either leaf spring type for heavy vehicle and helical spring type for light vehicle.

With advancement of technology all this systems are designed in such way that vibration induced in main body should damped out as early as possible by using under-damped system. But all these systems are giving poor performance as all this system are design for certain bandwidth of input excitation frequency w as well this system performance cannot be altered as prevailing conditions of input excitation. Now days with the introduction of smart materials and electronic controls, variation in performance can be achieved either by varying damper speed to vary damping force to cope with exact amount of excitation force F0 or by varying damping coefficient of damper by varying viscosity of smart fluid with application of electric current or by magnetic field. Still option with velocity variation is viable at low cost but again it is local control means intermittent whereas if excitation frequency varies cyclically then it might not give good performance and second option can cover range of input frequency of excitation but it will be costly due to smart material involvement.

To address this problem with care concept of dynamic vibration absorber came into existence, in which secondary mass m2 attached by spring of stiffness K2 to main body and it will take entire vibration from support by itself and vibrate freely and provide zero vibration to main body theoretically. This concept has problem as mass m2 and spring constant K2 cannot be varied as per input frequency as this dynamic secondary vibration absorber gives good performance only when natural frequency of secondary vibration absorber is equal to frequency of external excitation. The main principle of this concept of secondary vibration absorber is as to increase this bandwidth of input frequency to give ideally zero vibration to main body but practically it would be minimal at least; there should be provision to vary the mass of secondary body m2 and stiffness of spring K2 with variation in input frequency w and this can be achieved with this secondary vibration absorber concept.

International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 7 Issue 08, August-2018

2. SECONDARY DYNAMIC VIBRATION ABSORBER CONCEPT

2.1 Layout

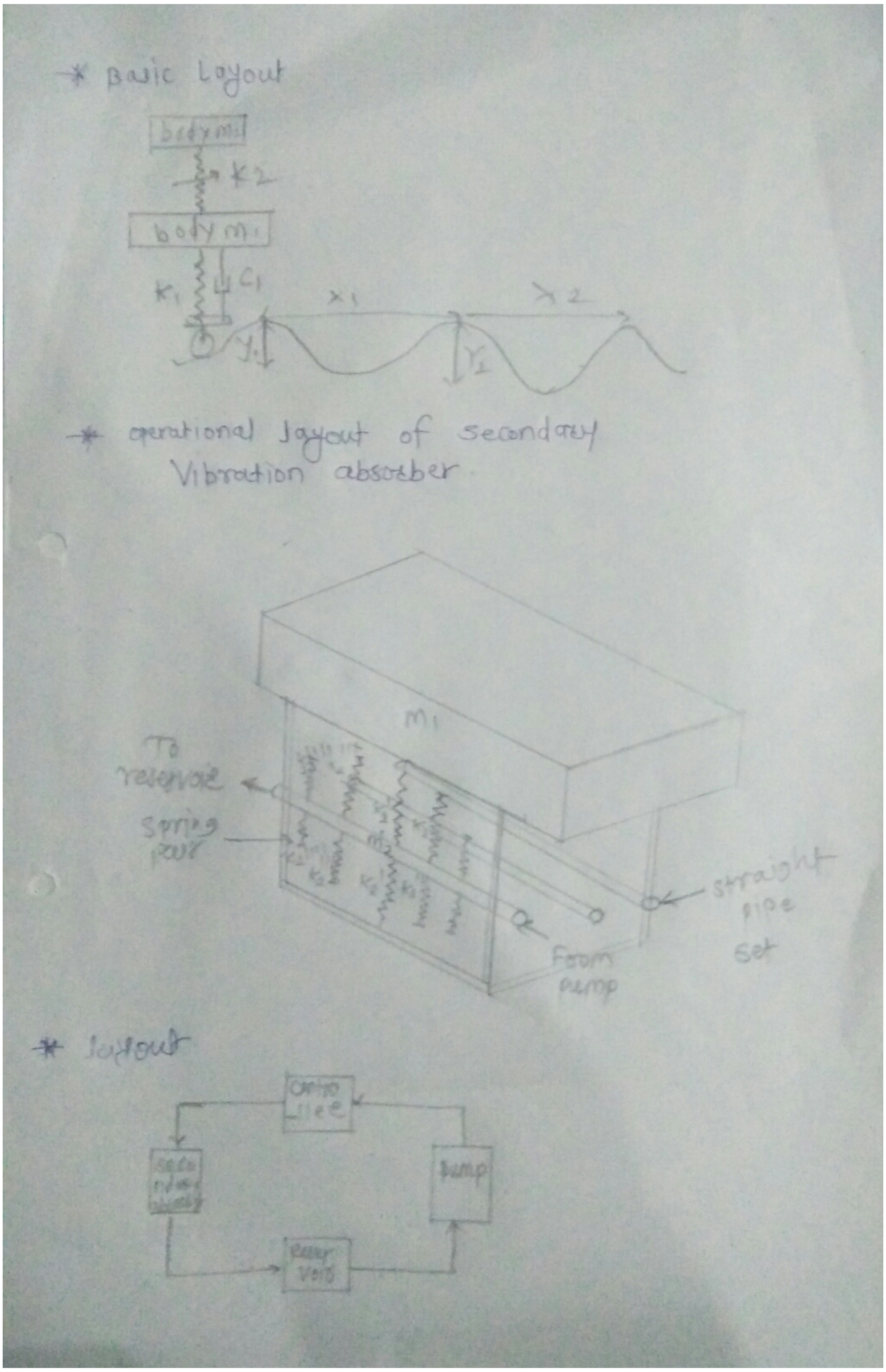

Figure 1. Modified Secondary vibration absorber concept and design layout

2.2 Construction

1. This secondary vibration absorber should be connected to main body of system.
2. It should have either hollow helical coiled body or straight pipes connected through pairs of springs of stiffness $K2^1$,$K2^{11}$,….. etc. in parallel between main body and secondary mass m2 in such way that each pair of spring will come into operation after certain displacement to increase stiffness of secondary system due to parallel arrangement and spring should be rigidly connected main body except lateral pair of spring, these spring will come into operation only when certain displacement has achieved as per following relation

$K2= 2K2^1+2K2^{11}$ +……….

3. This hollow helical tube or hollow straight pipe should be connected through pump and reservoir through flow control valve and variation in flow rate controlled by controller will bring change in mass m2 of secondary vibration absorber.
   Here $m2=m2^{1}*time$
   As $m2^{1}$= mass flow rate through system
   Here working fluid could smart material or any fluid available at low cost like water.
4. In this way this small unit coupled with controller taking input from speedometer to sense speed variation and sensor to sense change in excitation amplitude if excitation source is road. In this case velocity variation and wavelength variation will give current excitation frequency.

2.3 Working

1. When excitation frequency change dw is noted from sensor then controller will adjust flow rate by actuating flow control valve to maintain the natural frequency dwn2 of secondary vibration absorber as equal to dw.
2. In this way by varying mass flow rate bandwidth of input frequency will enhance.
3. Frequency change of excitation source can also be control by either varying mass flow rate through secondary straight pipes or by varying damping coefficient of damper C1 between main body and support as theoretically calculated in this paper.
4. Either combination of variation in dC1 and variation in dm2 or by using any of them alone vibration of main body can be damped out completely.
5. Displacement X2 of secondary vibration has relation to excitation force as excitation force will completely neutralize by spring force as
   Fo=Excitation force= K2*X2
6. As Fo changes with wavelength variation and amplitude variation of road and to compensate for this spring force should be changed by either varying K2 or varying X2. Displacement X2 has some limit for displacement so to change K2 with requirement will be favorable option and which will be done by engaging different spring for different displacement.
7. In this way input excitation frequency maintain equals to natural frequency of secondary mass m2 by varying flow rate $m2^{1}$ and by varying spring stiffness K2 and excitation force Fo variation is achieved by varying K2. So no force will be acting on main body m1 as well cycle gives best performance for resonance of w and wn2 and main body will have vibration if all system combinations are executed properly.
8. Here Fo value is mostly taken by current available damper, which are using two fluid to counteract this force and variation above certain level can be balanced by varying K2 and m2. In this way complete balance can be obtained and vibration would reduce to minimal.

2.4 Mathematical formulation for secondary vibration absorber and Results

1. Secondary vibration absorber system is two mass system and governing equation for it as follows:

Here for mass m1, $m1*X1^{11}+K1*(X1-X2)+C1*X1^{1}=Fo*Sin(w*t)$
For mass m2, $m2*X2^{11}+K2*(X2-X1)=0$

2. For better performance of this secondary absorber and to give X1=0
   Excitation frequency of support= natural frequency of mass m2
   Here w=wn2=(K2/m2)^.5 …equation 1
   That is if excitation source is road with amplitude Y and wavelength λ then angular frequency of excitation is given as w=2*pi*V/ λ
   On differentiating we get that
   As dw={ λ*(2*pi*dV)-2*pi*V*d λ}/ λ^2 ….. equation 2
3. Here for better performance and to meet condition posed by equation 1, we must change mass m2 and stiffness K2 as follows
   On differentiating equation 1, we get that
   As dw={[-K2*dm2/2*w*m2^2]+[dK2/2*m2*w]} …..equation 3
4. As well displacement of main body X1with excitation Y from support then X1 is given as
   X1=Y*(1+2*zeta*(w/wn1))^.5
   For X1=0 we get that
   As w=±(1/2*zeta)*wn1
   As w=±(Cc/2*C1)*wn1
   Cc=critical damping coefficient
   On differentiating we get that
   As dw=(wn1*Cc/2)*(-1/C1^2)*dC1 …….equation 4
   As practically C1 can only be vary by varying viscosity of fluid as in smart material or by changing damper dimension.
5. In this way to satisfy the condition of equal frequency of support and secondary body, we can achieve dw=dwn2 by following practice of using either equation 3 and 4 together or any of them alone.
   From equation 3, equation 4 and equation 5, we can write as
   As dw=dwn2={ λ*(2*pi*dV)-2*pi*V*dλ}/ λ^2 ={[-K2*dm2/2*w*m2^2]+[dK2/2*m2*w]} = (wn1*Cc/2)*(-1/C1^2)*dC1

Hence combine by controlling mass flow rate m2 or by controlling variation in C2 we can match frequency and frequency of main body can turned to zero.
In this way by controlling this entire variable shown with yellow background as sensing change in variable with red background and controlling few will bring steady state to main body as X1=0

6. Whereas to avoid unbalance of forces on main body arise from excitation support Fo should be damped out by spring force of secondary mass system, at steady state relation as
   Fo = -K2*X2
   On differentiating we get
   As dFo=-X2*dK2-K2*dX2 ….. equation 6
   As well excitation force Fo due to amplitude Y of support would be as follows
   Fo=Y*(K1^2+C1^2*w^2)^.5
   On differentiating we get that

As
dFo={[2*Y*dY*(K1^2+C1^2*w^2)]+[(2*C1^2*Y^2*w*dw)]+[2*C1*w^2*Y^2*dC1]}/2*Fo

On putting value of dw from support in terms of velocity and wavelength we get it as

dFo={[2*Y*dY*(K1^2+C1^2*w^2)]+[(2*C1^2*Y^2*w*(λ*(2*pi*dV)-2*pi*V*dλ}/ λ^2))]+[2*C1*w^2*Y^2*dC1]}/2*Fo ……equation 7

7. As well current technology has excitation force damped out by damping force as follows $Fo=C1*X1^{1}$
   On differentiating we get that $dFo=C1*dX1^{11}+X1^{1}*dC1$ …..equation 8
   As far now we can just control acceleration of damper and by doing that we are opposing excitation force Fo
8. In this way equation 7 tells that wavelength, velocity, damping coefficient and amplitude can control and act as input to produce excitation force Fo and it can be opposed by secondary absorber by varying spring stiffness K2 and displacement X2 as predicted from equation 6. As well by controlling damper speed can also opposed excitation force but secondary vibration absorber will do it in cyclic mode. So we can write it As
   dFo={[2*Y*dY*(K1^2+C1^2*w^2)]+[(2*C1^2*Y^2*w*dw)]+[2*C1*w^2*Y^2*dC1]}/2*Fo=
   $C1*dX1^{11}+X1^{1}*dC1= -X2*dK2-K2*dX2$

3. CONCLUSIONS

1. As in secondary vibration absorber by varying mass flow rate by sensing speed and wavelength, we can maintain change in external excitation frequency w as equal as natural frequency wn2. As this condition is satisfied then main body will have no vibration and performance will be high depends upon how fast mass flow rate variation achieved and to get this done many sets of straight hollow pipe should be installed in system with booster.
2. As external excitation force is completely damped out by spring force and that to in cyclic manner by varying stiffness K2, then main body will have no unbalanced force acting on it and complete balanced steady system can be achieved.
3. This system can be develop as small package along with required sensor for speed and road profile reading along with controller to damped out vibration of system induced by support excitation in Automobile, machine mounting without vibration isolation, precision machine and many more application where vibration are linear in motion.

4. ACKNOWLEDGMENT

I want to extend my sincere thanks to Dr. Amar pandhare and Mr. Paresh Kothawade for their constant guidance throughout my work. As well I want to extend my sincere thanks to my parent, Rohit, Rohan, Eknath, Akshay Waghchoure, Varsharani Patil and my friends for cheering they gave me during my work.

5. REFERENCES

[1] M. Brennan, 1997, “Vibration control using a tunable vibration neturalizer,” Journal of Mechanical Engineering Science, 211, 91–108.
[2] M. Brennan and J. Dayou, 2000, “Global control of vibration using a tunable vibration neutralizer,” Journal of Sound and Vibration, 232, (3), 585–600.
[3] T. D. v. D. type DB data sheet, “Produced by Dulmison,” .
[4] L. Mocks, 1984, “Damping of high-voltage overhead line conductor vibrations,” VDEVerlang Gmbh, Berlin und Offenbach, Bismackstrasse, D-1000 Berlin12.
[5] J. Tompkins, L. Merrill and B. Jones, 1956, “Relationships in vibration damping,” AIEE Winter General Meeting, New York, January 30 - February 3.
[6] P. Walsh and J. Lamancusa, 1992, “A variable stiffness vibration absorber for the minimisation of transient vibrations,” Journal of Sound and Vibration, 158, 195–211. 7. B. Korenev and L. Reznikov, 1993, Dynamic Vibration Absorbers, Wiley: West Sussex.