VIBRATION TRANSMISSIBILITY CHARACTERISTIC IN HYDROSTATIC BEARING

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ABSTRACT

Engine vibrations are transferred through the gear box and coupled with the gear mesh frequency vibrations results in noise being emitted to the surroundings. One approach to address this issue is to use hydrostatic bearing as filter the unwanted vibratory energy in gear set. The objective of this paper is to expect the hydrostatic bearing has good transmissibility characteristic across the bearing by preventing the vibrational energy transmission to gear housing. An experiment utilizing single recess circular hydrostatic bearing placed midway along a roller bearing support the shaft. The investigation examines the time response of the single recess circular hydrostatic bearing under externally pressurized fluid trapped in a recess. The results shows that as the pressure increased in recess at constant rotating frequency of the shaft, the transmissibility (T) value as a ratio of output signal amplitude to input signal amplitude become lesser than one ($T < 1$). The exhibited behaviour is similar to the potentially prove that the hydrostatic bearing could attenuate the unwanted vibratory energy across the shaft point-to-point end. This characteristic of dynamic behaviour of the hydrostatic bearing can be potential for vibration mitigation.

Keywords: Hydrostatic bearing, time response, transmissibility, vibration, pressure, circular recess

INTRODUCTION

In a typical vehicle gearbox, gear meshing noise is normally the predominant source of noise in a vehicle [1]. Mesh-frequency vibration from the gear set emits the vibrational energy passing through the shaft from the support bearings to the housing as acoustic noise [2]. The transmitted energy serves to excite the housing and other radiating surfaces, resulting in emitted noise. Alternatively, the gear mesh noise can be reduced by improving the conjugate actions of the gears [3]. This can be achieved by reducing the gear tolerances, resulting in smoother gear operation, but unfortunately at the cost of increased manufacturing costs. Modification of the gear tooth profile can also be carried out to improve the conjugate actions between the gears [4-5]. Alternatively, a method for reducing this emitted noise is to filter or

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disrupt the transmission of the vibratory energy to the housing. One proposal involves utilizing the filtering effect of a hydrostatic bearing as a method to reduce the transmission of the vibratory energy in gear sets [6]. Previous research on mitigating gear-mesh noise, have shown that a hydrostatic bearing can act as high pass filter for gear-mesh frequency vibration [7]. Building on that work, this research aims to examine the ability of hydrostatic bearing to act as a filter of gear-mesh frequency vibrations transmitted through shaft. It consists of a set of hydrostatic bearing placed radially and supporting a shaft supporting roller bearings. This should allow the gear-mesh frequency filtering effect of the hydrostatic bearing to be accurately determined.

METHODOLOGY

Experimental setup

Figure 1, shows the experimental setup used to study the behavior of a single pad hydrostatic bearing. A shaft with a diameter of 19 mm is supported by two identical roller bearings at each end. The distance between the two bearings is 23 mm, are measured from between the inner ends of the support bearings. The hydrostatic bearing, with a circular recess is filled with externally pressurized hydraulic oil type German Institute for Standardization (DIN) 515224 at 40 ^{0}C with kinetic viscosity of 22 mm^2/s, and was attached to the shaft at a point midway between the two support bearings. The shaft is driven at 1950 rpm throughout the experiment using 150W, Direct current (DC) motor. Two piezoelectric accelerometer (Dytran, model 3097A2) were used along with a National Instrument (NI) 9234 4-channel +/-5V, 24 Bits module to measure the response. Each accelerometer was mounted with threaded stud on the hydrostatic bearing. One accelerometer was placed on the top of hydrostatic bearing, at a location denoted by THB, and one at the bottom of hydrostatic bearing, at a location denoted by BHB. The experimental data was obtained from the NI module connected by USB to the data collection software, Labview^{ä}. The program was also used to analyse the data using the sound and vibration assistant platform.

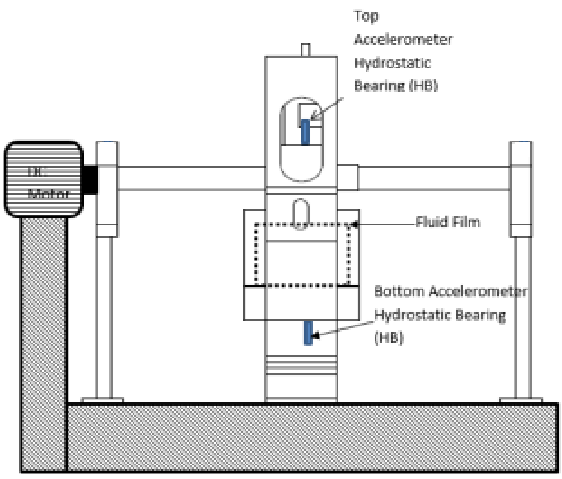


Figure 1. Experimental setup with Input Signal and Output Signal

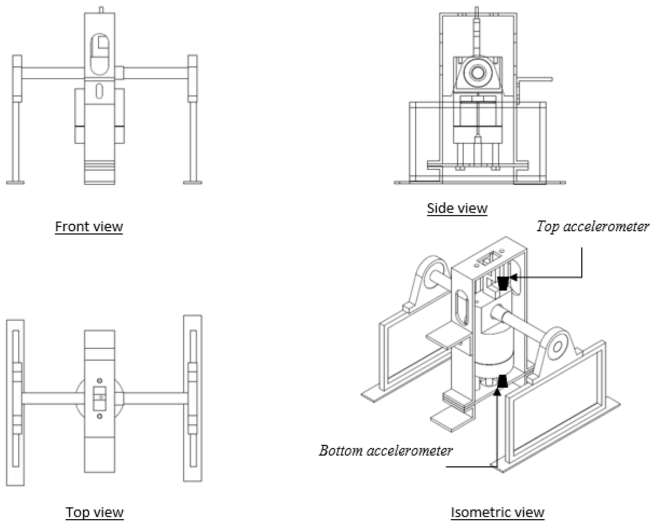


Figure 2. Orthographic and Isometric views

Experimental Procedure

The aim of the experiment was to measure the input and output signals across the hydrostatic bearing for different values of recess pressure. The test rig is fitted with a DC motor used to drive the rotor. The DC motor was turned on and allowed to run for 5 minutes to allow the system to settle to its steady state operation at zero pressure. The Labview[ä] was used to collect and display the experimental vertical y-axis acceleration values from the accelerometers at THB and BHB. The data collection was carried out for 60 seconds. Then a hydraulic pump supplying pressurized fluid to the recess was turned on and set at specified value of pressure. The system was allowed to run for 5 minutes to reach steady state operation. The data from the two accelerometers was then collected. The experiment was then repeated for different values of pressure. Once all the data has been collected the DC motor was turned off. The experiment was repeated for a total of 15 times, 5 recorded data for each pressure condition were taken.

RESULTS AND DISCUSSION

Time Response

Five sample readings of the time response obtained from the experiment, from 0 to 10 bar recess pressure, are shown in Figure 3-7. The figures show that there was a significant reduction in the measured output amplitude compared to the measured input signal amplitude. A window of 1 second's peak-to-peak amplitude stabilization for both input and output signals was taken after five minutes the system reaches steady state. Figures, also shows that the attenuation of the output signals amplitude, being observed once the recess pressure been supplied. The response for other recess pressure showed the same general behaviour as shown in Figure 3-7. As the pressure of the fluid in the recess was increased from 0 bar to 2 bar, the amplitude of the output signal reduces to 7 mm. Increase in the recess pressure to 6 bar the amplitude for input signal remain at the same amplitude, where else the output signal gradually reduced to 5 mm peak-to-peak. The same proportion of reduction took place for output signal for 8 bar to 3 mm and almost below 1 mm for output signal as the recess pressure increased to 10 bar.

Zero bar

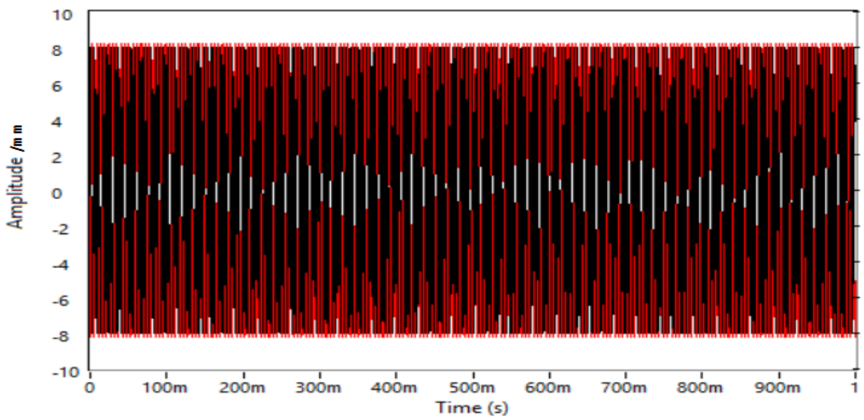


Figure 3. Time response of the input and output for 0 bar recess pressure

2 bar

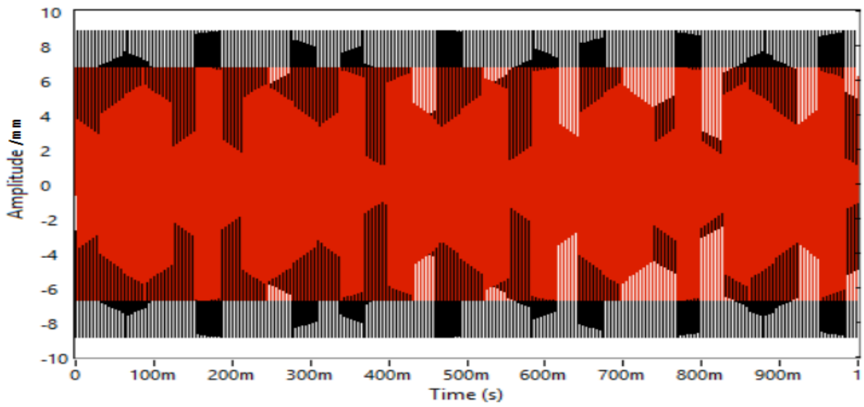


Figure 4. Time response of the input and output for 2 bar recess pressure

6 bar

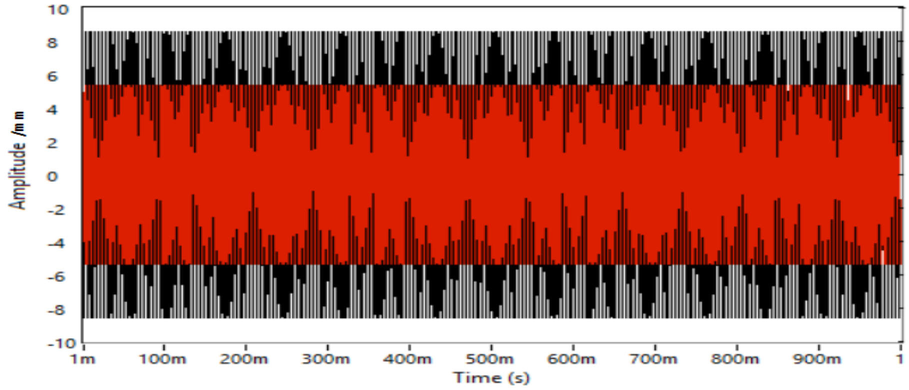


Figure 5. Time response of the input and output for 6 bar recess pressure

8 bar

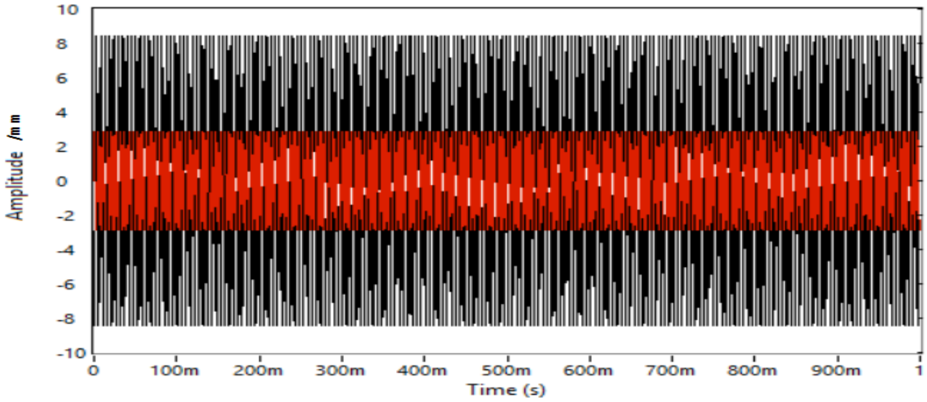
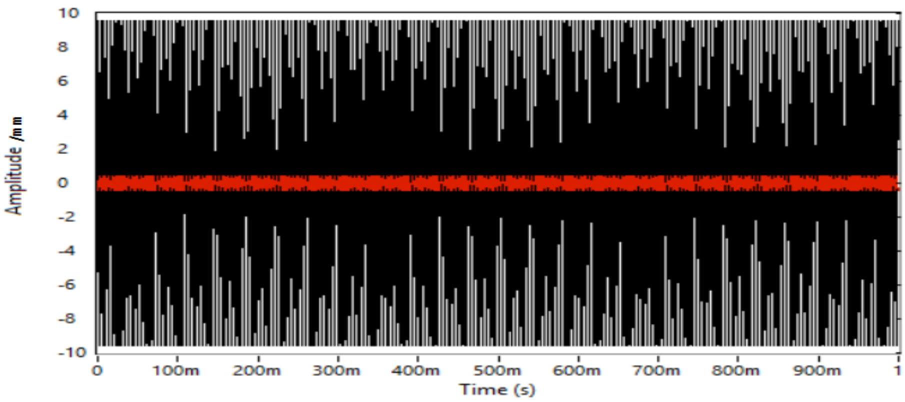


Figure 6. Time response of the input and output for 8 bar recess pressure

10 bar



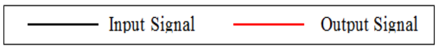


Figure 7. Time response of the input and output for 10 bar recess pressure

Table 1 shows the input signals, output signals, and transmissibility ratio for the different pressure. The data shows that the transmissibility ratio is reduced as the recess pressure is increased. Figure 8, shows reduction of 22% starts at 2 bar recess pressure, as the recess pressure increased gradually to 6 bar the output signal reduced to 44% and subsequently 66% for 8 bar and 88% for 10 bar, an average of 20 % decrease at each stage of increase in the recess pressure. Figure 8, shows relation of inversely proportional linearity as the recess pressure increase with the attenuation of output signal. Figure 9 also indicate inversely proportionally linearity of transmissibility ratio as the function of recess pressure, whereas the recess pressure increased, the ratio becomes less than one (T < 1).

Table 1. Transmissibility (T) ratio of output signal to input signal

Pressure	*0 Bar*	*2 Bar*	*6 Bar*	*8 Bar*	*10 Bar*
Amplitude of Input Signal	9 mm	9 mm	9 mm	9 mm	9 mm
Amplitude of Output Signal	9 mm	7 mm	5 mm	3 mm	1 mm
Transmissibility	1	0.77	0.55	0.33	0.11

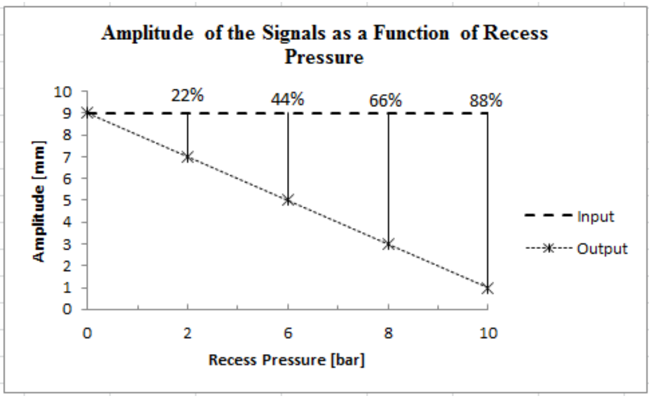


Figure 8. Percentage reduction amplitude of the Input signal when compared to the Output signal

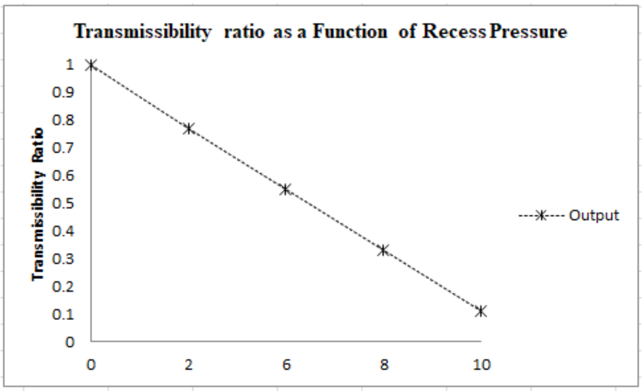


Figure 9. Transmissibility Ratio reduction against recess pressure

Frequency spectrum

The frequency spectrums of the input and output signals for, 0-10 bar recess pressure are shown in Figure 10-14. A window of 60 second's stabilization for both input and output signals was taken after the system reaches steady state Comparison of the two-amplitude spectrum shows that the main spike for both spectrums occurs at 32 Hz as the rotating frequency of the shaft. Figure 10 shows the equal magnitude for both input and output signals, this indicates that there is no frequency shift in the input signals as well as output signal, transmitted across the hydrostatic bearing. Similar response for other recess pressure was observed, input signal predominant the highest peak and corresponding attenuation takes place on the output signals. As the recess pressure increased to 2 bar till 10 bar the output signal across hydrostatic shows significant attenuation. This shows the output signal is discernible from noise of other sources due to filtering effect of the hydrostatic bearing

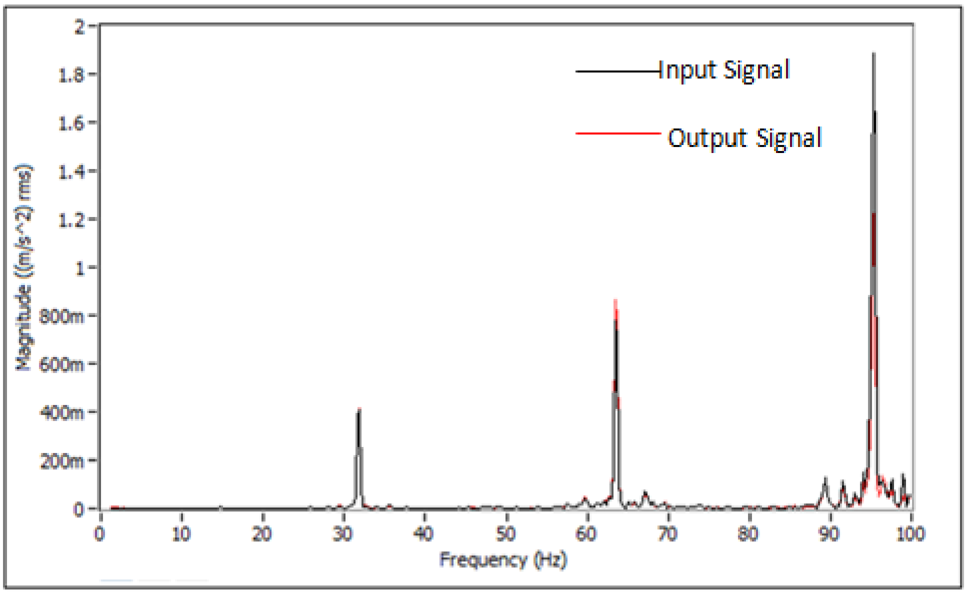


Figure 10. Frequency spectrum of the input and output signals for 0 bar recess pressure.

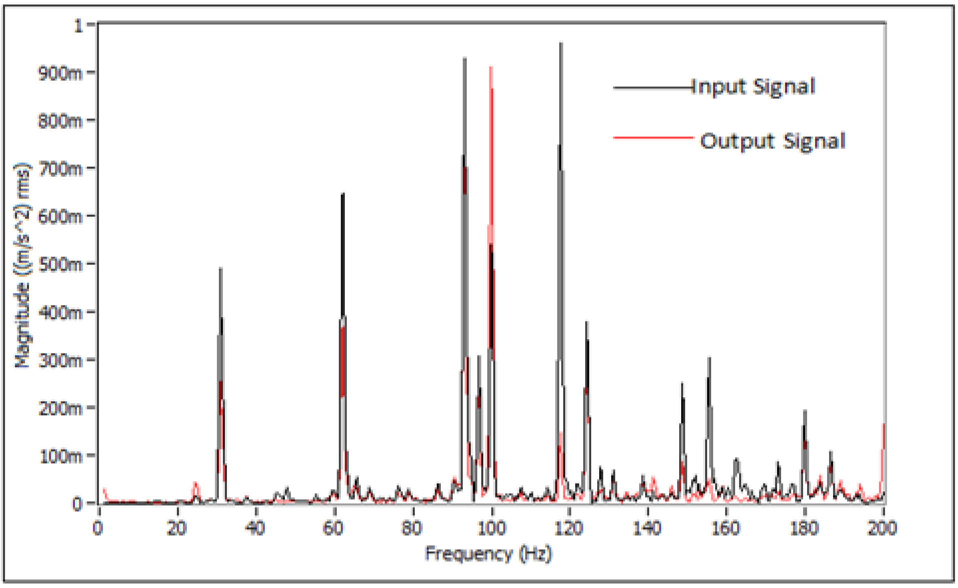


Figure 11. Frequency spectrum of the input and output signals for 2 bar recess pressure.

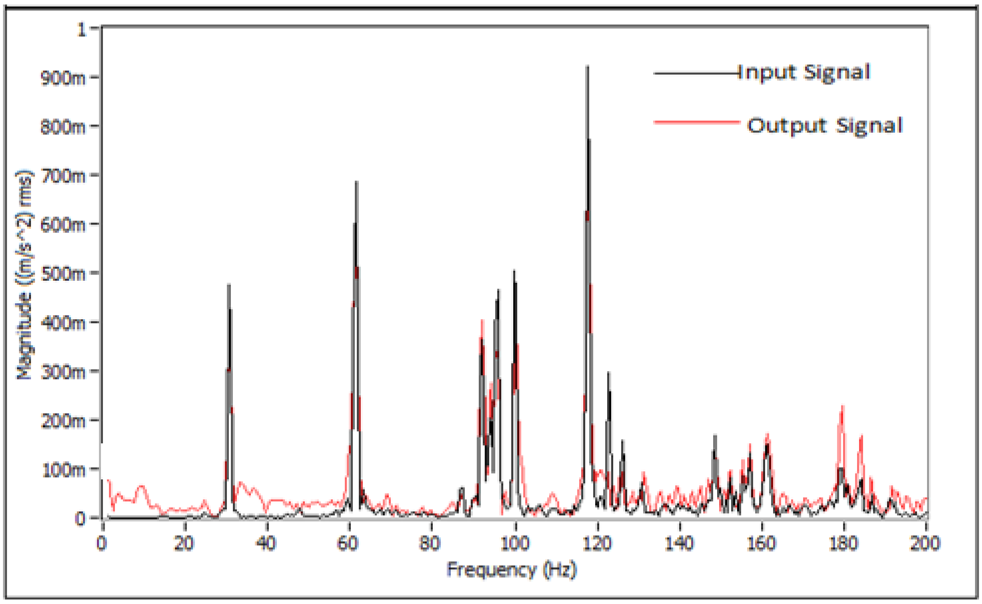


Figure 12. Frequency spectrum of the input and output signals for 6 bar recess pressure.

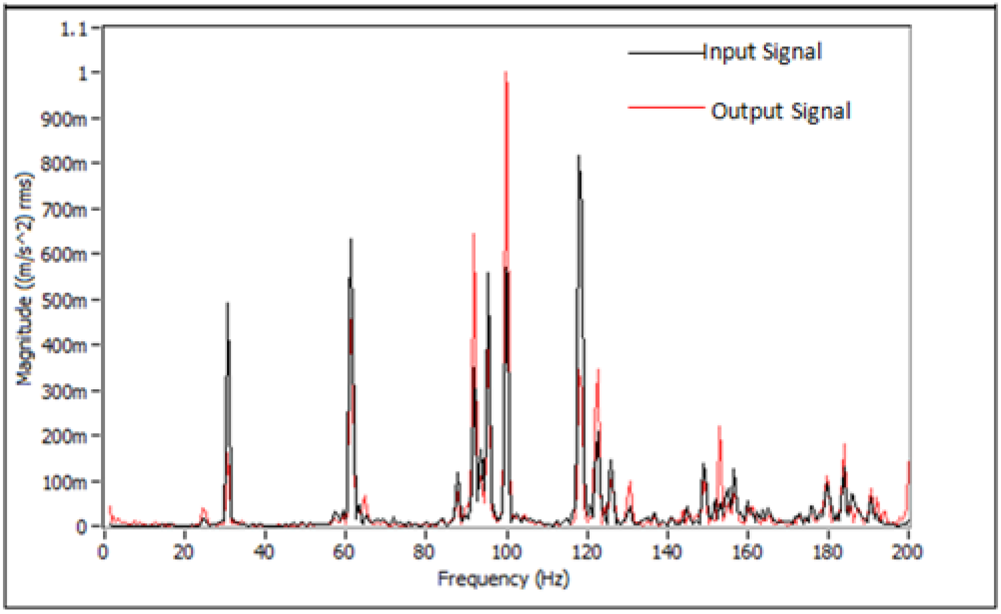


Figure 13. Frequency spectrum of the input and output signals for 8 bar recess pressure.

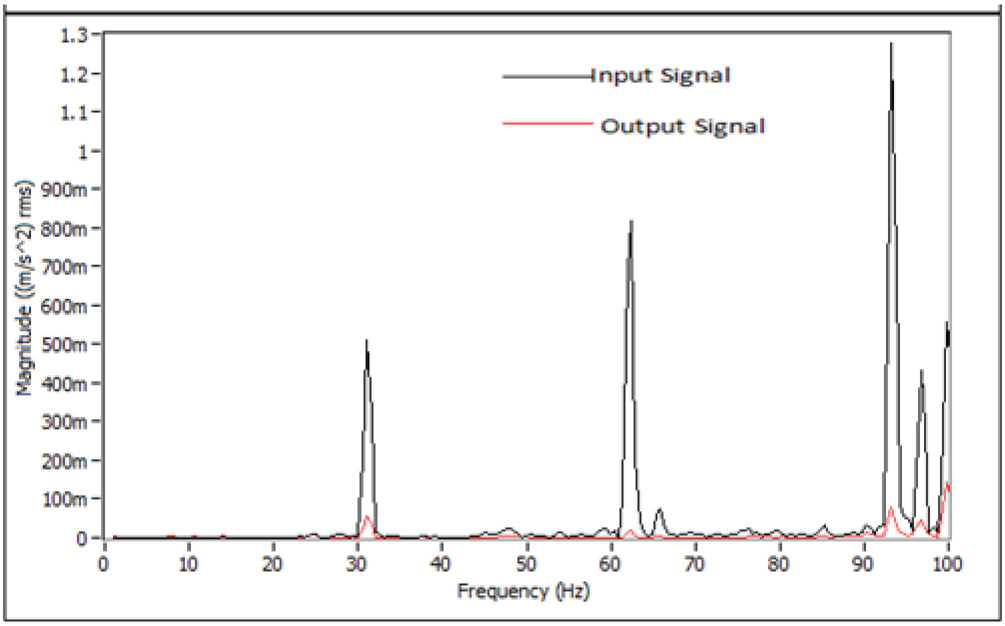


Figure 14. Frequency spectrum of the input and output signals for 10 bar recess pressure.

CONCLUSION

In the present study, experiment setup described successfully the transmissibility effect across the hydrostatic bearing with single pad configuration. The time response of these measurement shows transmissibility less than one (T < 1) as the recess pressure increased, for in that regard the experiment was successfully meets its main objective to understand vibration transmissibility characteristic across hydrostatic bearing on rotor. Similarly, the frequency response also indicates the reduction on the output signal amplitude as the recess pressure increased form 2 bar to 10 bar. The observed effect could potentially prove that the hydrostatic bearing could attenuate the unwanted vibratory energy across the shaft point-to-point end. However, in addition the experimental setup is also expected to measure the time varying value of the inflow and outflow and time-varying inlet and outlet temperature be examined in detail.

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