Resonant frequency of car brake disk by MATLAB

1.1MATLAB Analysing

The main aim in this section is to analyse the model (mathematically description of the process vary with time; ordinary deferential equation) numerically. Moreover, comparing the results to the previous practice and theory methods described in previous sections. One method of stability in a control system is known as bode plot; which is specifies the weak states of the open loop process when subjects to a load and shows it in a peak of dome as shown in figure 2, this peak is known as resonant frequency of the process. The natural frequency in second degree model (as in this study) is approximately equal to the resonant frequency due to small value of damping coefficient c. Furthermore, the stiffness is defined from practical data of frequency and the resonant frequencies determined in bode plot analysis of stability; provided by the software, is carried out in the following steps:

1. Creating the Simulink model (block diagram type) of the process as depicted in figure 7.
2. Implementing the values of the constants m, c, and k.
3. Defining the input and output signals of the open loop model.
4. Then, from a Simulink control analysis tool, the linear analysis will be selected. It is clear the model not includes the feedback loop. Since the feedback is related to the driver pressing on the brake pedal (Melons 2003[3]).
5. The bode plot code will apply for the process, and then the MATLAB Simulink creates a plot to show a resonant frequency.

The above steps will be repeated for different modes; i.e. mass reducing and gap creating between shoes and disk. Figure 7 shows the responses and highlights the resonant frequency of the model. In addition the results data are depicted in table 3.

1.2 Results and Discussion

Table 2 & 3 show overall extracted parameter obtained by FRF & MATLAB Simulink respectively. First identified modes of vibration from FRF curve in figure 5 identified by four of vibration modes. FRF plots shows how modes can cause the disc brake structure to vibrate and produce noise. At small input force from the hammer can cause a very large response of the natural frequencies. This is clearly indicated from the narrow peaks in FRF plots. Therefore, when the disc is excited at one of the peak frequencies, the response of disc brake per unit force will became large. As results the brake disc modes will acts like amplifier.

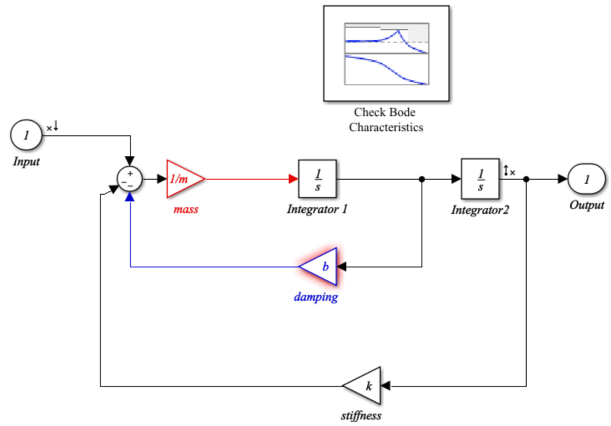

Figure 7. The MATLAB Simulink block diagram presentation of the disk process

The damping ratio obtain from modal data indicates that the structure of disc brake exhibit relatively low damping. As the damping ratio is very small (<< 1), therefore the damped natural frequency of disc brake is equivalent to the natural frequencies as $f_n = \sqrt{1 - \xi f_d}$. The term Q-factor that determines the sharpness frequencies was originated from the field of electrical tuning where sharpness the narrow peaks of resonant peak is desirable thing. The value increased along with increasing value of natural frequencies in which the damping ration becomes smaller and smaller.

Second, in figure 8 and table 3 demonstrate a MATLAB Simulink responses when tested by bode plot method of stability for different modes (changing in disk mass or increasing the air gap between shoes and disk). The damping ratio of the system is too small; where practically is 0.0032 table 2 and according to bode plot the resonant frequency is $3.093*10^{-7}$. In this case the two frequencies are equal. Also, this small values of the damping ratio makes the process exhibits sharp peak as in figure 8. Furthermore, the model has small value of damping coefficient (c); which is 0.00657, and large value of stiffness which is revealed in table 3.

These data show the decreasing in disk mass the elastic behaviour (or stiffness) will reduce; from the data the slop between stiffness and reduction in thickness has a slop of 4.6. Since the air gap between these parts created. Consequently, the resonant frequency reduced as the thickness decreased by a slop of 11.2. Therefore, due to the effect of creating air gap more displacements (x) required by the process [4]. However, mathematically the natural frequency is proportional to the square root of stiffness and inversely behaves with square root of the mass as described in equation 5.

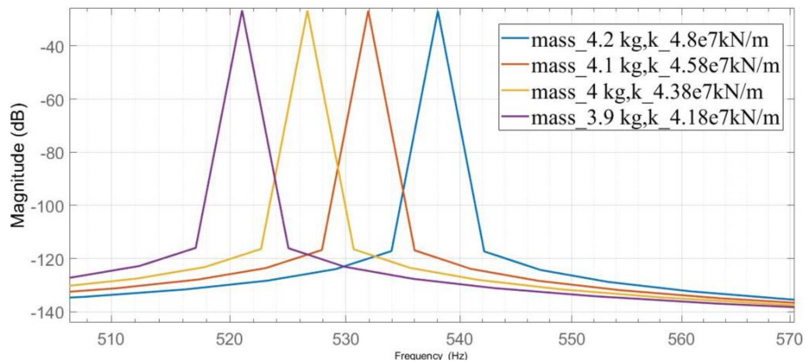

Figure 8. The effect of changing in mass on the resonant frequency

It is clear from curves of bode plot the process exhibits higher overshoots at the resonant frequency due to small value of damping coefficient c; hence small value of damping factor ζ. Consequently, high and sharp overshoot occurs (approximately 100 dB) in any transient case. In accordance to the sources of control system [11, 12, and 13] to optimize such process the damping factor ζ should be 0.707, then the damping coefficient must be 15 kN.s/m. In this case, the process exhibits smooth transient and zero overshoot (or 0 dB) as depicted in figure 9; which is optimized for mass of 4.2 kg.

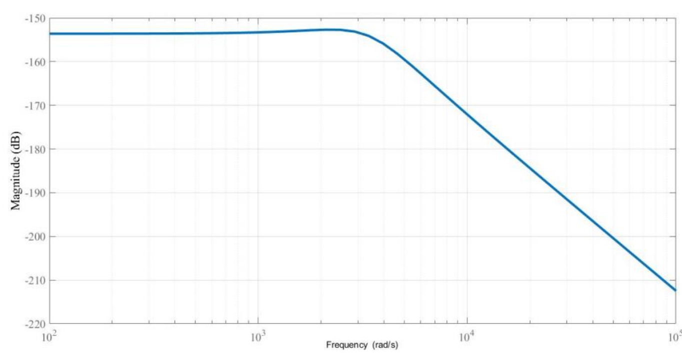

Figure 9. the optimised case of the disk brake process