CONDITION MONITORING OF CONTROL VALVE IN POWER PLANT

A DISSERTATION

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By

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Hivarkar

(Hivarkar Mahesh Padmakar)

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CCI	Control Component Incorporation
CVCS	Chemical Volume Control System
DVC	Digital Valve Controller
ERPI	Electrical Power Research Institute
FFT	Fast Fourier Transform
IEC	International Electrotechnical Commission
ISA	Instrumentation Society of America

CHAPTER 1
LITERATURE SURVEY

1.1 Introduction

Historical experience has documented the limitations of traditional component removal, based upon "statistical safe life" estimates. With the aid of emergent diagnostic and prognostic techniques, the obvious choice to improve the accuracy of component retirement is through its condition monitoring and automated health management.

Benefits of this approach include: improved safety, heightened system readiness, and reduced life cycle costs resulting from better understanding of the timing of critical system failure modes. A critical step towards accomplishing some of these economic and logistic goals is through the development of efficient condition monitoring systems. For different process components from existing data one can get knowledge base of acceptable parameter responses over the operational regime. Replacing conservative threshold limits of measurement with component that are updated on regular intervals can produce much more accurate predictions of a component's health and failure progression. This approach also benefits the maintenance decision process by improving fault classification/diagnosis. Predictions of the component future health status will provide the lead-time necessary to schedule maintenance proactively and avoid catastrophic failures.

After considering the benefits of Condition Monitoring techniques it has been decided to do the dissertation work in this field only.

While doing literature survey on this topic; following points are taken into account:

- Present development in condition monitoring.
- Selection of component for which condition monitoring has to be done.
- What are present methods to do it? And which method should be chosen.

Different papers, thesis, articles, transcriptions related to this topic have been searched on different sites like IEEE, Google, Altavista, etc.

1.2 Present status of condition monitoring

Industry and government are constantly under economic pressures to reduce costs while increasing service and productivity. To this endeavor, condition monitoring of critical plant machinery and associated components has become a very important tool over recent years. In the past, critical machinery would require scheduled shutdown periods for preventive maintenance. This approach is not only costly but has actually been know to reduce reliability by disturbing elements which were working perfectly well or by introducing faults during dismantling and reassembly.

Condition monitoring involves the continuous or periodic analysis of operational equipment and the identification of problems before component breakage or machine failure. Predictive maintenance helps to estimate the time remaining before machine breakdown, thus enabling equipment to be maintained as required and taken off-line only as necessary.

The benefits of Condition monitoring and Predictive maintenance techniques include,

- Condition and nature of faults is quickly identified without having to resort to visual inspections.
- Increasing machine availability and performance.
- Reducing overall maintenance requirements.
- The period between shutdowns can be confidently extended.
- Reducing emergency shutdowns and lost production.
- Corrective action can be implemented to prolong machine operation.
- Increasing equipment and machine life.
- Early detection of problems can help to plan and effectively apply downtime, material and labor.
- Parts or components with long lead times can be ordered well in advance, helping to reduce inventory costs.
- Significant cost savings, as highlighted by the above-mentioned benefits.

1.2.1 Condition monitoring by vibration measurement

With the increasing application of Condition monitoring and Predictive maintenance it is extremely important that the machinery and equipment characteristics are well understood, that the operation of machinery be optimized to reduce costs while improving product and service quality, and that a reliable monitoring system be incorporated within the equipment. Condition monitoring requires a measurement to be taken from a machine and used to indicate the current working condition of that machine. There are many measurements that can be taken, for example temperature, pressure or load, but the most commonly used parameter is vibration [1]. The moving elements of a machine will produce vibrations and by analyzing these it may be possible to obtain a clearly warning of developing faults.

Vibration in machinery has two definite identifying characteristics, vibration frequency and vibration amplitude. The principal component of the vibration frequency has same value as natural frequency of component or as the machine rpm for rotary component. The vibration amplitude can be measured in three different ways: displacement, velocity, and acceleration. An increase in vibration amplitude almost always indicates some mechanical fault. Operation criteria representing vibration boundary levels for stationary and rotating running condition depends upon type of machine or component.

During the literature survey it has been found that condition monitoring systems are well developed for rotating machines like pumps, motors, turbines and power transformers, bearings, gearboxes. While going through searched literature it is found that all over the world lots of research have been done in condition monitoring of all above listed components. Many systems have been proposed for vibration monitoring. Simple system involves hand held data collectors, which are carried to various points on the plant and finally returned to central computer where the data is retrieved and processed [2]. This system cannot provide real time monitoring and suffers from poor accuracy and repeatability. Alternative systems have been built using general – purpose data acquisition boards linked to a large computer. These systems provide real time operation but often require huge volumes of raw data to be transferred and processed and sometimes do not provide high accuracy.

But the principal of all the conditioning monitoring systems more or less are same. One system differ from other may be in sensor, signal conditioning techniques, signal processing methods or algorithm used. The figure 1.1 shows general block diagram of condition monitoring system.

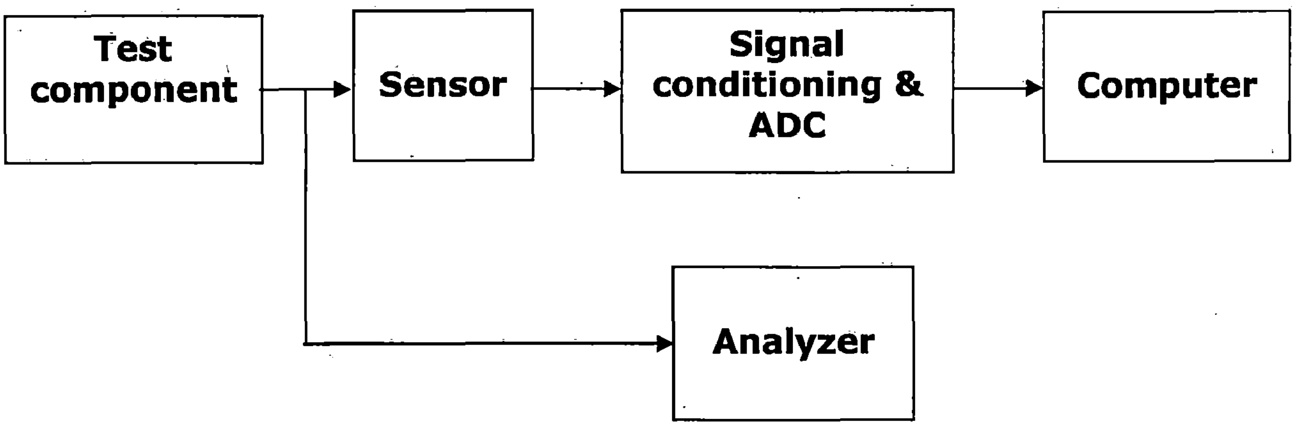


Fig 1.1 Block diagram of condition monitoring system

The main part of condition monitoring is to choose correct the vibration sensor depending upon amplitude and frequency range of vibrations. Then it consists of signal conditioning block, which may contain charge amplifier, or some techniques, that are adopted for anti-interference. System has good data acquisition system meeting the sampling criteria for given application. The acquired data is processed in PC using different techniques [3]. Generally frequency analysis, time analysis, power spectrum analysis and cestrum analysis have been preferred. Even different techniques like genetic algorithm, artificial neural network, and fuzzy logic are used for automatic fault detection in continuous condition monitoring. In many portable condition-monitoring systems, analyzer is used that have in built data acquisition, signal conditioning and analyzing part.

Nuclear plant, power plant, pulp & paper mill are the some industries where condition monitoring is more popular. Some of the examples are,

- All power plants are equipped with an online vibration monitoring system [4]. Existing systems are designed to monitor the shaft line of the turbine and steam inlet valves. Using this system for vibration monitoring of the stator end-windings gave significant benefits. On the one hand, the total cost of the monitoring program was reduced, and on the other hand, maintenance operator already knew what is problem and how to handle it.
- NTPC utilizes a vibration monitoring system at its power plants to evaluate machine performance. This system functions as an alarm monitor, displaying timely alarm data to control area. Diagnostic assessment of problems through the use of condition monitoring system has helped identify problems in system. It in all helped a lot to get prior alarms for failures in components and increased the operational efficiency of total Generating Station.

1.3 Component selection

From literature survey it has been found that less work has been done on condition monitoring of control valve, which is very important part of any process. As control valve is final element in control loop, its performance degradation will disturb the process in which it is working. In this dissertation control valve is chosen for condition monitoring.

The most common method for influencing the behavior of chemical processes is through the flow rate of process streams.

Usually, a variable resistance in the closed conduit or pipe is manipulated to influence the flow rate and achieve the desired process behavior. A valve with a variable opening for flow is the standard equipment used to introduce this variable resistance. The valve is selected because it is simple, reliable, relatively low cost and available for a wide range of process applications. Thus control valve is most important, but sometimes the most neglected, part of a control loop. The reason is usually the instrument engineer's unfamiliarity with the many facets, terminologies, and areas of engineering disciplines such as fluids mechanics, metallurgy, noise control, and piping and vessel design that can be involved depending on the severity of service condition.

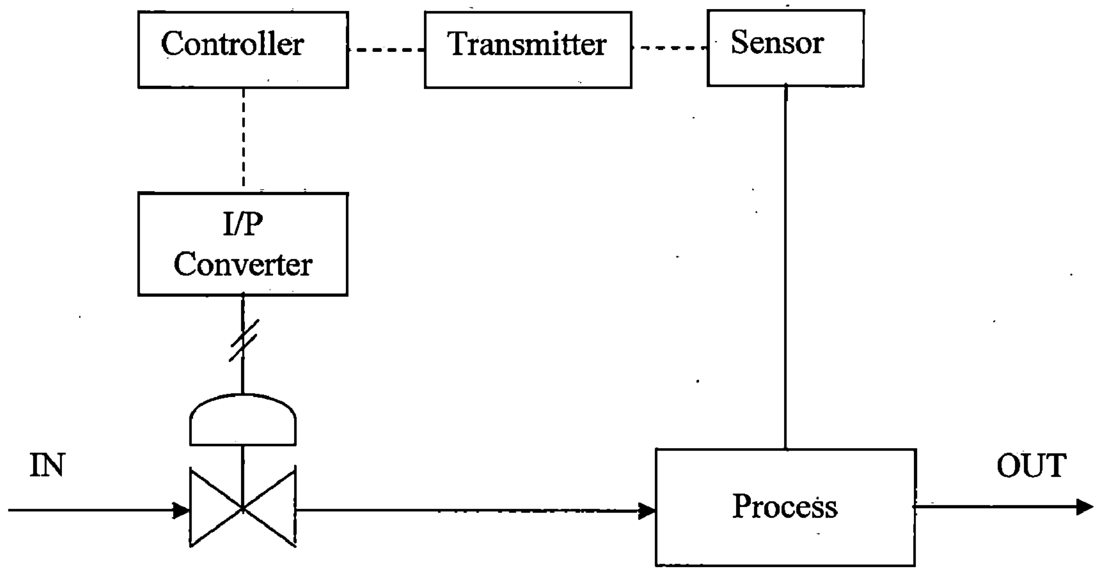


Fig 1.2 Block diagram of control loop

Any control loop usually consists of a sensor of the process condition, a transmitter and controller. Controller that compares the "process variable" received from the transmitter with the "set point" i.e. the desired process condition. The controller, in turn, sends a corrective signal to the "final control element", i.e. control valve. It is the last part of the loop and the "muscle "of the process control system.

Control valves affect the performance of plant in terms of output, reliability and availability because they are the final control elements in the operation. Therefore, critical evaluation of control valves must to be an integral part of any plant betterment program because the ultimate goal of such efforts is to improve the efficiency and reduce costs. Even control valves in the few severe service applications, which affect efficiency more than the rest of the valve population, have traditionally not been included in condition monitoring efforts.

Following some facts underlines the need for condition monitoring of control valve.

- Electrical Power Research Institute (EPRI) studies and Nuclear Regulatory Commission docket records indicate that over 1.5 million MW of energy production are lost each year because of valve-related shutdowns [6]. Table 1.1 shows the main steam and contaminated isolation valves related losses. In terms of lost energy, valve-related losses rank third behind turbine/generator and steam generator losses; 12 to 15 percent of the total outage time can be traced to valve problems.

Maintenance costs for valve components vary widely from plant to plant. In coal fired plant, between 20 and 60 percent of maintenance beget during outage is devoted to valves. In nuclear plant, up to 20 percent of the valves required repair each year.

Most plants have evolved to calendar scheduling of maintenance. Valves are attended to on a cyclic basis even if no signs of trouble can be detected. The schedules are deliberately conservative because unscheduled outages are so expensive. Cyclic scheduling, however, may leave out components that will fail before their planned service date if sign of deterioration are not clearly visible at an earlier shutdown. A less expensive and more effective valve maintenance schedule can be implemented only if the selection of valves to be serviced is made on a justifiable priority- of – need basis. The purpose of the control valve condition monitoring is to provide the means to do that.

Table 1.1 Summary of energy production losses by valve

Valve Category	Number of Events	Outage Time(hr)	Energy Equivalent (MWh)
Main steam/pressurizer overpressure protection	18	1,010	793,326
Other overpressure protection	07	329	247,896
Main steam isolation	15	2,131	1644,738
Containment isolation	12	1,286	1,040,619
Pressurizer spray	08	435	402,443
Check	08	547	458,518
Turbine control	02	46	3,727
Feed water regulating	02	36	27,156
Pressure/flow regulating	03	126	125,794
Other system stop valves	30	1,500	1,170,264
ALL	105	7,446	5,914,481**
** At $0.10/kWh; this represents $150 million per year.			

- Recent studies indicate that eliminating control valve problems alone can improve the heat rate of power plants in the range of 2% to 5% [7].

Control valves are the final control elements in the operation of a power plant. Therefore, plant efficiency is directly affected by non-performance of the valves, either in terms of output or in terms of reliability and availability. While contributions to plant efficiency loss from individual valve applications may be small, together they all can add up to be a significant value. When the invisible effects of the valve problems are taken into account, the net impact is even greater. The most critical factor used to judge the performance of a power plant is the heat rate of the unit. To achieve operation at lower heat rates the effect of controllable parameters on the plant heat rate should be quantified. Controllable losses typically are monitored and efforts are made to reduce these. Examples of severe service control valve applications in the main power-generating loop in gas power plants are: Boiler feed pump (BFP), minimum flow control, feed water control, turbine bypass, atmospheric dump and emergency heater drains. In addition, there are many other severe applications in various other systems in gas power plants, such as the Waste Heat Recovery Boiler (WHRB), Water Injection (WI), service water, DM Water Plant.

The typical problems caused in severe service valves are:

- Premature trim and body erosion due to lack of control of fluid velocity along the flow path.
- Poor shutoff capability, i.e. leakage through the valve under closed condition, because of inadequate actuator thrust, damage to the sealing surfaces caused by high velocities and improper calibration.

- Process controllability problems with the valve operating at lower openings because of wear.
- Poor dynamic response.

Most often, the visible effects of control valve problems are:

- Loss in production capacity,
- Occasional plant trips,
- Frequent maintenance, and
- Safety concerns.

In addition, there are other costs, which may sometimes be invisible, because of the valve problems:

- Penalty in heat rate/ high operating cost,
- Longer time for startup,
- Lower Unit availability,
- Collateral damage to other expensive plant equipment (e.g., turbine, heater, boiler tubes) because of occasional transients that cause operation beyond normal operating conditions, and
- Low flexibility in operation, e.g. part-load operation, or sliding pressure mode, in fossil power plants may not be possible even when it is desirable.

Thus all above facts indicate that condition monitoring of valve is very essential for betterment of plants.

1.4 Condition monitoring of control valve

Literature survey has been shown that comparatively less research is going on condition monitoring or diagnosis of control valve. Still companies like Emerson Process Management and organization like Instrumentation Society of America have been doing research in this area from many last years. But still most of the work is in experimentation stage only.

Emerson has introduced a major improvement – the Flow Scanner, a suitcase sized instrumentation package to test the valve assembly while it's operating in the field [8]. This allows a skilled valve analyst to determine whether the valve can be left in the line or whether it needs to be removed, disassembled and repaired. With the introduction of the digital valve positioner diagnostic technology migrated to the valve positioner. The Emerson has developed a digital positioner, which renamed as DVC (digital valve controller). Any valve assembly in the plant can be subjected to condition monitoring initiated directly from the control room or maintenance shop. Emerson's DVS requires proprietary software. Most of features that provide true valve diagnostic information still require the control valve assembly to be isolated from the process during testing.

CCI Valve, Avenida Empressa, Rancho Santa Margarita, CA has earned their distinguished reputation in the severe service valve industry through developing advanced technologies, strong application knowledge, and great customer relationships. It has been working on many areas in control valve including condition monitoring. Sanjay V. Sherikar, Ph.D., P.E., CCI, has presented paper on 'Evaluation of Control Valve Performance is Necessary in Plant Betterment Programs' [7]. In this paper he addressed many problems in power plant related to control valve and given some solutions to address them.

Herbert L. Miller, P.E., Vice President, Technology & standards, CCI has worked on Piping Vibration Involving Control Valve [9]. In this paper he made main focus on different reasons of vibrations and how they affect the health of control valve.

J.G. MacKinnon of CCI has presented paper on 'Recent Advances in Standardizing Valve Noise Prediction' [10]. David Minoofar of CCI & A.V. Karvelis of Babcock & Wilcox Research Center have presented a paper 'Practical Considerations in Noise Testing of Quiet Valves' at Inter Noise 80 in Miami, Florida. All these papers helped to know the different methods for prediction of noise, their merits and demerits. Noise analysis helps to know what is going wrong in the control valve. It is one of the major parameter that tells about health of control valve.

Some peoples are working individually on condition monitoring of control valve. Basically they are trying to focus on one or other type of control valve or its parts like actuators. Takeki Nogami of Shikoku Research Institute Inc, Takamatsu-si, Japan & Yoshihide Yokoi of The university of Tokushima, Takamatsu-si, Japan presented a paper on 'Failure Diagnosis system on Pneumatic control Valve' they developed condition monitoring system by using neural network technology for actuators of pneumatic control valve [11]. The data of 30 failure patterns are experimentally collected using more than ten sensors. A fast Fourier transform (FFT) is carried out on the time series of the sensor signals. The data of the magnitude spectrum, phase difference and others are used as the characteristic parameters in the failure diagnosis. Appropriate failure diagnosis information is extracted from the data. Similarities among the failure characteristics are established using fuzzy clustering and statistical analysis. The prototype consists of plural sub networks and one specific sensor signal, and deals with the magnitude spectra from the sensor signal. The main network makes the final decision according to the output from the sub-networks and other data. The number of network connections can be reduced by approximately 40% without degradation of the recognition capability, in comparison with a conventional system where only one neural network is used.

There is one other paper 'Fault diagnosis in industrial control valves and actuators' in Instrumentation and Measurement Technology Conference Proceedings, IEEE, by Sharif M.A, Grosvenor R.I., from Univ. of Wales, Cardiff. This research paper reports on experimental work carried out on an industrial type globe valve with a diaphragm type actuator [12]. The effects of a blocked vent hole, which is in the upper diaphragm casing, on the overall valve performance, are reported. A high resolution pressure sensor together with the latest valve diagnostic system, developed by Fisher-Rosemount, were extensively used to monitor the variation in the pressure that was developed in the upper diaphragm casing as a result of different amounts of vent hole blockage. Furthermore, the effects on several parameters including valve travel, spring rate, frictional force and the available seat load were established. Valve internal leakage is a common malfunction with industrial control valves. The causes of such leakage are numerous, including damaged plug or seat, insufficient seat load or reduced spring rate. The reduction in the available seat load due to the increased packing friction was considered in detail and significant results are reported. The paper concludes by highlighting some of the limitations of the existing diagnostic system and outlines the future work, which involves the development of a user-friendly expert system based on fuzzy rules, and is capable of diagnosing faults in control valves more accurately.

1.5 Significance of study

- From literature survey it has been realized that condition monitoring of plant component is must because it,

- Minimize total downtime during turnaround. Knowing what to do before the turnaround occurs is an essential part of reducing down time.
- Maximize the elapsed time between turnarounds. This directly affects process up time and profitability.
- Sustain reliable performance reducing or eliminating unscheduled downtime enhances reliability & profitability.

➢ Control valve is final element in any control loop, so its condition monitoring is necessary in process plant betterment program.

➢ It is very difficult to do complete condition monitoring of control valve in all respect because the methods developed up till now do it for some part of control valve or test only some properties of control valve.

➢ There are different methods for condition monitoring. Every method has its own merits and demerits. Selection of the method should be done depending upon requirement of plant. Some criteria like cost, complexity should be considered while doing so.

➢ Condition monitoring of control valve can be done by measuring parameters like vibration, noise. Vibration monitoring is common trend in rotatory machines. Same practices can be implemented in control valve.

1.6 Scope of the work

1. Methodology to be adopted

As it has been seen till now many peoples have tried their ways to do condition monitoring. The first part of dissertation is to find methodology that will be applicable for maximum number of control valves. At the same time it should be efficient to give at least broad idea about the health of control valve. So that maintenance people can make decision regarding its maintenance. In this stage of dissertation firstly the parameters are selected which will give maximum information about developing faults in control valve.

2. Sensor selection

Sensor selection is important for condition monitoring. Reliability & accuracy of condition monitoring mainly depends upon appropriate sensor selection. The appropriate sensor selection is vital part of this dissertation.

3. Selection of Analyzer

Appropriate analyzer is selected which meets required specification like amplitude, frequency, signal processing capabilities etc.

4. Software

Analysis of acquired signal is main part of any condition monitoring. Signal processing is done to extract the information from the signal. Software is developed to extract maximum data from signal and condition monitoring is made easier. Software has following modes of operation,

- Estimation: Software estimates the signal value (noise signal) using available standards method (ISA).
- Digital Filtering: The main feature of the data processing algorithm is digital frequency filtering of the time domain data. Filters are used to filter out unwanted or interfering signals. The measured signal will be over a wide frequency band. In order to reveal the individual frequency components making up the wide-band signal frequency analysis is performed.
- Fast Fourier Transform: The another feature of the data processing algorithm is the fast Fourier transform (FFT) which

transforms the digital data in time domain into its frequency components. Because of FFT engineers have sought ways to simplify their work by looking for the frequency content of signal, which he should, able to correlate to physical phenomenon in component.

5. Experimentation & Result

With selected sensor and analyzer set up is built to take reading. The reading is analyzed. It also compared with available standard data and depending on result obtained comments are made on the condition of tested valve. This experimentation has been done for different control valves.

Chapter 1 has covered the literature survey part. Chapter 2 deals with design aspects of condition monitoring system. It contains methodology to be adopted, parameter selection, sensor selection, vibration analyzer and software part. Chapter 3 tells about experimentation carried out with developed system and results obtained. And the last 4th chapter discusses the future scope and concludes on dissertation work.

CHAPTER 2
DESIGN

2.1 Introduction

While designing a condition monitoring system for control valve following reasons should be taken into consideration:

- Eliminate Disassembly: The system should able to show the fault in the control valve without disassembling it.
- Reduce Unscheduled Downtime: If particular control valve is beginning to exhibit signs of incipient failure, it is possible to schedule its repair during a convenient time. Provision can be made to ensure that the proper replacement parts, tools, equipment and manpower will be available at that time. This kind of planning will obviously reduce annual maintenance budget and help to ensure that repair will be completed on time.
- Avoid Wrecks: By following the condition monitoring of each control valve as it ages, one can predict its failures modes before they occurs. By not operating control valve to total destruction, the chances of making rapid repair are greatly enhanced and the probability of injury is removed.

Many peoples are interested in the advantages of condition monitoring fail to properly delineate the different aspect of condition monitoring. And as they set up a program using the wrong type of equipment, improper instrumentation, poor data, they don't get fruits of condition monitoring. Therefore care must be taken during design stage.

2.2 Methodology to be adopted

While adopting any methodology it has been seen that the developed system will be:

- Universally Applicable,
- Effectiveness,
- Cost Effective, and
- Easy in interpretation.

Condition monitoring requires measurement of different parameter of control valve. So in the system development first step is selection of parameter which helps in efficient condition monitoring.

2.2.1 Parameter selection

For effective condition monitoring it is always important to measure the parameter that will give maximum information about the fault developing in control valve. In case of control valve one can measure speed of response, flow response, temperature variation at outlet to get health status of control valve. Measurement system required for some of these parameters changes with type of control valve. So these systems will not be portable and universal. Measurement of temperature variation will not able to give idea about health of control valve on its own. Basically for complete monitoring of control valve one has to measure all these parameters. But it is not practicable.

Control valves are generally present whenever fluid flow regulation is required and are therefore found in a wide variety of industrial applications. Control valves regulate flow by increasing or decreasing the fluid pressure drop across an element. These pressure drop adjustments are usually accompanied by noise generation. The vibration is another parameter that is very difficult to avoid. It usually occurs because of dynamic effect of turbulence of fluid in control valve.

Thus noise & vibration will be measure of control vales health. So it has been decided to use these two parameters for condition monitoring.

2.2.2 Advantages of noise & vibration measurement

- They give us at least broad idea about the health of control valve.
- The system developed for measurement of these parameters is portable
- It is applicable to all type of control valve.
- As we are using one system only for all control valves available in plant, it is cost effective also.
- The sensors required for measurement of these parameters are well developed and easily available. Also well-developed analyzer recorders are also available to analysis these signals.

2.2.3 Valve noise & vibration sources

There are five basic sources of valve noise & vibration [13]. These are follows:

- Mechanical vibration,
- Resonant vibration,
- Inner- valve instability,
- Hydrodynamic, via turbulence or cavitations, and
- Aerodynamic.

Mechanical Vibration

Mechanical vibration noise in a valve is stem and plug vibrations caused by fluid turbulence and unsteady flow generated by velocity and / or large mass flows. This noise source is generally unpredictable in advance. Noise levels are under 90dBA and the 50 to 1500 Hz range. Valve damage is predominating problem in this case. This problem may be a manufacture's design problem, and it should be referred to them if it is encountered.

Resonant Vibration

Resonant vibration is a form of mechanical vibration caused by the resonant excitation of the valve internals caused by the fluid flow. It is characterized by a usually annoying, narrow -band, single pitch tone in the range of 2000 to 8000 Hz. The noise levels of this kind of vibration are much higher, often 90dBA or more, and are commonly called a screech or whistle.

Valve whistle occurs when the characteristic one or several frequencies generated by the turbulence that occurs at the throttling point happen to acoustically match the resonant frequencies of the system, usually the downstream piping whistle tends occur only at specific valve opening, generally over a band of 10% stroke. Screech, on the other hand, occurs when the frequency of the disturbance also happens to match a mechanical natural frequency of valve internals and on occasion of the piping structure. This situation, while less common, is far more destructive than whistle. Whistle and screech are required to be corrected because valve is likely to fail as result of accelerated mechanical wear, localized metal overheating, or even the failure of downstream piping attachments.

Inner valve instability

The final type of mechanical vibration is inner – valve instability, which is usually due to mass flow turbulence. The relationship between velocity and static pressure forces across the plug face and the force balance relationship in the vertical direction varies with time. This can produce vertical stem oscillations via the valve plug and actuator force balance. Noise is usually a low level rattle, rumble or chugging. It is usually in the 30 to 60 Hz frequency and almost always under 100 Hz and less than 90 dBA. This instability can accelerate the wear of valve stem packing & guides and can also lead to trim damage in some cases.

Hydrodynamic noise

Hydrodynamic noise is associated with either liquid turbulence or cavitations. In many case noise may not be troublesome, although severe cavitations can generate noise levels well in excess of 90 dBA. The noise may range from a low frequency rumble or crackle, to a high frequency squeal.

Cavitation noise increases with flow rate and pressure drop to some maximum level and then decreases in magnitude. This is due to a reduction in downstream pressure recovery, which reduces the rate of bubble implosion. If the pressure recovery is reduced enough the valve goes into flashing mode. The main problem is severe damage to the valve.

Flashing is rarely a source of high valve noise that requires correction. It does result in the increase of the velocities at the valve's exit and in the downstream piping because of the higher specific volume of the vapor or two phase flow. These higher exit velocities can generate some level of noise from moderate to high and in some cases can cause valve trim or body erosion damage by high velocity liquid impingement.

Aerodynamic Noise

Aerodynamic noise is the chief source of valve noise and the main problems we have to deal with. The mechanisms of this noise generation are complex. It is known that valve noise basically is a result of the interaction of the high velocity turbulence of the fluid passing through the valve orifice with the boundary fluid in the mixing region, and/ or the generation of pressure cells immediately downstream of the orifice. The amount of acoustic power or sound energy generated is related to mass flow rate, upstream – to – downstream pressure ratio, valve geometry, and physical properties of fluid. Noise can be generated to a significant degree from subsonic velocities of Mach 0.4 to sonic Mach 1.0 and higher. Even at relatively low-pressure drops, substantial noise generation can occur with large mass flows.

Noise can increase even more dramatically as the pressure ratio continues to increase. Roughly three regimes are related to the internal pressure ratio across the orifice where the noise producing mechanisms increase in severity.

Thus noisy & vibrating valves can have induced damage, mechanically or acoustically induced vibration and valve instability. In severe cases valve service life is drastically reduced and in some cases, the complete destruction of the inner valve may occur in a matter of hours or even minutes. There have been cases where valves internals disintegrated in as little as 15 minutes due to the resonance of internals. At the same time because of this noise downstream piping and equipment may also be subject to acoustic vibration-induced damage.

2.2.4 Merits and limitations

➢ Noise measurement

- Noise measurement is non-contact type. So it is not subjected to damping effect as in case of vibration measurement.

- It can be made portable. Sensors are easily available. Sound level meters directly give readings in "dBA"
- Different theoretical models are available for predicting the noise levels like IEC, ISA. Thus using these models standard noise data for control valve is calculated. And measured data from actual control valve is compared with this standard data. Any deviation from calculated data will indicate the condition of control valve. Thus these methods avoid need of recording of noise data over long period of time.
- Noise measurement is always prone to interference. It is very difficult to measure noise of single control valve in presence of many other noise sources present in running plant. So it is very difficult to determine what the measured values actually indicate.
- The U.S. Occupational Safety and Health Act of 1970 established maximum permissible noise levels for all industries whose business affects interstate commerce. These maximum sound levels, given by OSHA, have become the accepted noise standard for most regulatory agencies. There are many noise sources in industrial plants, but major contributors are control valves operating under condition of high mass flow and / or pressure drop. Apart from condition monitoring, the measurement of noise will also help to know whether control valve is following the OSHA standards. It is necessary for betterment of plant operator's health.

➢ Vibration Measurement

- Main advantage of vibration measurement is it is less prone to interference.
- It can be also made portable. But special attention should be given while mounting the sensor. Sensors are easily available. Different

commercial analyzer are available that can be used directly for analysis of vibration signals.

- There is no model that can calculate vibration level in control valve. Thus there is no standard data is available for comparison purpose.
- Several years ago, a gas plant in Saudi Arabia experienced cracking of an Acid Flare Header. The site engineers carried out vibration tests to determine the cause and proposed several changes to the piping system to eliminate the cause of the damage. The changes were implemented, but the problem continued and eventually led to cracks in the piping as well as the header supports. A historical review of the problem was conducted once again. The analysis revealed that 10″ valves in the header system were the main contributors to the excessive vibration.

In Simonen and Gosselin (2001) piping vibration fatigue was reported as the cause of piping failures 29 percent of the time in US nuclear plants between 1961 and 1996. In small bore pipes, 2 inch and less, Vibration fatigue accounted for 45 percent of the piping failures. And they found that control valve is one of the main sources of vibration in piping.

Thus measurement of vibration is also important for the downstream components, which are susceptible to vibration.

After considering merits and demerits of both parameters obvious question come in mind is that whether there is any relations between noise and vibration generated by control valve. Acoustic energy generated by fluid flow through a control valve propagates through the piping end creates a fluctuating pressure field which forces the pipe walls to vibrate. These vibrations in turn cause pressure disturbances outside the pipe that radiate as sound.

2.2.5 Noise calculation using vibration measurement

It is very difficult to measure the sound generated by a single control valve, in presence of multiple noise sources and reflected sound. In these instances, converting the vibration levels of the pipeline in which the valve is installed to an equivalent sound pressure level eliminates many of the measurement problems. A study of sound transmission loss through the walls of commercial piping indicated the feasibility of converting pipe wall vibrations to sound levels.

Basic to the vibration-to-sound conversion technique is the relationship between acoustic power and radiation efficiency [14]. Ideally, the pressure of an acoustic wave is proportional to the particle velocity of the medium, through which the wave passes, with the constant of proportionality being the acoustic impedance of that medium. At the surface of a pipe, particle velocity is assumed equal to the velocity at which the pipe wall is vibrating. From this, acoustic wave pressure at the wall can be related ideally to wall velocity by:

$$P = \rho_0 c_0 v \qquad \text{... (2.1)}$$

Where P = rms acoustic pressure,

ρ_o = ambient density,

c_o = ambient wave speed ,

v = rms wall velocity,

It is helpful when discussing the transfer of acoustic energy from one location to another to utilize the parameters acoustic power and acoustic pressure. Acoustic pressures exist as a result of a net acoustic power flow through a finite area. Given a power level, an increase in area through which the power flows results in a decrease in pressure acting on that area.

Acoustic power is related to acoustic pressure by the following general formula:

$$W = P^2A/(\rho_0 c_0) \qquad \text{... (2.2)}$$

Where W =acoustic power

Substituting equation (2.1) into equation (2.2) yields the ideal acoustic power radiated by the vibrating pipe surface where the area of interest (A) is the surface of the pipe. Therefore:

$$W_I = \pi D l \rho_0 c_0 v^2 \qquad \text{... (2.3)}$$

Where W_I =ideal acoustic power

D = O.D. of pipe

l =pipe length

An actual acoustic power can be calculated from equation (2.2) based on sound pressure level measurements at a point away from the pipe surface. The area term would be that of a cylinder with a radius equal to the distance (r) from the observer to the centerline of the pipe. The actual power may be written from equation (2.2) as:

$$W_A = 2\Pi r l P^2/\rho_0 c_0 \qquad \text{... (2.4)}$$

Where W_A =actual acoustic power

A radiation efficiency term (σ) can be defined as the ratio of the actual acoustic power to the ideal acoustic power (W_A/W_I). Using this definition and equations (2.3) and (2.4) the relationship between the velocity of the pipe wall and the acoustic pressure at a point in space is:

$$P^2 = \rho_0^2 c_0^2 v^2 D\sigma/2r \qquad \text{... (2.5)}$$

Where σ = radiation efficiency

r = radial distance from centerline

It now is evident that if the radiation efficiency is known, the conversion from wall vibration velocity to acoustic pressure at any point in a free field can be made easily. The efficiency with which a surface radiates sound is a function of frequency. A coincident frequency (f_c) can be defined as the frequency at which the propagation velocity of a flexural wave in the pipe surface equals the velocity of sound in the acoustic medium. Coincident frequencies for many common steel pipes (air as the acoustic medium) are provided in Table 2.1.

Earlier studies indicate that radiation frequency is equal to unity above the coincident frequency and is directly proportional to the frequency in the region below coincidence.

In summary:

$$P^2 = \rho_0^2 c_0^2 v^2 Df/2rf_c \qquad f < f_c \qquad \ldots (2.6a)$$

$$P^2 = \rho_0^2 c_0^2 v^2 D/2r \qquad f \geq f_c \qquad \ldots (2.6b)$$

Acceleration measurements are often made rather than velocity measurements. The relationship

$$v^2 = a^2/w^2 \qquad \ldots (2.7)$$

where a = rms wall acceleration

can be used in equations (2.6a) and (2.6b) to yield

$$P^2 = a2\ \rho_0^2 c_0^2 D/4\pi^2 2rff_c \qquad f < f_c \qquad \ldots (2.8a)$$

$$P^2 = a2\ \rho_0^2 c_0^2 D/4\pi^2 2rf^2 \qquad f \geq f_c \qquad \ldots (2.8b)$$

Table 2.1 Coincidence Frequencies (f_c in Hz) of Steel Pipe in Air

PIPE DIA.	PIPE SCHEDULE		
	40	80	160
2	3241	2290	1455
4	2106	1451	939
6	1783	1155	695
8	1550	998	551
10	1367	998	-
12	1331	998	-
16	1331	998	-
24	1331	998	-

In order to use the preceding formulation it is necessary to convert the mean-square values to decibels. This can be accomplished using the following definitions.

Sound pressure level in dB—SPL = $10 \log P^2/P_0^2$.

Velocity level in dB—VdB = $10 \log v^2/v_0^2$.

Wall acceleration level in dB— AdB = $10 \log a^2/a_0^2$.

Widely accepted values for the various reference parameters are:

p_o = .0002 dynes/cm2.

v_o = 10–6 cm/sec.

a_o = 10 –3 cm/sec2.

Using the above definitions, equations (2.6a) and (2.6b) can be changed to decibel notation.

$$SPL=10\log v^2/10^{-12} + 10\log D/2r + 10\log f/f_c - 13.7 \quad f < f_c \quad \ldots (2.9a)$$

$$SPL=10\log v^2/10^{-12} + 10\log D/2r - 13.7 \quad f \geq f_c \quad \ldots (2.9b)$$

If the absolute value of the wall velocity is obtained with a vibration meter for example, then this value may be substituted directly in the first term (v) on the right hand side of the equations. This velocity must be expressed in cm/sec. When velocity measurements are taken in decibels referenced to 10^{-6} cm/sec, this velocity dB may be substituted for the entire first term ($10\log v^2/10^{-12}$)

In the same manner, equations (2.8a) and (2.8b) may be changed to decibel notation for acceleration measurements.

$$SPL=10\log a^2/10^{-6} + 10\log D/2r - 10\log ff_c + 30.4 \quad f < f_c \ldots (2.10a)$$

$$SPL=10\log a^2/10^{-6} + 10\log D/2r - 20\log f + 30.4 \quad f \geq f_c \ldots (2.10b)$$

Absolute values of the acceleration in cm/sec2 can be substituted directly for (a) in the first term of equations (2.10a) and (2.10b). If acceleration levels are taken in decibels relative to

10^{-3} cm/sec2 then this level may be substituted for the term $(10\log a^2/10^{-6})$.

When vibration levels are taken in decibels relative to a reference value other than presented here it is then necessary to equate an absolute value using the definitions of velocity-dB or acceleration-dB and substituting for (v) or (a) in equations (2.9) or (2.10), respectively.

The equations presented for velocity (9a, 9b) and acceleration (2.10a, 2.10b) allows conversion as a function of frequency. Ideally, to gain an overall equivalent sound pressure level from vibration measurements a summation of the corrected levels from each frequency band would be made. This is essential when acceleration is the quantity measured. However, when converting overall wall velocity to overall acoustic pressure, equation (2.9b) serves as a reasonable approximation as long as the velocity spectrum is not dominated by low frequency components.

So finally it has been decided that vibrations of down stream pipe wall of control valve is measured. Using above formulae these levels are converted to noise levels at different frequencies. And this noise levels can be compared to noise values calculated from standard data. The proposed block diagram of condition monitoring system is as shown in below

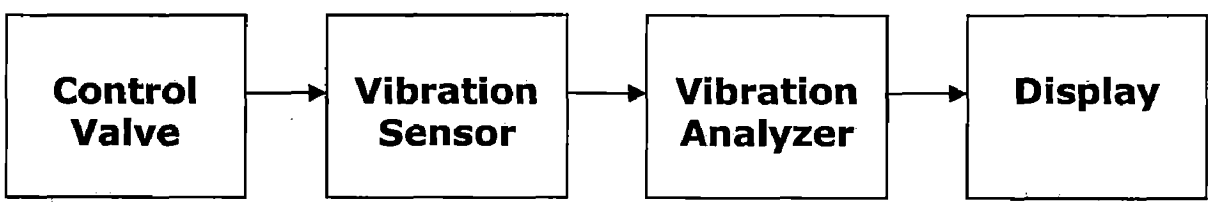


Fig 2.1 Block schematic of proposed system

2.3 Sensor selection

Sensor selection is important for control valve condition monitoring. Reliability & accuracy of condition monitoring mainly depends upon appropriate sensor selection. To accomplish this, a sensor selection methodology is adopted. Many sensors are available to meet today's demand. For condition monitoring of control valve sensors that are most sensitive to the measured parameters need to be selected. However, the problem of selecting the suitable sensors requires thorough understanding of:

- the process to be monitored;
- the failure types most likely to be encountered;
- Frequency range of parameters
- Amplitude range of parameters

With such an understanding, an appropriate sensor can then be selected for a given condition monitoring.

2.3.1 Vibration Sensor

Critical to vibration monitoring and analysis is the vibration sensor. A body is said to vibrate when it describes an oscillating motion about a reference position. This vibration signals in practice usually consist of many frequencies occurring simultaneously and the amplitude of vibration level also changes drastically over small period of time. So selection of appropriate sensor covering these frequencies & amplitude is very important.

We can measure vibration in 3 different parameters viz. displacement, velocity, acceleration. These parameters are mathematically related and can be derived from each other also. Selection of a sensor proportional to displacement, velocity or acceleration depends on the frequencies of interest and the signal levels involved [15].

Displacement

Displacement may be preferred for low speed applications (less than 10 Hz) Measurement of displacement will give the low frequency components most weight and conversely acceleration measurements will weight the level towards the high frequency components.

Velocity

Velocity may be preferred for applications with speeds ranging from 10-1000 Hz It has been proven that the overall RMS value of vibration velocity measured over the range 10 to 1000 Hz gives the best indication of a vibration's severity. A probable explanation is that a given velocity level corresponds to a given energy level so that vibration at low and high frequencies are equally weighted from a vibration energy point of view. In practice many components have a reasonably flat velocity spectrum.

Acceleration

Acceleration may be preferred for high frequency applications (10 - 10000 Hz and above) Acceleration measurements are weighted towards high frequency vibration components, so this parameter tends to be used where the frequency range of interest covers high frequencies.

It is found that the frequency range for control valve varies from few Hz to around 10 kHz. Even higher values of frequencies can be present in some cases. And maximum amplitude range is around 100g (all these values are taken from ISA handbook of control valve). Higher values of vibration levels can only be found because of cavitations only.

As measurement of acceleration for our application gives flat spectrum response over wide frequencies; acceleration is used to measure vibration.

Thus most suitable sensor for the application here is Piezoelectric Accelerometer [16].

The advantages of this sensor are:

- Extremely wide dynamic range, low output noise - suitable for shock measurement as well as for almost imperceptible vibration.
- Excellent linearity over their dynamic range.
- Wide frequency range.
- Compact yet highly sensitive.
- No moving parts - no wear.
- Self-generating - no external power required.
- Great variety of models available for nearly any purpose.
- Acceleration signal can be integrated to provide velocity and displacement.

Accelerometer Mounting

The method of mounting the accelerometer to the measuring point is one of the most critical factors in obtaining accurate results from practical vibration measurements. Sloppy mounting results in a reduction in the mounted resonant frequency, which can severely limit the useful frequency range of the accelerometer. The ideal mounting is by a threaded stud onto a flat, smooth surface. A thin layer of grease applied to the mounting surface before tightening down the accelerometer will usually improve the mounting stiffness. The tapped hole in the machine part should be sufficiently deep so that the stud is not forced into the base of the accelerometer. The resonant frequency attained is almost as high as the 32 kHz. But as we are using only one sensor (to make system more economic) and mounting it at different place of component of our interest we are using commonly used alternative mounting method. It is the use of a thin layer of bees-wax for sticking the accelerometer into place. Its resonant frequency is only slightly reduced (to 29 kHz) from that of stud mounting. Because bees-wax becomes soft at higher temperatures, the method is restricted to about 45°C. For higher temperature different adhesives are necessary depending on the temperature of the application.

We have accelerometer of KOSHA AX-22 whose specifications are as follows.

Table 2.2 Accelerometer specifications

Dynamic		
Reference sensitivity	mV/g	23.46
Voltage sensitivity	mV/g	22
Frequency range	Hz	20-20000
Charge sensitivity	PC/g	54.36
Amplitude linearity	%	0.1
Mounted Resonant frequency	kHz	>37
Maximum shock	G	2000
Electrical		
Capacitance	PF	2317
Physical		
Weight	Gm	40
Sensing element		Piezoelectric
Mounting torque	Nm	1.8
Case Material		Ss 316
Environmental		
Temperature	^{0}C	-20 to +180
Sealing		Epoxy

2.4 Vibration analyzer

The DI-2203 analyzer is used for vibration analysis. It is also called as structural analyzer because of its popular application in structure analysis. It is a powerful, dual-channeled, real-time FFT (Fast Fourier Transform) analyzer [17]. It is capable of measuring, processing, displaying and storing a wide range of analysis function. This combined with its portability, rugged structure and ability to operate from its internal NiCd batteries, makes the analyzer equally suitable for use in the laboratory or in the field.

Input Channels

It has two input channels CH1 & CH2. Signals input are made via the BNC connector. The maximum measurement voltage range is ±10v and minimum voltage range is ±10mV. The inputs are protected against higher voltage transients. Voltage inputs can be DC or AC coupled, while a third option; 'ACCEL/ICP' is available for direct connection of piezoelectric transducer. When ACCEL/ICP is selected, an internal power supply and signal condition circuit is switched into the corresponding channel input(s). The power supply is a 4mA constant current source with an open circuit voltage of 24V. This is used to power transducers without any external signal condition. The accelerometer input is also AC coupled with a −3dB point at approximately 0.34 kHz.

Processing

It has process screen to specify what function the analyzer will compute and what data formats will be used for display. This analyzer can compute FFT of signal, power spectrum, difference in two signals in time domain and many more. We are using frequency spectrum for our analysis. It gives vibration level for individual frequency over selected frequency band.

Display

The analyzer can show two displays one above the other in one screen only. So it is possible to view both time and spectrum data for channel 1 and channel2. Similarly, it is possible to view both time and spectrum data for channel1 and channel2.

Memory Storage

DI-2203 analyzer has internal battery backup memory. One can also use PCMCIA memory card to store data. When saving, both the setup state and corresponding data traces (signal, dual or map) are stored. However, when recalling it is possible to recall just a setup state, just data traces or data setup state and data traces.

2.5 Software

In this dissertation software is developed to make calculation and analysis part easier. MATLAB 6.5.1 is used platform for this software. MATLAB integrates mathematical computing, visualization, and a powerful language to provide a flexible environment for technical computing. The open architecture makes it easy to use MATLAB and its companion products to explore data, create algorithms, and create custom tools that provide early insights and competitive advantages. So MATLAB made programming easier and efficient.

Software is developed for

2.5.1 Noise calculation

It has been decided to measure vibration for condition monitoring. So next obvious question come in the mind is what acceptance level for vibration is? There are some methods to decide acceptance level. One is to keep recorded vibration data at critical points on control valve over the years. But for new control valve this will not possible. Second method of determination is available if plant has several identical control valves operating at identical condition. If three control valves have similar vibration spectra and forth control valve exhibits higher level of vibration, operating at similar condition. Then you can decide what may be acceptance level for these control valves. And third method is to determine the acceptance level depending upon standard data available. This standard data may be theory proven, or experimentally proven. Or there may be some models available to calculate these standards data. For vibration in control valve such standard data is not available. Equation relating noise & vibration are used to get acceptance level of vibration, because for noise calculation some theoretical models are available. These equations are already mentioned in previous paragraph. There has been progress on the development of a theoretical model of control valve noise prediction. Two of these are,

IEC noise prediction model

The first theoretical model is presented as a draft standard by the International Electrotechnical Commission, IEC, Reference 4. The draft is based on a German VDMA Association Task Force on Control Valve Noise Measurement and Prediction. The method predicts the energy generated its efficiency of conversion into sound energy, and then its transmission through the downstream pipe wall.

This method require acoustic coefficient. And there is no theoretical relation for acoustic coefficient; it must be implied from experimental measurements of sound pressure level with different type valves. The coefficient is also a function of pressure ratio, pressure recovery factor, number of orifices, valve opening, and other velocity distribution influences.

The fact that the method is so strongly dependent upon an experimental result detracts from the statement that it is a fundamentals based method [18]. The use of this method would require the valve manufacturers to measure and publish this coefficient for a large number of different types of valves. Based on the earlier display of uncertainty in prediction methods it would be expected that the published values for the acoustical coefficient would be no less definitive.

With the limited published information on the acoustical coefficient it is not possible to fully explore the accuracy of the IEC method.

ISA noise prediction model

The second theoretical method will be referred to as the ISA method. The ISA method is most theoretically based prediction method available. Its major benefit is its independence from an experimental database. ISA published a reasonably accurate prediction method in ISA-S7517, Control Valve Noise Prediction, which was based on modified free jet noise theories [19]. This method is used to calculate noise for control valve under test. It gives total noise L_a by following equation

$$L_a = 5 + L_{pi} - T_L + L_G \text{ dBA} \quad \ldots (2.11)$$

$$L_{pi} = 147 + \eta m + 10\log(C_V F_L P_1 P_2 / D_i^2) + 10\log(C_0/1128) \quad \ldots (2.12)$$

Where ηm = a modified acoustic efficiency factor

P_1= inlet pressure, psia

P_2= outlet pressure, psia

D_i= the inside diameter of the outlet pipe in inches

C_0 =speed of sound of fluid, ft/sec

$C_0 = 223*(K(T+460)/M)^{1/2}$; ft/sec

K= specific heat ratio(1.4 air, 1.33 steam)

T = fluid temperature, ^0F

M = molecular weight (29 air, 18 steam)

If P_1/P_2 is less than $P_1/P_{2\,critical}$, i.e. subsonic

In regime I i.e. $P_1/P_2 < P_1/P_{2\,critical}$:

$\eta m = 26\log[(P_1/P_2)/(P_1F_L^2-(P_1+P_2))]+10\log(F_L^2) - 38.6$ dB ... (2.13)

For Regimes II and III, i.e. P1/P2 > P1/P2critical, flow is sonic up to Mach 1.4:

$\eta m = -40+ 37\log[(P_1/P_2)/(P_1/P_{2critical})] + 10\log(F_L^2)$ dB ... (2.14)

for Regime IV, i.e.$P_1/P_2>P_1/P_{2break}$, jet velocity is above mach 1.4:

$\eta m = -31.2 + 10\log[(P_1/P_2)/(P_1/P_{2break})] + 10\log(F_L^2)$ dB ... (2.15)

for Regime V, i.e.$P_1/P_2 > 22\alpha$, jet velocity is above Mach 3

$\eta m = \eta m$ Max of Regime IV ... (2.16)

$$P_1/P_{2critical}=P_1/(P_1-0.47F_L^2P_1)$$

$$\alpha=(P_1/P_{2critical})/1.89$$

Now the transmission loss is

$$T_L = T_{Lfr} + \Delta T_{Lfp} \qquad \text{... (2.17)}$$

Where

$T_{Lfr}=138.4+10\log(rt_{p2}/D_i^4)+10\log((P_2GgC_0/14.7*1185)+1)-20\log(C_0)$

... (2.18)

D_i=pipe inside diameter, inches

G_g = specific gravity (air = 1, steam = 0.49)

r= distance from pipe = 39in+1/2D_i

t_p=pipe wall thickness, inches

and ΔT_{Lfp}

if $f_p \le f_0$ then $\Delta T_{Lfp} = 20\log(f_o / f_p) + 13\log(f_0 /f_r)$ ... (2.19)

if $f_p > f_0$ and $f_p \le f_r$ then $\Delta T_{Lfp} = 13\log(f_0 / f_r)$ ... (2.20)

if $f_p > f_r$ then $\Delta T_{Lfp} = 20\log(f_0 / f_r)$ ... (2.21)

where

f_r=62420/D_i, cps

f_0=14C_0/D_i, cps ... (2.22)

f_p=0.2Co*M_j / D_j; if $P_1/P_2<3$, cps ... (2.23)

$f_p=0.28\ C_0 / D_j (M_j^2-1)^{0.5}$; above $P_1/P_2=3$, cps ... (2.24)

$M_j=\{ (2/(K-1) [(P_1/P_2\alpha)^{(K-1)/K} - 1] \}$ ... (2.25)

The jet diameter, D_j is a function of the flow area and the valve style modifier F_d.

$$F_d = 1/(N_0)^{0.5}$$

Where N_0 is number of flow path.

$$D_j = 0.015\ F_d (C_v F_L)^{0.5} \text{ in feet} \quad \text{... (2.26)}$$

Finally L_G, the pipe velocity correction factor, is:

$$L_G = 16 \log[1/ (1 - 0.02\ P_1\ C_v\ F_L / D_i^2\ P_2)] \quad \text{dB} \quad \text{... (2.27)}$$

It is very difficult to solve these so many equations. So software is developed to make life easier. This software take required parameters from user and solve these equation and give direct calculated value of noise.

The main draw back of this method is that it gives total noise level and cannot break the noise into its frequency component. So advance method of noise prediction can be used for this purpose.

Advance method

Procedure of calculating noise from valves can be broken down into two processes. The first is noise generation in the turbulence downstream of the valve. This noise travels essentially unimpeded through the downstream pipe exciting the pipe walls. The second process is the transmission of the noise within the pipe through the pipe wall to the surrounding space.

To simplify the formulation of noise prediction formulas, noise generation processes are divided into three modes depending on the flow Mach number. Below Mach 1 the flow is definition subsonic. In this mode noise generated by a turbulence interaction dominates noise generation. Between these two modes is a transition mode where the total noise is a complex combination of mechanism.

In the subsonic mode the sound power behaves according to the proportionality

$$W \alpha \; wM^5C^2 \quad (2.31)$$

The magnitude of the sound power is often conveniently tied to experiments through the use of a term called the acoustic efficiency factor. This factor is the ratio of the sound power to experiments through the use of a term called acoustic efficiency factor. This factor is the ratio of the sound power to the mechanical power of the jet.

$$\eta = W/(wV^2) \quad \ldots (2.32)$$

Experiments have shown that the value of the acoustic efficiency at the point where the jet is exactly sonic is approximately 4*10(-4). This constant allows changing expression (1) to an equation.

$$W = 2.00*10^{-4}wM^5C^2 \quad \text{for } M<=1.0 \quad \ldots (2.33)$$

In the high sonic mode where noise is dominated by shock cell turbulence interaction the sound power is given by the expression:

$$W = 2.07*10^{-3}wC^2(M^2-1)^{1/2} \quad \text{for } M>= 1.41 \quad \ldots (2.34)$$

Noise generation in the transition mode between these two modes is complex. While it may be theoretically possible to predict the noise generation, from a practical standpoint, the noise is simply approximated by interpolation from tentatively chosen by the Instrument Society of America is

$$W = 2.00*10^{-4}M^{6.8}C^2 \quad \text{for } 1.0<M<1.41 \quad \ldots (2.35)$$

This equation meets the values of the high sonic & subsonic mode at their common borders.

Predicting the sound power is done by solving the appropriate equation 2.33, 2.34, or 2.35. The mass flow rate can be determined from conventional valve sizing methods. The speed of sound is determined from the fluid conditions in the downstream pipe.

$$C = (kR_uT/m) \quad \ldots (2.36)$$

Ru is the Universal gas Constsant(831.5 Joules/Kg mole k). Calculating the Mach number is somewhat more involved. The freely expanded Mach number is found from the expression.

$$M=[(2/(k-1))[(P_1/P_2a)(k-1/k)]-1]^{1/2} \quad \text{... (2.27)}$$

The parameter "a" is used to account for the recovery characteristics of the valve so that the product P_2a is equal to the vena contracta pressure when the flow is subsonic. In terms of the familiar pressure recovery term, FL, used in valve sizing:

$$a=1/(F_L^2)+P_1/P_2(1-1/F_L^2) \quad \text{... (2.28)}$$

Sound pressure

To develop expression to calculate the transmission loss through pipe walls and to predict sound levels in the traditional units, the sound power must be converted to sound pressure. Assuming that the sound field is uniformly distributed across the pipes diametric area the acoustic pressure P^2 is related to the sound power by

$$P^2 = \rho wc/\pi r^2 \quad \text{... (2.39)}$$

Since the acoustic pressure can vary over several orders of magnitude it has traditionally been converted to a term that varies logarithmically called the sound pressure level. This conversion is made using the equation:

$$L_p = 10 \log_{10} (P^2/P_0^2) \quad \text{... (2.40)}$$

Where L_P is measured in decibels (dB) is simply a reference quantity. Its value is $4 * 10^{-10}$ pascals^2. Combining equation 2.39 and 2.40 yields the expression:

$$L_P = 89+10\log_{10}W-10\log_{10}\rho+10\log_{10}C - 20\log_{10} r \quad \text{... (2.41)}$$

Frequency dependent sound pressure level

The transmission loss through the pipe wall is a complex function of the frequency of the noise, also "A" weighting factors used to develop the dBA scale are frequency dependent. For these reasons it is necessary to breakup the sound pressure into its frequency spectrum and analyzes the noise in each frequency band. After correcting for transmission loss and weighing the sound pressure, the frequency components are summed to final overall produced by the valve.

Test have confirmed theory that the frequency spectrum of control valve noise decrease as the fourth power of frequency well below the peak frequency and decrease as the inverse square of the frequency well above the peak.

The ISA has proposed a curve that approximates the shape of the frequency spectrum of noise. A relationship based on the one proposed by the ISA committee is

$$L_{fi} = -10\log_{10}\{[1+(fi/2fp)^2][1+(fp/2fi)^4]\}-7.9 \qquad \text{... (2.42)}$$

L_{fi} is used to correct the overall sound pressure level for the sound pressure in the one-third-octave band whose center frequency is fi. The peak frequency depends on the valve geometry i.e. the effective jet diameter and the noise generation mode. For the sub critical and the transition modes the peak frequency is given by:

$$f_p=0.2mC/D \qquad \text{... (2.43)}$$

For the high sonic mode the peak frequency is given by:

$$f_p=0.4C/(1.25D(M^2-1)^{1/2}) \qquad \text{... (2.44)}$$

where D is the diameter of the jet. The diameter can be calculated using:

$$D = 0.0046\ (C_v\, F_L / n)^{1/2} \qquad \text{... (2.45)}$$

Where n is the number of separate flow passages.

Transmission loss

The transmission loss is a function of the pipe geometry, Mach number of the flow in the pipe, frequency of the sound, and pressure in the downstream pipe. Instrument Society of America has published tables of transmission loss depending on pipe size and schedule, frequency of the noise, and the speed of the gas in the downstream pipe. These values must be corrected for downstream pressure according to the equation:

$$TL_{ci}=TL_i+10\log10[(P_2/101000+1)2.616+0.0315fit/(5.232+.0315f_it)] \quad \text{... (2.46)}$$

Where TL_i is the uncorrected transmission loss. Preliminary values for TL_i extracted from the draft of the ISA standard. These values are given for Mach numbers in the downstream pipe of 0 and 0.3. For match numbers between these two points the transmission loss terms can be interpolated. The Mach number in the downstream pipe can be calculated from the equation:

$$M_2 = w / \rho\pi r^2C \quad \text{... (2.47)}$$

Distance term

Beyond the pipe the sound level falls off with distance. The term correcting for distance is:

$$L_R=10 \log_{10} ((R+r)/r) \quad \text{... (2.48)}$$

Where R is distance from the pipe wall to the point of interest.

A scale

Finally the sound level must be weighted according to the "A" scale. The values of Lai are given as function of one-third-octave frequency band. Combining all these factors leads to the frequency dependent "A" weighted sound pressure level at a point outside the pipe downstream of the control valve.

$$L_{pi} = L_p + L_{fi} - TL_{ci} - L_R - L_{Ai} \quad \text{... (2.49)}$$

The overall "A" weighted sound pressure level is

$$SPL=10\log_{10}[\Sigma 10Lpi/10] \quad \text{...(2.50)}$$

The summation should be made over the frequency bands from 25 Hz to 20000Hz.

Table 2.3 "A" WEIGHTING FACTORS

Band Center Frequency Hz	LAi	Band Center Frequency Hz	LAi
10	-70.4	630	-1.9
12.5	-63.4	800	-0.8
16	-56.7	1000	-0
20	-50.5	1250	+0.6
25	-44.7	1600	+1.0
31.5	-39.4	2000	+1.2
40	-34.6	2500	+1.3
50	-30.2	3150	+1.2
63	-26.2	4000	+1.0
80	-22.5	5000	+0.5
100	-19.1	6300	-1.0
125	-16.1	8000	-1.1
160	-13.4	10000	-2.5
200	-10.9	12500	-4.3
250	-8.6	16000	-6.6
315	-6.6	20000	-9.3
400	-4.8		
500	-3.2		

The second program is written to calculate frequency dependent sound pressure level. This software take required input form user for control valve under test. It calculated "A" weighted sound levels which are functions of one-third octave frequency band. This spectrum forms database required for comparison in condition monitoring. It displays calculated noise spectrum. The calculated noise spectrum then stored and used for comparison with data acquired from control valve under test.

2.5.2 Frequency analysis

The measured signal will be over a wide frequency band. In order to reveal the individual frequency components making up the wide-band signal we perform a frequency analysis. For this purpose we use a filter, which only passes those parts of the vibration signal, which are contained in a narrow frequency band. Digital filters, or filters implemented in software, have several attractive advantages over analog filters such as, no temperature drift, no component tolerance, and wide frequency range.

In our application the pass band of the filter is moved sequentially over the whole frequency range of interest so that we obtain a separate signal level reading for each band.

Filter design

There are two basic types of filter used for the frequency analysis of vibration signals.

1. Constant bandwidth filter

A constant bandwidth filter has fixed bandwidth, for example 3 Hz, 10 Hz etc. If one has center frequency of 10 Hz and bandwidth of 5 Hz, the energy of two signals at say 8Hz and 11Hz will be added together and considered to be a single peak at 10 Hz. In this type of filter the error is for low frequencies than higher frequencies for same value of bandwidth. But in this case if more resolution is needed then it would be possible to go for small band width also.

Constant bandwidth analysis gives better frequency resolution at high frequencies and when plotted on a linear frequency scale is particularly valuable for sorting out harmonic patterns etc.

2. Constant percentage bandwidth filter

Bandwidth of this type of filters can be tuned to a given center frequency and will see all of the frequencies that exist in the region of +/- some percentage of the center frequency. There is no concise answer to the question of which type of frequency analysis to use.

Constant percentage bandwidth analysis tends to match the natural response of mechanical systems to forced vibrations, and allows a wide frequency range to be plotted on a compact chart. It is subsequently the analysis method, which is most generally used in vibration measurements. So it has been decided to use constant percentage bandwidth filter for vibration analysis.

Most constant percentage bandwidth filters are designed in terms of octave, for example, 1/3 octave, or 1/10 octave bandwidth. An octave is doubling of frequency. From 2 Hz to 4 Hz is an octave, from 4 to 8 is an octave, from 8 to 16 is an octave, and so on. The effective frequency bands for octave filters can be calculated as follows:

Lower corner frequency = 0.707 * center frequency.

Upper corner frequency = 1.414 * center frequency.

The problem of constant percentage bandwidth filters is that, a center frequency gets large number of frequencies that can pass through the filter increases and therefore resolution drops. Octave band filters give less resolution. A more detailed analysis can be obtained by using bands more narrow than octaves. If the octave is split into three parts, each of these three parts is referred to as a one-third-octave band, or often just a third-octave band. This 1/3 octave band has 3 times more resolution than octave band. The effective frequency bands for 1/3-octave filters, which are 23% wide, can be calculated as follows:

Lower corner frequency = 0.89 * center frequency.

Upper corner frequency = 1.12* center frequency.

Table 2.4 Standard octave and 1/3- octave center frequencies (in Hz).

Octave Bands	1/3 Octave Bands		
31.5	31.5	315	3,150
63	40	400	4,000
125	50	500	5,000
250	63	630	6,300
500	80	800	8,000
1,000	100	1,000	10,000
2,000	125	1,250	12,500
4,000	160	1,600	16,000
8,000	200	2,000	20,000
16,000	250	2,500	25,000

Another reason to select the 1/3-octave band filters is that ISA model calculates the noise levels at center frequencies of 1/3-octave band filters. These 1/3- octave band filters are designed in MATLAB. We are considering the vibration signals up to 20 kHz, so sampling frequency is very high. In DI-2203 analyzer for 20 kHz frequency band sampling frequency is given as 2.56 * 20 kHz, i.e. 51.2 kHz. So at this high sampling rate it is very difficult to develop filters at low frequencies. Different types of IIR filter like Butter worth, chebyshevI and II, elliptical has been tried. Elliptical filters have shown satisfactory result for low frequency filter.

Natural Frequency of Vibration

The valve has its own natural un-damped frequency. If valve is oscillating at this frequency then maximum damage will take place.

This un-damped natural frequency of vibration is dependent on valve size. Table 2.5 shows natural frequency for different size of valve.

Table 2.5 Un-damped natural frequency of control valve

Valve Size		Un-damped Natural Frequency
Inch	**Mm**	**Hz**
1	25	32
1 ½	37.5	27
2	50	22
3	75	16
4	100	14
6	150	10
8	200	9

Depending on the size of control valve filter is deigned for un-damped natural frequency of vibration to see whether valve oscillate at this frequency or not & if oscillating then at what amplitude it is oscillating.

Standing Waves

Within every flowing pipe there will be a sonic wave moving axially back and forth in the pipe. This is referred to as a standing wave. The frequency of this wave will be dependent upon the length of pipe and the sonic velocity of the fluid in the pipe. The length of the pipe is not the total length of pipeline with all of its valves, pumps, orifices, branches and so on, but it is the length between obstruction or acoustic barriers. Examples of obstructions would be valves, pumps and orifices. An acoustic barrier would be an opening into a larger pipe, reservoir, and the end of a pipe run such as "T" intersection where the branch of interest requires a right angle turn.

The frequency of standing wave can be calculated from the following equations and then compared with natural frequencies of valve to determine if there is a potential for this to be the root cause of the vibration.

Closed End Pipe f = i*c/(4*L) (2.51)

Opened End Pipe f = i*c/(2*L) (2.52)

As we are measuring control valve vibration on the downstream pipewall, standing waves act as interference for vibration signal. The filter is deigned to filter out standing waves from vibration signal measured form control valve. Also this filter is useful to get magnitude of standing waves which may cause damage to control valve.

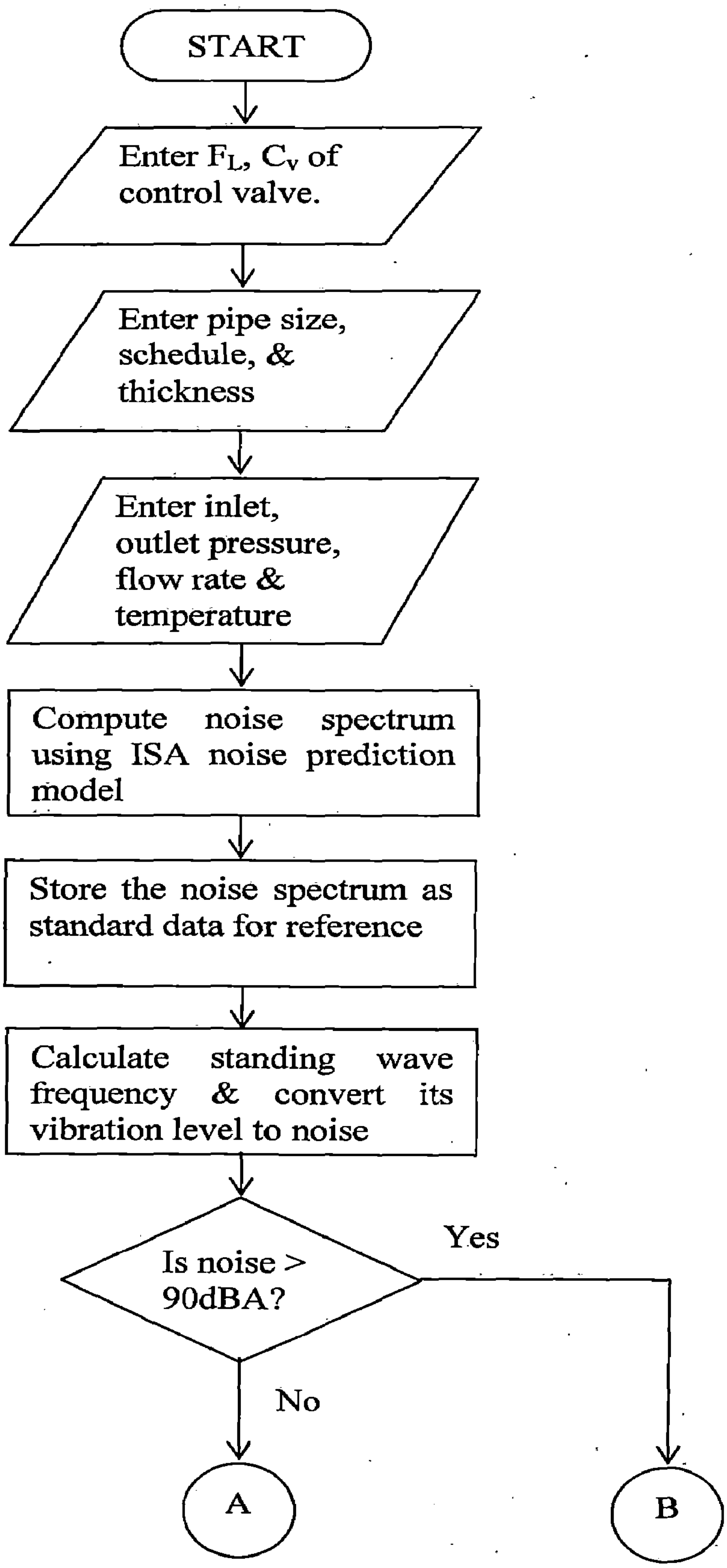
Spectrum Analysis

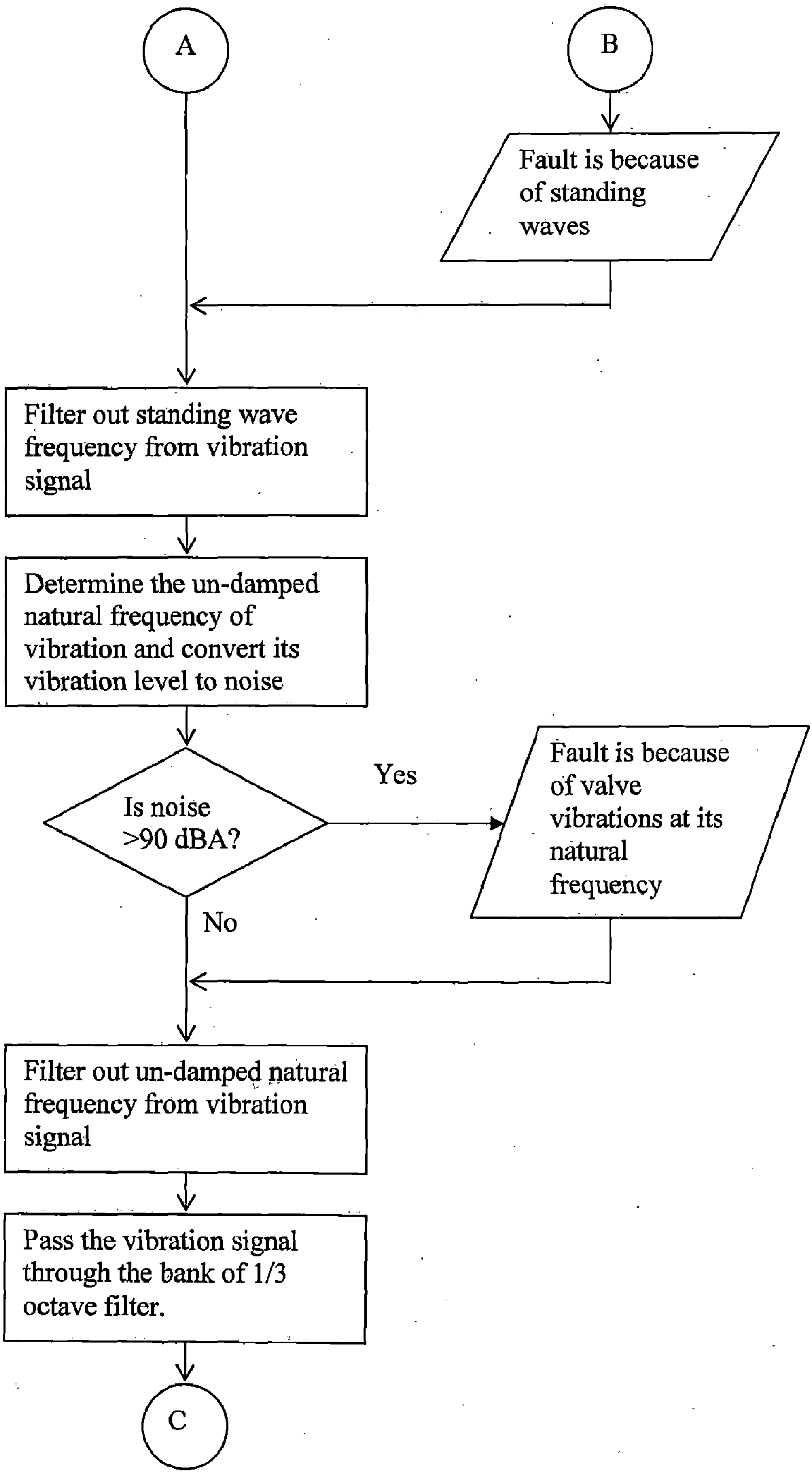
Spectrum is plot of amplitude verses frequency. The spectrum of the single sine wave A*sin(2 * pi * t * f) is a single spike of amplitude A and frequency f. the Fast Fourier Transform (FFT) algorithm is used which make the calculation of the frequency spectra of signal more efficient and rapid. 1024 point FFT is obtained for entire signal and also for specific frequency bands.

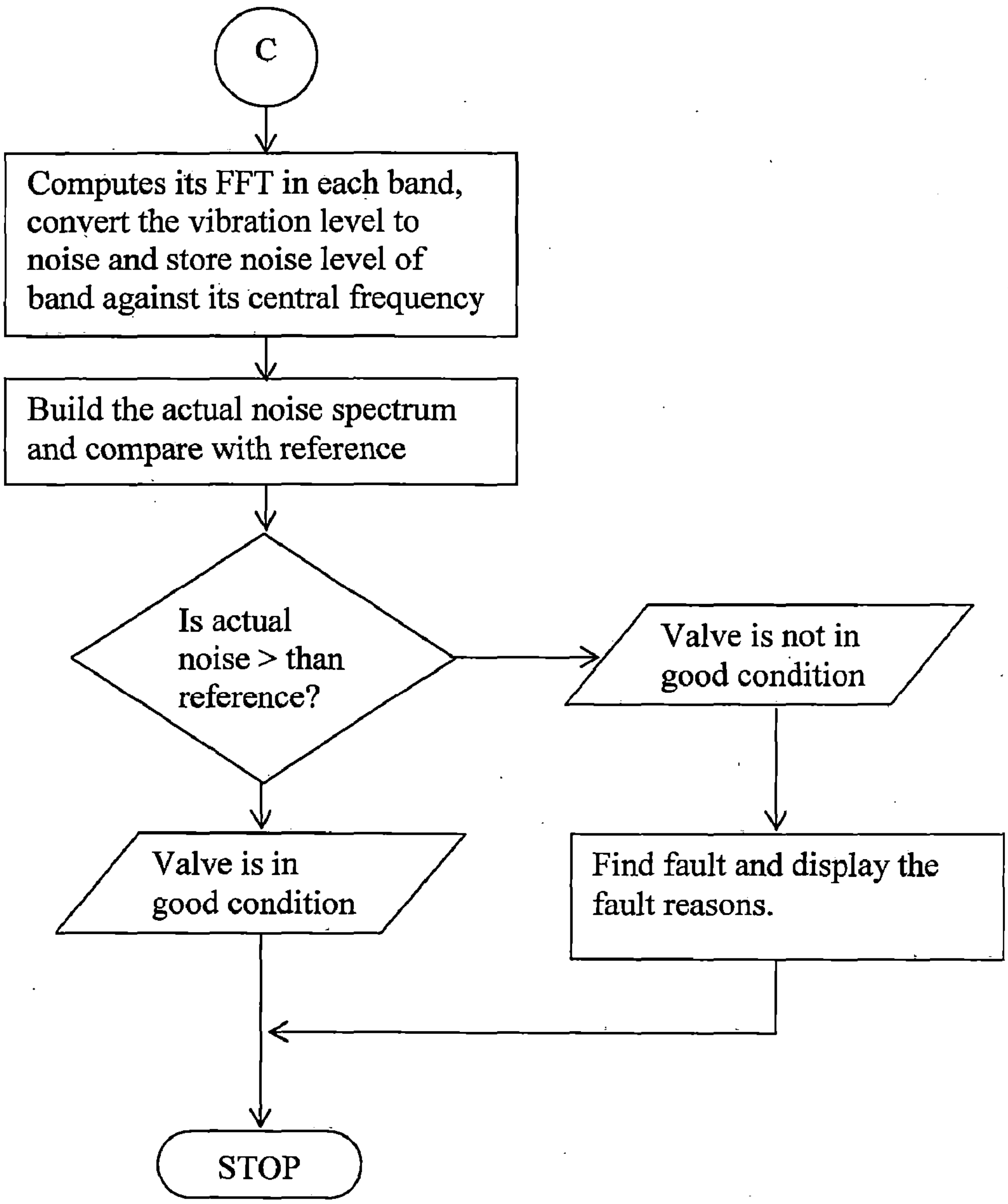
Software for condition monitoring of those valves for which noise can be predicted is developed so far. Valves for which we cannot predict the noise using standard models or for which we don't know the data required for noise prediction can also be condition monitoring by above developed software. In this case vibration signal is passed through the bank of 1/3 octave filter. In each band noise level is calculated and is compared with given threshold. If level is above 90 dBA then valve is becoming noisy. If level is above 110 dBA then the valve is noisy and it requires maintenance. If noise level is below

90dBA then valve is in good condition. In this case also we filtered out standing frequency and check for un-damped natural frequency of vibration of valve.

Flow chart of software




A
B
Fault is because of standing waves
Filter out standing wave frequency from vibration signal
Determine the un-damped natural frequency of vibration and convert its vibration level to noise
Is noise >90 dBA?
Yes
Fault is because of valve vibrations at its natural frequency
No
Filter out un-damped natural frequency from vibration signal
Pass the vibration signal through the bank of 1/3 octave filter.
C


C
Computes its FFT in each band, convert the vibration level to noise and store noise level of band against its central frequency
Build the actual noise spectrum and compare with reference
Is actual noise > than reference?
Valve is not in good condition
Valve is in good condition
Find fault and display the fault reasons.
STOP

CHAPTER 3
EXPERIMENTATION & RESULTS

3.1 Introduction

After successful completion of design part, it has been decided to start the monitoring of different control valves. Main aim of experimentation was to collect the vibration data and analysis it and try to comment on condition of control valves.

3.2 Prerequisites

For gaseous medium like air, steam; standard noise prediction methods are available. This predicated noise data could be used as standard database for condition monitoring. If medium is gaseous and you want to predict the noise for installed control valve using standards methods then it is required to know the following data:

1. Flow Coefficient (C_v) (dimensionless): A constant (C_v) related to the geometry of a valve, for a given travel, that can be used to establish flow capacity. It is the number of U.S. gallons per minute of 60^0F water that will flow through a valve with a one pound per square inch pressure drop.
2. Pressure Recovery Factor (F_L) (dimensionless):The liquid pressure recovery factor, F_L, is a measure of the valve's ability to convert the kinetic energy of the fluid at the vena contracta back into pressure. The internal geometry of the valve determines the value of F_L. It is a function of the direction of flow through the valve, the valve position, and whether the valve has a full or reduced seated trim.
3. Ratio of specific heats (k) (dimensionless) of medium
4. Molecular Weight of medium (kg/kg-mole)
5. Pipe Schedule
6. Pipe Size (inch or meter)

7. Pipe wall thickness (inch or meter)
8. Number of flow paths

Non-Vibration Parameter Measurements

For noise prediction it is required to measure

1. Inlet Pressure, P1 (Pa)
2. Outlet Pressure, P2 (Pa)
3. Flow Rate (kg/s)
4. Temperature of downstream fluid (^{0}F)

To measure all these parameters the setup required should have flow meter, pressure sensors and temperature sensor.

3.3 Experimental Setup

Experimental setup consists of accelerometer (KOSHA AX-22), connecting cables and DI-2203 structural analyzer. The block schematic of experimental setup is shown in figure 3.1

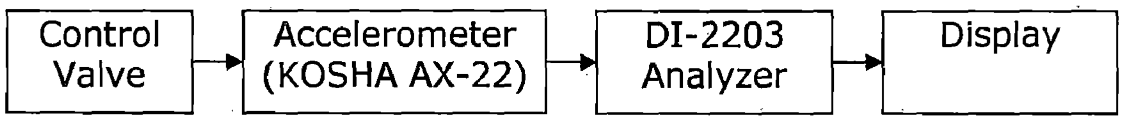


Fig 3.1 Experimentation Setup

Procedure

Steps followed while doing condition monitoring are as follows,

1. Obtain the data of valve for which condition monitoring has to be done. Get its C_v, F_L, no of flow paths. Get the molecular weight & ratio of specific heat of the fluid in use.

2. Mark spacing on the downstream pipewall where you want to take vibration reading. Be sure that surface is flat enough to hold accelerometer rigidly. If we assuming that all equipment is operating properly, then the important variable in making a measurement is the attachment of the accelerometer to the pipe wall. Rigid attachment to the pipewall is critical to accurate field results.

3. Don't locate the accelerometer on a flange, elbow, valve body, or other pipefitting. Measurements should be taken a minimum of two diameters from the end of a straight run of pipe. It is advantages if you locate it within few feet's from down stream of control valve.

4. Accelerometer Mounting: Mount the accelerometer on pipewall using thin layer of bee wax. If the application temperature is above the 45^0C then different adhesive which stands at higher temperature is necessary. One can also use treaded stud for mounting.

5. Different vibration frequencies can be present at different openings of control valve. One can take vibration reading at different flow opening. So make settings accordingly. For field situated control valve one can also take reading at operating conditions.

6. Power on the DI-2203 analyzer. Wait for some time because during power-up process, the analyzer resets its internal processors and carries out a self-test to ensure that internal circuitry is functioning correctly. It also checks the integrity of the data and setups stored to internal SRAM memory has been maintained. If any problems are encountered they are reported in the form of error messages displayed on the screen.

7. After successful starting of analyzer go into "Calibration Screen" by pressing the SHIFT then STRAT hardkeys. Calibration screen has,

A. Input Screen - This screen has setting for range of input parameters, coupling (AC, DC, and ACCEL/ICP).

B. Trigger Screen - This screen has acquisition mode, trigger source, and trigger setting.

C. Frequency Screen - This screen has setting for analysis bandwidth, FFT block size (samples/spectral line), X-axis scaling, anti-alias filters.

D. Processing Screen - This screen has setting for functions like spectrum, power density spectrum and so on.

E. Display Screen - This screen has setting for display format.

So do the proper setting as per the requirement. The important settings are coupling set to ACCEL/ICP, analysis function set to spectrum with linear Y axis scale and display format setting should be such that time and spectrum plots are one above the other.

8. After completion with setting part, connect the accelerometer to input channel of analyzer. Now pressing of STRAT hard-key starts the vibration signal acquisition.

9. Store the display screen in which vibration signal shows maximum level.

10. Measure the down stream pipe length till next obstruction or acoustic barriers come. This pipe length is used for calculation of standing wave frequencies whose first harmonic is at

$f=i*c/(4*L)$ for closed valve; $i=1$

$f=i*c/(2*L)$ for open valve; $i=1$

11. Enter the parameters required in the developed software and predict the noise spectrum.

12. Calculate the energy coefficient. If it is less than 1000 then the noise usually will not exceed 90dBA. If it is greater than 1000 then noise will be 90dBA and above. This will easily classify valves between the "white" categories and "gray/black". This will also helps to know

whether noise generation is because of installation/sizing fault or because of other faults.

13. If there is any standing frequency in vibration signal then filter it out. Noise at this frequency should not be grater than 90dB, otherwise it cause damage to downstream components and valve also.

14. Check the whether natural frequency of vibration is present in signal. If it is present then it will cause more damage to the valve.

15. Do the 1/3octave filter analysis. Convert vibration levels to noise levels using software. Compare predicted noise levels with actual observed noise levels. And depending upon deviation of actual from predicted levels comment can be made on condition of control valve.

Observed noise in each band of frequency is compared with given threshold.

If noise is less than predicted then valve is in good condition.

If noise is greater than predicted noise but deviation is not greater than 10%predicted noise; then valve is becoming noisy in this band.

If noise is greater than predicted noise with deviation is greater than 10% of predicted noise; then valve is noisy in this band.

16. In order to find probable fault the actual signal is passed through different types of band pass filters.

A. if summed noise level in the frequency range 1-100 is greater than 90dB then valve may be suffering from internal valve instability

B. If summed noise level in the frequency range100-1500 is greater than 90dB then valve may be suffering from mechanical vibration.

C. If summed noise level in the frequency range2000-8000 is greater than 90dB then valve may be suffering from resonant vibration.

17. Finally all graphs and results are displayed.

Many times for some applications pressure tapping for control valves cannot be allowed. For the pipe lines carrying cryogenic fluid or situated in hazards area making holes in them to insert thermowell is also not possible. In such case we can not measure parameters required for noise prediction. Even for liquids mediums and some of the gaseous medium noise prediction methods are not available. In these cases standard data base for comparison is not available. But generally noise above 90 dB is considered as abnormal. And noise above 110dB is considered as dangerous noise. So these two thresholds are used to determine valve's condition.

3.4 Observations

Experimentation has been done with three different control valves. In three cases medium is water so standard noise prediction methods cannot be used for noise prediction. In these cases vibration signal are converted to noise levels and these levels are compared with thresholds to do the condition monitoring. While doing experimentation the above mentioned procedure is strictly followed.

1. The first tested control valve is pneumatically actuated globe control valve. The figure 3.2 shows picture of condition monitoring setup with control valve.

Data: Prestige Control Valve-series 240

Pneumatic Control Valve Type 241-1

Single- Ported Globe Valve

Flow Coefficient (C_v) =3

Pressure Recovery Factor (F_L)=0.95

Pipe Schedule=40

Pipe size = 1inch

Pipe wall thickness=0.00818m

Medium=Water

Inlet Pressure=2.8 Kg/cm2

Outlet pressure=.2Kg/cm2

Downstream pipe is open

Downstream pipe length till acoustic obstruction =6.1feet.

Therefore standing frequency =c/2*L= 1498/(2*1.83)=409.289Hz.

The vibration readings are taken at full opening of control valve. Values of inlet and outlet pressures are noted down (these values are listed above data). This flow loop doesn't have flow meter to measure flow. So flow is measured by first principal method.

15 liters of water is collected in 15 sec.

Flow rate=3600 liter/hr;

1 m^3 = 1000liter

Flow rate =3.6 m^3/hr=3600kg/hr=1kg/s

Energy coefficient =113.47

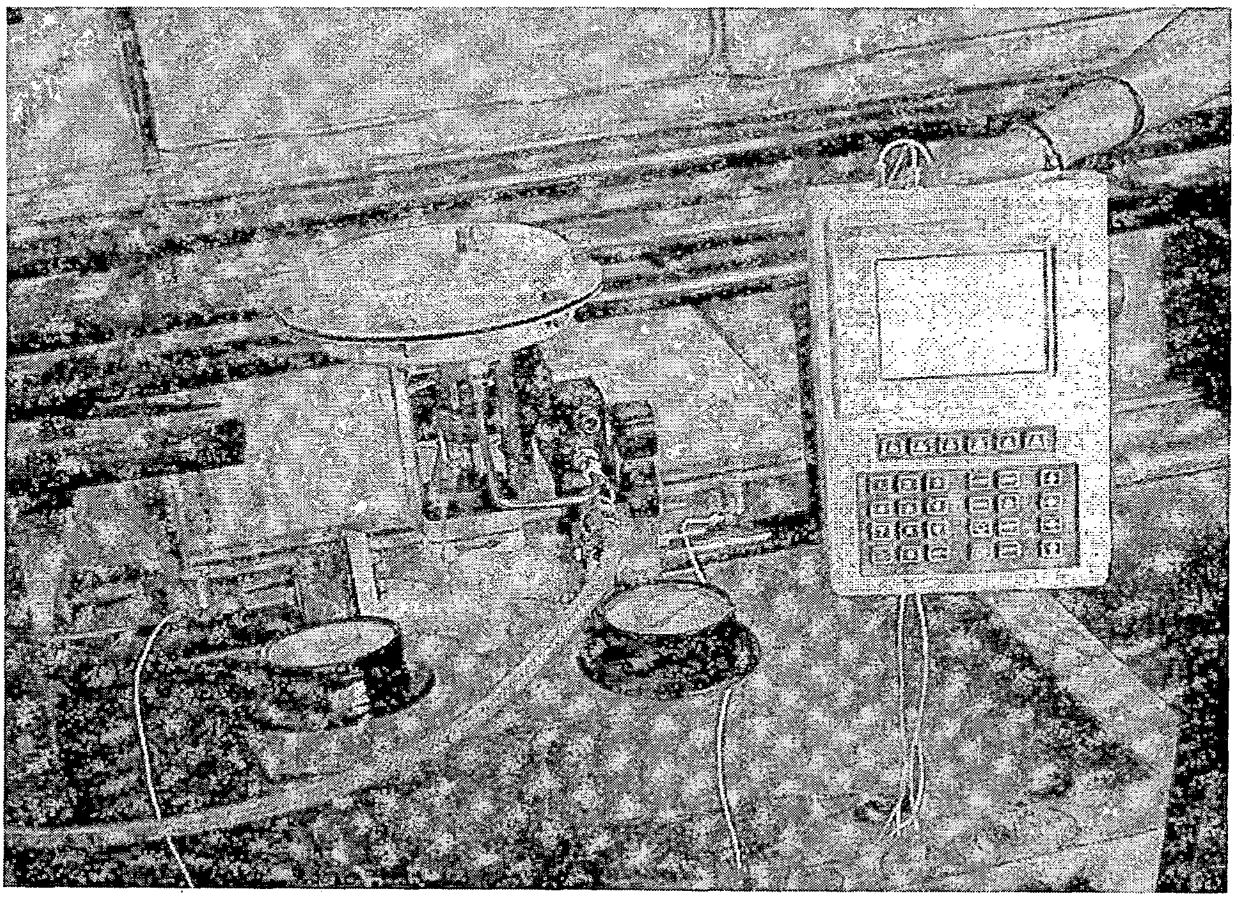


Fig 3.2 Condition monitoring setup along with control valve

The acquired vibration signals along their spectrum are displayed in following figures. Corresponding noise levels, calculated using software are also shown.

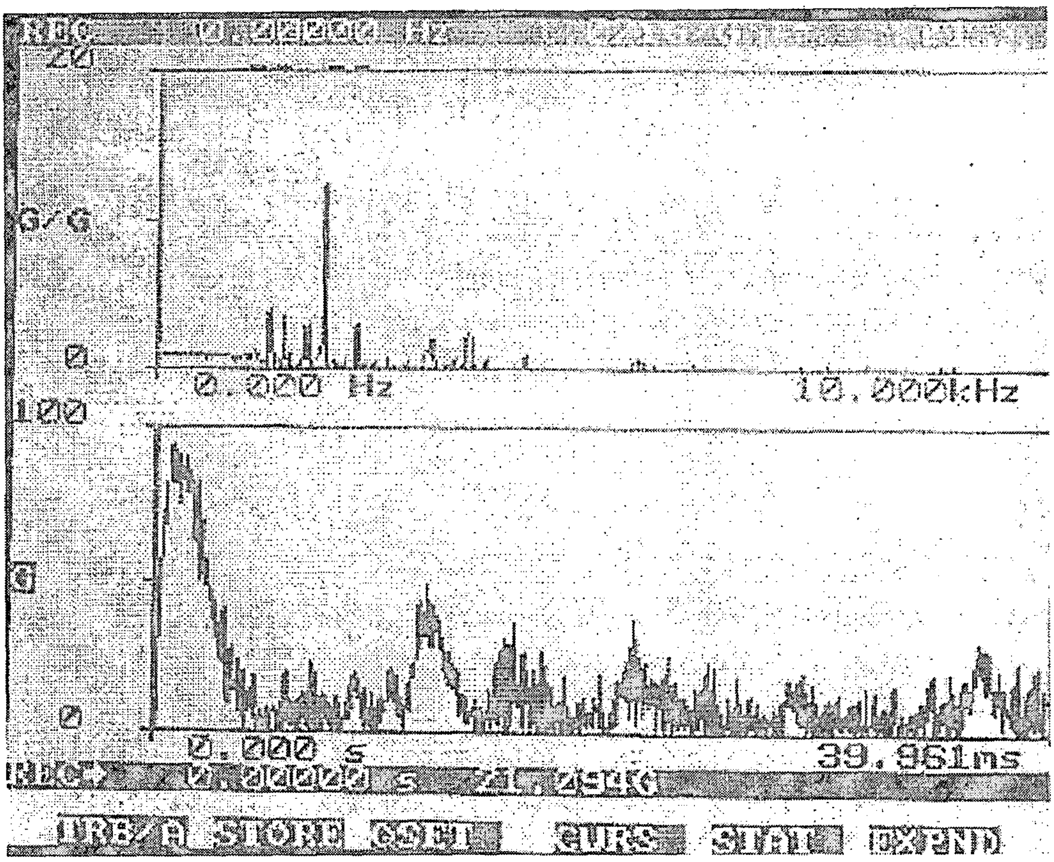


Fig 3.3 Vibration signal acquired from control valve 1

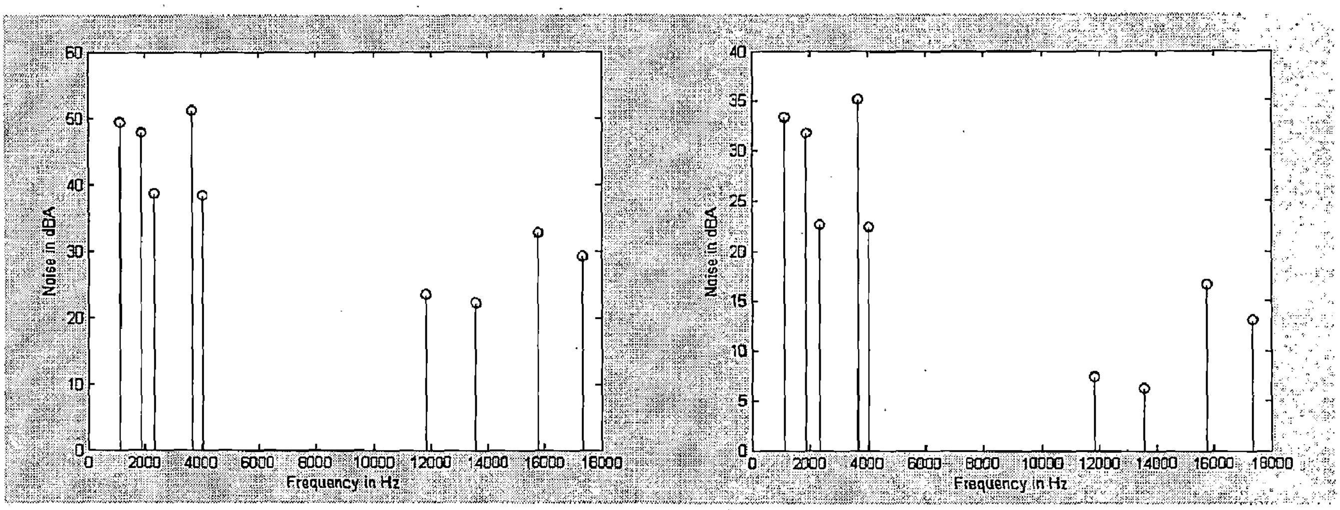


Fig 3.4(a) Noise level calculated for 1inch distance (54.7086 dBA)

Fig 3.4(b) Noise level calculated for 1meter distance (38.66dBA)

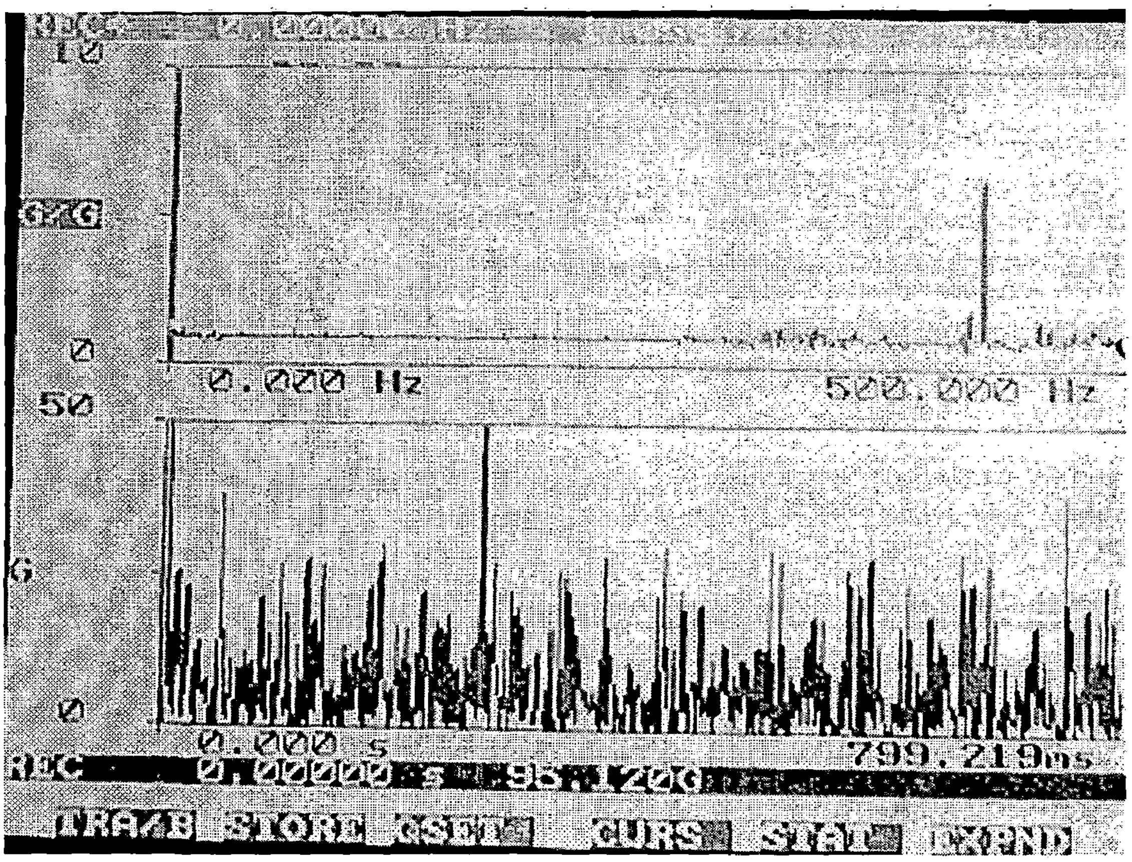


Fig 3.5 vibration signal taken for 500 Hz frequency Band

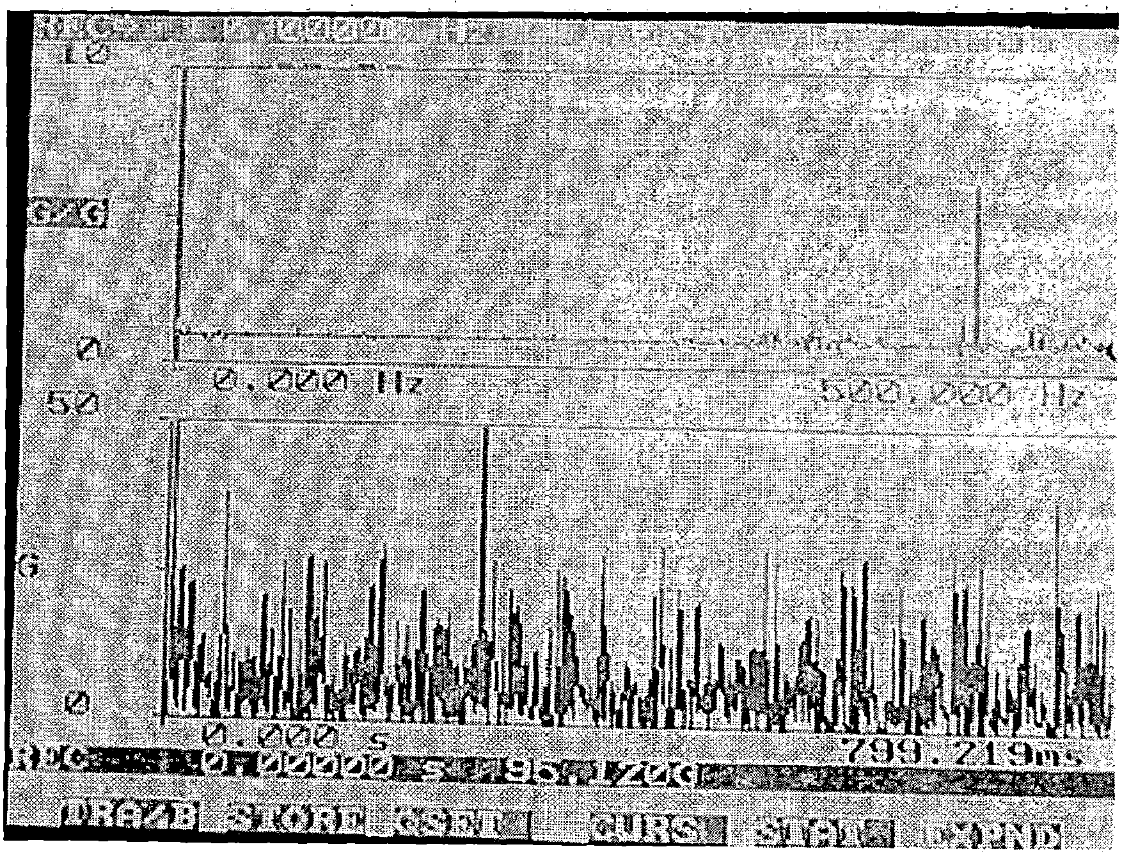


Fig 3.6 Vibration signal taken for 1000Hz frequency Band.

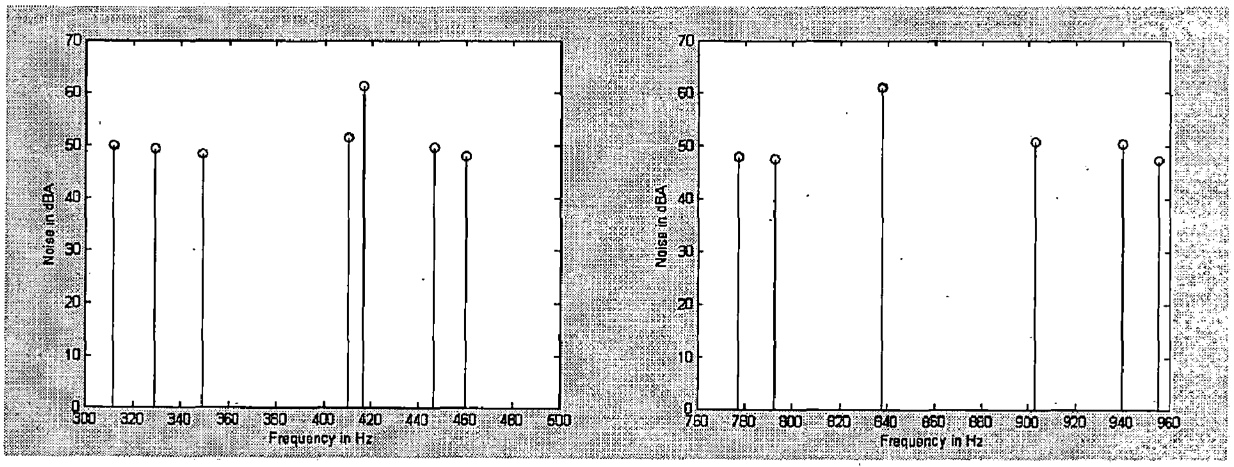


Fig 3.7(a) Noise calculated for 500Hz (62dBA)
Fig 3.7(b) Noise calculated for 1000Hz (62.26 dBA)

2. The second tested control valve is also pneumatically operated Valve. It is present in flow loop built for cooling purpose. This control valve is installed many years before, therefore it is expected that condition monitoring of this valve give us some useful readings. The medium is water and there is no provision for measurement of inlet and outlet pressure measurement. The vibration signals are collected at different flow openings of control valve.

Pneumatic Control Valve

Bottom Guided Globe Valve

Pipe Schedule=40

Pipe size=2 inch

Pipe wall thickness=0.00818inch

Medium=Water

Readings are taken at different flow rates

Flow meter =Rotameter

Flow rate in LPH=2700, 2500,2000,1300,1000,500

Flow rate Kg/s =0.75, 0.69, 0.55, 0.36, 0.27, 0.14

Accelerometer Location=6inch from control valve at the downstream.

Downstream pipe is close.

Downstream pipe length till acoustic obstruction =2feet 7inch.

Therefore standing frequency =c/4*L=483.22Hz

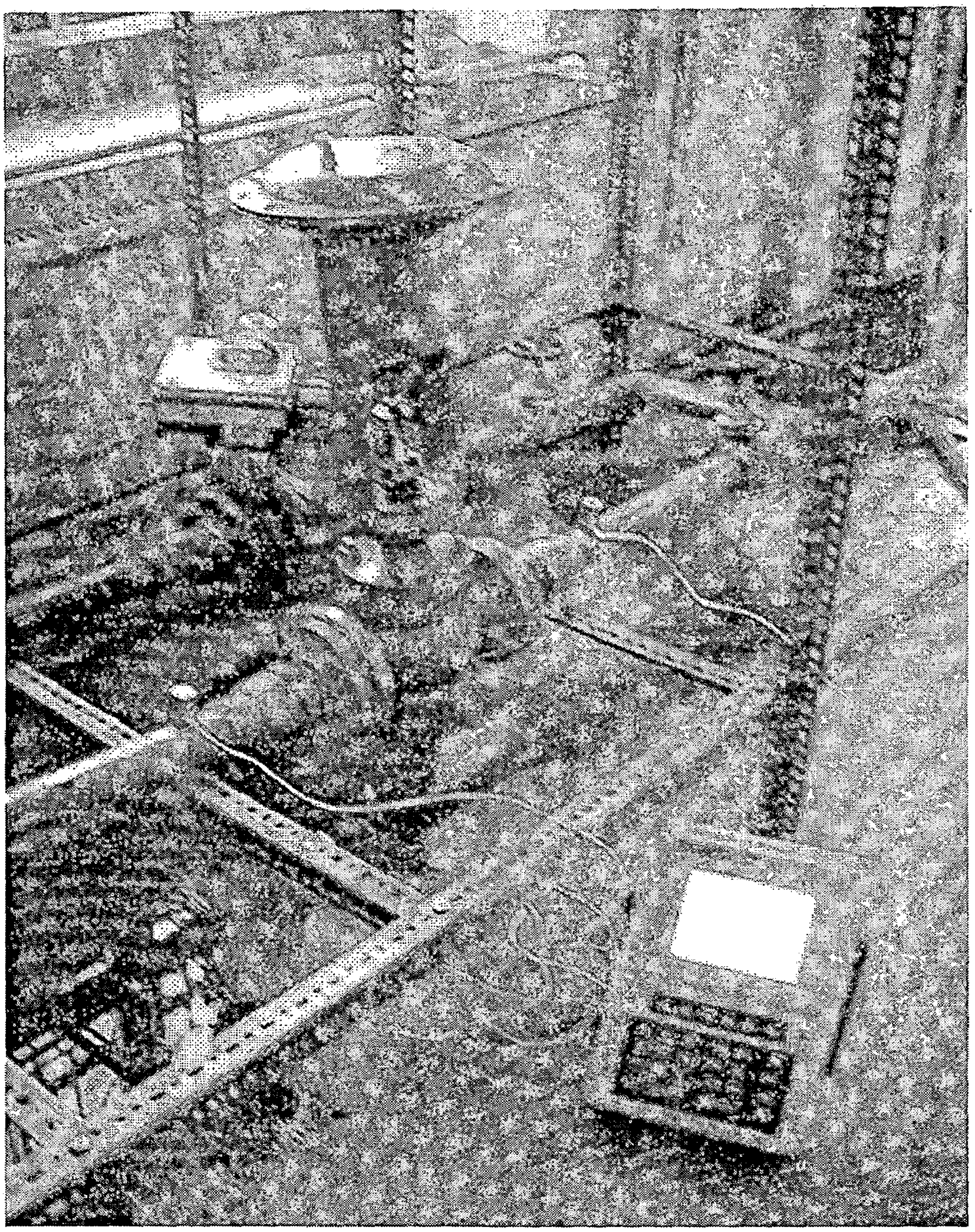


Fig 3.8 Second Control Valve with condition monitoring setup

The vibration signals of this control valve at different opening are shown in figures given below,

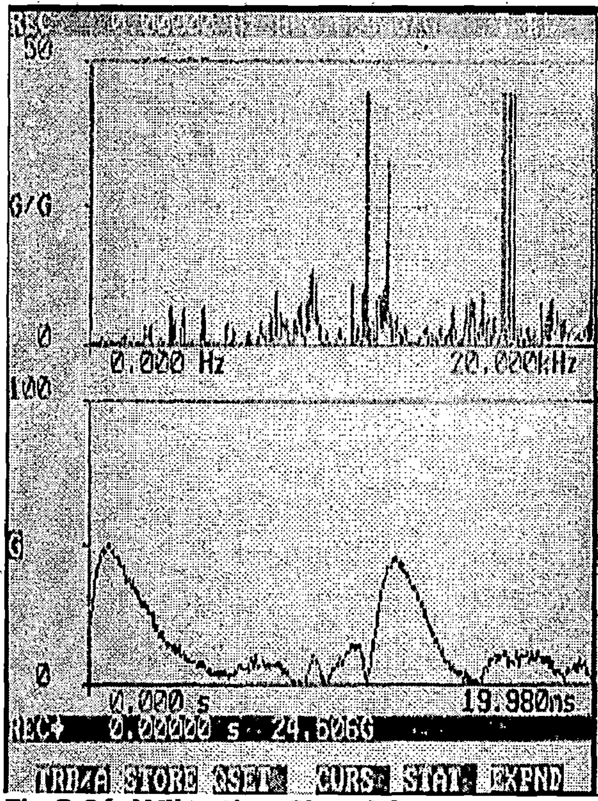


Fig 3.9(a)Vibration Signal for 2700 LPH

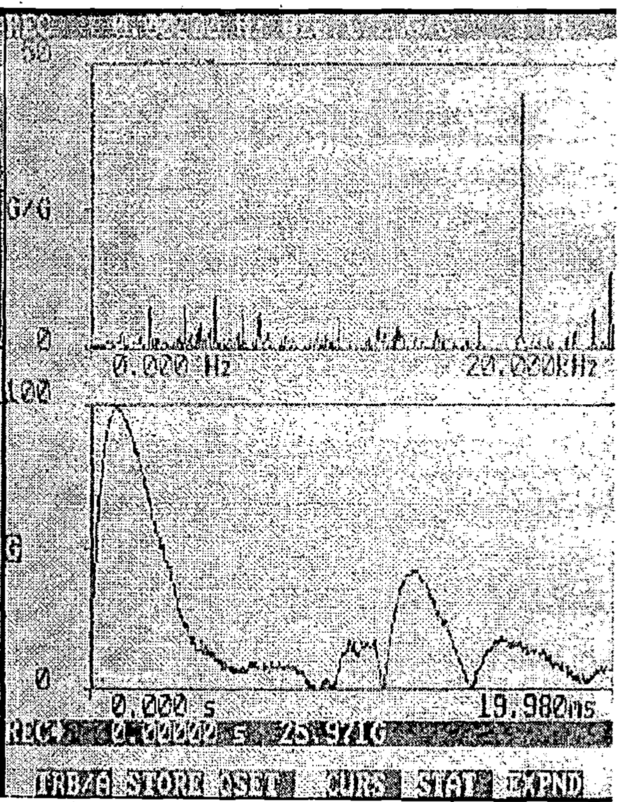


Fig 3.9(a)Vibration Signal for 2500 LPH

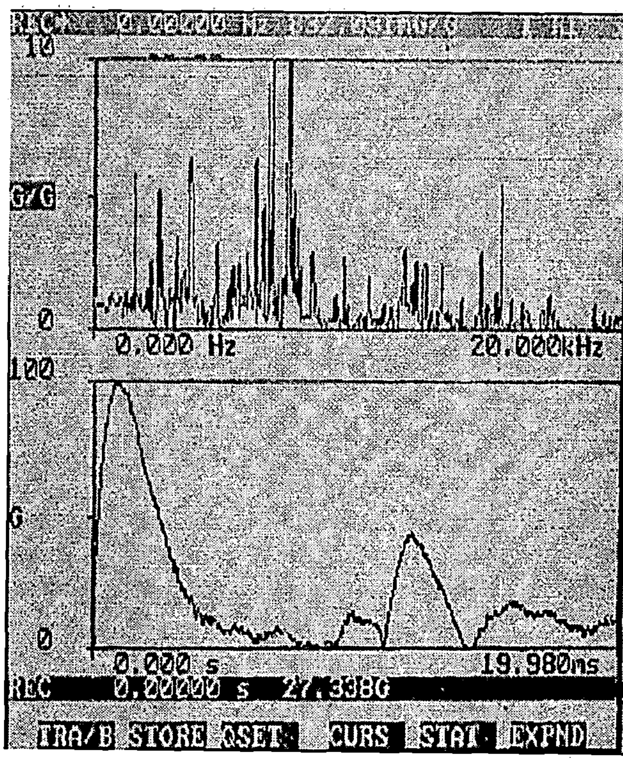


Fig 3.10(a) Vibration Signal for 2000 LPH

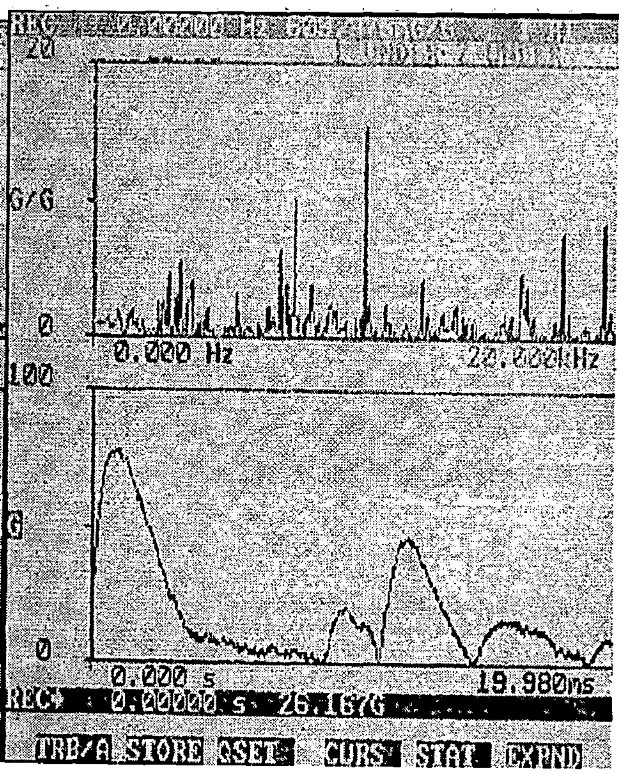


Fig 3.10(b) Vibration Signal for 1300 LPH

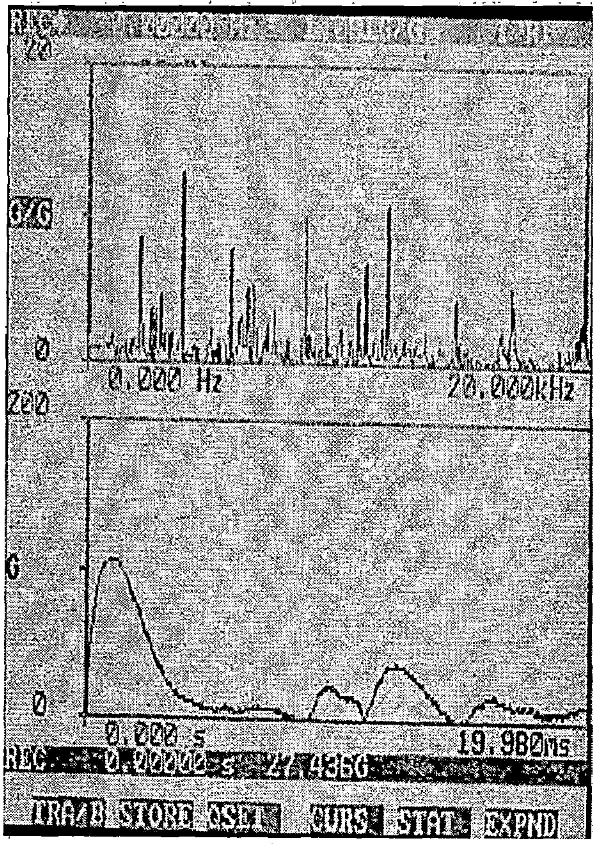


Fig 3.11(a)Vibration Signal for 1000 LPH

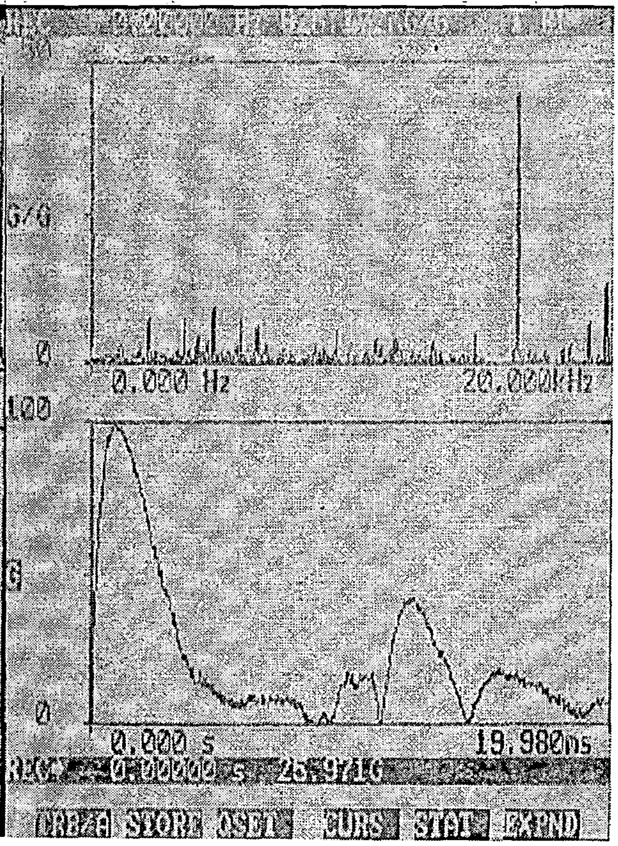


Fig 3.11(b)Vibration Signal for 500 LPH

For all these vibration signals noise levels are calculated using software and these noise signals are shown in figures below,

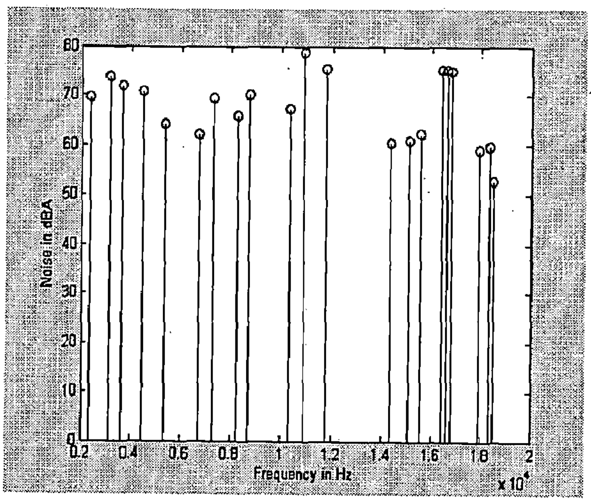


Fig 3.12(a) Noise Signal for 2700 LPH

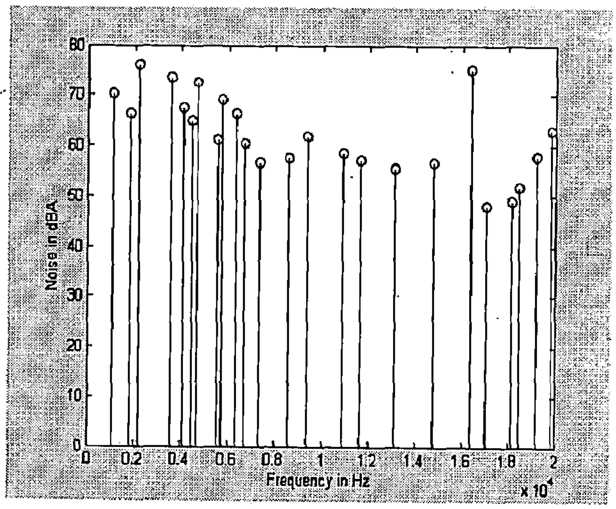


Fig 3.12(a) Noise Signal for 2500 LPH

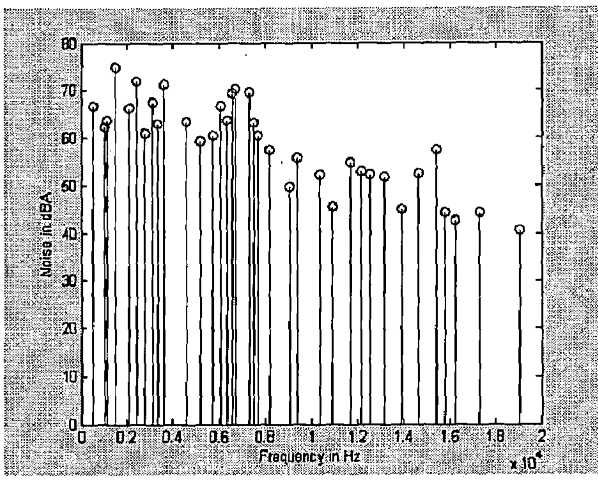


Fig 3.13(a) Noise Signal for 2000 LPH

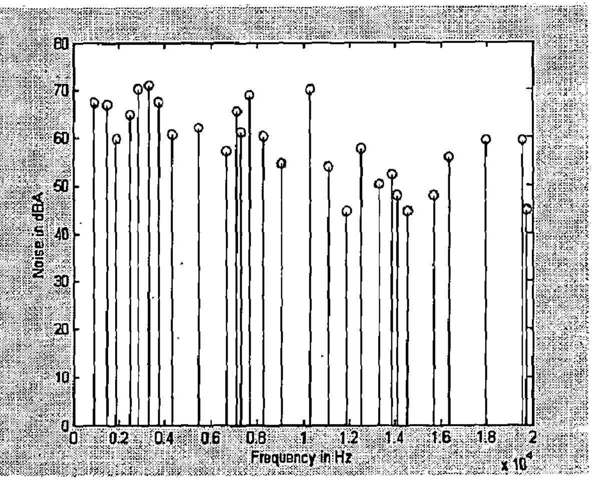


Fig 3.13(a) Noise Signal for 1300 LPH

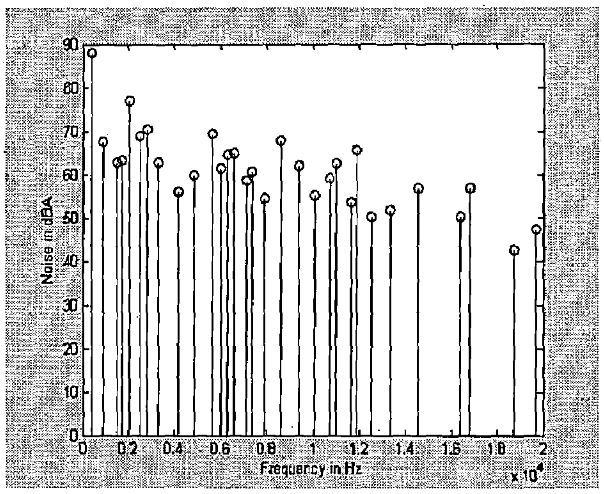


Fig 3.14(a) Noise Signal for 1000 LPH

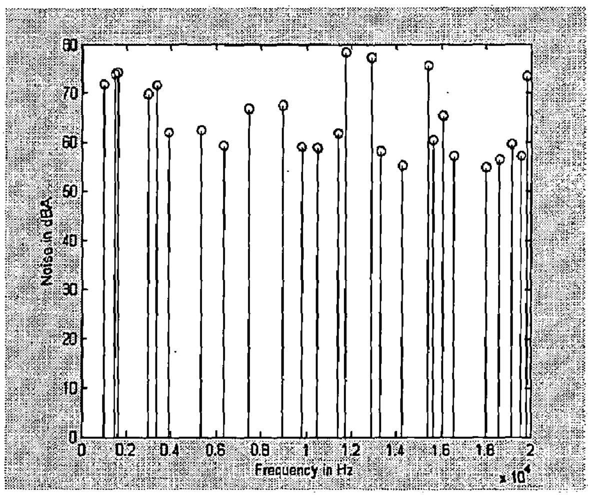


Fig 3.14(a) Noise Signal for 500 LPH

3. Third tested control valve is electrically operated valve. As compared to second valve the age of this valve is less.

Electrically operated Control Valve

Regeltechnik-kornwestheim Gmdh

West Germany

Model -STS102, Sr No-78182

Normally full open

Pipe Schedule=40

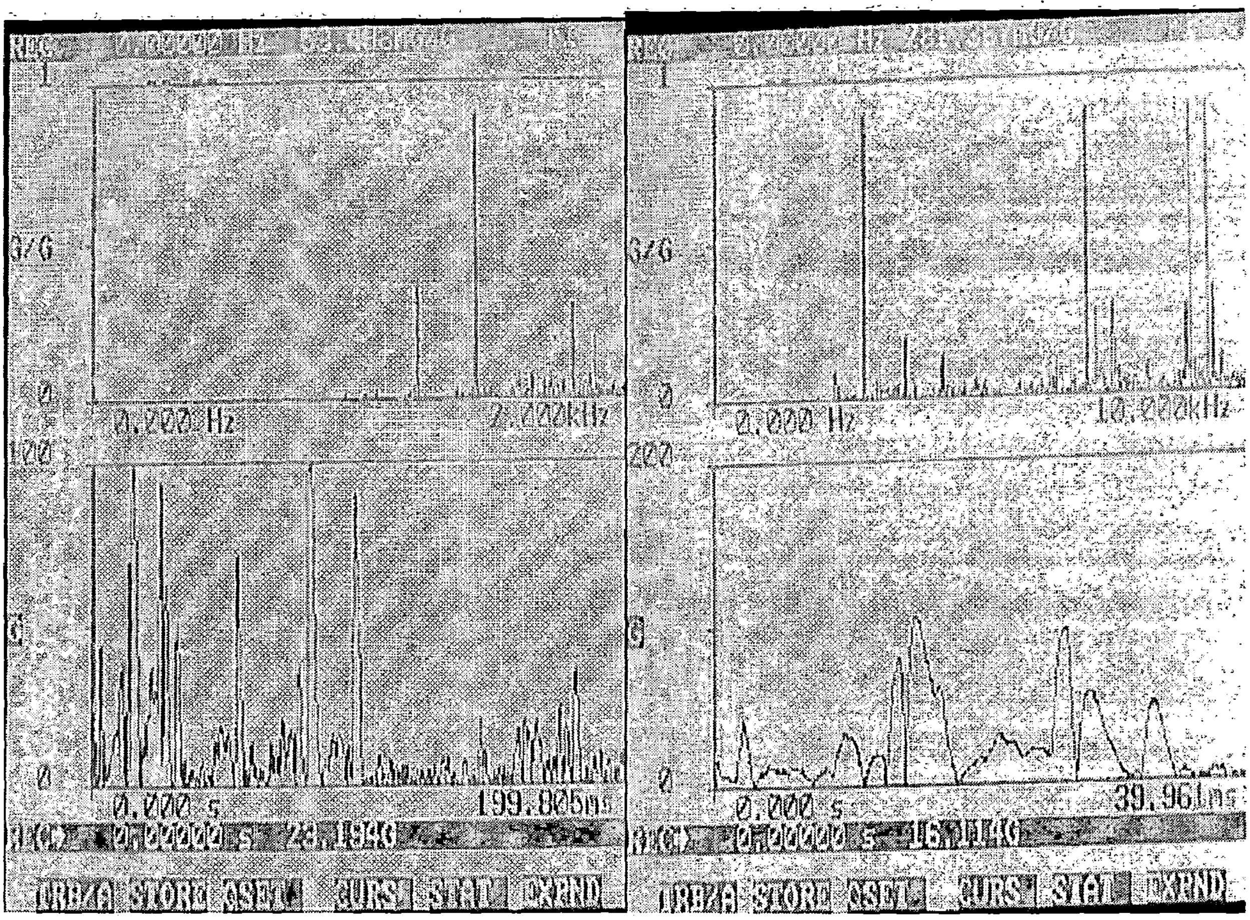
Pipe size=1/2 inch

Pipe wall thickness=0.00818inch

Medium=water

Flow rate=2.2 Kg/s for travel 0

The vibration signals noted for this control valve are shown below,



(a) For frequency range 2000Hz (b) For frequency range 10000Hz

Fig 3.15 Vibration Signal measured at full opening

Noise levels for these signals are shown in the figure below,

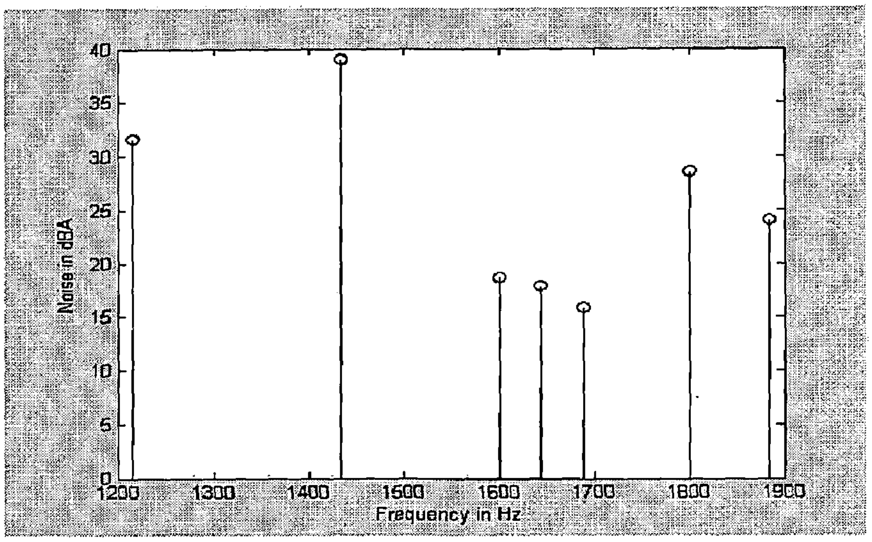
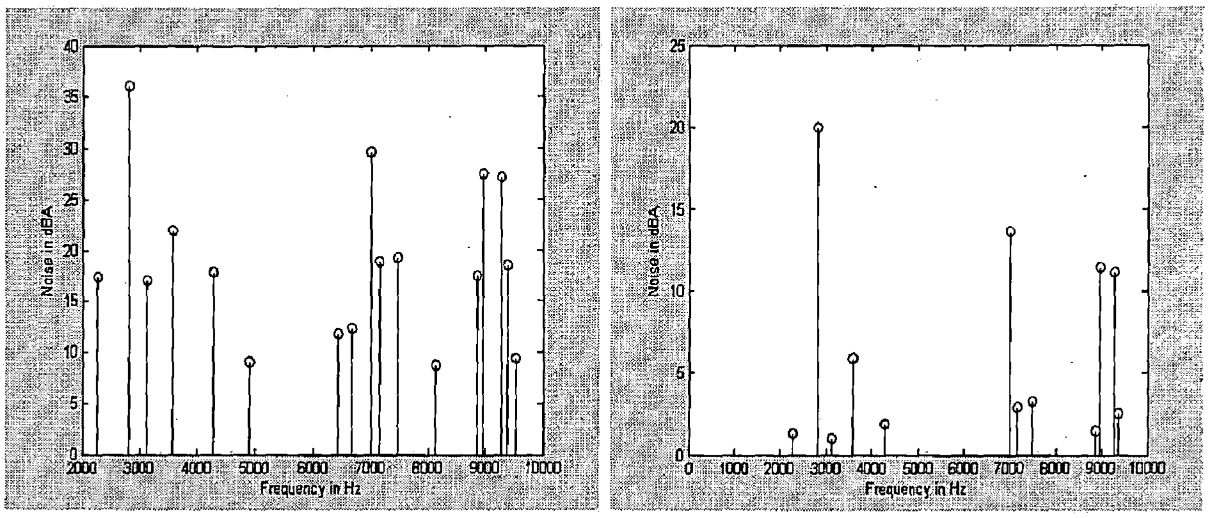


Fig 3.16 Noise signal for 2000Hz frequency band



(a)At distance 1inch (38.244dBA) (b)At distance 1meter (22.22dBA)

Fig 3.17 Noise signal for 10000Hz frequency band

3.7 Results

1. First Control Valve

The energy coefficient of this valve is calculated. It is 113.7 which is less than 1000, so definitely this valve will not produce the noise above the 90dBA. After actual measurement noise level calculated is 54.70 dBA for 1inch distance from and 38.66 dBA for 1 meter distance. Calculation of 1 inch distance is made because we have tried to measure noise very near to pipewall in order to minimize the interference of other noise sources in the vicinity. The calculated noise 38.66 dBA is very less than 90 dBA thresholds so valve is in good condition. As it is newly installed it shows low magnitude for vibration and noise. But after seeing the frequencies in spectrum (which varies from 1150Hz to 4050Hz), it can be said that it could be suffer from Mechanical & Resonant vibrations.

Another important reading observed for this valve is that it shows almost harmonic frequencies i.e. 417.5 Hz & 834Hz. The calculated first harmonic standing wave frequency of pipeline is 409.289 Hz which is very near to 417.5 Hz. Thus standing wave also has its effect on vibration of pipewall. So it must be removed while analyzing the vibration signal.

2. Second Control Valve

This valve has been installed many years ago. May be because of this it shows the more noise level than any other tested valve. Wearing of this valve may be taken place. Vibration readings are taken for different valve openings and it has been seen that control valve shows different vibration levels for different opening.

From the table it can be see that this control valve shows higher noise at 1000 LPH flow and it gives low noise 81.0905 for reasonable flow. It gives minimum noise for 1300 LPH flow, but this flow is not sufficient.

The vibration signals have almost all frequencies of vibration. But these frequencies are more crowded in resonant vibration frequency band. It may suffer from resonant excitation of valve internals as time passes.

Table 3.1 Noise Levels for different flow rates

Flow (LPH)	**Over All Noise** (dBA)(1inch distance)
2700	84.74
2500	82.0753
2000	81.0905
1300	78.7323
1000	88.8187
500	84.8792

The noise calculated for this valve for 1 meter distance is less than 90dBA thresholds so this valve is also in good condition.

3. Third Control Valve

The noise calculated for this valve is very low. It is 38.244dBA for 1 inch distance and 22.22 dBA for 1meter distance. The frequencies of vibration are less than 10 kHz. Considering the noise level we can say that valve is in very good condition.

CHAPTER 4

FUTURE SCOPE & CONCLUSION

4.1 Future scope

Due to safety and economical reasons diagnostic and condition monitoring systems are of growing interest in all complex industrial productions. Key components of any process plants are control valves. So condition monitoring systems are required which detect, diagnose and localize faulty operating conditions at an early stage in order to prevent severe failures and to enable predictive and condition oriented maintenance. To sustain in today's market competition and to increase profitability many plants like power plants, paper mills are adopting condition monitoring system for control valve. So there is great scope for this technique in future.

The present system is developed by considering that it is used for periodic monitoring. Basic reasons behind this were to use one system for all control valves and also reduce the cost of continuous monitoring system. But this is only possible for those control valves in which faults due to vibration & noise after started developing, takes many days to convert them to hard failures. But in the higher pressure & flow applications vibration & noise can cause failure to valves in some minutes. For such severe control valve one can design continuous monitoring system. In this system continuously vibration signal is measured, converted to noise signal, compared with standard data and display is upgraded continuously. Even in this system one can provide alarm if vibration level cross the certain threshold. Provision for storage of faulty signal should be made so that it can be used for comparison in future.

Because of fluid's nonlinear dynamic, it is very difficult to predict the control valves performance for different conditions of pressure &

flow. Very few methods are available for prediction of noise. And these are mainly for gaseous fluid. Only lots of experimentation and analysis will help to make accurate condition monitoring of control valve.

In present system faults in actuator and accessories of control valve are not considered. Fluctuation in inlet pressure, sluggish response of valve, diaphragm leakages have their own effect on performance of control valve. Thus one can develop the system along with vibration monitoring to get good idea about control valve's health.

In case of software one can use advanced filtering method, analysis methods or algorithm to analysis the vibration signal most effectively to extract maximum useful information.

Conclusion

The developed system is effective to get idea about condition of control valve. It is capable of telling amount of noise generated by control valve. It can give idea about fault developing in control valve in advance. It also briefly tells the reasons of developing faults. In case where noise prediction is possible this system does the more accurate condition monitoring.

From experimental result it is found that it reconfirms the theory on which whole condition monitoring system is developed. The methodology used in this system also takes care of fault developed due to standing waves of pipewall. History tells that standing waves are one of the reasons causing failures in valve and vibration sensitive components in nuclear plants. So the ability of this system to detect these waves is advantages.

If vibrations in control valves are because of improper sizing or installation, then this system is prone to give faulty results. This is its limitation. To avoid this it is always advised to calculate energy coefficient for the control valve under monitoring and confirming that it is less than 1000.

In short this system meets minimum requirements for reliable, portable, low cost and simple in use and understanding condition monitoring system for control valve.

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