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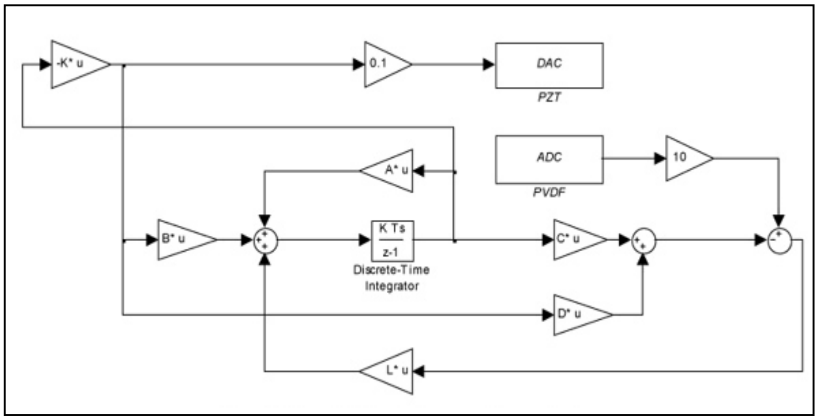
SYSTEM IDENTIFICATION AND VIBRATION CONTROL OF A MECHANICAL SYSTEM (SINGLE DEGREE OF FREEDOM) BY PID CONTROLLER



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ABSTRACT:

Whenever an object is set into motion, vibrations are produced. And in real life, these vibrations pose a serious effect to the system. As we know that layman efforts to curb the vibrations according to need are manual (like spring damper etc), which are ineffective, imprecise, and difficult. So we thought of developing an intelligent control system, which can deviate the vibrations almost instantly and precisely and could also control the degree of deviation. That's why, to control the vibrations, a PID controller was proposed in order to increase the life of a desert cooler and further, decrease the noise pollution. In this experiment, a single degree of freedom system was employed and parameters of the system (desert cooler) at that one particular point were found with 'system identification technique'. After calculating the system parameters, a second order differential equation was obtained, then by taking its laplace, transfer function was obtained. Then this equation's transfer function was used in PID control toolbox (Matlab 2014a) to control and stabilize the vibrations. To be specific, gains, namely proportion, derivative and integration were altered or tuned with the help of PID tuning toolbox to get the top notch values. This technique will prove to be a milestone in designing a self stabilising and a better acoustics desert cooler. This can be also used in customising various appliances comprising of moving parts such as washing machines, dish washers, mixer grinders and many more. In order to accomplish these, real-time actuators to be interfaced with the PID control systems will be used.

KEYWORDS- *Vibration, PID, Matlab, PID tuning, System identification, oscillations.*

I. INTRODUCTION

Vibrations are oscillations in mechanical dynamic systems. Although any system can oscillate when it is forced to do so externally, the term “vibration” in mechanical engineering is often reserved for systems that can oscillate freely without applied forces. Sometimes, in engineered systems, these vibrations cause minor or serious performance or safety problems. For instance, when vibrations of an airplane are excessive, people in the airplane become uneasy, when the natural vibrations of the human body and organs resonate with frequencies of vibration of an airplane. All the bodies are capable of vibration, which possess mass and elasticity. To some extent, many engineering appliances and buildings encounter vibrations and their design needs modifications.

Similarly, a mechanical device, desert cooler which is widely used in subcontinent was chosen. It is responsible for a lot of noise pollution and it often gets weared out easily, because of fatigue loading caused by constant usage. The main sources of vibrations are rotation of motor, design of wings and air drift.

We used single degree of freedom of system to employ PID, as SDOF is easy to visualize and analyze, and it is the most basic problem. PID Controller works on the basis of how much steady state error is there and fix it to the desired value with the help of proportionality, integration and derivation [1]-[3]. Suppose there is an error between the input we supplied and the output we got. Now from the feedback system this error will be made to go through PID until the error gets minimised. In PID, proportionality to reach the steady state increases with increase in the error. More the error more faster the error will reduce. Proportionality is given by error multiplied by proportional gain. Output will be reached fast i.e. rise time will be less but its max peak overshoot rises and there is a problem of offset. Now coming to integral, when it is employed, the output of the system depends upon the integration of the error signal. By adding integral controller, 'type' of system increases and steady state error increases but stability decreases. Derivative controller produces an output which is derivative of error signal. It gives output according to the slope of the error signal. By adding derivative controller, signal decreases. Due to which steady state error may increase but stability increases.

The trickiest part in this whole process was to calculate the system parameters by system identification method [4]. System identification is a graphical technique in which a plot between amplitude and angular frequency is made to calculate the system parameters. Amplitude is calculated with the help of a tri-axial accelerometer and values in the most dominant axis were averaged to get the desired value for a given frequency. Whereas, frequency is calculated from RPM, which in turn was calculated with the help of a tachometer.

However, desert cooler is just an example, this particular method can also be used on many other machines to control vibrations.

I. THEORY AND CALCULATIONS

First of all, mathematical model of the selected system is to be generated .Various techniques can be adopted for this purpose. One of them is System identification technique [4]. To apply this technique the amplitudes of the vibrating desert cooler, at different revolutions per minute, were required. This data was collected using a 3 axis accelerometer which was, further connected to the vibration data acquisition system.

An accelerometer is an electromechanical device employed to acquire degree of acceleration. Such forces can be gravitational, static, forces on mobile instruments etc. The accelerometer used can measure around 800 readings of vibration in 1 second.

The data for the vibration amplitude was collected at various positions on the desert cooler. Then using this data, a position was located where the vibrations were dominant as compared to the rest of the body. Thus, focus was laid on reducing the vibrations of that point until that particular point becomes stable and other areas in its ambience.

But the problem was that we needed multiple vibrating frequencies of the cooler. This could be done by varying the rpm of the motor. But the cooler uses a three speed motor which can rotate at only three different rpm. Thus, regulator of the cooler was replaced by a continuous variable resistance regulator. Using it, numerous points of rpm as required for the purpose were obtained.

The next step was to measure readings at that position with varying rpm. A tachometer was used for the measurement of rpm.

The following readings were obtained:

Avg amplitude(X) (mm)	R.p.m.(rev/min.)	Ang.Freq.(rad/s)
0.239870864	1290	135.1428571
0.420734382	1210	126.7619048
0.4349583	1115	116.8095238
0.646132253	1045	109.4761905
0.613383734	1022	107.0666667
0.292248498	955	100.047619
0.115615293	912	95.54285714
0.081558045	800	83.80952381
0.097561599	650	68.0952381
0.090983548	596	62.43809524
0.105153338	512	53.63809524
0.103397849	430	45.04761905
0.118305388	265	27.76190476

Fig. 2. Readings of amplitude for a given R.P.M.

A. Transfer Function from system identification-

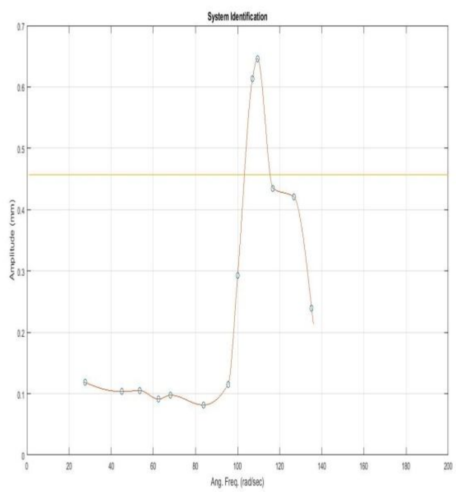


Fig. 1. Plot for system identifaction

The graph obtained for system identification is as shown above:

From this graph, using the values of peak amplitude, peak frequency; an iterative formula for damping ratio (denoted by zeta) is obtained. Afterwards a series of iterations are performed to calculate the approximate value of zeta. The final value of zeta is then used to find system parameters required to define a system. For the desert cooler taken for the purpose of experiment, mass for the system is taken as 29kg. The entire task of finding the system parameters was done using an user interactive MATLAB program[5]. The program written in m-file is shown below.

```
xp=0.646132253*(10^(-3)) %peak amplitude
wpk=109.4761905 %peak angular frequency
xc=0.456884498*(10^(-3)) %xc=xp/sqrt(2)
w1=103
w2=115 % w1 and w2 are ang.freq. corresponding to xc
for t=1:50
z(1)=0.1; %zeta
p(t)=wpk*(1+(z(t))^2); %natural frequency
r1(t)=w1/p(t);
r2(t)=w2/p(t);
xr(t)=xp*sqrt(1-(z(t))^2); % amplitude
n(t)=xr(t)/xc;
    z1(t)=(1-(r1(t))^2)/(2*sqrt(n(t))^2-(r1(t))^2);
    z2(t)=((r2(t))^2-1)/(2*sqrt(n(t))^2-(r2(t))^2);
z(t+1)=(z1(t)+z2(t))/2;
```

Fig. 3. Matlab program to calculate system parameters

The following system parameters were obtained:
k=3.499763403101679e+05N/m
c=3.748639203592120e+02Ns/m
m=29kg
F=26.56kN
After calculating the system parameters, using

$$mx'' + cx' + k = F\,Sin\,(pt)$$

the following second order differential equation was obtained:

$$29x'' + 374.86x' + 349976.34 = 26.56\,Sin\,(109.855t)$$

Where natural frequency, p=109.855 rad/sec.

Taking laplace, the following transfer function was obtained [6]:

G(s) = $1/(29s^2 + 374.86\,s + 349976.34)$

B. Vibration control

The above transfer function was tested for controlling and stabilizing the system using PID controller. For this purpose, firstly MATLAB PID control toolbox was used [1]-[3].

The following block diagram represents problem formulation and the controller attached for its control.

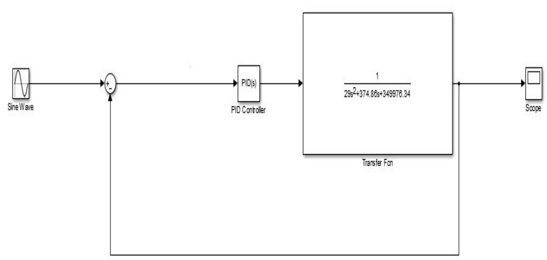
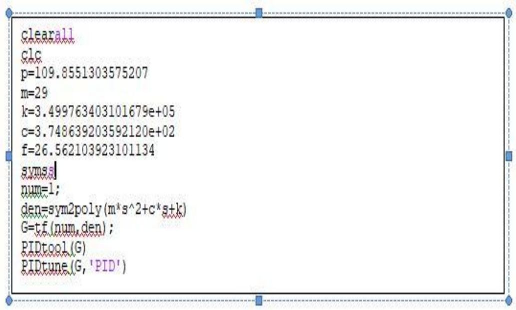


Fig.4 Block diagram for problem formulation

In order to control the system initially, the effect of changing the key components: P, I & D in a PID controller were obtained. Various parameters such as rise time, peak time, settling time, percentage overshoot essential for the stability of the system were observed by changing the values of Kp, Kd & Ki.

>>PIDtool() command was used for generating the controller.

>>PIDtune() chooses a value, based on the system dynamics, that achieves a balance between response and stability.



```
clearall
clc
p=109.8551303575207
m=29
k=3.499763403101679e+05
c=3.748639203592120e+02
f=26.562103923101134
symss
num=1;
den=sym2poly(m*s^2+c*s+k)
G=tf(num,den);
PIDtool(G)
PIDtune(G,'PID')
```

Fig. 5. Program to control the system

II. RESULTS AND DISCUSSION

The various effects on the properties of the system were observed which are very much clear from the self-explanatory figures shown below:

The step input was given to the system without providing any control and the lot obtained is as follows [2]:

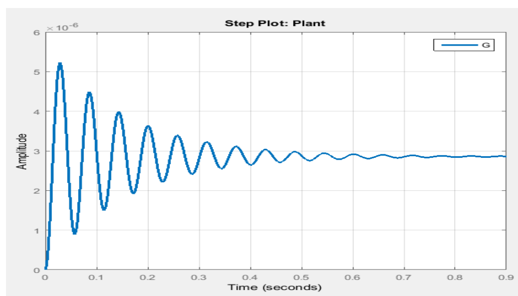


Fig. 6. Output of the system without any control applied

Graphs, after applying PID are shown below:

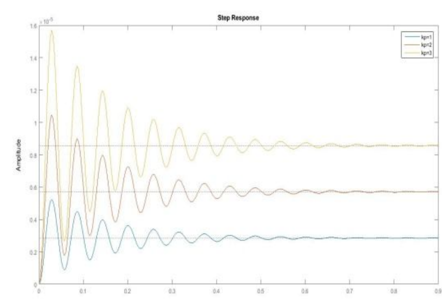


Fig. 7 Graph obtained when just Kp is changed. It can be clearly interpreted that for a step input of unity, by increasing the value of Kp from 1 to 3 in a stepwise manner, the response obtained achieves a better response closer to the actual state of the step input function i.e.unity.

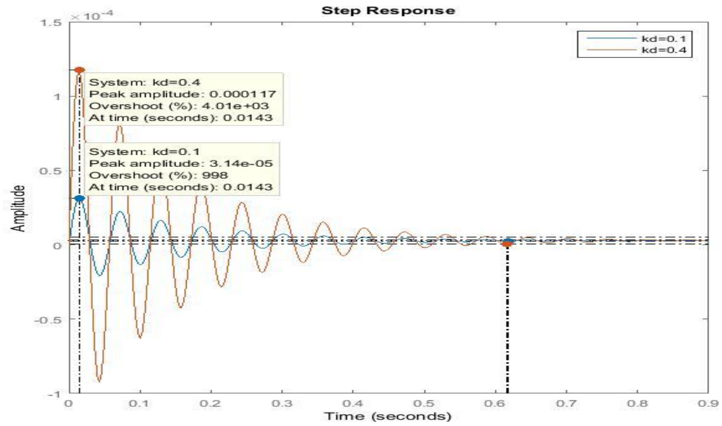


Fig. 8 When just Kd is changed. Graph elucidates that at 0.0143 seconds, by reducing the value of Kd from 0.4 to 0.1, there is a decrease of percentage overshoot (4.01e+03% to only 998%) which is a better for stability of the system.

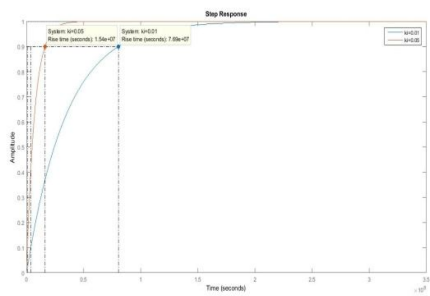


Fig. 9 When just Ki is changed. In this figure on increasing the value of Ki from 0.01 to 0.05, there is a decrease in rise time (from 7.69e+07 seconds to 1.54e+07 seconds) which would decrease the time required for the system to attain a steady state.

As we can see, graphs shown above are oscillating and/or take a great deal of time to achieve stability. So, optimality has not reached yet. There is a need to manage Kp, Kd and Ki in such a manner that all the properties of response systems such as rise time, peak time, percentage overshoot etc. are optimal.

After achieving a balance between stability and response, an optimum value of PID gains was adopted using PID tuning for a step input being applied to the system. And, this graph below shows, tuned response

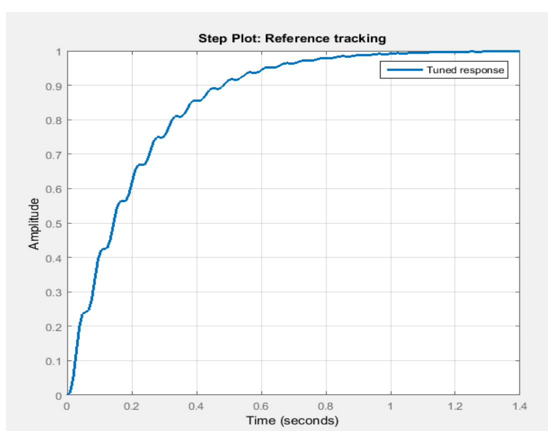


Fig. 10 Tuned response. This gives us the most feasible value for rise time, peak time, stability etc.

And the tuned values of gains are as follows

```
Kp = 2.55e+06 Ki = 1.22e+08 Kd = 1.33e+04
```

III. CONCLUSION

In a nutshell, we can say that PID toolbox along with its PID tuning is one of the most versatile and convenient method to control a well-defined system. This approach is being used to control the vibrations and stabilise the mechanical system proposed above. It can be concluded that this work justifies the manner in which a mechanical system can be formulated into a mathematical model and its response can be stabilised according to the input conditions, as in this, step input has been used. For the purpose of control, PID control system being used can adjust the response of the system with varying values of input function by changing gain values..This opens up many possibilities in various projects, where vibration is needed to be controlled entirely or at a certain level. System Identification plays a major role to find system parameters, which are imperative to implement the PID controller . In future, to introduce this control in real-time, electromagnetic dampers and/or electromagnetic springs, etc. can be used. Also, there can be a provision of real-time actuators which will impose certain boundations to limitise the magnitude of oscillations.

IV. REFERENCES

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