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Virtual Instrumentation based comparative technique for Vibration analysis and Energy loss calculation

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ABSTRACT

THIS PAPER proposes a modern approach for acquiring and analyzing vibration signals and to estimate the energy loss effectively during vibrations. The vibration signals acquired from the heavy machines with the help of vibration accelerometer will be transferred to a computer through a Data acquisition card and analysis is being carried out by using LabVIEW. Frequency domain analysis is used to study the nature of the vibrations and to estimate the energy loss during vibration. Comparative study is made with conventional vibration analyzers for performance validation of virtual systems. The performances of identical machines can be compared and steps can be taken to improve the performance of less efficient ones. Web publishing feature has been used to create a centralized virtual instrumentation server to acquire the signals from the sensor through a remote terminal in real time via networking to analyze and evolve recommendations over the network. The testing panel can be easily implemented in any location and if required, can be controlled via Internet or LAN.

Key Terms: Data acquisition, Energy loss calculation, Vibration analysis, Virtual Instrumentation

INTRODUCTION

Vibration analysis is the most important tool in industrial automation in economically designing the higher operating speed machineries with assurance of safety and reliability. Vibration is caused due to bearing faults, angular and parallel misalignments, loose joints, improper mounting, oil whip and whirl and many reasons include smaller misalignments and material[1]. Vibration analysis is further useful in identifying the natural frequencies to avoid resonance and hence structural instability. Having the knowledge about different frequency conditions and the forces caused are required for developing vibration isolation system. Study on input, output characteristics[2] of the given system is used to identify the components in terms of its mass, stiffness, and damping[3]. Vibration is suitably sensed using accelerometers by considering the frequencies and amplitudes expected to be measured, operational condition of the machine and data processing method[3,4].

Machine vibration is to be monitored periodically to avoid machine damage and cost involved within. Condition monitoring helps to identify the power consumption due to vibration. As well as the power required for the machine to perform its intended function and to avoid occupational hazards[4]. To sustain higher operating speed, improved efficiency and productivity, vibration monitoring is done with a preventive maintenance schedule. For proper monitoring of the vibration the conceptual design of a low-cost multipurpose stand-alone DAQ device has been provided [5]. Spectral analysis the suitable methodology for estimation to understand the cause for vibration[7]. Analytical method and texture analysis will be helpful for finding energy loss and to evolve recommendations [8]. Proposed method describes about the network based remote server configuration for the estimation of vibration profile and hence the energy loss.

MATHEMATICAL MODELING

Vibration

A reciprocating or back-and-forth movement involving a continual interchange of kinetic energy and potential energy and also an oscillation of repetitive motion around a mean equilibrium position. Vibration is the combination of many oscillatory motions simultaneously. The vibration of a mass supported by a spring is an up-and-down motion that involves continual interchange of kinetic energy associated with motion of the mass and potential energy associated with distortion of the spring [7].

Amplitude describes the severity of vibration, and frequency describes the oscillation rate of vibration. The combined spectra will illustrate the root cause for vibration. Vibration is related to the size of the vibratory movement, speed and the force associated with the movement. Since vibration is combination of multiple oscillatory motions, spectral Analysis is being carried out for studying the individual frequencies at which a machine component vibrates, as well as the amplitudes corresponding to those frequencies, to get inferred the condition of the machine. Vibration analysis involves data collection method, fast data collection, data processing and data display.

Sensors

Since the paper approaches the instrumental method with virtual instrument technique selection criteria and sensors are significant. Piezoelectric transducers, which generate electrical charge when subjected to a mechanical stress, are suitable for vibration measurements.

The charge generated in the crystal due to a force is given by in eq. (1)

$$Q= kF= kAp \qquad (1)$$

where k is called the piezoelectric constant, A is the area on which the force *F* acts and *p* is the pressure due to *F*. The output voltage of the crystal is given by in eq.(2)

$$E= \nu tp \qquad ---(2)$$

where *v* is called the voltage sensitivity and t is the thickness of the crystal.

When a transducer is used in conjunction with another device to measure vibrations, as a vibration pickup which has mass(m) spring(k) damper(c) as a system mounted on the vibration body, then the vibratory motion is measured by finding the displacement of the mass relative to the base on which it is mounted. The bottom ends spring and the dashpot will have the same motion as the cage and their vibration excites the suspended mass in to motion. Then the displacement of the mass relative to the cage, z = x-y, where x denotes the vertical displacement of the suspended mass[3].

The vibrating body is assumed to have a harmonic motion is expressed in eq.(3):

$$y(t)= Y \sin\omega t \qquad (3)$$

The equation of motion of the mass *m* can be written as

$$m(x-y)+c(x-y) = k(x-y) \ --- \qquad (4)$$

By defining the relative displacement z as in eq.(4)

$$z = x-y \quad --- \qquad (5)$$

eq.(4) can be written as by substituting eq.(5)

$$mz+cz+kz= 0 \qquad (6)$$

Equations (3) and (6) lead to

$$mz+cz+kz=m\omega^2Y \sin\omega t \qquad --- (7)$$

This equation is identical to the equation of motion, hence the steady-state solution is given by

$$z(t)=Z\sin(\omega t-\ddot{O}) \qquad \text{--- (8)}$$

where Z is given by

$$Z=\frac{y\omega^2}{[(k-\omega^2)^2+c^2\omega^2]^{2/z}}=\frac{r^2y}{[(1-r^2)^2+2(r)^2]^{2/z}} \qquad (9)$$

$$Ø=\tan^{-1}(c\omega/k-m\omega^2)=\tan^{-1}(2\zeta r/1-r^2) \text{ ---} \qquad (10)$$

$$r=\omega/\omega_n \qquad (11)$$

And

$$\zeta=c/zm\omega_n \qquad \text{--- (12)}$$

eq.(11) and eq.(12) thus involves natural frequency.

Accelerometer

An accelerometer is an instrument that measures the acceleration of a vibrating body. From the accelerometer record, the velocity and displacements are obtained by integration. Eq. (5) and (6) yield

$$-z(t)\,\omega_n^2=\frac{1}{[(1-r^2)^2+2(r)^2]^{2/z}}\{-Y\omega^2\sin(\omega t-\ddot{O})\} \qquad (13)$$

This shows that if

$$\frac{1}{[(1-r^2)^2+2(r)^2]^{2/z}}\approx 1 \qquad (14)$$

eq. (11) becomes

$$-z(t)\,\omega n^2\approx -Y\omega^2\sin(\omega t-\ddot{O}) \qquad \text{--- (15)}$$

By comparing eq. (15) with $\ddot{Y}=-Y\omega^2\sin(\omega t-\phi)$, the term $z(t)\omega_n^2$ gives the acceleration of the base $\ddot{Y}$, except for the phase lag, hence instrument reads the value of $\ddot{Y}=-z(t)\ \omega^2_n$. The time by which the record lags the acceleration is given by t' $=\phi/\omega$. In single harmonic component, time lag is insignificant. The value of the expression on the left-hand side of eq. (14) . It can be between 0.96 and 1.04 for $0 < r < 0.6$ if the value of lies between 0.65 and 0.7. Since r is Small, the natural frequency of the instrument has to be large compared to the frequency of vibration to be measured. From the relation it is found that the mass needs to be small and the spring needs to have a large value of k (i.e., short spring), so the instrument[10] will be small in size. Due to their small size and high sensitivity, accelerometers are preferred in vibration measurements[10]. In practice, eq (11) may not be satisfied exactly; in such cases the quantity

$$\frac{1}{[(1-r^2)^2+(2\ \zeta\ r)^2]^{1/2}} \tag{16}$$

can be used to find the correct value of the acceleration measured and hence eq.(16) describes about acceleration component.

Mathematical modeling of vibrations is done by acquiring the vibration signals from the vibration source and then shifting it to the frequency and power spectrum.

Energy in the ideal mass-spring system

The potential energy (Ep) of the ideal mass-spring system is equal to the work done stretching or compressing the spring is given in eq.(17):

$$E_p= -\int_0^x F dx=\int_0^x kx dx = \frac{1}{2}kx^2= \frac{1}{2}kA^2\cos^2(\omega_0 t+ Ø) \tag{17}$$

The kinetic energy (E_k) of the ideal mass-spring system is given by in eq.(18) the motion of mass.

The total energy of the ideal mass-spring system is constant:

$$E= Ep+Ek=\frac{1}{2}kA^2 =\frac{1}{2}mv_{max}^2 \tag{18}$$

$$E_k=\frac{1}{2}mv^2 = \frac{1}{2}m\omega_0 A^2 \sin^2(\omega_0 t+ Ø) = A^2\sin^2(\omega_0 t+ Ø) \tag{19}$$

eq.(19) describes at the extremes of its displacement, the mass is at rest and has no kinetic energy and stores potential energy when the spring is compressed. Due to external enforcement, when the mass reaches equilibrium with spring, it is converted into kinetic energy.

TYPES OF VIBRATIONAL EFFECTS

Vibration classification is required to position the sensor[10]. Basic classes of vibration are lateral, torsional and axial. Since mechanical stress cause charge development in sensor, the sensor will experience the following effect viz. longitudinal, shear and traverse.

- *Longitudinal Effect:* The size of the charge depends only on the force applied. The only way to increase this charge is to connect several plates mechanically in series and electrically in parallel. If this is done, the charge is:

$Q_x = d_{11} \times F_x \times n$ ---(20)

where: d_{11} = piezoelectric coefficient (for quartz crystal, d_{11}=2.30pC/N) ---(21)

F = applied force, n = number of crystal plates,
x = direction of applied force

- *Shear Effect:* This effect, too, is independent of the size and shape of the piezoelectric element as well as of the charge distribution. The charge occurs at the surfaces under load; at n elements connected mechanically in series and electrically in series is:

$Q_x = 2 \times d_{11} \times F_x \times n$ ---(22)

- *Transverse Effect:* In this case, a force in the direction of one of the neutral axes produces a charge on the surfaces of the corresponding polar axis. The magnitude of the charge is dependent on the geometrical dimensions of the piezoelectric element.

Assuming dimensions a, b, and c, the charge is:

$Q_y = d_{11} \times F_y \times b/a$ ---(23)

where y = a neutral axis.
eq. (20), (21), (22) and (23) describe about different and crystal constants.

EXPERIMENTAL SET UP

Vibration exciter is an Electrodynamic type of device. It consists of a powerful magnet placed centrally, surrounding which is a suspended exciter coil. This assembly is enclosed by a high permeability magnetic circuit for optimum performance.

The experimental set up of vibrational analysis where LabVIEW with relevant data acquisition unit will integrated to perform vibration analysis instead of hardware based systems is shown in figure 1.. When an electrical current is passed through the exciter coil a magnetic field is created around the coil, these field interacts with the field due to the central permanent magnet and this results in the upward or downward movement of the suspended coil depending upon the direction of current flow in the coil, if an alternating current is injected in the coil, it moves up and down continuously. Thus controlling the frequency of the coil current, the frequency of vibration is controlled. By controlling the amount of current, the amplitude of vibration is controlled. Power Amplifier is the control unit for the exciter. This unit consists of a tunable sine wave oscillator, a power amplifier to inject current into exciter coil and protection circuits[15]. A tunable sine wave oscillator is designed around a voltage controlled oscillator using an integrated circuit which produces triangle wave oscillations basically. A special circuit converts the triangular wave into sine wave. Suitable buffer amplifiers are incorporated to produce distortionless sinewave output with good amplitude stability and frequency response[12, 17].

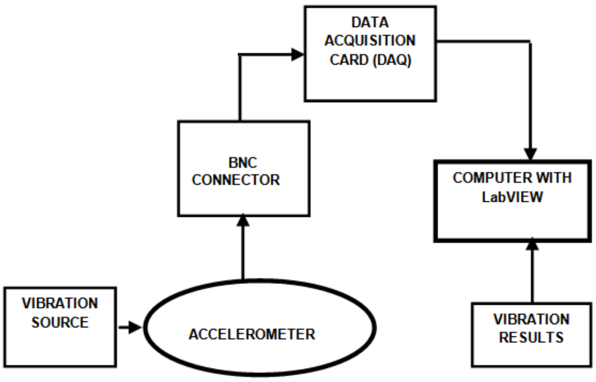


Figure 1: Block Diagram of Proposed Workbench BLOCK DIAGRAM

Accelerometer is mounted where the vibration is to be monitored as per the standards and signals are transferred through a charge amplifier to the terminal block BNC 2120 which in turn connects to DAQ card PCE 6024E which 200kS/s. It provides continuous signal to LabVIEW based analyser over the steady state vibration behavior.

Certain care are generally taken for the selection of sensor viz: natural frequency of accelerometer, frequency range, minimum and maximum amplitude of vibration, environmental conditions and mounting method.

The sensor is a low G Micro-machined Accelerometer with a capacity of acquiring signals ranging from +40g to -40g. This is a capacitive and surface-micro-machined integrated-circuit accelerometer. The device consists of a surface micro-machined capacitive sensing cell (g-cell) and a CMOS signal conditioning ASIC contained in a single integrated circuit package. The sensing element is sealed hermetically at the wafer level using a bulk micro-machined “cap” wafer. The g-cell is a mechanical structure formed from semiconductor materials (polysilicon) using semiconductor processes (masking and etching). It can be modeled as a set of beams attached to a movable central mass that moves between fixed beams. The movable beams can be deflected from their rest position by subjecting the system to acceleration.

LabVIEW and DATA ACQUISITION SYSTEM

Virtual Instrumentation is novel technique as a combination of software and hardware onto the industry standard computer technologies to create user defined, reconfigurable instrument system to reduce development time since graphical programming environment in user interface and source code. Stand alone executables are also possible with signal conditioning and processing tools with network based automation. This technique helps in designing discrete component based design and has tools for Circuit desing, Control system, Real time I/O, VLSI, Embedded system which includes mathematical modeling.

It is possible to channelize the signal through microcontroller based configuration; DAQ cards mounted on PCI slot as shown in figure. 2 serve the purpose when multiplexed channels are preferred. Front panel interface of the vibrobench is shown in figure 3. It consists of four main sections like Sensor section to prompt between different types of sensors, Operation type to select standardization, measurement and Analysis and recording.

Graph section consists of four different graphs to illustrate the vibration parameters like displacement, velocity and acceleration[6].

Power spectral analysis shows the frequency composition of vibrating segment which finds the magnitude and its reoccurrences during vibration[9].

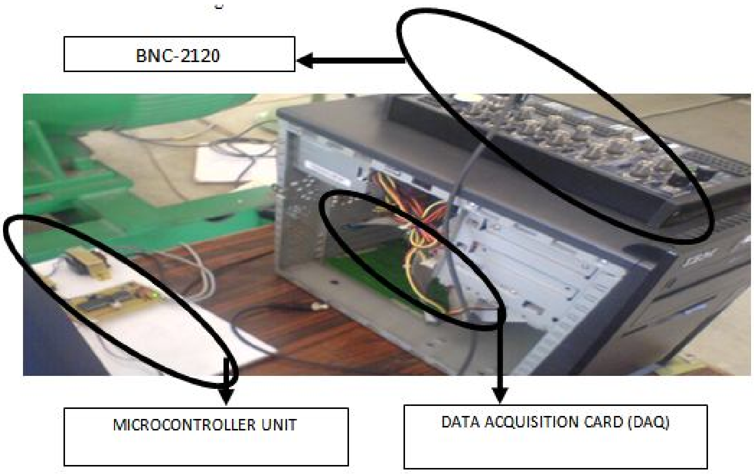
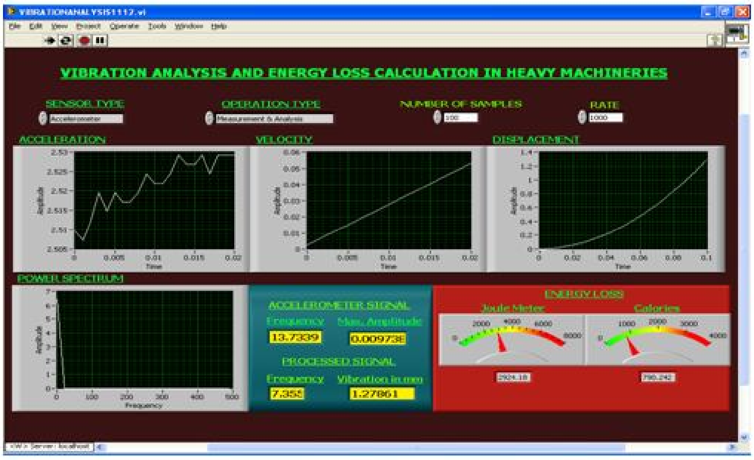


Figure 2: DAQ Card Connection and Signal Conditions Unit



VIBRATION ANALYSIS AND ENERGY LOSS CALCULATION IN HEAVY MACHINERIES

SENSOR TYPE: Accelerometer

OPERATION TYPE: Measurement & Analysis

NUMBER OF SAMPLES: 100

RATE: 1000

ACCELEROMETER SIGNAL
Frequency: 13.7339
Max. Amplitude: 0.009738

PROCESSED SIGNAL
Frequency: 7.355
Vibration in mm: 1.27861

Figure 3: Front Panel of Proposed LabVIEW based Workbench

The block diagram of the vibrobench or source code has been explored in figure 5. Depending on the front panel the block diagram is also divided into many sections based on the hardware connections shown in figure 2.

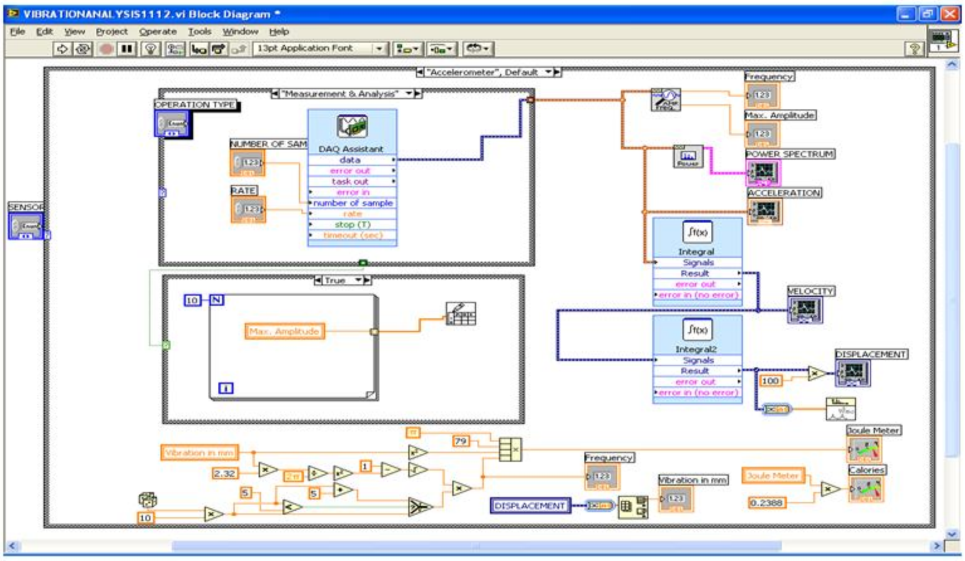


Figure 5: LabVIEW Source Code

ENERGY LOSS CALCULATION

Energy loss in a machine means a decrease in efficiency of that machine. Energy loss can occur in many ways. The energy loss may be due to sound, vibration, heat, friction etc. Here, the energy loss is calculated by taking vibration into consideration.

The rate of change of energy with time (dW/dt) is given by in eq.(25) and (26)

$$\begin{aligned} dW/dt &= \text{force} \times \text{velocity} \\ &= Fv \\ &= -\,c\,v^2 \\ &= -\,c\,(dx/dt)^2 \qquad \text{---(24)} \end{aligned}$$

Or

$$\begin{aligned} \Delta E &= \int F_D\,(dx/dt)\,dt \\ &= \int F_D\,X\,\omega\,\cos\omega t\,dt \qquad \text{---(25)} \end{aligned}$$

The negative denotes that energy dissipates with time. Consider a motion as $x(t) = X \sin\omega_d t$, where X is the amplitude of the motion, and the energy dissipated in a complete cycle is given by

$$\Delta W = \int c\ (dx/dt)^2\ dt$$
$$= \int cX^2\ \omega_d\ \cos^2\omega_d t\ .\ d(\omega_d t)$$
$$= \Pi\ c\ \omega_d\ X^2 \qquad \text{---(26)}$$

This shows that the energy dissipated is proportional to the square of the amplitude of motion. We know that energy loss occurs due to the damping given to the machine. The frequency of damped vibration is given by

$$\omega_d = (\sqrt{1} - \zeta^2)\ \omega_n \text{ ---} \qquad (27)$$

The logarithmic decrement is given by

$\Delta = 2\ \Pi\ \zeta = 2.32\ X$

Now let us take a sample reading. From the experiment we acquired the following values:

$X = 1.2615$ mm, $c = 79$, $\omega_d = 8.51$ Hz,

According to the formula:

$\Delta W = \Pi\ c\ \omega_d\ X^2 = 3.14 \times 79 \times 8.51 \times (1.2615)^2 = 3363.53$ Joule

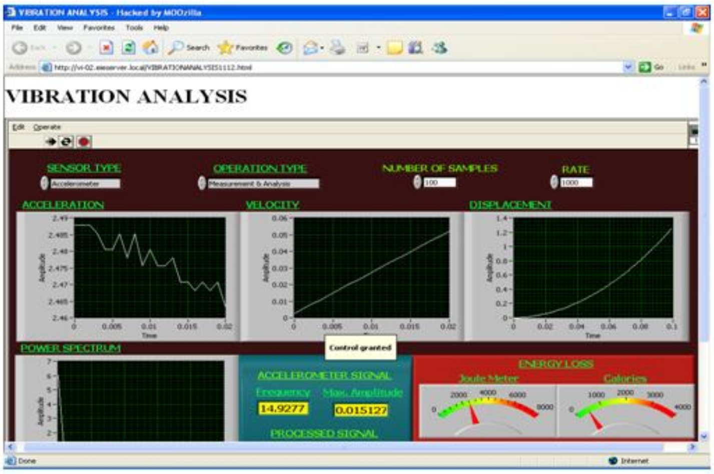
And if it is converted joule to calories then the value of Energy loss will be 1 Joule = 0.2388 Calorie

3363.53 Joule = .2388 x 3363.53 Calorie = 803.21 Calorie.

WEB PUBLISHING AND COMPARISON

The web publishing tool is yet another feature of LabVIEW. It allows the user to acquire readings remotely. The user can view and control a VI front panel remotely, either from within LabVIEW or from within a Web browser, by connecting to the LabVIEW built-in Web Server.

When the user opens a front panel remotely from a client, the Web Server sends the front panel to the client, as shown in figure 4, but the block diagram and all the subVIs remain on the server computer[13, 14]. User can interact with the front panel in the same way as if the VI were running on the client, except the block diagram executes on the server. Use this feature to publish entire front panels or to control remote applications safely, easily, and quickly.



VIBRATION ANALYSIS

SENSOR TYPE: Accelerometer
OPERATION TYPE: Measurement & Analysis
NUMBER OF SAMPLES: 100
RATE: 1000
Frequency: 14.9277
Max. Amplitude: 0.015127

Figure 4: Web published Front Panel as VI Server

Readings of a conventional Vibration System is shown in below Table 1 and corresponding plot is shown in figure 6 for the vibration parameters.

Table 1. Conventional Vibration System Values

S.No	Frequency (Hz)	Displacement (mm)	Velocity (mm/sec)	Acceleration (mm/sec^2)
1	5	1.8	45	16
2	6	1.7	41	22
3	7	1.5	40	28
4	8	1.6	46	30
5	9	1.4	36	32
6	10	1.2	44	36

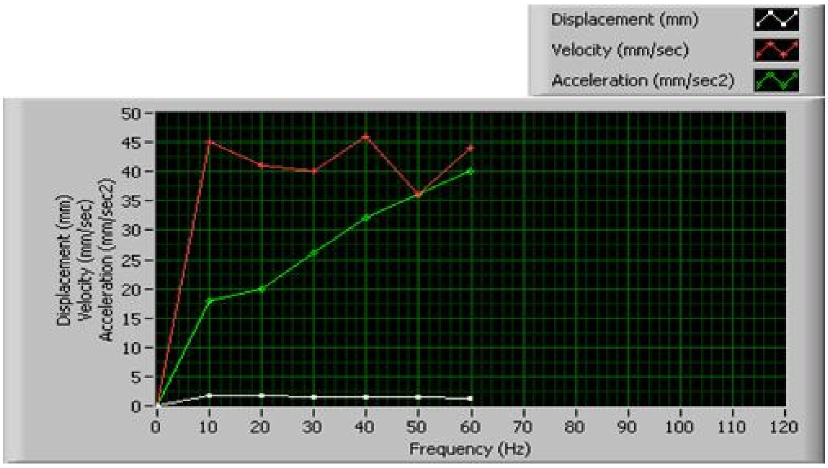


Figure 6: Vibration Parameters in Conventional Instruments

Virtual Instrumentation based Vibration System readings are illustrated in Table 2 and the graph is following as in figure 7.

Table 2. VI based Vibration System Values

S.No	Frequency (Hz)	Displacement (mm)	Velocity (mm/sec)	Acceleration (mm/sec^2)
1	5	1.5	40	16
2	6	1.48	41	22
3	7	1.46	38	28
4	8	1.42	40	30
5	9	1.4	36	30
6	10	1.38	40	30

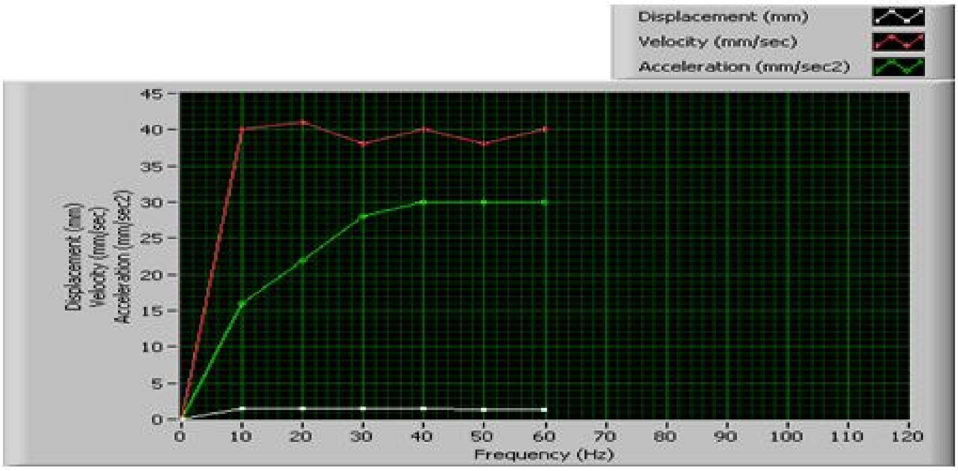


Figure 7: Vibration Parameters in Virtual Instruments

Based on the readings acquired it can be concluded that in accordance with the standard readings the VI based system is much more accurate, reliable and efficient than the conventional system.

CONCLUSION

This paper provides a new instrumental approach to analyze heavy machinery vibration. Proposed system is user friendly and cost effective over conventional system to implement for remote system analysis. Interactive front panel and graphical source code leads to faster development time and graphical displays of different parameters involved. Spectral analysis leads to evolve recommendations over the energy lost, machine to machine with VI server enability for network based automation.

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