

Signature analysis of the primary components of the Koeberg nuclear power station

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Dissertation submitted in partial fulfilment of the requirements for a degree Master of
Engineering at the Potchefstroom Campus of the North-West University

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November 2010

ABSTRACT

In line with its commitment to safe nuclear power generation, the Koeberg Nuclear Power Station (KNPS) replaced the outdated vibration monitoring system with a modern on-line vibration monitoring system. This will allow plant personnel to monitor components on a continuous basis which will provide faster response time in the scenario of excessive vibrations of the primary components.

This study focuses on the analysis of the vibration of the primary components of the KNPS by analysing the frequency spectra of the vibration signals of the primary components and comparing these to reference signatures obtained during similar operating conditions. The condition of the vibration sensors will also be evaluated.

In order to obtain a deeper understanding of the vibration behaviour and hence vibration signatures of the KNPS primary reactor components, a simplified mathematical model of the primary components is developed, based on the system of elasto-dynamic equations. The equations are solved numerically and used to simulate the KNPS vibration monitoring system. The mechanical system is modelled. Time series are generated and Fast Fourier Transforms (FFT) are calculated to simulate the new KNPS monitoring system. In the simulation mechanical degradation of the primary components as well as sensor degradation is simulated.

The purpose of this study is to indicate whether mechanical degradation has occurred in the primary components of the plant and to validate the vibration signals. At the same time the study aims to lay a foundation for future monitoring and interpretation of vibration signatures by simulating the vibration and the monitoring signals.

It was found that the primary components had not been affected by mechanical degradation as no deviations in resonances were detected in the frequency signatures. A small number of vibration sensors were found to have deteriorated; hence replacement / maintenance was proposed.

The mechanical model and the simulation of the monitoring signals proved to be useful to understand and interpret the vibration of the KNPS primary components.

KEYWORDS

Frequency signatures

Koeberg Nuclear Power Station

Mechanical degradation

Primary system

Vibration sensors

PREFACE

Firstly, I would like to thank God Almighty. The privileges in my life are endless and I would especially like to thank Him for blessing me with the “gift of comprehension” to allow me to interpret and represent my version of one of His creations.

To my study leader, Prof. Frikkie van Niekerk; Without your constant guidance and motivation this dissertation would not have been possible. Your dedication to hard work has inspired me to deliver a standard of work which until now has been unknown to me.

I would also like to thank Mr. Mark Gordon for providing me with his expertise and allowing me to focus the majority of my time on finishing this dissertation. Many thanks and gratitude goes towards Mr. Merick Hoole, Mr. Heinrich Penzhorn and Dr. Vernon Marshall for providing me with the necessary information to complete this dissertation.

Finally to the love of my life, I thank you for the constant motivation, inspiration and understanding. The invaluable part you played in this dissertation will not be forgotten.

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ACCRONYMS AND ABBREVIATIONS

Abbreviation	Full Description
KNPS	Koeberg Nuclear Power Station
FFT	Fast Fourier Transform
APSD	Auto Power Spectral Density
CPSD	Cross Power Spectral Density
CAD	Computer Aided Design
MEMS	Micro-Electric-Mechanical System
SiO ₂	Silicon dioxide
ADC	Analog-to-digital converter
DAC	Digital-to-analog converter
HPF	High pass filter
LPF	Low pass filter
BPF	Band pass filter
Im	Imaginary value
Re	Real value
COMOS	Condition Monitoring System
PHWR	Pressurised Heavy Water Reactor
RPV	Reactor Pressure Vessel
rpm	Revolutions per minute
MathCAD™ 14	Mathematical Computer Aided Design version 14
km	Kilometres
MPa	Mega Pascal
Eskom	Electricity Supply Commission (South Africa)
UO ₂	Uranium dioxide
KIR	Reactor Loose Part Monitoring System
LPMS	Loose Part Monitoring System
RVMS	Reactor Vibration Monitoring System
rads	Absorbed radiation dose
BF ₃	Boron Trifluoride
mm	Millimetres
GE	General Electric

Hz	Hertz
MCP	Main Coolant Pump
LED	Light Emitting Diode
PCI	Peripheral Component Interconnect
PXI	PCI extension for Instrumentation
DSA	Dynamic Signal Acquisition
DAQ	Data Acquisition
PC	Personal Computer
DVD	Digital Video Disc
rms	Root Mean Square
PWR	Pressurised Water Reactor
FEM	Finite Element Model
AMS	Advance Monitoring System
2D, 3D	Two -, Three dimensional
FEA	Finite Element Analysis
USA	United States of America
TMI	Three Mile Island
USSR	Union of Soviet Socialist Republics
INES	International Nuclear Event Scale

NOMENCLATURE

ω	Angular frequency
k	Stiffness coefficient
m	Mass of the system
f	Frequency
c	Damping Coefficient
$F(t)$	Force dependent on time
x	Displacement
$\dot{x}$	Velocity
$\ddot{x}$	Acceleration
π	Pi-value (Ratio of circle's circumference to its diameter)
t	Time in seconds
g	Earth's gravitational acceleration in m/s^2
τ	Period
δ	Kronecker delta function
$\mathbb{F}$	Fourier Transform
ϕ_{xy}	Phase angle between function x and y
γ_{xy}	Coherence between function x and y
MW_{th}	Mega watt thermal
MW_e	Mega watt electrical
$^{\circ}C$	Degrees Celsius
N/cm^2	Neutrons/centimetres square
$\underline{\underline{M}}$	Mass in matrix format
$\underline{\underline{C}}$	Damping in matrix format
$\underline{\underline{K}}$	Stiffness in matrix format
Δt	Time between to successive data points

1. INTRODUCTION

World wide nuclear reactors have proved to be reliable and to operate extremely safely. This is partially due to the fact that the reactor systems are designed to be safe and reliable and that additional measures are taken to guarantee safety, such as vibration monitoring of systems and components. Generally mechanical components could be classified into two types of components for vibration monitoring purposes:

- Passive components include the reactor pressure vessel (RPV), core barrel, steam generators etc. Mechanical degradation typically develops at a slower rate in these components than in rotating components. Typical instrumentation used to monitor the vibration of these components include accelerometers, velometers, displacement transducers and ex-core neutron detectors.
- Active components, also referred to as rotating components, include the main coolant pumps (part of the primary loop) and turbines (part of the secondary loop). Instrumentation for vibration monitoring purposes include proximity probes, velocity transducers and accelerometers. In order to measure the phase angle of rotating shafts a key phasor is usually used as reference point.

In terms of the dynamic behaviour of components, the stiffness of a component is associated with its natural frequencies which could be monitored to indicate the onset of mechanical failure.

Vibration monitoring of a system can provide essential information on the structural integrity and possible degradation of a system. This is important for reasons including:

- Improved plant availability (decreased down-time)
Vibration monitoring does not influence the normal operation of the plant. Hence vibration monitoring output can be obtained without it being necessary to shut down the reactor. Predictive maintenance often leads to components eligible for replacement / repair being ordered in time and maintenance shut-downs being shortened.

- Cost saving (increased up-time)
Avoiding severe failure by monitoring mechanical degradation not only decreases down time, it also enhances significant cost saving.

In nuclear power plants stringent safety criteria apply; hence predictive maintenance is used to optimise plant economics and to circumvent failure. This is where online vibration monitoring plays an important role in determining potential degradation of a system. The main goals of vibration monitoring are to enhance safety, for investment protection purposes as well as for optimal plant up-time.

This study will focus on the vibration monitoring of the reactor internal components of the KNPS. This consists of analysing data, obtained from the KNPS by means of neutron detectors and accelerometers and converting time waveforms into FFT frequency spectra to produce Auto Power Spectral Density (APSD) functions and Cross Power Spectral Density (CPSD) functions, referred to as “signatures” of the associated components. The signatures contain peaks associated with the resonance frequencies of some of the components being monitored.

The system of elasto-dynamic equations will be solved numerically and will be used to simulate the vibration signals of KNPS primary components. Simplified mass, stiffness and damping matrices will be used. The force vector will be assumed to be a flow-induced wide band white noise spectrum with a 25 Hz component due to the shaft rotation of the reactor primary pumps. In the system mechanical degradation and signal degradation will be simulated.

The purpose of the simulation is to provide a better understanding of the vibration of the reactor primary components of the KNPS and similar reactors. Signal validation of the sensors used will also be done.

2. VIBRATION AND VIBRATION MONITORING

2.1. Introduction

Vibration is the response of an elasto-dynamic system subjected to an external force. Vibration can either be useful or destructive. An example of useful vibration is the vibration in musical instruments to produce music. Destructive vibration includes excessive machine vibration due to misalignment, imbalance and excessive vibration due to bad design or incorrect operating such as when excitation forces and resonances overlap in the frequency domain.

Two types of vibration occur (Timoshenko *et al.*, 1974:1):

- Free Vibration

In this type of vibration an initial force is applied to the system after which the system is allowed to vibrate without any external forces acting on the system. An example is shown in Figure 1. The mass is displaced from the equilibrium position by an initial force and allowed to vibrate freely until it comes to rest. The explanation of the symbols in the figure will follow in the text.

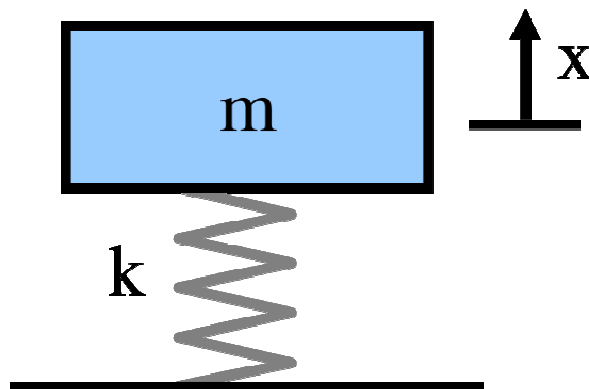


Figure 1: Free vibration of a system (Source: www.arab-eng.org)

- Forced Vibration

This type of vibration results from continuous excitation by an external force. An example of this type of vibration is a fan driven by a motor. The fan has its own natural frequencies but it is forced to operate at another frequency (unless the

fan and the motor have the same frequency) because of the force applied to the system by the motor.

Every system has a set of natural frequencies, depending on the degrees of freedom. In a single degree of freedom system, the system's angular frequency can be determined by the following equation:

$$\omega = \sqrt{\frac{k}{m}} \quad (2.1)$$

and from the above equation the undamped natural frequency can be determined,

$$f = \frac{\omega}{2\pi} \quad (2.2)$$

Here ω is the angular frequency in rad/sec, k is the stiffness in N/m and m is the mass in kg of the system. The system's natural frequency is indicated by f and is measured in Hertz.

2.2. Vibration equation

A vibration system is a system where kinetic energy, stored in the moving mass of the system, is continuously transformed to and from potential energy, stored in the spring. Damping in a system determines the rate at which energy is being lost to the environment. In mechanical machines the damping in a system is caused by friction or designed dampers (Blake, 2010:44).

Figure 2 displays a single degree of freedom system with viscous damping c and stiffness k connected to a wall on the one side and to a mass m on the other. The displacement of the system due to an external force $F(t)$ is x .

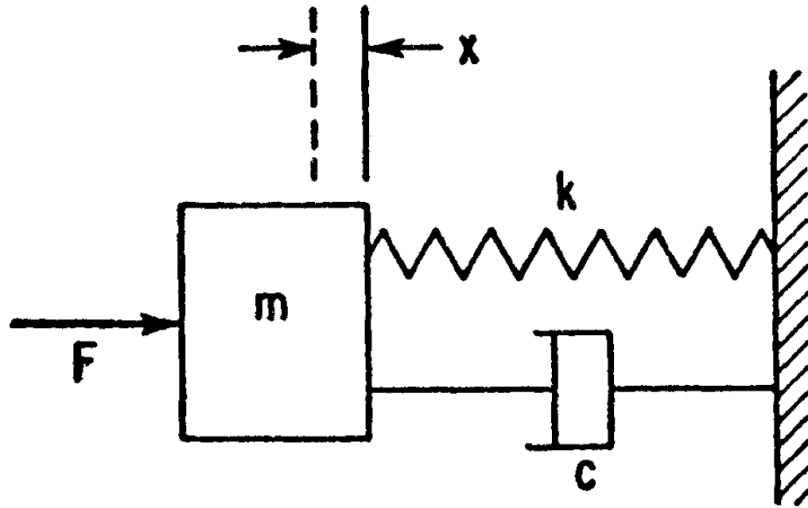


Figure 2: Single degree of freedom system (Source: Blake, 2010:52)

According to Newton's third law of motion with every action force on a system a reaction force in the same magnitude but in opposite direction is associated. Newton's second law of motion states that a force of a system is equal to the mass of that system times the acceleration. Taking these two laws into consideration the elasto-dynamic vibration equation can be defined as follows (Blake, 2010:44):

$$m\ddot{x} + c\dot{x} + kx = F(t) \quad (2.3)$$

where

m is the mass of the system (kg),

c is the damping of the system (N.s/m),

k is the stiffness of the system (N/m),

$\ddot{x}$ is the acceleration or second derivative of x to time (m/s^2),

$\dot{x}$ is the velocity or first derivative of x to time (m/s) and

the force applied at time t is $F(t)$ in N.

Multi degree of freedom systems often result in more complex problems. Therefore it is often more convenient to write equation 2.3 in matrix format. This method is discussed in detail in chapter 7.

Detailed analysis on these systems as well as the explanation of the energy equilibrium of the system can be found in Blake (2010:48).

2.3. Vibration measurement

Vibration measurement is done by either using a continuous on-line measurement system or by performing specific measurement campaigns. Different components contribute to the vibration measurement system and will be discussed in this section.

Sensors

Vibration sensors are used to measure vibration in a system. Three types of vibration measurement sensors are available:

- Displacement in m,
- Velocity in m/s and
- Acceleration in m/s^2 .

The different attributes the sensors have should be taken into account in selecting sensors for vibration monitoring. Displacement sensors are preferably used to measure the shaft motion and the internal clearances in for example sleeve bearings. Velocity sensors are the preferred choice when vibration monitoring in rotating machines are necessary. Unlike the displacement sensors which are more sensitive in the low frequency region, the velocity sensors are more compatible with accurate reading in the low to medium frequency range. A more detailed discussion regarding the displacement and velocity sensors is presented in section 5.2.1.

An accelerometer is an acceleration measurement device usually mounted directly on a component to measure the acceleration of that component. The accelerometer is the most popular vibration measurement device due to its wide applicability. This device is able to calculate vibration signals in the low to very high frequency ranges with very high accuracy. Modern accelerometers have very small dimensions and do not need a power source to operate. Accelerometers measuring acceleration in more than one direction (x, y and z direction) are available; however single axis accelerometers are preferred because of simplicity and accuracy.

In Figure 3 the basic design of an accelerometer is explained. In area A of Figure 3 a small mass (number 1) is kept at equilibrium by two springs. As soon as gravity acts on the mass or a force acts on the system the mass will move out of its equilibrium position, changing the reading on the indicator (number 3). The springs will stretch and contract (numbers 6 and 5), acting against the movement of the mass, and will force the mass back to its equilibrium position.

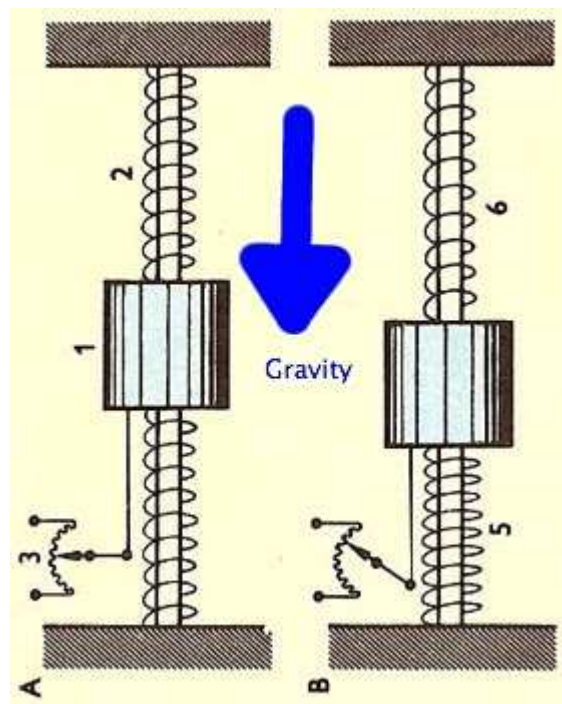


Figure 3: Basic accelerometer principle (Source: <http://epx.com>)

Accelerometers are designed to measure and indicate the acceleration experienced on the earth's surface. This means that the earth gravitational force which is approximately 9.81 m/s^2 or 1 g will cause the accelerometer to indicate, when it is at rest, a measurement of approximately 1 g upwards. Different types of accelerometers are mentioned below:

- Piezoelectric Sensor
- Shear Mode Accelerometer
- Surface Micro machined Capacitive or MEMS
- Capacitive Spring Mass Base etc.

Currently accelerometers are used in a wide variety of home appliances. It ranges from cell phones to video game controllers, e.g. the tilt detectors in regular smart phones which are used to tilt displays.

The piezoelectric accelerometer will be discussed in detail due to the applicability in this study.

Piezoelectric accelerometers

Piezoelectric accelerometers use, as the name would suggest, the piezoelectric effect to determine dynamic acceleration. The piezoelectric effect was first observed in 1880 by two brothers Pierre and Jacques Curie (APC International Ltd, 2010). Piezo which was adopted from the Greek language meaning to squeeze, is the effect which results from a mechanical stress on a piezoelectric material: an electrical charge is generated. Figure 4 illustrates the measurement method which is used to operate a piezoelectric accelerometer. In the first part the accelerometer is not subjected to any acceleration thus displaying zero voltage as output. However as soon as the accelerometer is subjected to any acceleration, the mass will apply a pressure to the piezoelectric material which will generate an electric output signal.

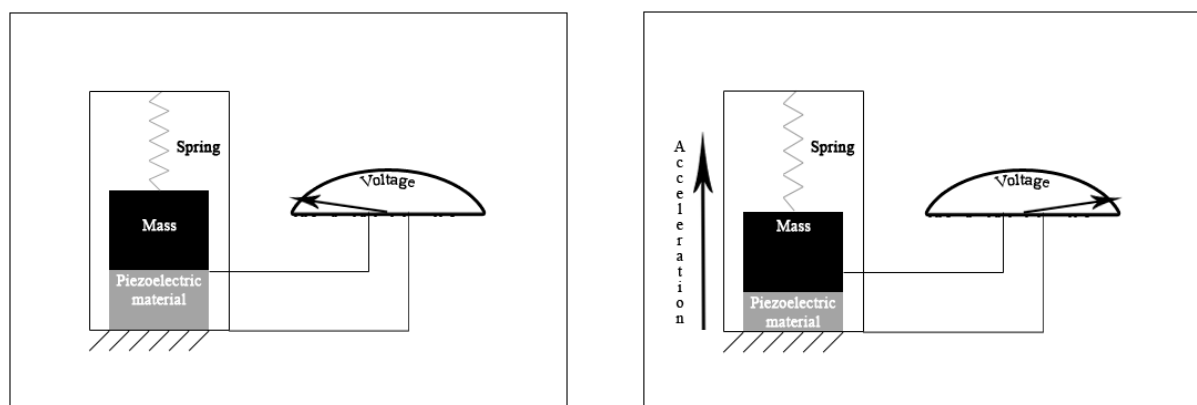


Figure 4: Piezoelectric accelerometer (Source: www.piezoaccel.com)

Similarly, by reversing the above mentioned scenario and connecting the piezoelectric material to an electrical charge, will result in a mechanical stress in the material (APC International Ltd, 2010). Two types of piezoelectric materials are typically used:

- Natural materials which include Rochelle Salt, Cane Sugar, Tourmaline-group materials and the most common crystal Quartz (SiO_2) (Nanomotion, 2006).
- Man made polycrystalline materials which are produced from a process called artificial polarization. The material used in the KNPS accelerometers is called lead-zirconate-titanate (Molyneux, 2007:8).

Sensor mounting

The mounting of vibration sensors plays an important role as it directly influences the accuracy of the readings. There are three types of configurations when mounting a vibration sensor. The frequencies and amplitudes that need to be measured will also influence the mounting method. The three types are listed below.

- Stud mounting

This configuration, also known as bolt mounting, is the preferred choice when mounting sensors on a permanent basis. This allows for high frequency testing in harsh environments. The surface must be clean and paint free. Meggitt (2009) states that for accurate readings the surface roughness of the component being measured should not exceed approximately 812 micro millimetres.

- Adhesives

This method allows vibration monitoring without extensive machining. This method reduces the frequency range of the sensor due to the damping coefficient of the adhesive used. A clean mounting surface is of great importance in order for the adhesive to bond correctly. Finally, removal of the sensor due to upgrades or replacements, is difficult when compared to the other methods.

- Magnets

This method is used when measurement is done on a non permanent basis. The sensor is connected to a magnet and the unit can be moved to the position of choice. Either the flat magnets for flat surfaces or the pole magnet configuration for

non flat surfaces can be used. The operating frequency of this type of mounting is considerably lower if compared to that of stud mounting.

Signal conditioning

Vibration sensors measure physical changes in components in analogue form, which is for example an electrical voltage or electrical current. These signals are converted into digital form (binary code) so that it can be manipulated by a computer. An analog-to-digital converter (ADC) is a device that converts an analogue signal from a sensor into a digital signal. The sampling frequency of an ADC is defined as the number of data points per second used to sample the analogue signal. The dynamic range is the accuracy of the measurements. Figure 5 displays a simple sinusoidal function (blue line). To convert this analogue signal into digital format sufficient data points will have to be plotted to represent the original periodic function when they are connected. This is illustrated in red. According to the Nyquist theorem, the resolution or sampling rate will need to be greater than twice the highest frequency in order to obtain the correct frequency spectra. Assuming that a frequency spectrum with a highest frequency of e.g. 20 Hz that needs to be converted, it would be necessary to use a sampling rate of at least 40 Hz (Stremler, 1992:129).

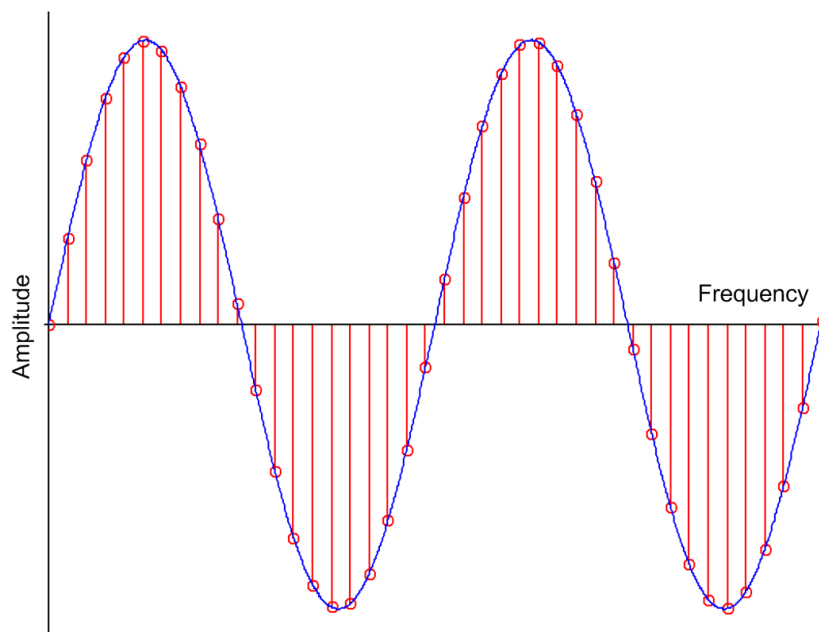


Figure 5: Analog-to-digital converter (Source: Figure by the author)

If the sampling rate does not exceed twice the highest frequency an incorrect representation of the original analogue signal is obtained once the digital signal is converted back to an analogue signal. The phenomenon is called aliasing and is illustrated in Figure 6. In Figure 6 the data points are insufficient to represent the original signal (blue graph) and by converting from digital format to analogue by using a digital-to-analog converter (DAC) the incorrect presentation is obtained (red graph). Should a lower sampling rate be used to sample signals containing high frequency signal components, anti-aliasing filters are used to remove the high frequency content.

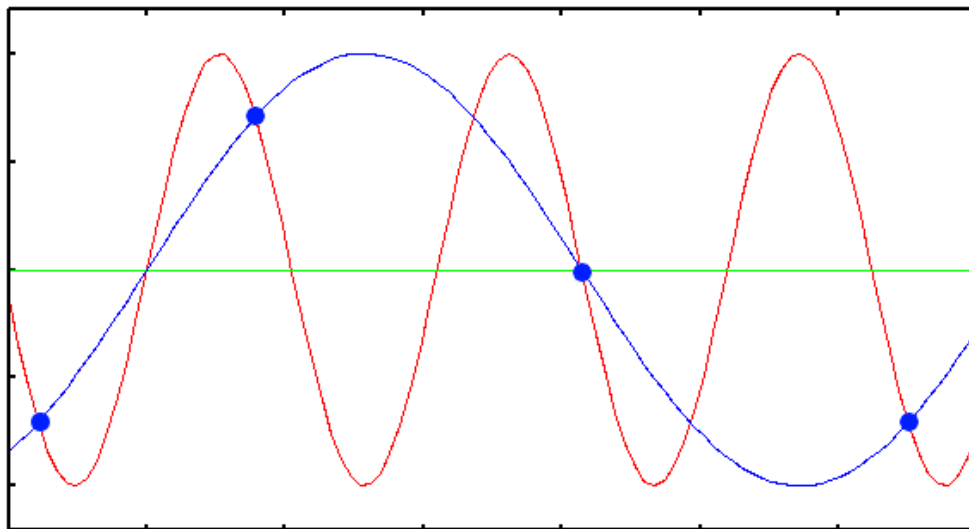


Figure 6: Signal aliasing (Source: www.svi.nl)

A good signal dynamic range is needed for the analysis of vibration signals. It is therefore important to obtain a good signal to noise ratio in addition to using a sensitive ADC. The following steps are typically used in signal conditioning:

- Amplifier

Amplifying the signal from accelerometers is important because the voltage outputs generated by these vibrations sensor are very small. It is also important to amplify the signal as close as possible to the vibration sensor. The vibration signal is constantly influenced by external noise and amplifying the signal very close to the sensor improves the signal to noise ratio.

- Filtering

Signal filtering is done to remove unwanted frequencies from a vibration signal. This is important to eliminate aliasing in signals by eliminating unwanted frequencies. A band pass filter allows certain frequencies to pass through whilst rejecting other. The following filters are used:

- o High pass filter (HPF)

The filter is designed to allow high frequencies to pass through the filter whilst removing all the low frequency components.

- o Low pass filter (LPF)

This is directly the opposite of the HPF and allows the low frequencies to pass whilst removing all the high frequency components.

- o Band pass filter (BPF)

This filter serves as a combination of the HPF and LPF. This filter will allow frequencies between the frequencies specified to pass through and will remove all the frequency components outside the band-pass limits.

Once these procedures are completed, the signal is ready for further processing by performing the ADC step and further analyses such as determining the FFT to determine the frequencies within the signals.

2.4. Time waveform / Frequency spectrum

The Fourier transform is used to convert time series into frequency spectra. Well known scientists like Daniel Bernoulli, Leonhard Euler etc. contributed to this analysis but in the end it was Jean Baptiste Joseph Fourier who was credited with the discovery. It was in 1807 in a paper titled “Mémoire Sur la propagation de la chaleur dans les corps solides” where Joseph Fourier modelled all functions by trigonometric series (Keston, 1998). This was the birth of Fourier analysis.

2.4.1. Fourier series

This method enables the user to convert any periodic function and represent that function in terms of the sum of harmonic functions (Rao, 2004:53). Sine waves for even functions and cosine waves for odd functions are used to replicate a periodic function which enables the calculation of the different frequencies necessary to replicate the original function. These frequencies can then be displayed as a frequency spectrum.

Figure 7 illustrates an example of a triangular periodic function with period τ where the period is defined as the time for one cycle to be completed.

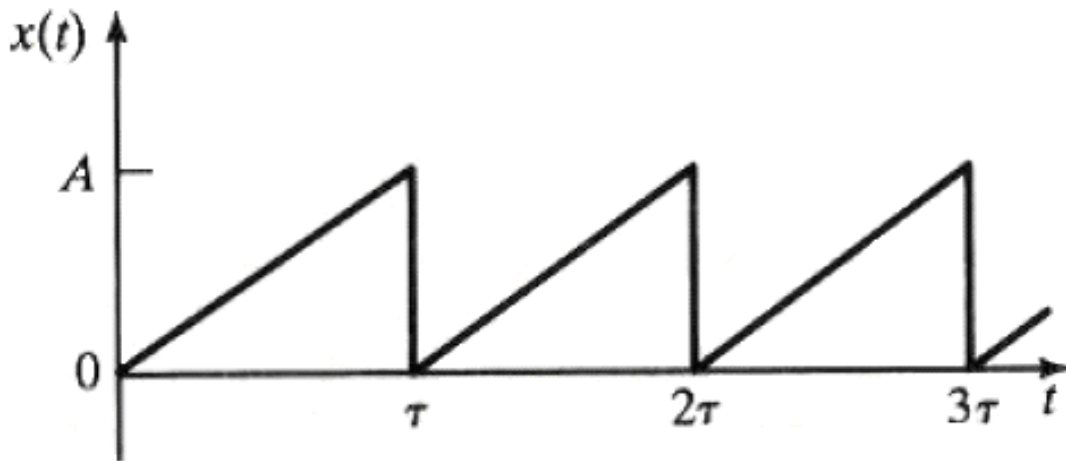


Figure 7: Periodic function (Source: Rao, 2004:54)

In Figure 8, the Fourier method is used to represent the periodic function in the form of sine and cosine functions. By adding the terms it becomes clear that the sum of these functions approximate the original periodic function; the more terms added, the better the approximation.

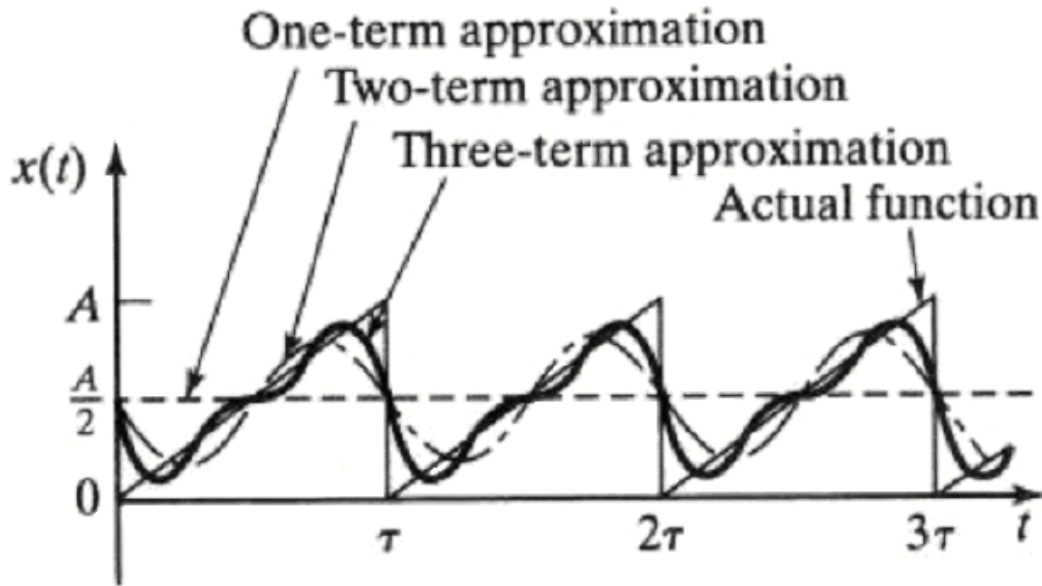


Figure 8: Simulating the periodic function (Source: Rao, 2004:54)

According to Rao (2004:54), the Fourier series can be defined as follows if $x(t)$ is the periodic function with a period τ :

$$\begin{aligned}
 x(t) &= \frac{a_0}{2} + a_1 \cos \omega t + a_2 \cos 2\omega t + \dots + b_1 \sin \omega t + b_2 \sin 2\omega t + \dots \\
 &= \frac{a_0}{2} + \sum_{n=1}^{\infty} (a_n \cos n\omega t + b_n \sin n\omega t)
 \end{aligned}
 \tag{2.4}$$

where:

t is the time in seconds,

ω is the fundamental frequency defined by $\omega = \frac{2\pi}{\tau}$ and

$a_0, a_1, a_2, \dots, b_1, b_2, \dots$ are constants.

To determine the constants, algebraic manipulation is used to simplify the equations. Firstly equation 2.4 is multiplied by $\cos(n\omega t)$, followed by another multiplication with $\sin(n\omega t)$. Integration of the new equation over one period $\tau = \frac{2\pi}{\omega}$ will render an equation with all but one term in the equation equal to zero (Rao, 2004:54).

Thus

$$a_0 = \frac{\omega}{\pi} \int_0^{\frac{2\pi}{\omega}} x(t) dt = \frac{2}{\tau} \int_0^{\tau} x(t) dt$$

$$a_n = \frac{\omega}{\pi} \int_0^{\frac{2\pi}{\omega}} x(t) \cos n\omega t dt = \frac{2}{\tau} \int_0^{\tau} x(t) \cos n\omega t dt$$

$$b_n = \frac{\omega}{\pi} \int_0^{\frac{2\pi}{\omega}} x(t) \sin n\omega t dt = \frac{2}{\tau} \int_0^{\tau} x(t) \sin n\omega t dt$$

Once the constants have been calculated the values can be substituted and the different sine and cosine functions can be determined. Figure 9 illustrates a periodic function (red) and the different sine and cosine harmonic functions generated from equations 2.4. It should be noted that all the cosine functions (F2(t), F3(t) and F4(t)) are zero due to the fact that these are odd functions and the periodic function F1(t) starts at zero and therefore the sum of cosine functions will not positively influence the replication of the periodic function.

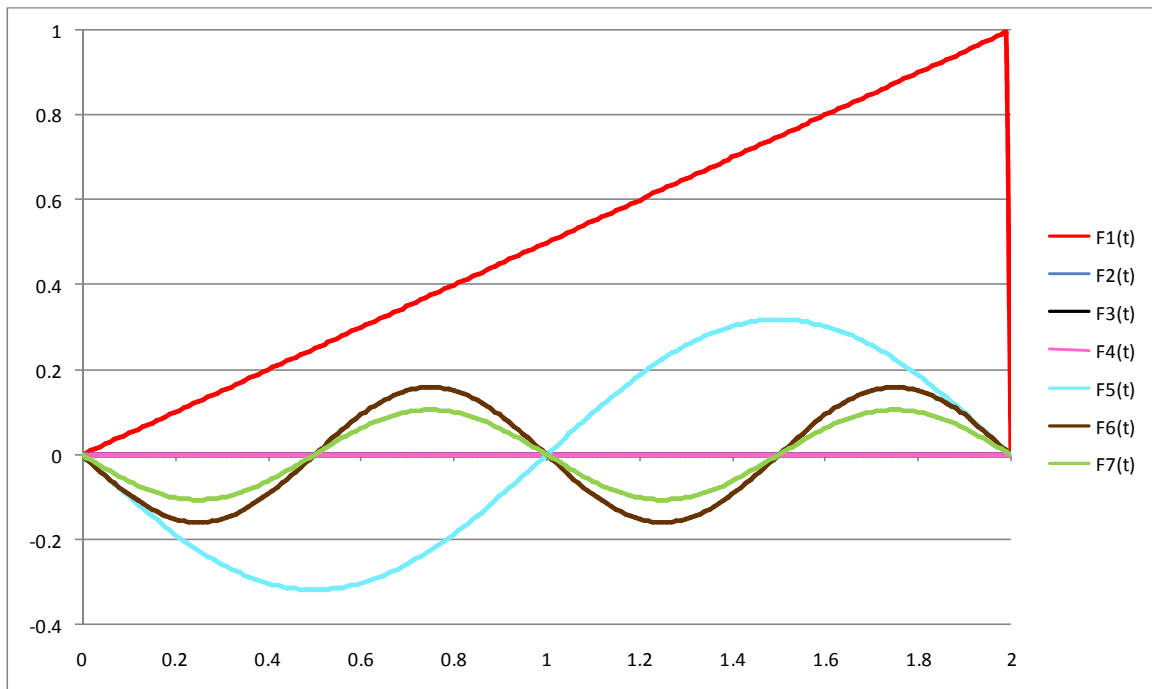


Figure 9: Harmonic functions (Source: Figure by the author)

As previously mentioned, the sum of these harmonic functions (G2(t)) is calculated to replicate the original periodic function. In Figure 10 this is illustrated. With only three harmonic functions the sum of these functions are beginning to approximate the original

function. However as the number of harmonic functions used increases to infinity, the difference between the original periodic function and the harmonic function becomes smaller, except in the area at the point of discontinuity (In Figure 10 this is in the area of $t = 2$). This principle is known as the Gibbs phenomenon (Rao, 2004:55). Rao (2004:55) states that the error at that point of discontinuity will remain at 9% even if the harmonic functions are increased to infinity.

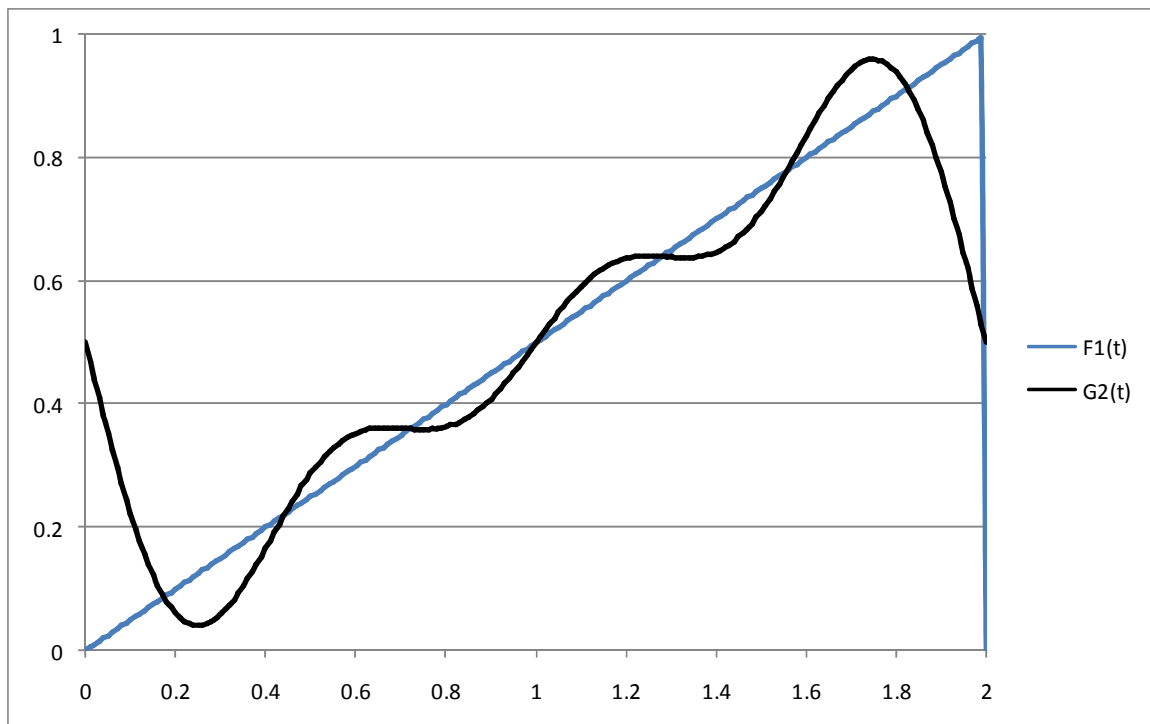


Figure 10: Representation of the original periodic function (Source: Figure by the author)

After the different frequencies have been identified the unit impulse function or the Kronecker delta function can be used to identify the different frequencies on a frequency spectrum. The Kronecker delta function can be defined as follows (Weisstein, 2010):

$$\delta_{\omega, x} \equiv \begin{cases} 1 & \text{if } \omega = x \\ 0 & \text{if } \omega \neq x \end{cases}$$

Figure 11 depicts the frequency spectra according to the different time waveforms.

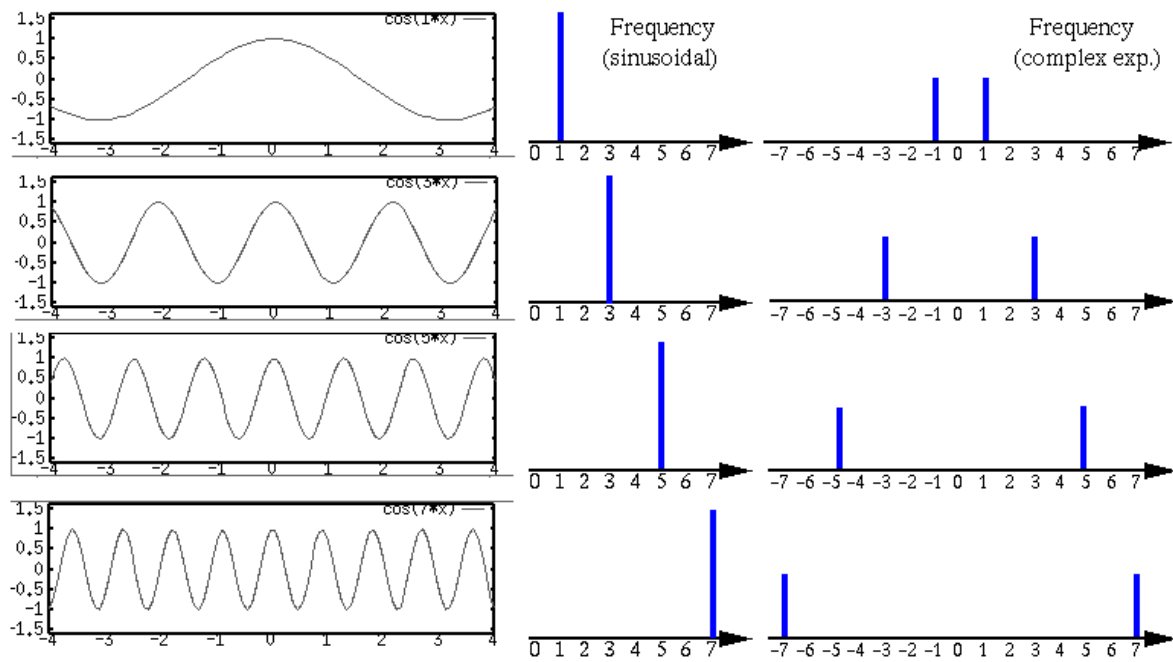


Figure 11: Coefficients of the Fourier series (<http://fourier.eng.hmc.edu>)

2.4.2. Fourier Transform

A Fourier transform can be described as a method similar to that of the Fourier series but a much more powerful method because the function is integrated from $-\infty$ to $+\infty$. The Fourier transform is also the preferred method to use when dealing with non periodic signals. Assume a function $F(f)$ similar to that of equation 2.4 which contains both sine and cosine series. However we define the sine series as an imaginary component (Palm, 2007:564). Therefore:

$$F(f) = \int f(t) \cos(ft) dt - i \int f(t) \sin(ft) dt \quad (2.5)$$

where f is the frequency.

By manipulating equation 2.5 we obtain

$$F(f) = \int f(t) (\cos(ft) - i \sin(ft)) dt \quad (2.6)$$

and by using Euler's formula and integrating from negative infinity to infinity we obtain

$$F(\omega) = \int_{-\infty}^{\infty} f(t)e^{-i\omega t} dt \quad (2.7)$$

which is known as the Fourier transform $F(\omega)$ of $f(t)$.

Converting a function from the frequency domain into the time domain the inverse Fourier transform is used and defined by

$$f(t) = \frac{1}{2\pi} \int_{-\infty}^{\infty} F(\omega)e^{i\omega t} d\omega \quad (2.8)$$

Assume two vibration sensors with time functions $x(t)$ and $y(t)$ respectively. The APSD and CPSD for the system can be calculated as follows (Natke, 1983:1):

$$APSD_x(\omega) = \langle \mathbb{F}^*(x(t)) \cdot \mathbb{F}(x(t)) \rangle \quad (2.9)$$

$$APSD_y(\omega) = \langle \mathbb{F}^*(y(t)) \cdot \mathbb{F}(y(t)) \rangle \quad (2.10)$$

$$CPSD_{xy}(\omega) = \langle \mathbb{F}^*(x(t)) \cdot \mathbb{F}(y(t)) \rangle \quad (2.11)$$

where $\mathbb{F}$ represents the Fourier transform of the function, $*$ denotes the complex conjugate and $\langle \rangle$ denotes the expectation value.

The APSD function provides magnitude whereas the CPSD function provides both magnitude and phase. The phase angle ϕ_{xy} between the functions $x(t)$ and $y(t)$ can be calculated with the follow equation (Natke, 1983:1):

$$\phi_{xy}(\omega) = \arctan \frac{\text{Im}\{CPSD_{xy}(\omega)\}}{\text{Re}\{CPSD_{xy}(\omega)\}} \quad (2.12)$$

where $\text{Im}\{CPSD_{xy}(\omega)\}$ and $\text{Re}\{CPSD_{xy}(\omega)\}$ represent the imaginary and real values respectively. Finally the magnitude of the CPSD can be calculated as follows:

$$|CPSD_{xy}(\omega)| = \sqrt{\{\text{Im}(CPSD_{xy}(\omega))\}^2 + \{\text{Re}(CPSD_{xy}(\omega))\}^2} \quad (2.13)$$

The coherence function is defined by

$$\gamma_{xy}^2(\omega) = \frac{CPSD_{xy}(\omega) \cdot CPSD_{xy}^*(\omega)}{APSD_x(\omega) \cdot APSD_y(\omega)} \quad (2.14)$$

and is used to indicate whether the vibration at a particular frequency is related or not. The values of the coherence function lie between zero and one for all frequency values. Zero indicates no relation between the vibrations and one indicates that the vibrations are strongly related.

2.5. Vibration analysis overview

The timeous detection (and prevention) of mechanical degradation in active and passive components is the primary goal of condition monitoring and predictive maintenance. Components in industrial plants are usually expensive. As a consequence it is a requirement that the maximum useable lifetime be obtained from these components. Spare components are not always kept on site and due to the fact that the manufacturing and transport of large components are often time consuming, an optimised condition monitoring and predictive maintenance programme could be beneficial. Predictive maintenance entails the ability to predict when components should be replaced. According to Reimche *et al.* (2003:1) vibration monitoring of components started intensifying in the 1960s with very basic monitoring equipment and methods. Over the years these types of equipment and methods were improved to basic handheld vibration monitoring equipment and in the 1980s computers were introduced for the first time to analyse and store vibration data captured from a system.

Online monitoring systems have become more prevalent due to the developments in monitoring equipment and techniques and decreasing cost as well as the realisation that such on-line systems offer significant cost-benefit advantages in terms of plant protection and improved safety measures.

A vibration monitoring system should fulfil the following requirements in order to operate a nuclear plant with confidence (Jirsa, 1996:1):

1. The system should be online for continuous vibration monitoring.
2. The system should be able to monitor the integrity of individual internal components in normal operation.
3. The system should be capable of monitoring and predicting anomalous internal component vibration in the event of an accident.

Most nuclear power plants are equipped with on-line vibration monitoring equipment which automatically monitors the vibration behaviour of the components. This minimises the input of plant personnel during normal operation which will in effect minimise human error. These systems can automatically detect deviations in vibration behaviour by monitoring e.g. frequency signatures and will provide reports and alarms to the user (Reimche *et al.*, 2003:2).

One of these systems is called COMOS, which according to van Niekerk and Sunder (1988:155) is an online condition monitoring system for the main coolant pump (MCP) and passive primary components vibration. This system operates on the principle of collecting reference signatures and comparing these with frequency signatures obtained whilst monitoring the system online. This system will then indicate any changes in frequencies which will be an indication of possible component degradation. The system includes a number of user-definable diagnostic rules to assist the operator in diagnoses in case of unforeseen deviations in the signatures.

In France a data base called SINBAD has been created for the storage of processed frequency signatures to assist nuclear power plants with significant information to detect mechanical degradation. This signature processing has, according to Trenty (1995:347) allowed the user to:

- Scrutinize frequency signatures and be able to see the difference between the frequency signatures of mechanical degradation and variation in signatures for normal operation.
- Observe mechanical degradation; predicting mechanical wear of measured component based on frequency signatures obtained.
- Finally, this enabled automatic recognition of different types of vibration behaviour for the measured components.

Accelerometers, velocity and displacement transducers measure the vibration during the normal operation of different components. The sources of the vibration for active and passive components are of different origins. Vibrations in active (rotating) components are primarily caused by the forces acting on that component to enable it to rotate at a given frequency while passive components vibrate due to external forces. Flow-induced vibration is common in both active and passive components. Fluctuations in pressure and temperature also influence the vibrations of components in a system (Reimche *et al.*, 2003:1).

The Atucha-I nuclear power plant, situated in Argentina, is a pressurised heavy water reactor (PHWR). According to Belinco *et al.* (1997:22) it was designed and built without online vibration monitoring equipment. However in 1990 it was decided to install three velocity transducers together with one piezoelectric accelerometer. Both the transducers and the piezoelectric accelerometer were placed on the top of the RPV head.

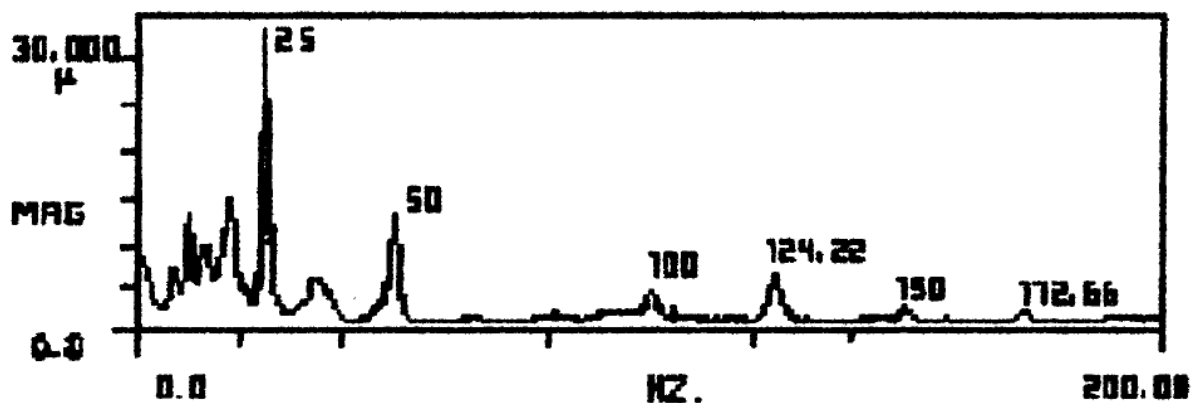


Figure 12: 0 – 200 Hz band spectrum from a velocity transducer (Source: Belinco *et al.*, 1997:24)

Figure 12 indicates resonances caused by components in the primary system. This spectrum was obtained using a velocity transducer. Figure 12 depicts a number of resonances in the region of 0 Hz to 25 Hz. According to Belinco *et al.* (1997:24) the low frequency range from 0 Hz to 25 Hz contained three different frequency bands. The first band (6.8 Hz – 11 Hz) that was identified contained distinctive resonances at 7.1 Hz, 9.9 Hz and 10.45 Hz respectively. The 7.1 Hz and 9.9 Hz resonances were identified as the natural frequencies of the RPV while the 10.45 Hz peak was identified as the fluid resonance frequency caused by the MCP. Research still needed to be done

on the second frequency band (12 Hz – 14 Hz) to clearly define each resonance; however resonances that were identified were the natural frequencies of the moderator vessel and moderator vessel upper head at 12.9 Hz and 13.4 Hz respectively. In the third frequency band (16 Hz – 20 Hz) the only resonance that could be clearly identified was the natural frequency of the moderator vessel. The resonance at 25 Hz is the rotating frequency of the MCP which rotates at roughly 1500 revolutions per minute (rpm). Belinco *et al.* (1997:24) states that the resonances at 124.22 Hz are the pressure pulsation of the MCP and the 172.66 Hz is the pressure pulsation of the moderator pumps. The latter is caused by unsteady flow behind the blades of the pump causing pressure pulsations and stress concentrations to occur. The so called blade pass frequency is calculated by multiplying the rotational speed by the number of pump blades. At the Atucha-I nuclear power plant the MCP consists of 5 blades and the moderator pumps of 7 blades and both rotate at a speed of approximately 1500 rpm.

During operation, minor deviations in the vibration behaviour of the component may develop, caused by for e.g. misalignment, resonance or fatigue.

Resonance should be carefully considered in machine design and operating regime. When the frequency of an external influence (forcing function) coincides with the natural frequency, resonance is amplified. This often results in failure within a short time span. A typical example is running a rotating machine at a rotation speed coinciding with a natural frequency of the shaft (the so-called critical speed). This will cause severe vibration in the rotating machine. Running a machine with the frequency of excitation forces close to machine resonances is one of the causes of high vibrations in mechanical machines.

Shigley and Mischke (2001:396) state that fatigue failure is caused by the continuous fluctuation of stresses. The initial sign is typically a small crack but in due time will gradually increase until a visible crack can be seen. Thereafter crack propagation will increase considerably because of stress concentrations that will occur in that region.

Typically signals from the accelerometers, velocity and displacement transducers are converted into a waveform in the time domain. As an example, Figure 13 depicts the different states in the lifetime of a gear wheel. The graphs on the left hand side indicate

the signals in the time domain from the sensors while the graphs on the right illustrate the frequency signatures.

The first spectrum (graph A) is used as reference to illustrate normal vibration of the gear wheel. On the right the frequency signature is shown for normal vibration (Reimche *et al.*, 2003:5).

In graph B a minor deviation which is indicated with red develops on the frequency spectrum while the reference condition is still indicated in green. This minor deviation will act as a point where stress concentrations will start to develop and from this point the fault line will gradually increase through the whole component. Both the time domain and the frequency domain on graph C displays the onset of fault development from a small fault into a distributed fault. The fault will gradually increase over time and the deviation from the reference condition in the time wave and the frequency spectrum will gradually increase as shown in graph D and E. Finally, if not repaired in time, this will lead to catastrophic failure of the component.

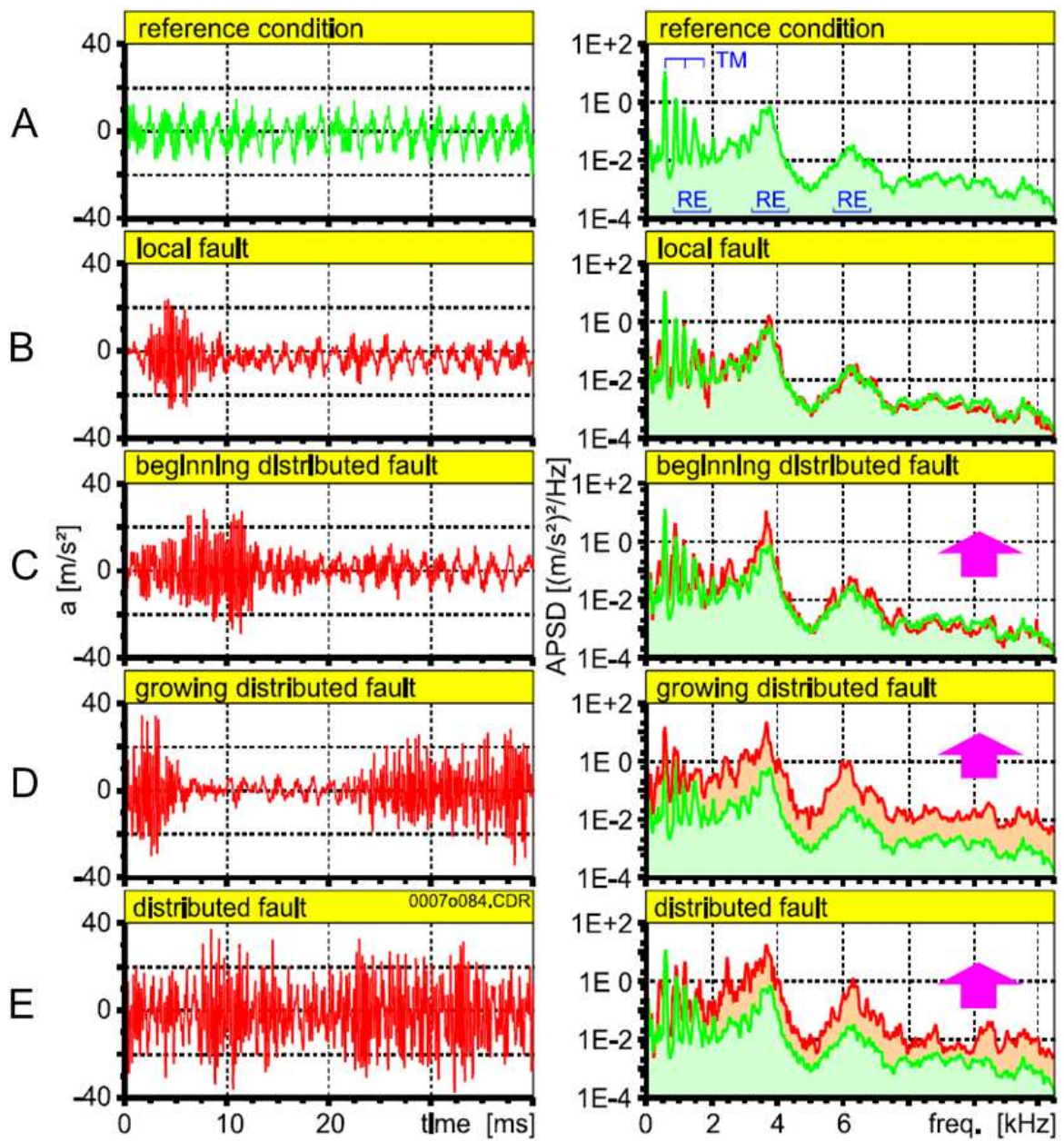


Figure 13: Time waves and frequency spectra for a gear wheel (Source: Reimche *et al.*, 2003:5)

3. PROBLEM STATEMENT

The engineering problem can be summarized as follows:

“The purpose of the study is to obtain primary reactor component vibration data and analyse this data, by i.a. converting time waves into frequency signatures (using the Fast Fourier Transform). Investigate the long term signatures of the KNPS in order to indicate possible mechanical degradation. Simulate the vibration behaviour of the mechanical system and the measurement chain by virtue of solving the elasto-dynamic vibration equation.”

A nuclear power reactor is continuously subjected to vibration excitation forces such as the forces generated by the MCP shaft rotation speed at approximately 25 Hz as well as random flow-induced forces. Monitoring the vibration of plant components will enhance the safety of the plant and will increase plant efficiency by virtue of reduced plant down-time and planned maintenance.

Vibration data from the KNPS will be converted from the time waveform into the frequency spectrum by means of Fourier analysis. A set of reference frequency spectra, or reference signatures, will be calculated and used for subsequent comparison of newly obtained signatures. The results will be studied to determine whether any deviations from the reference pattern are found.

Resonances in the frequency spectra represent frequency resonances of specific components within the system. These components will be identified in the frequency spectrum. Deviations in the resonances need clarification and interpretation since these deviations may be indicative of mechanical degradation of that component. The latter will have to be replaced to eliminate failure of the component which will lead to reduction of down time.

Sensor decay in nuclear reactors is a cause of concern due to the ongoing high radiation level within the primary system where the sensors are located. Therefore, while monitoring deviations in the frequency spectra, the degradation of the sensors will also be investigated to determine whether replacement is necessary. This is done by

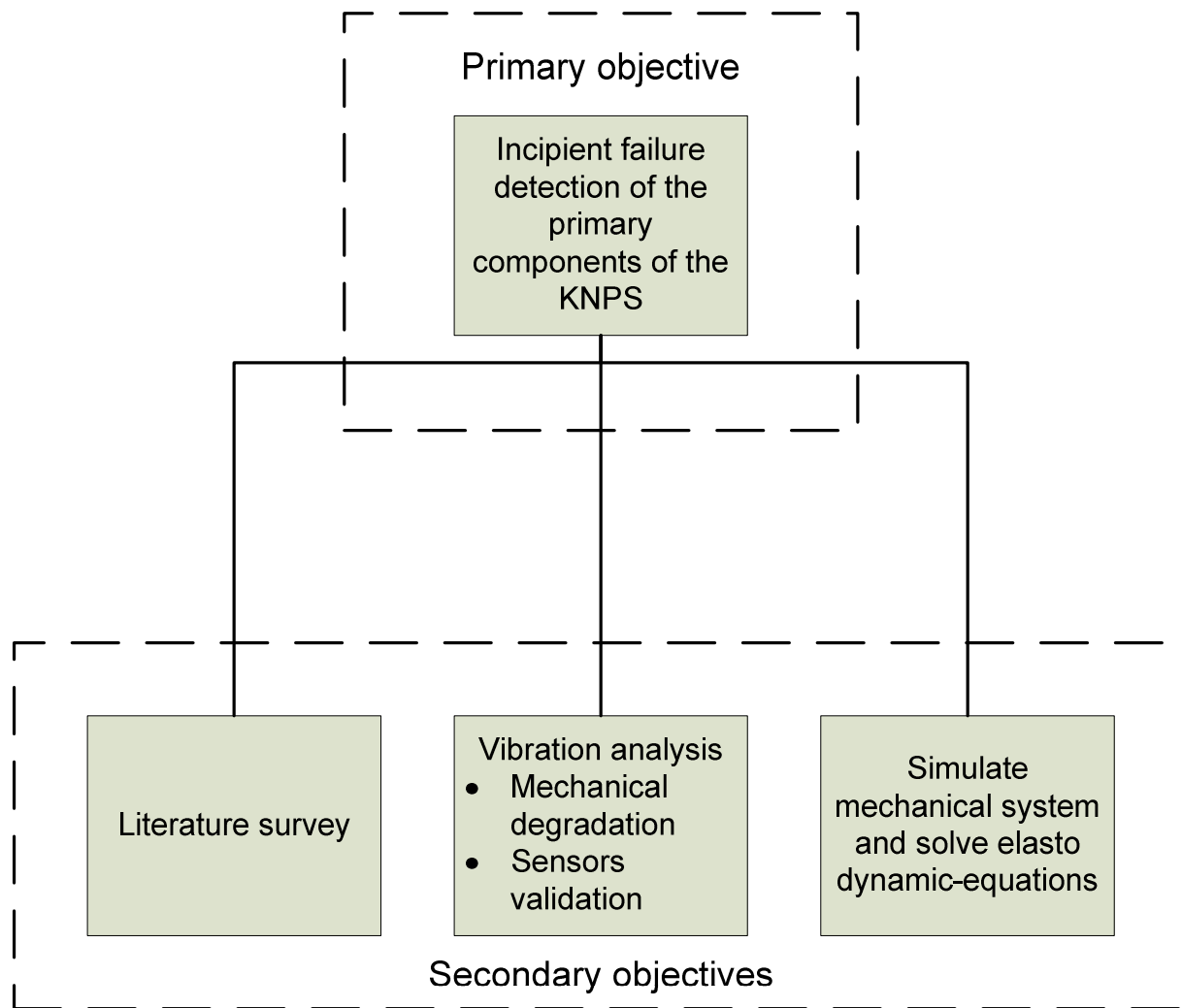
studying signals (e.g. frequency spectra); access to these accelerometers is not allowed during reactor operation due to high radiation levels in the reactor containment.

The primary system of the KNPS will be modelled using a simplified elasto-dynamic model simulating the mass, stiffness and damping of a number of components. This model, incorporating the mass, damping and stiffness matrices, will be used in the elasto dynamic equations to generate displacement time series. It will be shown that this simplified model can be used to simulate a number of resonances of different components.

The above mentioned equations will be solved numerically from first principles. Programming will be done in the software package MathCADTM 14.

The study will be concluded with a summary of findings regarding the mechanical integrity of the KNPS primary components and the signals used in the study. Finally recommendations will also be made regarding further studies in this field.

4. PROJECT OBJECTIVES



Methods employed:

Primary objective

The primary objective of this study has been identified as determining whether the primary components of the KNPS have undergone any mechanical degradation (as could be measured by the existing vibration sensors) which could lead to unexpected failure of component and downtime of the plant. This will provide plant personnel with the necessary information to do predictive maintenance. To achieve this goal a few secondary objectives have been identified.

Secondary objectives

1. Literature survey

The literature survey contains relevant aspects of the field of this study. The literature overview covered aspects relating to vibration equipment, vibration sources and plant parameters of the KNPS. An aspect related to signature analysis is to study cases of vibration interpretation and failure development in similar plants.

2. Vibration analysis

Vibration data from the KNPS will be analysed to determine mechanical degradation in the primary components. This involves analysing the time waveforms from the accelerometers and neutron detectors and converting this by means of FFT into frequency signatures in order to determine any resonance deviation. From these signatures the sensor degradation will also be determined.

3. Elasto-dynamic equations

The primary system of the KNPS will be simulated using a mass, stiffness and damping model. From this model elasto dynamic-equations will be drawn and solved numerically from first principles. Programming will be done in the software package MathCADTM 14. This model will then be used to simulate mechanical degradation in a component. The model will also indicate the output signal from the vibration sensors when the sensors have experienced advance decay due to radiation exposure etc.

5. VIBRATION IN THE PRIMARY SYSTEM OF THE KNPS

5.1. The Koeberg Nuclear Power Station

The KNPS is a 2 unit, 3 loop nuclear power reactor producing 2785 MW_{th} and 909 MW_e per unit. The KNPS is situated just outside Duynefontein, approximately 30 km North of Cape Town, and is also the only nuclear reactor in Africa generating electrical power. The first unit went online on the 21st of July 1984, followed by unit 2 which was commissioned on the 9th of November 1985. The power generation contributes roughly 6 percent of the energy demand in South Africa (World nuclear association, 2010).



Figure 14: Aerial view of the Koeberg Nuclear Power Station (Source: www.eskom.co.za)

The KNPS uses water as primary coolant to remove heat generated in the core by the fission process. Cold primary water coolant flows into the inlet of the RPV at 286.7 °C where it is heated by the nuclear core to 323.2 °C. The coolant exits at the outlet to the steam generator where the heat is removed by the secondary cycle as can be seen in

Figure 15 (International nuclear safety center, 1997). The steam generator consists of 3330 inverted U-tubes which have a total heat transfer surface area of 4699 m². The function of the pressuriser, with a volume of 38.85 m³, is to keep the pressure in the primary cycle at 15.5 MPa, which is sufficient to keep the water coolant in the primary cycle from boiling. The colder coolant goes through the primary pump where it is pumped back into the RPV to complete the cycle (Eskom, 2000:19).

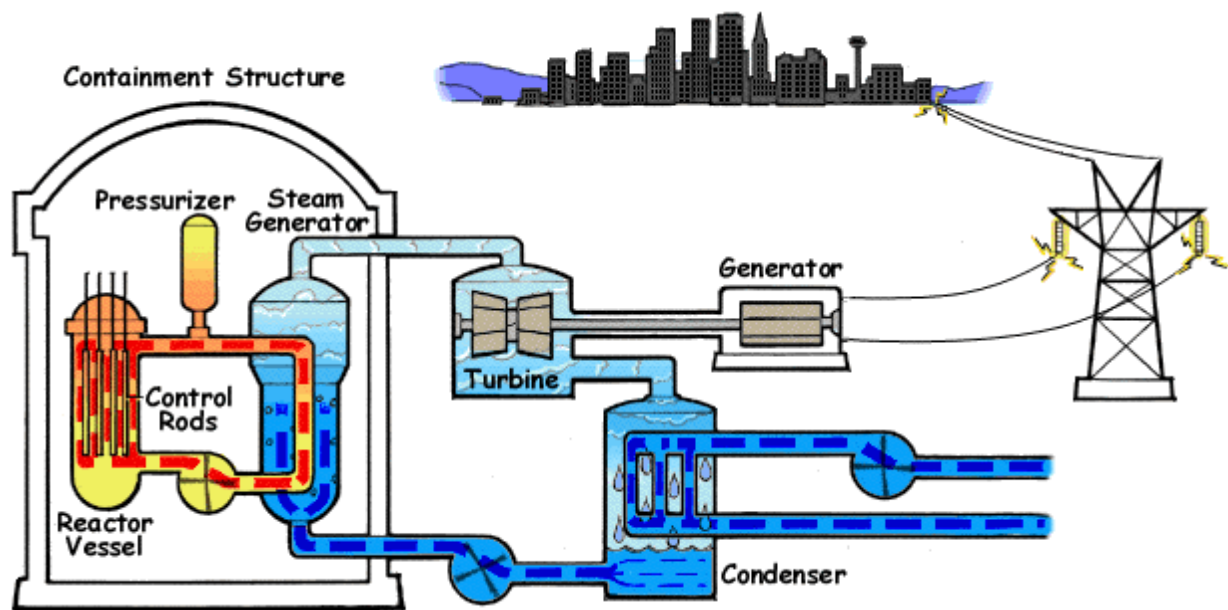


Figure 15: Koeberg Nuclear Power Station layout (Source: www.coal2nuclear.com)

The core of the reactor contains 157 fuel assemblies. The assemblies consist of an array of 17 x 17 elements which include 24 control rods. The function of the control rods is to control the criticality of the reactor thus also serving as a safety feature. The fuel rods are made of Uranium dioxide (UO₂) with a maximum enrichment limit of 4.95% (Eskom, 2000:17).

The plant is designed for a lifetime of 40 years but Eskom is in the process of extending the lifetime by 20 years. Steam generator upgrades are scheduled for 2015. The overall efficiency of the KNPS is 32.6% (Eskom, 2000:17).

In Appendix A a brief overview on the origin and development of nuclear energy is discussed. The latter also contains an explanation of the two major accidents which occurred in the history of nuclear power which influenced safety criteria associated with nuclear power generation.

5.2. Vibration monitoring of the primary reactor components of the KNPS

The KNPS reactor vessels for unit 1 and unit 2 are continuously subjected to dynamic conditions influencing its vibration behaviour. The vibrations occur in the reactor and reactor internal components which include fuel assemblies, fuel rods, control rods, core barrel and the RPV. The excitation forces in the primary system are primarily flow induced, as well as due to the forces caused by the rotating pumps and forces exerted through the piping systems.

In order to monitor the mechanical integrity of the primary reactor components, the KNPS installed sensors on the RPV as described in section 5.2.2. The reactor loose part monitoring system, referred to as the KIR system (based on a French acronym), consists of two functionally different sub-systems.

The Loose Part Monitoring System (LPMS) is an independent on-line monitoring system with the main function to detect loose parts in the primary system which could cause damage to components during normal operation of the plant. Sensors for this system are placed at both the bottom of the RPV and each of the steam generators. However the discussion and functions of this system fall outside the scope of this study.

The Reactor Vibration Monitoring System (RVMS) captures vibration data from accelerometers located on the RPV and neutron detectors located outside the RPV and stores this data in the form of time wave forms as well as APSDs (Molyneux, 2007:5). This data can be used to study and monitor deviations and trends in the vibration of components in order to detect possible mechanical degradation.

5.2.1. KNPS vibration sensors

Accelerometers

The accelerometers used in the KNPS for the vibration monitoring of the reactor primary components are Endevco piezoelectric accelerometers (Molyneux, 2007:11). In 1961, Endevco started to focus some of its attention in developing high temperature

accelerometers for extreme conditions. The first accelerometers with this unique feature contained a quartz crystal with a temperature limit of approximately 400 °C. Gradually over the years the temperature limit increased as technology developed. In 1988 Endevco released a 760 °C high temperature accelerometer (Walter, 2006:5).

Seven Endevco piezoelectric accelerometers model number 7703A - 200 are mounted on the RPV of the KNPS. These accelerometers weigh 62 grams each, which enables easy handling and mounting on the RPV without affecting the system. The accelerometers were also tested in a high radiation environment and can withstand 10^8 rads of gamma flux and 10^{10} N/cm² of neutron flux, making it suitable for measuring acceleration in the primary system of a nuclear plant without excessive degradation during an operating cycle. Operating temperatures for this accelerometer range from - 55 °C to approximately 288 °C (Endevco, 2008:2).

This small device is also self generating in that no external power supply is needed to power the accelerometer. The accelerometers are able to provide output in the high resonance frequency with great stability over long periods of time.

Neutron detectors

Four neutron detectors are mounted on the RPV of the KNPS. The primary use of the neutron detectors is to calculate the fission rate within the reactor core. This allows determination of the reactivity of the reactor which is essential for the safe operation of the plant. However the detectors, located 90 degrees apart on the outside the RPV, as depicted in Figure 16, can also be used for vibration monitoring of the primary components. The detector consists of independent upper and lower sections. Hence the detectors have viewing angles on the upper and lower sections. The neutron detectors can indicate movement of different components within the RPV. Some of these components include:

- Core barrel,
- Fuel assemblies and
- Fuel rods.

Figure 16 indicates the general layout of the neutron detectors at the KNPS. The upper and lower detectors of each of the four ionisation chambers detect the neutrons exiting the RPV. By means of ionization the neutrons are converted into charge particles by reacting with boron trifluoride (BF_3) gas releasing an energetic alpha particle. The number of alpha particles released is an indication of the total number of neutrons present. The signal from the detector is then amplified by a factor of 800 prior to feeding into the KIR system.

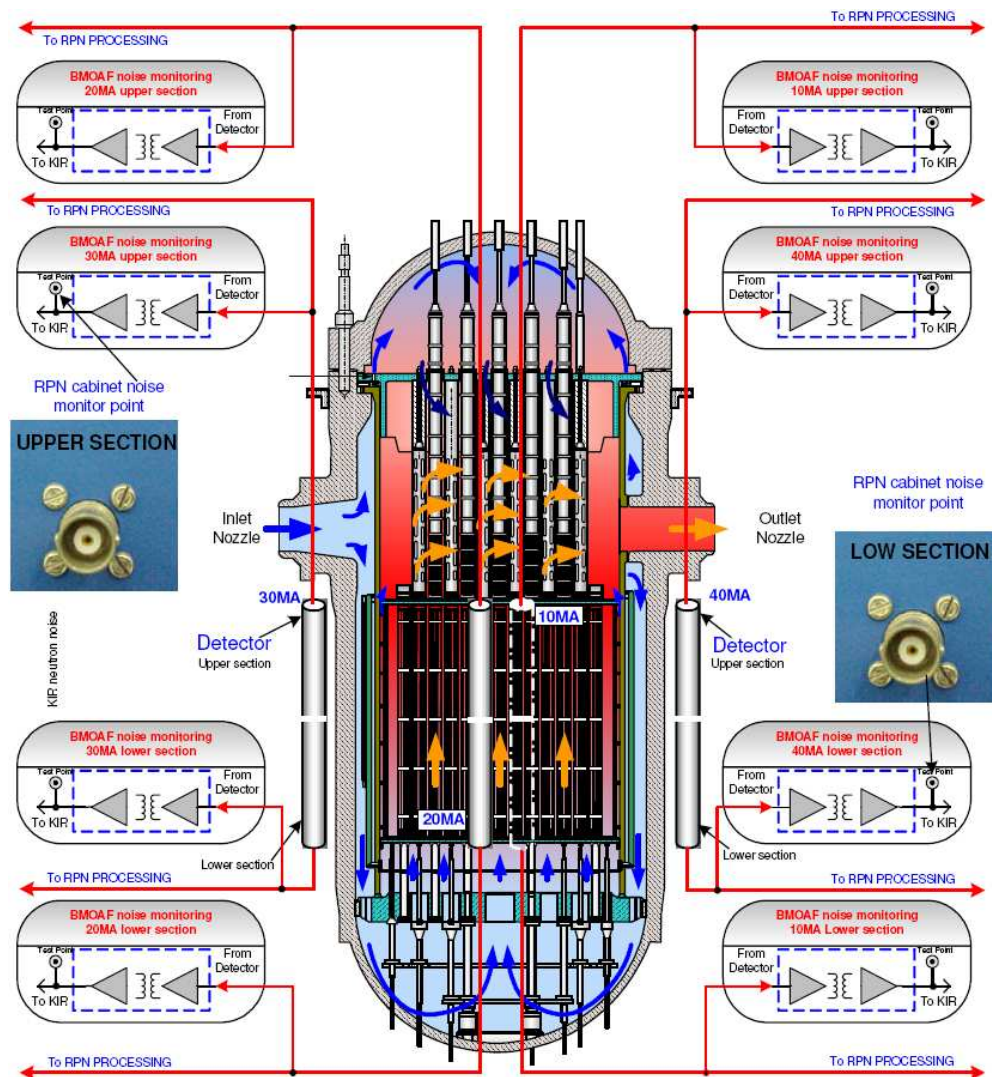


Figure 16: Neutron detector's positions around the RPV (Source: Molyneux, 2007:12)

When the reactor is operating under normal conditions water coolant is forced into the inlet nozzle and moves down between the RPV and the core barrel, then up into the reactor core, up through the fuel assemblies and out at the outlet nozzle. Figure 17 is a simplified illustration of coolant flow in the vessel. Due to the turbulent flow of the coolant, internal components are excited to vibrate. The relative movement of the core

barrel with regards to the reactor vessel, changes the water gap in the so-called down-comer area, resulting in a modulation of the neutrons arriving at the neutron chambers. The statistical fluctuations on the output of the neutron detectors will therefore contain a measure of the relative vibration between the core barrel and the reactor core.

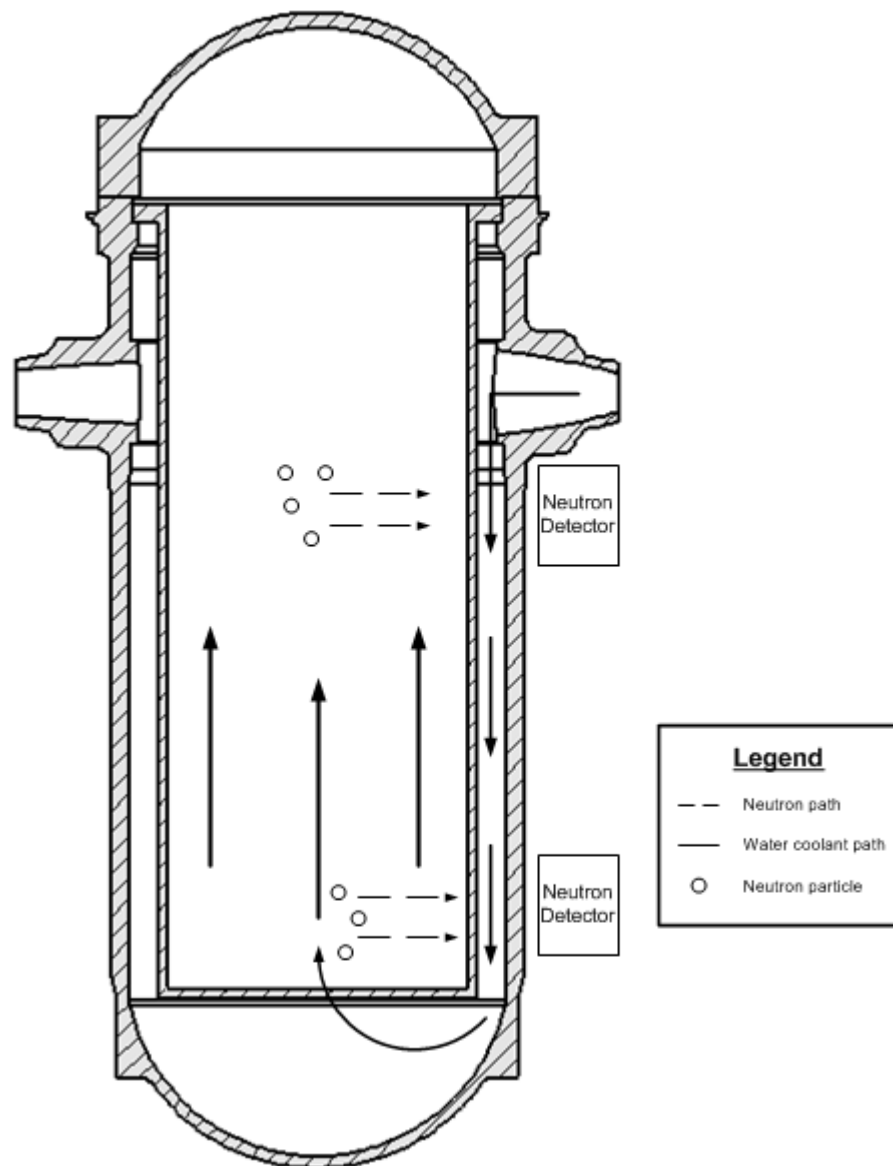


Figure 17: Frontal view of the RPV with core barrel (Source: Figure by the author)

Velocity and proximity transducers

Velocity and proximity transducers are devices used to measure the velocity and displacement respectively of a component or system. KNPS uses sensors from the Bently Nevada range.

Four 3300 XL Proximeter[®] sensors are installed on the motor / pump configuration. The locations of the sensors are as follows:

- Motor shaft and
- Pump shaft.

Proximity measurements are recorded in both the x and y direction for the above mentioned locations. The sensors are designed with zero moving parts and the sensors are radiation resistant to function in the nuclear environment.

Six radiation resistant 330530 Velomitor[®] sensors are installed on each of the pump and motor assemblies at the KNPS. The positions of the sensors are as follows:

- Motor non drive end,
- Motor drive end and
- Pump upper casing.

For the above mentioned positions, velocity changes are measured in the x and y directions. The velometers are designed with no moving parts, minimising mechanical wear and degradation thereby improving performance and lifetime of the device. The 330530 velometer is fitted with a piezoelectric crystal and operates on the same principle as the piezoelectric accelerometer. When the device is accelerated in a direction, a pressure is exerted on the piezoelectric crystal, resulting in mechanical stresses being generated inside the crystal which produces an electric charge. However the piezoelectric crystal is connected to on-board electronics containing both a low noise amplifier and an integrator. As soon as the acceleration signal is generated, the electronics integrate the signal and develops the first integral of acceleration which is velocity.

The firm casing of the velometer is manufactured from 304L stainless steel, having physical dimensions of 25.3 mm in diameter and 63.5 mm in height making the device very portable. The complete assembly weighs roughly 142 grams. The velometer is designed to operate in harsh radiation environments like the primary system of the KNPS and can withstand gamma radiation up to 3 Mrads. Operating temperatures range from approximately – 55 °C to 121 °C (GE Energy, 2 009:2).

5.2.2. Sensor positioning

The KIR system was installed to record and store vibration data of the primary components of the KNPS. This system's primary goal is to detect deviations in vibration signatures from normal operation in order to predict mechanical degradation.

As depicted in Figure 18 the one part of the KIR system called the LPMS consists of nine accelerometers, three of which are located on the bottom of the RPV (008MV, 009MV and 010MV). The remaining six accelerometers are located on the three steam generators (021/022MV, 031/032MV, and 041/042MV). The sensors monitor the system for loose parts enabling an alarm to be activated to indicate the detection of a problem. Data is continuously captured and stored on a hard drive to promote the concept of learning trends for future comparison (Molyneux, 2007:5).

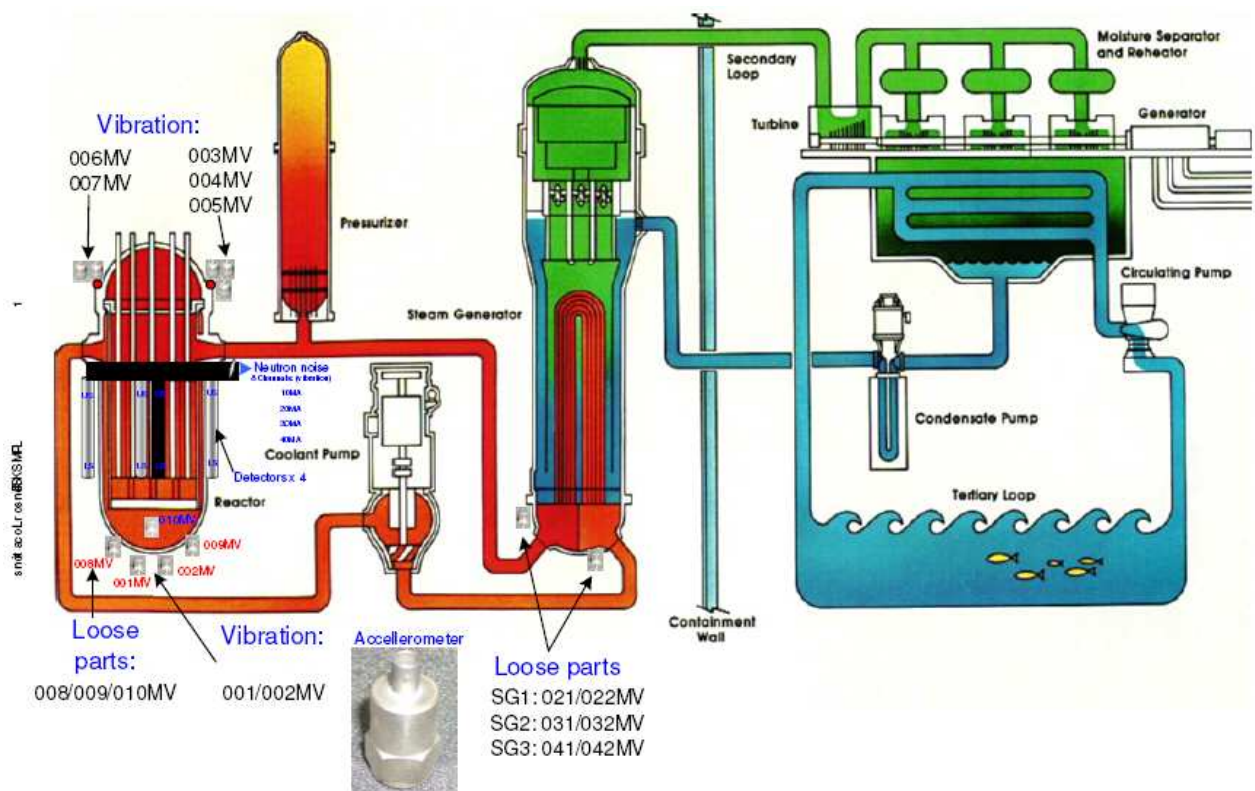


Figure 18: Reactor loose part monitoring system (Source: Molyneux, 2007:7)

The sensors of the RVMS system, including the accelerometers (001MV-007MV) and neutron detectors (RPN 10MA, RPN 20MA, RPN 30MA and RPN 40MA upper and lower), are depicted in Figure 18. Time signals are sampled with a frequency of 1024/8

Hz, rendering vibration spectra are captured with a resolution of 0.125 Hz from where the APSD is calculated before storage.

The sensors of the RVMS system are mounted on the RPV of unit 1 and unit 2. Figure 19 illustrates the layout of the sensors (van Niekerk, 2000:3):

- 4 Neutron detectors close to the upper half of the core, each of which is 90 degrees apart (1SH – 4SH),
- 4 Neutron detectors close to the lower half of the core, each of which is 90 degrees apart (1SB – 4SB),
- 3 Perpendicular accelerometers mounted on the top of the vessel flange (3MV – 5MV),
- 2 Perpendicular accelerometers mounted on the top of the vessel flange 180 degrees out of phase from above mentioned accelerometers (6MV – 7MV),
- 2 Perpendicular accelerometers mounted on the bottom of the core barrel (1MV – 2MV).

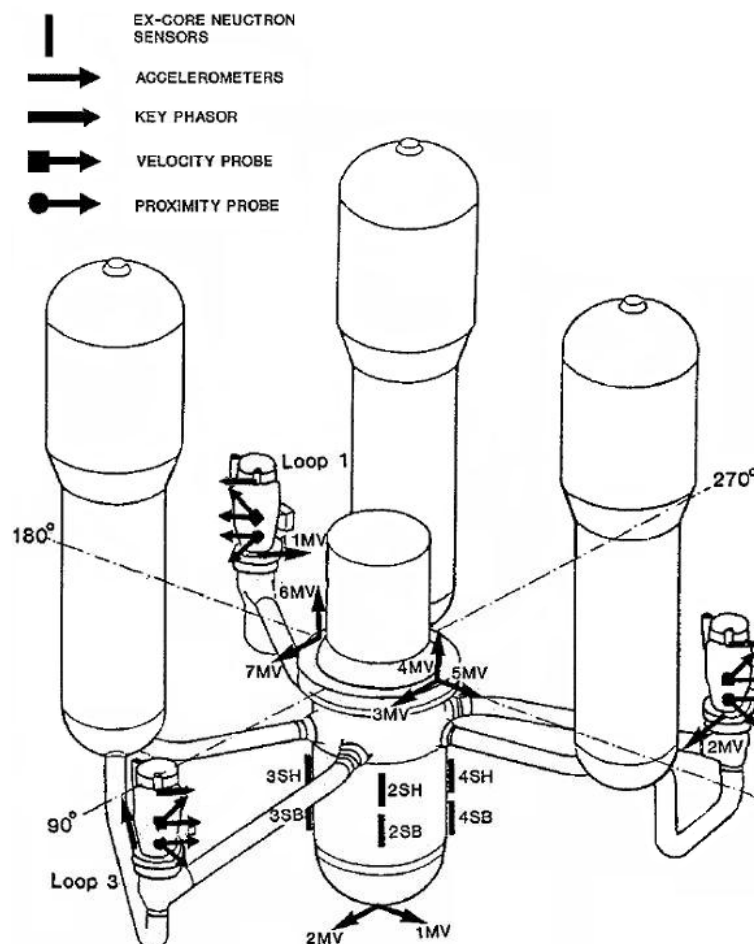


Figure 19: Vibration sensors positions for the KNPS (Source: van Niekerk, 2000:3)

Velocity probes are mounted on the MCPs to measure vibration of the pumps. A proximity probe measures the relative displacement of the MCP. The MCPs or primary coolant pumps are also fitted with one accelerometer each. The pumps rotate at approximately 1500 rpm. The monitoring of the MCPs are not part of the scope of this study.

5.2.3. Signal path

Figure 20 shows the signal path of an accelerometer from the actual detection of vibration on the component to and including the processing of vibration data. The accelerometer (1) is directly connected to the component or monitored system. From here the signal leads to a charge converter (2) which amplifies the signal and converts the signal from a charge output into a voltage. The converter is essential due to the low output charge generated by the piezoelectric accelerometer. Long cable lengths also negatively influence the signal quality (Molyneux, 2007:6).

Cables penetrate the containment walls (3) and feed into a signal conditioning module (4). The 16 channel conditioning module has two functions, firstly to amplify the signal from the charge converter and secondly to power the charge converter. It also serves as the interface between the vibration monitoring equipment and the computer system (Molyneux, 2007:6). Each channel is also fitted with two light emitting diodes (LEDs) to indicate the state of the input signals.

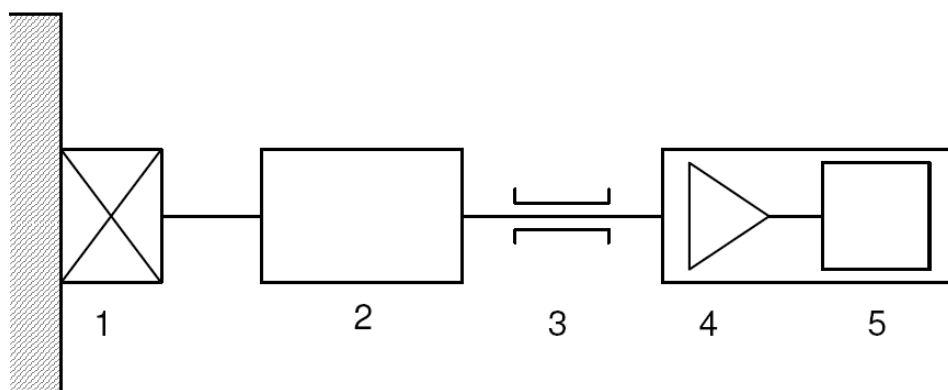


Figure 20: KIR system process (Source: Molyneux, 2007:6)

Finally the signal is imported into a computer system (5). Figure 21 reflects a more detailed overall layout of the vibration monitoring system installed by IST delkor. The data is then transmitted via a dedicated Ethernet connection to the peripheral component interconnect (PCI) extension for Instrumentation (PXI) system consisting of four dynamic acquisition (DSA) cards and a multifunction data acquisition (DAQ) card which converts the real time data into digital data for processing by the industrial PC. Also connected to the PXI system are four test hammers connected to each of the three steam generators and the RPV. The latter is part of the LPMS to test system functionality and to determine different sizes of loose parts. Finally audible signals are available from real-time data or recorded data by utilising the audio monitor.

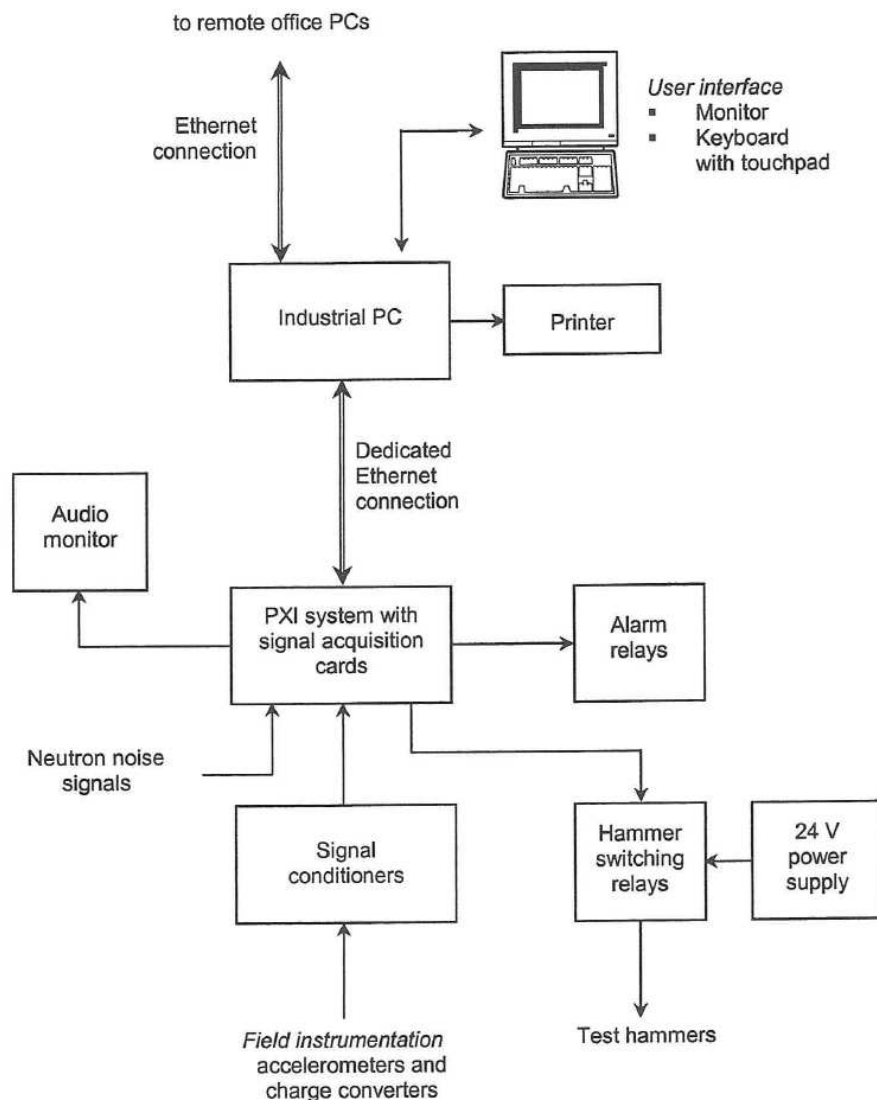


Figure 21: Vibration monitoring system architecture of the KNPS (Source: Penzhorn, 2005:3)

The user interface with the signal processing procedure is obtained by the use of the industrial PC fitted with Windows 2000 as operating system. It allows the user to evaluate incoming data as well as data stored on the hard drive. Data can be written on a DVD writer which increases data mobility and redundancy. A printer is also provided for hard copies.

5.3. Software used for monitoring purposes

LPMS Analysis Client is the software program used by the KNPS for the monitoring of the LPMS system and RVMS system. The program is used to display the vibration signals recorded by the accelerometers and neutron detectors mounted on the RPV, steam generators and MCPs.

Each signal recorded by the vibration sensors is filtered by a low-pass eight-order Chebyshev filter with a corner frequency of 50 Hz (Penzhorn, 2005:8). Dynamic signals from accelerometer and neutron detector channels are recorded and processed into APSD prior to storage.

Data samples are captured in 8-second blocks (containing 1024 data points) after which the APSD is calculated. The waiting period, that is the time between successive samples, can be defined by plant operators in hours. Once completed the next 8-second record is captured and APSD is calculated. This process is repeated x amount of times which can also be set by plant personnel. After the process is completed for x amount of samples, the APSD is averaged by dividing the sum of the APSD by x.

Following the above mentioned process, the average APSD is stored together with the appropriate time waveform. This enables the viewing of signals at later stages for diagnostic purposes.

For each sample recorded, the LPMS Analysis Client software package also stores the average plant conditions at the same time the data was recorded. This provides plant personnel with the necessary information to obtain an understanding of the plant's state at the time the measurement was taken. The water coolant inlet temperature, water coolant outlet temperature, core power and the primary system pressure are recorded.

Signal integrity can be validated using the LPMS Analysis Client. A set point is defined with no sensor decay or cable loss. While the system is running the root mean square (rms) value for each channel is monitored. Signal failure is indicated if at any time the signal drops below the set point.

6. SPECTRUM ANALYSIS

As shown in the introduction, the frequency spectrum reveals harmonic content of a time series. This makes frequency analysis a powerful and elegant method for vibration monitoring. The correct interpretation of the content of frequency spectra is needed in order to use spectra for (on-line) vibration monitoring.

The interpretation of frequency spectra consists of identifying resonances (i.e. the frequency and amplitudes of peaks in frequency spectra) and determining the components corresponding to these resonances. Frequency spectra measured at the onset of component life will be used as a reference. Subsequent measurements will be compared to relevant reference signatures. Changes in frequency peaks, associated with component resonances, may be indicative of mechanical failure and need to be investigated.

In this chapter the frequency spectra calculated from accelerometers and neutron detectors for both units of the KNPS will be investigated to determine whether any components within the primary system has experienced mechanical degradation. Data used consists of 2 parts. The first part contains vibration data from 1985 to 2000 which was analysed by van Niekerk (2000:4). The data and conclusions recorded will be compared to the new data set. The second section contains new vibration data. A compact disc containing the data for both units was received from the KNPS together with the LPMS Analysis Client software package. The FFT spectra for all signals included in the signal array will be analysed and interpreted. The findings from the two sets of data will be compared in order to obtain an overall interpretation of the state of the KNPS units one and two.

6.1. Interpretation of KNPS signatures

The following modes of vibration of the KNPS primary components have been identified and linked to peaks in measured spectra (van Niekerk, 2000:5):

- Core barrel beam mode

The beam mode of the core barrel can be defined as the movements of the core barrel, being fixed at the top and free at the bottom, similar to that of a pendulum moving from side to side. This mode is clearly distinguishable in the frequency spectra of the ex-core neutron detectors. Each neutron detector at the KNPS is 90 degrees apart and results in opposite detectors measuring core barrel frequencies 180 degrees out of phase. From this result the conclusion can be drawn that as the core barrel moves closer to the one detector the core barrel will move further from the opposite detector. Park *et al.* (2003:181) illustrated the beam mode of a standard Korean pressurised water reactor (PWR) using a finite element model (FEM), as shown in Figure 22.

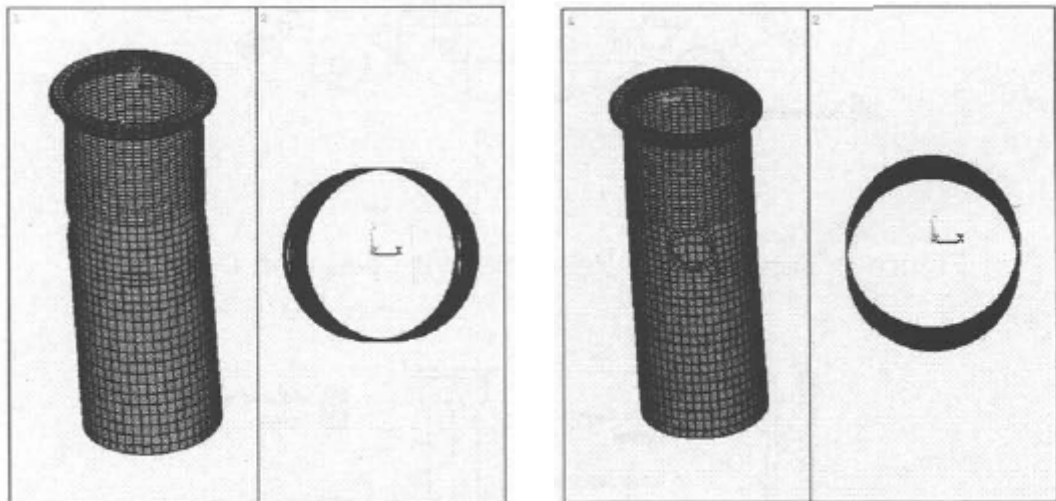


Figure 22: Core barrel beam mode movement (Source: Park *et al.*, 2003:181)

- Vessel rocking mode

The vessel rocking mode, also known as the vessel swing mode, is similar in movement to that of the core barrel. The RPV is a rigid body and movement in any direction is minimal. Movements are measured in three directions and are mentioned below:

- o Movements in the 0° - 180° plane,
- o Movements in the 90° - 270° plane and
- o Movements in the vertical direction.

- Fuel assembly beam mode

The vibrations active in the fuel assemblies are similar in motion to those of the core barrel beam mode. The water coolant entering the core barrel from the bottom of the RPV flows pass the fuel assemblies causing the assemblies to vibrate. The fundamental operating frequency ($n = 1$) is measured at the KNPS together with the higher nodes ($n = 2, 3$).

The natural frequency of the above mentioned fuel assembly beam modes tend to decrease through the fuel cycle due to the change in stiffness which is caused by i.a. high levels of radiation within the core.

- Core barrel shell mode

This movement refers to the vibration of the core barrel during normal operation due to the water coolant entering the RPV. This mode is not as clear as the core barrel beam mode but is still relatively easy to identify in the frequency spectrum. Park *et al.* (2003:181) illustrated the core barrel shell mode of a standard Korean PWR using a FEM (Figure 23).

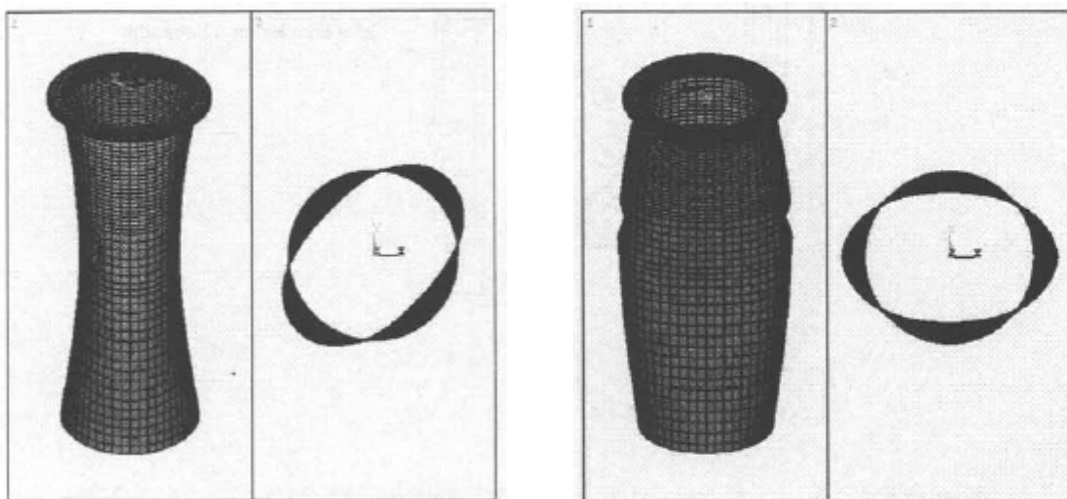


Figure 23: Core barrel shell mode of a standard Korean PWR (Source: Park *et al.*, 2003:181)

- MCP shaft rotation speed

The MCP at the KNPS is designed to operate at a constant speed of approximately 1500 rpm. This resonance is always visible on the frequency spectrum at approximately 24.75 Hz. The 1st sub harmonic, that is half the operating frequency at approximately 12.5 Hz, is visible in the signatures obtained from the KNPS.

- Fluid resonance

Fluid resonances in pipes are a common occurrence and are caused by a pump forcing fluid through the pipe. Wide-band flow-induced excitation occurs and when pressure waves are reflected between the ends of the pipe, a standing pressure wave emerges. The pressure waves travel at the speed of sound in the medium. Knowing the speed of sound or pressure waves in the medium as well as the lengths of pipes between sections, allows a calculation of the expected frequency peaks associated with standing pressure waves.

- MCP / Steam generator

The frequency resonance observed in the MCP and the steam generator is caused by the stiffness present within the pipe connecting the two primary components. There is still uncertainty regarding the interpretation of this resonance and further measurements will have to be completed to confirm.

- MCP water whirl effect

Water whirl (similar to oil whirl) occurs in thin-film lubricated bearings and is associated with unstable conditions. The frequency is usually just below one half of the operating frequency of the MCP.

- Control rod drive mechanism vibrations

The control rod drive mechanism can be seen as a large structure and research is still necessary for confirmation. However this vibration does not fall

under the scope of this study. There is still uncertainty regarding the interpretation of this resonance and further measurements will have to be completed to confirm.

- MCP / vessel vibration

This vibration is similar to that of the MCP / steam generator. These two primary components are connected by the primary coolant pipe. Further measurements will have to be completed to confirm.

The first vibration analysis data set was obtained on request by the KNPS. The vibration data was obtained using the instrumentation described in section 5.2.1. For the purpose of this study, only signals from the 7 accelerometers and the 8 neutron detectors were used. The long term trends were investigated and compared to detect possible mechanical degradation in the components of the primary system. The sensors in the RVMS were also investigated to determine whether signal degradation occurred over time.

Figure 24 and Figure 25 display the frequency signatures of the RPN 10 MA upper neutron detector for units 1 and 2 measured on the 28th of January 2000 and the 19th October 1999 respectively. In the spectrum in Figure 24 the fuel assembly beam mode ($n = 2$) at 6 Hz (light blue line), the core barrel beam mode at approximately 8.1 Hz (red line), the core barrel shell mode at roughly 19.9 Hz (yellow line) and the primary pump operating frequency at 24.75 Hz (dark blue line) indicate resonances and refer to the operating frequencies of the different components in unit 1. The spectrum in Figure 25 displays resonances which represent the fundamental fuel assembly beam mode at 3 Hz (red dotted line) and the fuel assembly beam mode second (purple dotted line) and third (light blue dotted line) modes at 6.25 Hz and 10 Hz respectively. The core barrel beam mode at 8.1 Hz (dark blue dotted line), core barrel shell mode (green dotted line) at 19.9 Hz and the rotating MCP at 24.8 Hz (light yellow line) are also visible.

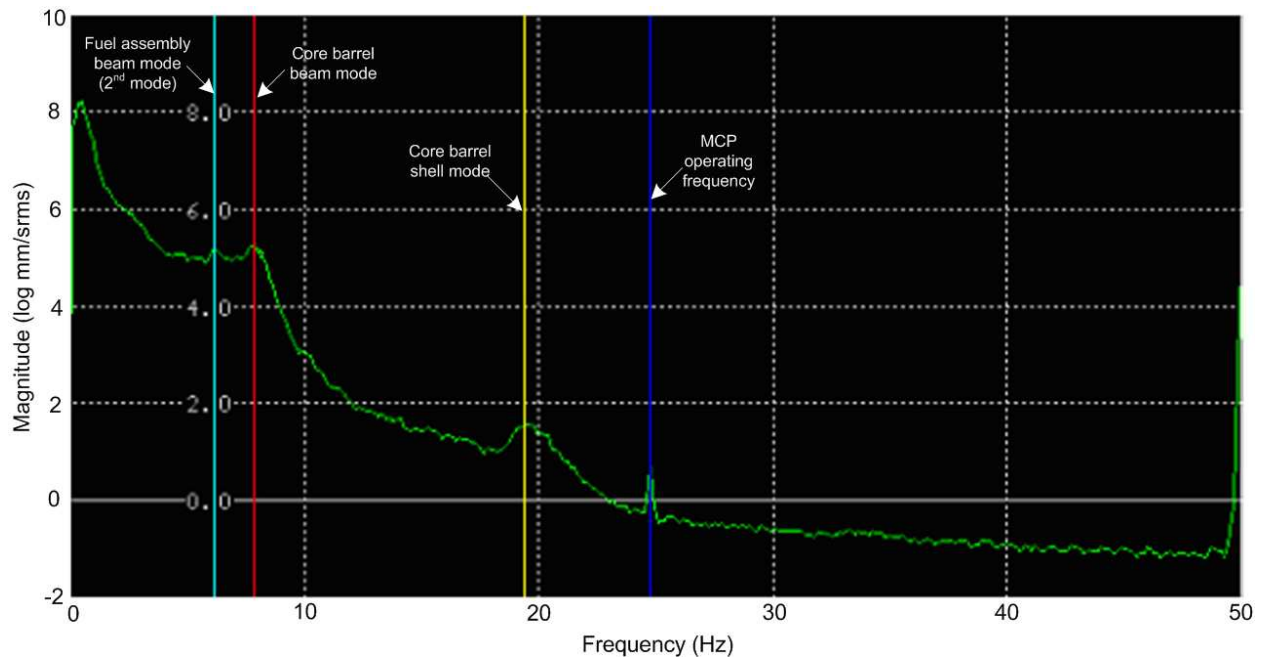


Figure 24: RPN 10 MA upper neutron signature for unit 1 measured on the 28 January 2000 (Source: van Niekerk, 2000:15)

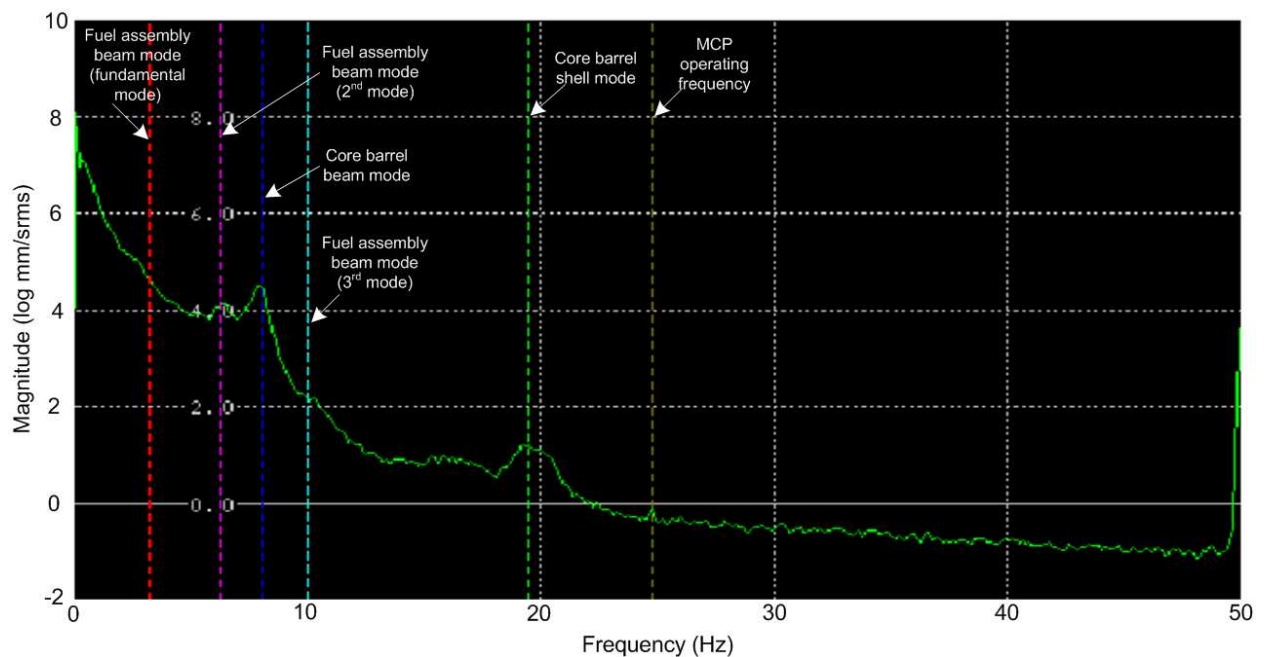


Figure 25: RPN 10 MA upper neutron signature for unit 2 measured on the 19 October 1999 (Source: van Niekerk, 2000:16)

In Figure 26 the frequency spectrum of the KIR 001 MV accelerometer of unit 1 is displayed. This measurement was taken on the 28th of January 2000. Again this spectrum provides the core barrel beam mode, core barrel shell mode and the primary pump operating frequency. The resonances of the vessel rocking mode (0°-180°) (red dotted line), vessel rocking mode (90°-270°) (Light blue dotted line) and vessel rocking

mode (Vertical) (Green dotted line) are identifiable at 16.9 Hz, 18 Hz and 18.7 Hz respectively.

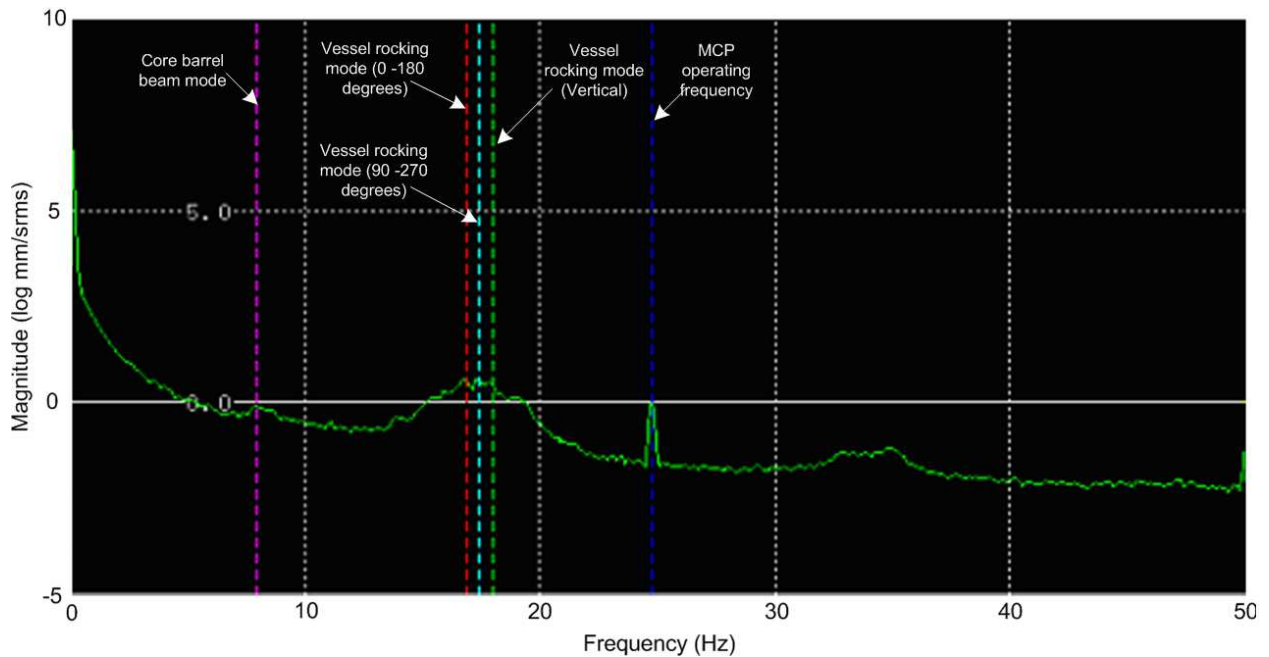


Figure 26: KIR 001 MV accelerometer signature for unit 1 measured on the 28 January 2000 (Source: van Niekerk, 2000:15)

Figure 27 displays a signal similar to that of Figure 26. This signal was measured on the RPV of unit 2 of the KNPS on the 19th October 1999. This spectrum displays resonances of the 2nd fuel assembly beam mode at 6.25 Hz (red dotted line) and core barrel beam mode at 8.1 Hz (yellow dotted line). The vessel rocking mode in the 0 to 180 degrees (purple dotted line), 90 to 270 degrees (dark blue dotted line) and vertical (light blue dotted line) direction can be seen at 17.5 Hz, 18.1 Hz and 18.70 Hz respectively. Although not indicated on the frequency spectrum by a vertical line, the peak associated with the MCP shaft rotation is found at 24.8 Hz.

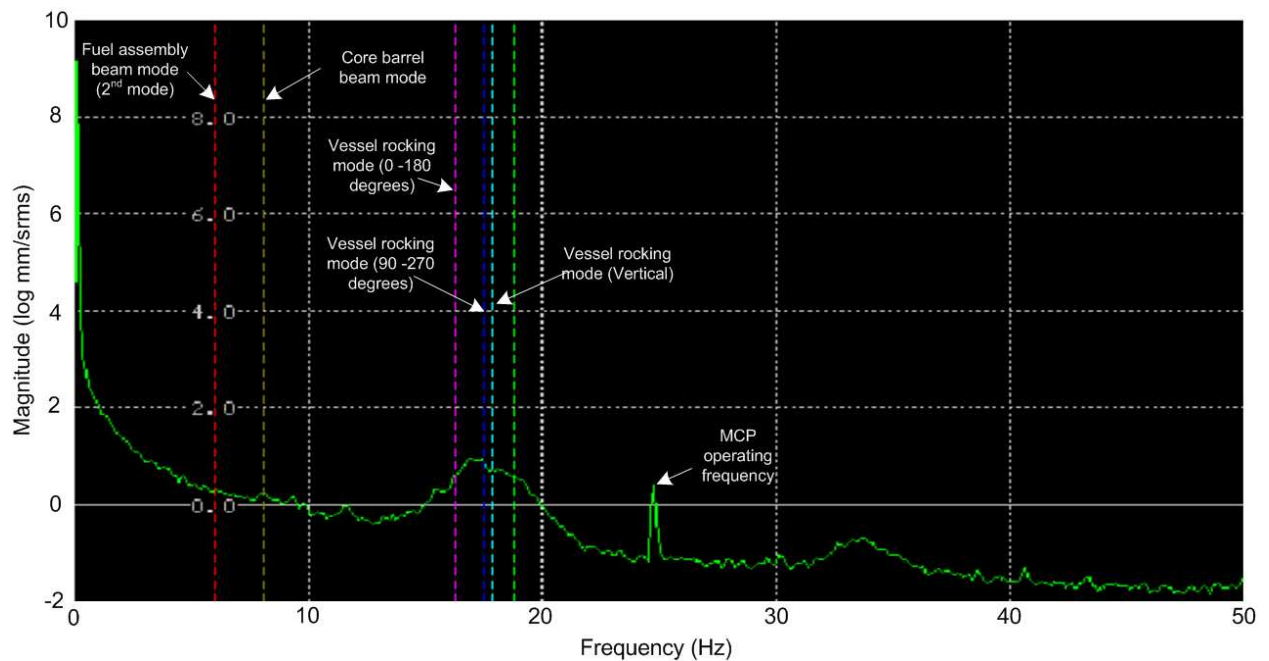


Figure 27: KIR 001 MV accelerometer signature for unit 2 measured on the 19 October 1999 (Source: van Niekerk, 2000:16)

No resonance shift in the known / interpreted resonances was found in the spectra from the two neutron detector and two accelerometer signals discussed above. A downward shift of a resonance frequency would be due to a change in stiffness, which in turn could be attributable to mechanical failure. The identified resonances for unit 1 and 2 were calculated on the 11th February 2000 and the 15th December 1999 respectively and will be used as reference for new data. The reference frequencies for unit 1 and 2 are indicated in tables 1 and 2 respectively.

Table 1: Operating frequencies for unit 1 as measured on the 11th February 2000

Frequency (Hz)	Interpretation of Resonance
2.6	Fuel Assembly Beam Mode (Fundamental mode)
5.0	MCP/Steam Generator (?)
5.6	Fluid Resonance
6.3	Fuel Assembly Beam Mode (n = 2)
8.0	Core Barrel Beam Mode
10.3	Fuel Assembly Beam Mode (n = 3)
11-12	MCP Water Whirl Effect
12.5	MCP Shaft Vibration Sub-harmonic
15.4	Vibration of Control Rod Drive Mechanism (?)

16.9	Vessel Rocking Mode (0-180)
18.0	Vessel Rocking Mode (90-270)
18.7	Vessel Rocking Mode (vertical)
19.9	Core Barrel Shell Mode
24.8	MCP Shaft Rotation
32.6	Fluid Resonance (?)
35.0	MCP/Vessel Vibration (?)

Table 2: Operating frequencies for unit 2 as measured on the 15th December 1999

Frequency (Hz)	Interpretation of Resonance
3.00	Fuel Assembly Beam Mode (Fundamental mode)
5.00	MCP/Steam Generator
5.60	Fluid Resonance
6.25	Fuel Assembly Beam Mode (n = 2)
8.10	Core Barrel Beam Mode
10.00	Fuel Assembly Beam Mode (n = 3)
11.70	MCP Water Whirl Effect
12.50	MCP Shaft Vibration Subharmonic
15.80	Vibration of Control Rod Drive Mechanism
17.50	Vessel Rocking Mode (0-180)
18.10	Vessel Rocking Mode (90-270)
18.70	Vessel Rocking Mode (vertical)
19.90	Core Barrel Shell Mode
24.80	MCP Shaft Rotation
33.00	Fluid Resonance
35.50	MCP/Vessel Vibration

The interpretation of the resonance frequencies accompanied by a (?) still needs confirmation.

6.2. Signal validation

Signal integrity plays an important role in the ability of the vibration monitoring system to produce accurate readings. However signal decay is a realistic problem in especially the primary system due to the high levels of radiation. Radiation within the primary system causes the sensor's material properties to change which directly affect the accuracy of the measurements.

Figure 28 illustrates a signal from the KIR 001 MA accelerometer following progressed decay. The measurement was taken on the 28th January 2000. A decaying signal is often characterized by a broad-band increase of the noise floor of the signal leading to a disappearance of resonances / peaks in the frequency spectra, as depicted in Figure 28.

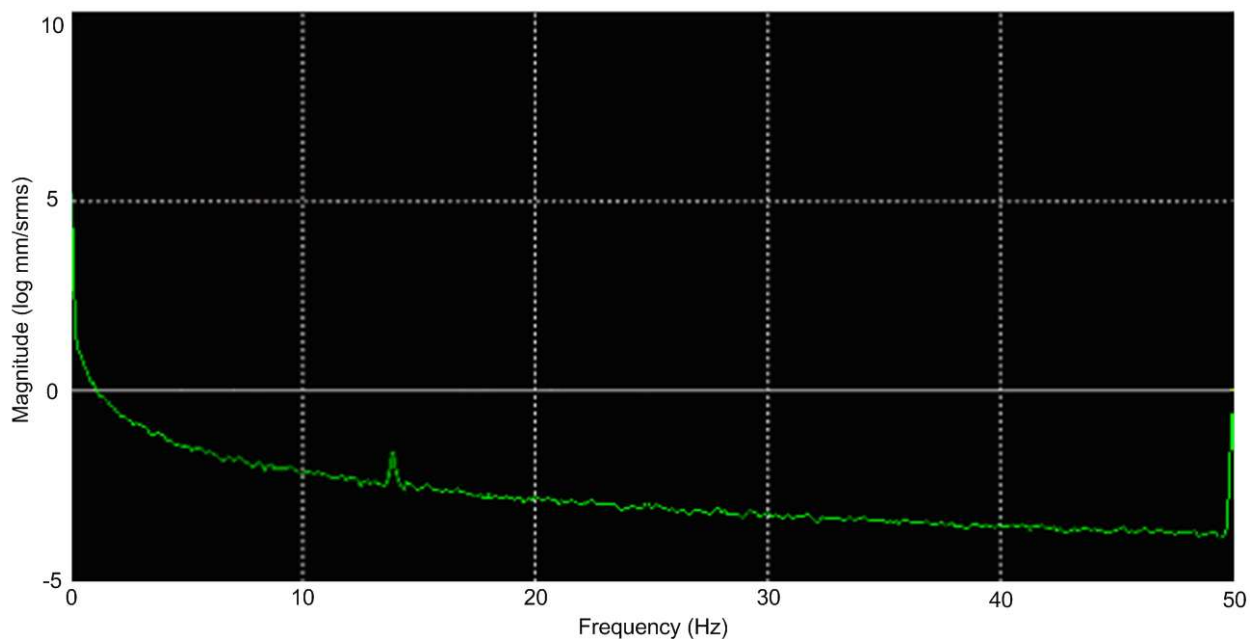


Figure 28: Progressed decay of an accelerometer signal (Source: van Niekerk, 2000:17)

6.3. Measurements from 2006 - 2010

The second part contains vibration data from the RVMS system, from July 2006 to May 2010. This data contains 884 frequency spectra for unit 1 and 890 for unit 2. The data was analysed and the different frequency spectra were compared. Figure 29 displays two signals from the RPN 10 MA upper neutron detector for unit 1 of the KNPS. The

first measurement indicated in blue illustrates the frequency spectrum as measured on the 31 August 2006 while the second measurement in red displays the frequency spectrum on the 26 April 2010. Due to excessive noise in the signals, vibration signals from the first of July 2006 to the end of August 2006 could not be used for analyses. The data measured on the 26 April 2010 was preferred because this is the last frequency spectrum available. All the resonances and shaft rotation peak specified in section 6.1 are clearly visible on the spectra. The peak at 50 Hz is due to electromagnetic influence from the grid frequency. As can be seen in Figure 29 the resonances measured on the 31 August 2006 are similar to the resonances measured on the 26 April 2010.

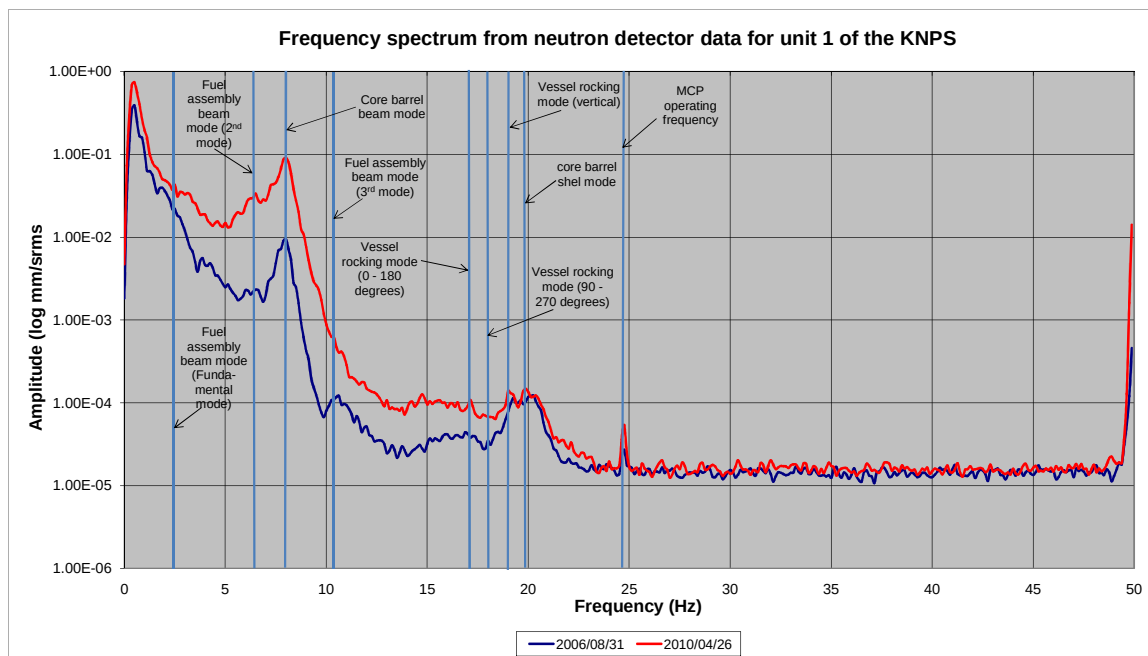


Figure 29: Frequency spectrum from neutron detector data for unit 1 of the KNPS
(Source: Figure by the author)

Figure 30 displays the frequency spectrum calculated from the KIR 001 MV accelerometer signal. Resonances on the accelerometer spectra are not as clear as on the neutron detector signals due to characteristics discussed in section 2.3. The first measurement was recorded on the 31 August 2006 (blue line) and the second (red line) on the 26 April 2010. The MCP 2nd harmonic is clearly visible at roughly 48.75 Hz for the measurement recorded on the 26 April 2010 (red line) but not for the measurements recorded on the 31 August 2010 (blue line).

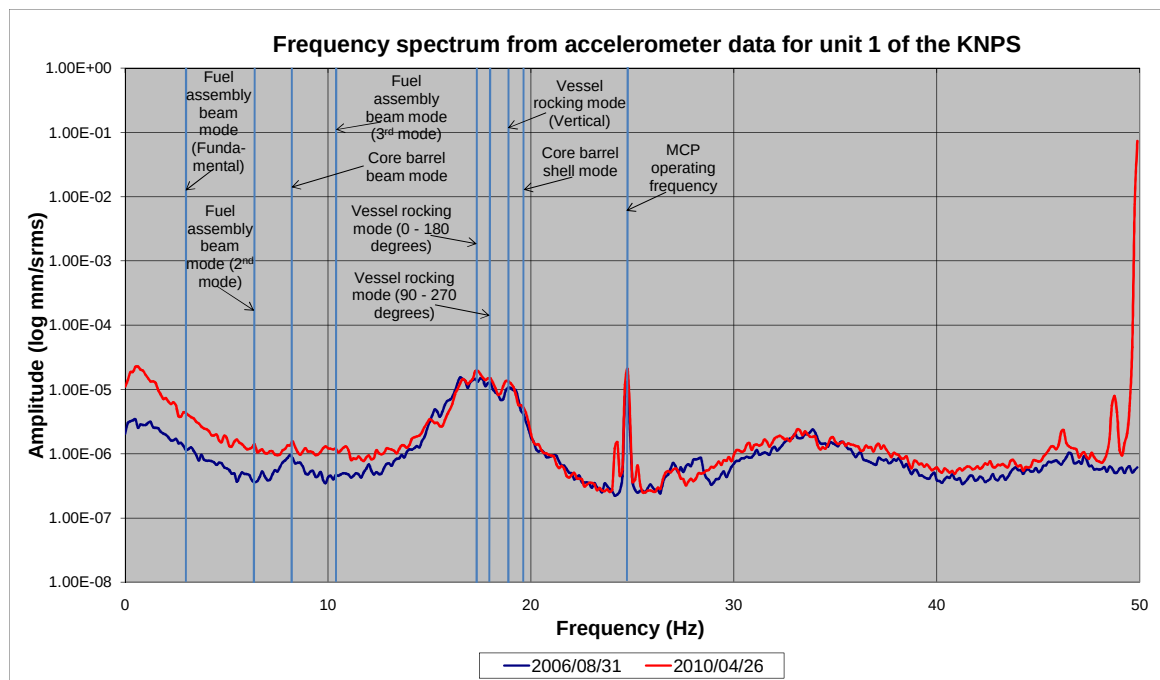


Figure 30: Frequency spectrum from accelerometer data for unit 1 of the KNPS
(Source: Figure by the author)

Figure 31 illustrates vibration data from the RPN 10 MA upper neutron detector of the KNPS unit 2. The measurements were recorded on the 31 August 2006 and on the 12 May 2010 respectively. As would be expected, the frequency spectra are similar to that of unit 1. The core barrel beam mode at approximately 8.1 Hz and the core barrel shell mode at 20 Hz are clearly visible. Further investigation also reveals the fundamental fuel assembly beam mode, 2nd and 3rd modes at roughly 3 Hz, 6 Hz and 10 Hz respectively.

The frequency spectrum also reveals the MCP shaft rotation frequency at 24.75 Hz and the three directions of vessel rocking mode at approximately 17 Hz, 17.5 Hz and 19 Hz.

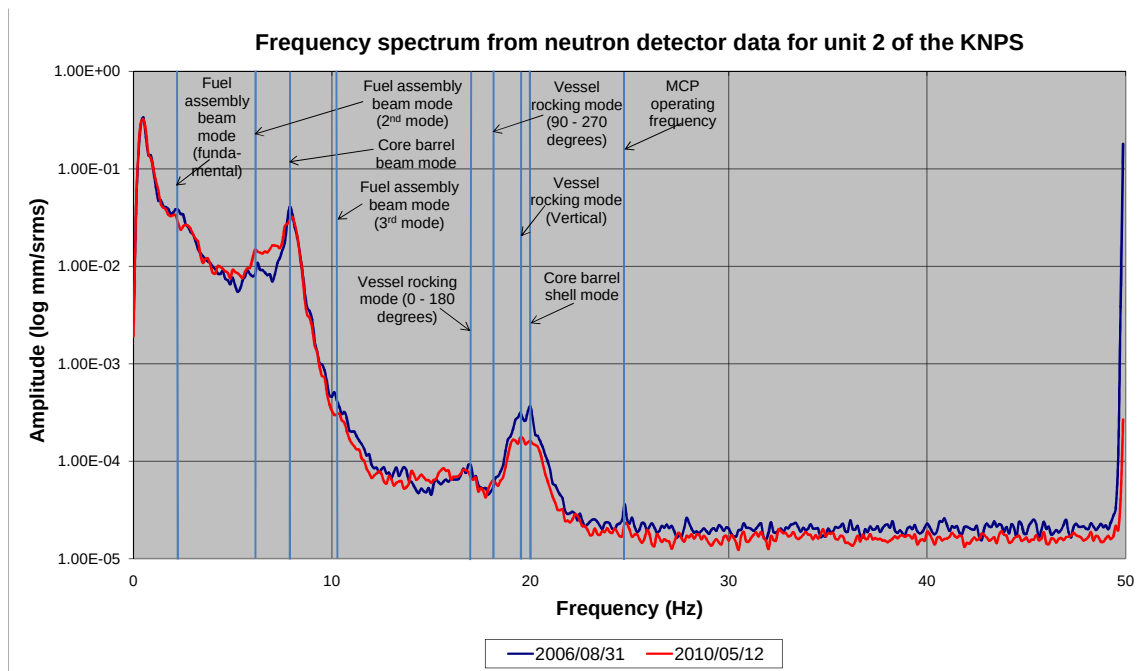


Figure 31: Frequency spectrum from neutron detector data for unit 2 of the KNPS
(Source: Figure by the author)

Figure 32 displays the KIR 001 MV accelerometers frequency spectra for unit 2 of the KNPS. Again this signal is similar in resonances to that of unit 1. The measurement recorded on the 31 August 2006 (blue line) is similar to the measurement recorded on the 12 May 2010 (red line).

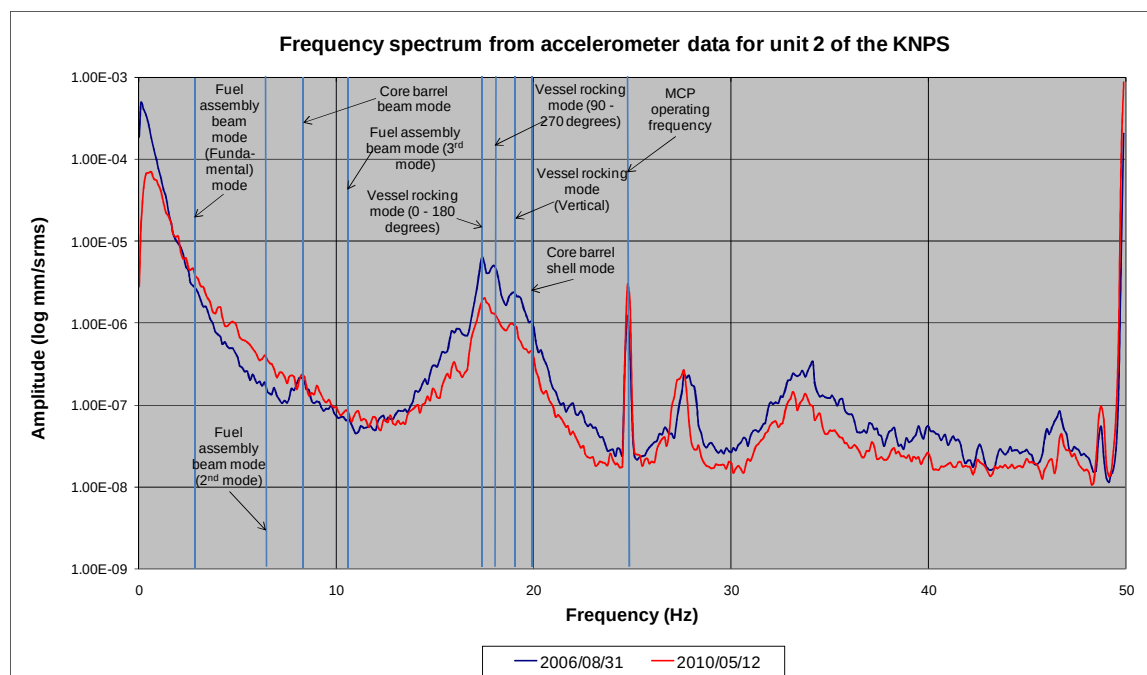


Figure 32: Frequency spectrum from accelerometer data for unit 2 of the KNPS
(Source: Figure by the author)

As discussed in section 6.2, signal integrity is of vital importance in the vibration monitoring process. The quality of the signals for unit 1 and 2 were investigated and are presented in tables 3 and 4.

Table 3: Sensor integrity of unit 1 as measured on the 26 April 2010

SIGNAL	SIGNAL QUALITY		
	Signal Good	Early Decay	Progressed decay
Neutron Detectors			
1RPN10MA U	x		
1RPN10MA L	x		
1RPN20MA U	x		
1RPN20MA L	x		
1RPN30MA U	x		
1RPN30MA L	x		
1RPN40MA U	x		
1RPN40MA L	x		
Accelerometers			
1KIR001MV	x		
1KIR002MV		x	
1KIR003MV			x
1KIR004MV		x	
1KIR005MV	x		
1KIR006MV	x		
1KIR007MV		x	

Table 4: Sensor integrity of unit 2 as measured on the 12 May 2010

SIGNAL	SIGNAL QUALITY		
	Signal Good	Early Decay	Progressed decay
Neutron Detectors			
1RPN10MA U	x		
1RPN10MA L	x		
1RPN20MA U	x		
1RPN20MA L	x		

1RPN30MA U	x		
1RPN30MA L	x		
1RPN40MA U		x	
1RPN40MA L	x		
Accelerometers			
1KIR001MV		x	
1KIR002MV	x		
1KIR003MV		x	
1KIR004MV			x
1KIR005MV		x	
1KIR006MV	x		
1KIR007MV			x

In tables 3 and 4 signals are grouped into three categories based on signal quality. The majority of the signals were found to be in good working order. Three sensors need replacement due to progressive decay in the signal whereas seven signals were showing signs of early decay. Signal degradation is more likely due to long cables between the accelerometer and the charge converter. Connector failure and the degradation of sensors due to the constant high rates of radiation within the primary system also contribute to signal degradation.

According to van Niekerk (2000:7) sensor degradation could be caused by i.a.:

1. In the containment area
 - Sensor failure
 - Signal conditioning failure
 - Signal cable/connector problems
 - Pre-amplifier failure
2. In the KIR room
 - Amplifier failure
 - Component failure in the AMS system which includes
 - o AMS power supply
 - o Channel filter cards

- o Channel ADC cards
- o Computer software
- o Cable and connectors

6.4. Long term trends

In this section the frequency spectra will be analysed and long term vibration trends will be identified. Vibration data from 1985 to 2000 were divided into one measurement per year while the new vibration data from 2006 to 2010 were divided into 4-monthly intervals. However this was not always possible as sometimes the unit outages coincided with the 4 months pattern. In these cases a frequency spectrum was either chosen on the first date available prior to the outage or after the outage. These intervals were found to be sufficient as mechanical degradation in component gradually happens over time as explained in section 2.5. All accelerometer and neutron detector frequency spectra were analysed and from these spectra the resonances were identified. The resonances identified for unit 1 and 2 are tabulated in tables 5 and 6 respectively.

As shown in tables 5 and 6, constant resonance frequencies were identified for the different components from 1985 to 2010. However small deviations were not sufficiently indicative to mechanical degradation in the primary component.

In Figure 34 and Figure 35 a summary of the resonances of the different components for unit 1 and 2 are illustrated on a time line graph. The spectral deviations found for the time period 2006 to 2010 are found to be similar to deviations found in the spectra for the time period 1985 to 2000. Due to the complexity of the KNPS conclusions regarding mechanical degradation should be made carefully.

The core barrel beam mode at 8.1 Hz is e.g. well known and documented and it has been found that e.g. a frequency decrease of less than 0.5 Hz indicated degradation of the hold down springs in the reactor (Sunder, 1992:6). Using the Biblis-PWR Unit A as an example, a reference signature was obtained in May 1979. In September the following year another signature was measured and it was found that the hold down springs of the core barrel had relaxed and this resulted in a resonance change of

approximately 0.3 Hz. This was the case for two later measurements as well. This is illustrated in Figure 33.

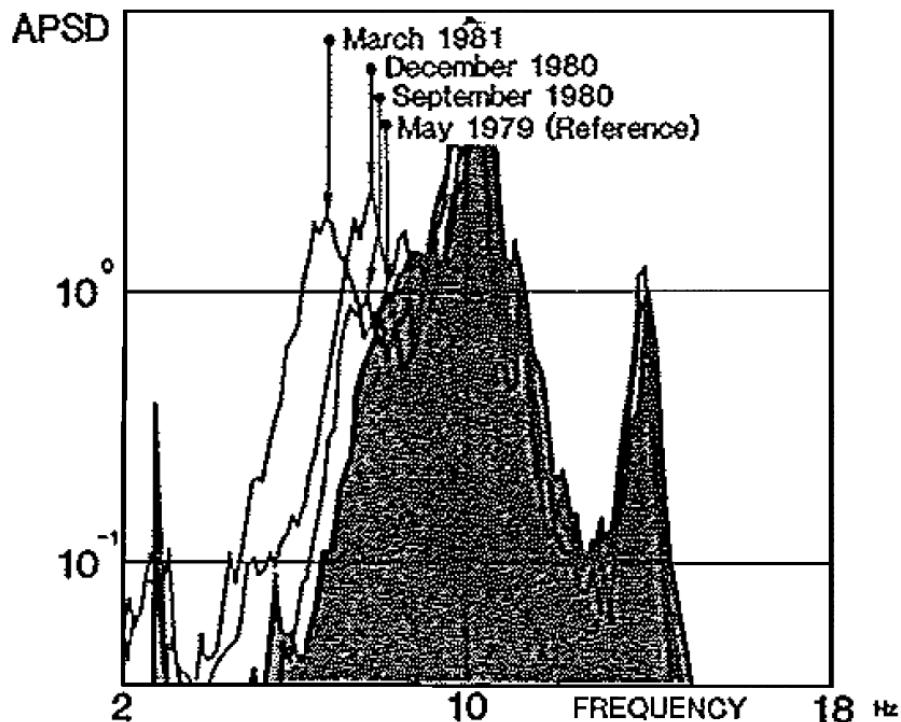


Figure 33: Relaxation of core barrel hold down springs (Source: Sunder, 1992:6)

Similarly, a broad band increase of the core barrel shell mode peak has been demonstrated to be indicative of increased baffle jetting.

The decrease of the resonances associated with the fuel elements are due to the irradiation of the fuel assemblies; during the fuel cycle the springs in the fuel assemblies loose stiffness due to the irradiation. These deviations are also illustrated in Figure 34 and Figure 35 and are considered normal. Until the unidentified peaks in the spectra are not clearly related to a particular mechanical resonance and hence component, the deviations observed in the long term trends cannot be related to potential mechanical degradation. The frequency spectra of the vessel rocking modes are not clearly distinguishable due to overlap of the peaks. Therefore the deviations of these peaks are not necessarily indicative of degradation.

Hence no unexpected deviations were recorded which can be clearly related to mechanical degradation of the components.

Table 5: Signal interpretation of unit 1

	Fuel Assembly Beam Mode (Fundamental mode)	MCP/Steam Generator	Fluid Resonance	Fuel Assembly Beam Mode (n = 2)	Core Barrel Beam Mode	Fuel Assembly Beam Mode (n = 3)	MCP Water Whirl Effect	MCP Shaft Vibration Subharmonic	Vibration of Control Rod Drive Mechanism	Vessel Rocking Mode (0-180)	Vessel Rocking Mode (90-270)	Vessel Rocking Mode (vertical)	Core Barrel Shell Mode	MCP Shaft Rotation	Fluid Resonance	MCP/Vessel Vibration
Measured on:	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz
16-Sep-85	3.51	4.93	5.48	6.41	8.11	10.6	11.5	12.5	15.6	16.9	17.8	18.6	20	24.8	32.2	35.3
30-Oct-86	3.01	4.93	5.75	6.11	8.05	10.4	11.5	12.4	15.5	17	18	18.4	20.2	24.8	32.5	34.8
29-Apr-87	2.44	4.93	5.53	6.3	8.08	10	11.5	12.5	15.5	17	18	18.4	20.3	24.8	32.5	34.9
25-Jul-88	2.93	4.93	5.48	6.3	8.08	10.4	11.4	12.4	15.5	16.9	17.9	18.4	20.3	24.8	32.3	34.8
15-Jun-89	2.9	4.85	5.53	5.95	8.08	10	11.5	12.4	15.1	17.2	18.1	18.7	20.1	24.8	32.3	35.3
07-Feb-90	2.19	4.79	5.59	6.38	8.08	10.6	11.9	12.4		17.5	18.1	18.4	19.7	24.8	33.3	34.1
16-Oct-91	2.71			6.3	8.08	10.3		12.5		17	17.8	18.2	20	24.8	32.9	34.9
14-Oct-92	2.62		5.74		8.23					18	18.7	19.5	19.5	24.8	33.4	34.2
18-Jul-93	2.74			6.73	8.23								19.7	24.8		35
30-Jun-94	2.99			6.48	8.23								20.5	24.8		
30-Jun-95	2.74			6.9	8.23								20	24.8		34.7
09-Nov-98	2.75			6.38	8					17.4	17.8	18.3	19.8	24.8		35.4
02-Nov-00	2.5			6.63	8	10				17.1	17.5	18.5	19.8	24.8		35
Vibration Data from December 2000 to August 2006 was not available to be analysed																
04-Sep-06	3.13			6.38	7.88	10.3				17.1	17.9	18.6	20.1	24.8		
30-Nov-06	3			6.38	7.88	10.4				17.1	17.8	18.9	20	24.8		
23-Mar-07	3.13			6.38	8	10.3				17.1	17.8	18.9	20.1	24.8		
24-Jul-07	3.13			6.38	8	10.3				17.3	17.9	19.4	20.1	24.8		
23-Nov-07	3			6.38	8	10.1				17	17.6	19.3	20.4	24.8		
23-Mar-08	3.13			6.25	8	10.1				17.1	17.8	19.1	20.1	24.8		
24-Jul-08	3			6.38	8	10.4				17.3	17.9	19	20.1	24.8		
23-Nov-08	3.13			6.38	7.88	10.4				17.1	17.9	19	20.1	24.8		
19-Apr-09	3.5			6.38	8	10.3				17.3	17.8	18.8	20	24.8		
24-Jul-09	3			6.38	8	10.3				17.3	17.8	18.9	19.9	24.8		
24-Nov-09	3			6.25	8	9.75				17.1	17.6	19	19.9	24.8		
18-Mar-10	2.75			6.38	8	9.88				17.1	17.6	19.1	19.9	24.8		
26-Apr-10	3.13			6.38	8	9.88				17.1	17.9	18.9	19.8	24.8		

Table 6: Signal interpretation of unit 2

	Fuel Assembly Beam Mode (Fundamental mode)	MCP/Steam Generator	Fluid Resonance	Fuel Assembly Beam Mode (n = 2)	Core Barrel Beam Mode	Fuel Assembly Beam Mode (n = 3)	MCP Water Whirl Effect	MCP Shaft Vibration Subharmonic	Vibration of Control Rod Drive Mechanism	Vessel Rocking Mode (0-180)	Vessel Rocking Mode (90-270)	Vessel Rocking Mode (vertical)	Core Barrel Shell Mode	MCP Shaft Rotation	Fluid Resonance	MCP/Vessel Vibration
Measured on:	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz	Hz
09-Dec-85	3.3	5.1	5.6	6	7.92	0	11.2	12.4	15.2	17	18.2	18.6	20.2	24.8	32.5	35.5
19-May-86	3.4	5	6	6	7.95	10.1	11.4	12.2	15.6	17	18.2	18.4	20.2	24.8	32.6	35.5
15-Jun-89	2.9	4.7	5.3	6.1	8.09	9.5	11.5	12.4	15.1	17	17.9	18.5	20	24.8	32.5	35.4
23-Nov-90	2.9	5.1	5.4	6.1	8	10.1	11.8	12.7	16	17.8	18.4	18.8	20.6	24.8	33.2	35.8
09-Dec-91	2.8	5.1		6.2	8.14	10.1	12.3	12.6	16.3	17.7	18.4	19.1	20.1	24.8		35.9
03-Nov-92	2.9		5.7	6.5	8.11				16.2	17.7	18.2	18.9	19.7	24.8	33.9	34.7
20-Nov-93	3			6.3	8.17								19.7	24.8		
03-Nov-94	3.2			6.7	8.23					17.2	17.7	18.7	20	24.8		
15-Jan-95	2.8			6.3	8.1					17.7	18	18.5	19.6	24.8	33	35.5
05-Jun-95	2.8			6.3	8.1					17.7	18	18.5	19.6	24.8	33	35.5
20-Nov-97	3		5.8	6.3	8.05	10.2	12		16.3	17.5	17.8	18.8	19.5	24.8	33	35.5
15-Dec-99	2.5			6.5	8	10.4				17	17.8	19.3	20.1	24.8		
Vibration Data from January 2000 to July 2006 was not available to be analysed																
11-Aug-06	2.88			6.5	8	10.1				17	17.8	19.1	19.8	24.8		
10-Nov-06	3			6.38	8.13	10.4				17	17.9	19.3	19.8	24.8		
10-Mar-07	2.88			6.38	8.13	10.3				17.3	18	19.4	20.1	24.8		
11-Jul-07	2.75			6.38	8.13	10.1				17.1	17.9	19.4	20.4	24.8		
11-Nov-07	3			6.38	8	10.5				17	17.8	19.3	20	24.8		
10-Apr-08	2.5			6.38	8	10.3				17.1	18	19.1	19.8	24.8		
10-Jul-08	2.63			6.38	8	10.4				17.1	17.5	19.3	20.4	24.8		
11-Nov-08	2.88			6.38	8	10.3				17.4	18.4	18.9	19.8	24.8		
11-Mar-09	2.88			6.25	8	10.4				17.1	17.5	18.5	19.4	24.8		
11-Jul-09	2.75			6.25	8.13	10.4				17.4	17.8	19.3	19.6	24.8		
14-Nov-09	2.63			6.5	8	10.4				17.1	18	18.6	19.5	24.8		
11-Mar-10	2.5			6.38	8	10.4				17.3	17.5	19.3	20	24.8		
12-May-10	2.63			6.38	8	10.3				17	17.5	19.1	20.3	24.8		

Summary of the long-term trend analysis of unit 1 of the KNPS

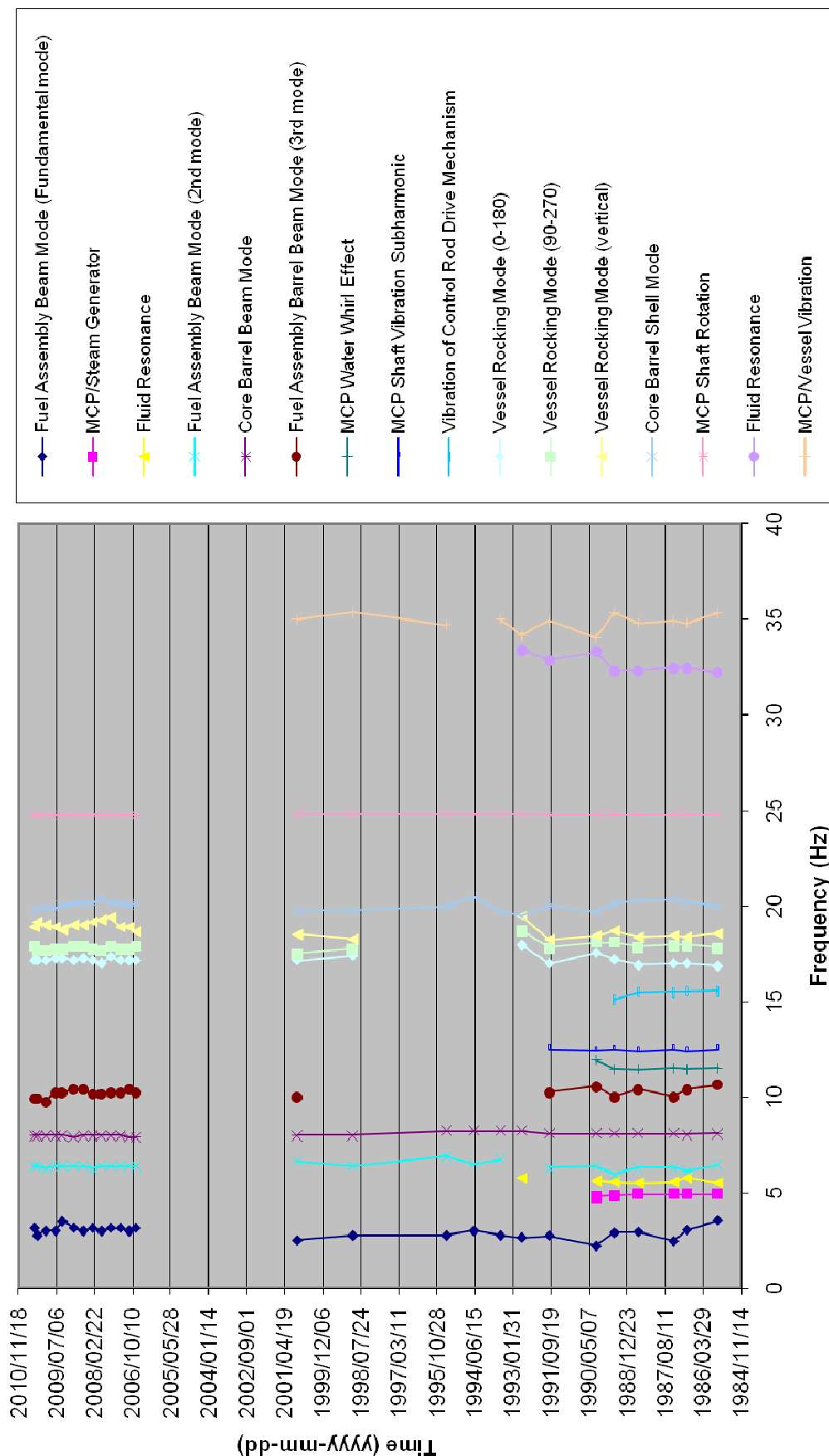


Figure 34: Long-term trend analysis for unit 1 of the KNPS (Source: Figure by the author)

Summary of the long-term trend analysis of unit 2 of the KNPS

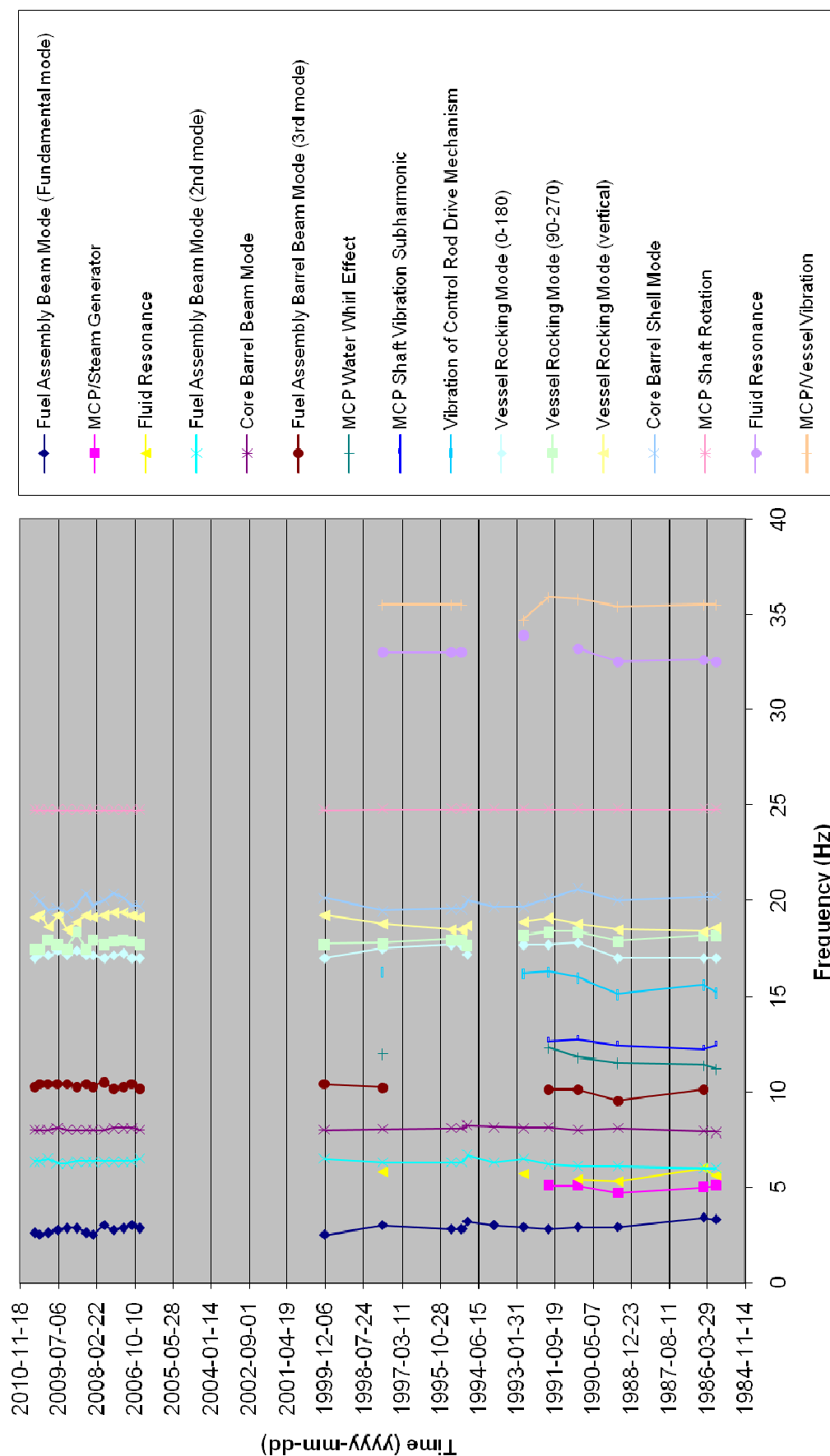


Figure 35: Long-term trend analysis for unit 2 of the KNPS (Source: Figure by the author)

7. SIMULATIONS

In this chapter the primary system of the KNPS will be modelled using a simplified elasto-dynamic model with simplified mass, stiffness and damping values of a number of components. This model, incorporating the mass, damping and stiffness matrices, will be used in the elasto dynamic equations to generate displacement time series.

The above mentioned equations will be solved numerically from first principles. Programming will be done in the software package MathCADTM 14. The simulated excitation forces act on a mass, spring and damping system creating displacement fluctuation within the system. In this model only the masses of the primary components and their operating frequencies were available and all other values were obtained by approximation.

The model is completed by calculating FFTs and APSDs on the time domain data and obtaining the resonances of all the primary components. The values of the stiffness components are obtained using a trail and error method due to the availability of the operating resonances of all the components. Mechanical degradation within the primary system will be discussed and simulated. Finally sensor degradation caused by for example radiation damage etc. will be simulated.

7.1. General overview

In reality mechanical systems are more complex than the simple single degree of freedom system shown in Figure 2 and need to be approximated by many degrees of freedom. Therefore it is necessary to formulate a multi degree of freedom general elasto-dynamic equation. For each degree of freedom the elasto-dynamic equation 2.3 has to be solved. An elegant and powerful method to solve a set of elasto-dynamic equations is by writing these equations in matrix form.

Assume a system with n degrees of freedom. The degree of freedom determines the size of the matrices. Therefore matrices that will be used to formulate the equations will contain n (rows) $\times$ n (columns) elements. The mass matrix used in the equation is given below:

$$\underline{\underline{M}} = \begin{pmatrix} M_{11} & 0 & 0 & \dots & 0 \\ 0 & M_{22} & 0 & \dots & 0 \\ 0 & 0 & \ddots & 0 & 0 \\ \vdots & \vdots & 0 & \ddots & 0 \\ 0 & 0 & 0 & 0 & M_{nn} \end{pmatrix}$$

The mass matrix only contains input on the diagonal while the rest of the matrix contains zeros.

Setting up the matrices for the stiffness matrix $\underline{\underline{K}}$ and damping matrix $\underline{\underline{C}}$ is much more challenging. Looking at each mass in the system individually, the sum of each damping and stiffness element connected to that mass must be entered into the corresponding space in the damping and stiffness matrices respectively. In other words the diagonal of the stiffness matrix will be filled with the sum of the springs acting on the relevant mass. The same rule applies to the damping matrix.

Following the previous step; the springs (representing stiffness) connecting any two masses should be identified. In order to place these negative values in the correct position in the matrix the masses are numbered from 1 to n as can be seen in the mass matrix. Assume spring k is connected to two masses 1 and 5 respectively. The correct position for k in the stiffness matrix is at the position in column 1, row 5. However these matrices are always symmetric and therefore the value will also have to be inserted at column 5 and row 1. Finally once all the values have been inserted and there are still open terms in the matrix those terms are equal to zero. In a similar manner the matrix for $\underline{\underline{C}}$ can be obtained.

The force vector has dimension n. The force acting on a mass corresponds to that row in the force vector. For example, if a force acts on mass five then the position of the force in the force vector will be that of row 5. The force vector is shown below:

$$\underline{F}(t) = \begin{pmatrix} F_1(t) \\ F_2(t) \\ \vdots \\ F_n(t) \end{pmatrix}$$

The general elasto-dynamic equation for a multi degree of freedom system can be formulated as:

$$\underline{\underline{M}}\ddot{\underline{x}}(t) + \underline{\underline{C}}\dot{\underline{x}}(t) + \underline{\underline{K}}\underline{x}(t) = \underline{F}(t) \quad (7.1)$$

where

$\underline{\underline{M}}$, $\underline{\underline{C}}$, $\underline{\underline{K}}$ are the matrices as specified above,

$\ddot{\underline{x}}$ is the acceleration vector,

$\dot{\underline{x}}$ is the velocity vector,

$\underline{x}$ is the displacement vector and

$\underline{F}(t)$ denotes the force vector.

According to Wilson *et al.* (1973:241) the acceleration vector $\ddot{\underline{x}}$ and velocity vector $\dot{\underline{x}}$ can be defined as follows:

$$\ddot{\underline{x}} = \frac{\underline{x}_1 - 2\underline{x}_0 + \underline{x}_{-1}}{(\Delta t)^2} \quad (7.2)$$

and

$$\dot{\underline{x}} = \frac{\underline{x}_1 - \underline{x}_{-1}}{2\Delta t} \quad (7.3)$$

where

Δt is the time step and

$\underline{x}_{-1}$, $\underline{x}_0$, $\underline{x}_1$ are the displacements of the system at the time steps -1, 0 and 1 respectively.

Substituting equations 7.2 and 7.3 into equation 7.1 yields the displacement of the system as follows:

$$\underline{x}_1 = \left[\frac{\underline{\underline{M}}}{(\Delta t)^2} + \frac{\underline{\underline{C}}}{2\Delta t} \right]^{-1} \left\{ \left[\frac{2\underline{\underline{M}}}{(\Delta t)^2} - \underline{\underline{K}} \right] \underline{x}_0 + \left[\frac{\underline{\underline{C}}}{2\Delta t} - \frac{\underline{\underline{M}}}{(\Delta t)^2} \right] \underline{x}_{-1} + \underline{F}(t_0) \right\} \quad (7.4)$$

From equation 7.4 the general time series can be defined as follows:

$$\underline{x}_{i+1} = \left[\frac{\underline{M}}{(\Delta t)^2} + \frac{\underline{C}}{2\Delta t} \right]^{-1} \left\{ \left[\frac{2\underline{M}}{(\Delta t)^2} - \underline{K} \right] \underline{x}_i + \left[\frac{\underline{C}}{2\Delta t} - \frac{\underline{M}}{(\Delta t)^2} \right] \underline{x}_{i-1} + \underline{F}(t_i) \right\} \quad (7.5)$$

where $i \in 1 \dots N$ and N can be defined as the number of data points.

If the first two displacement vectors, $\underline{x}_0$ and $\underline{x}_1$ can be found and using equation 7.5, the whole time series could be calculated for all n . The displacement vector $\underline{x}(t)$ can be defined as:

$$\underline{x}(t) = \begin{pmatrix} x_1 \\ x_2 \\ \vdots \\ x_n \end{pmatrix}$$

In a similar manner the vectors for the acceleration and velocity can be determined.

7.2. KNPS model

In Figure 36 the layout of the primary system of the KNPS is shown.

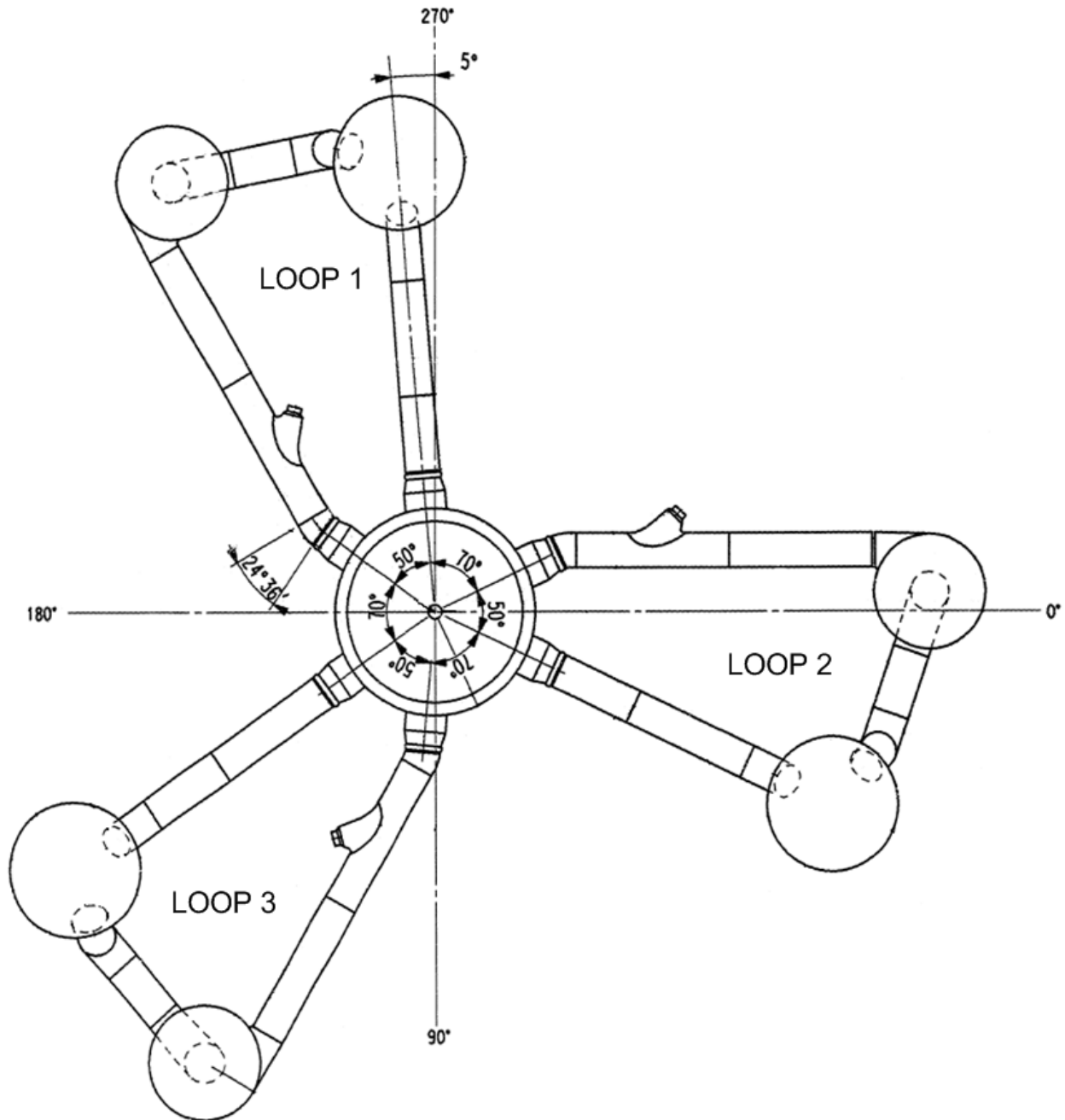


Figure 36: Primary system layout of the KNPS (Source: ESKOM, 2000:1 (Redrawn))

The KNPS primary system layout can be represented in terms of a simplified mass, damping and stiffness model. As shown in Figure 37, masses were allocated to the RPV (M_1), core barrel (M_2), three steam generators (M_3 , M_5 , M_7) and three MCPs (M_4 , M_6 , M_8). The pipe work connecting these components were all given damping values

Mass, damping and stiffness simulation of the KNPS primary system

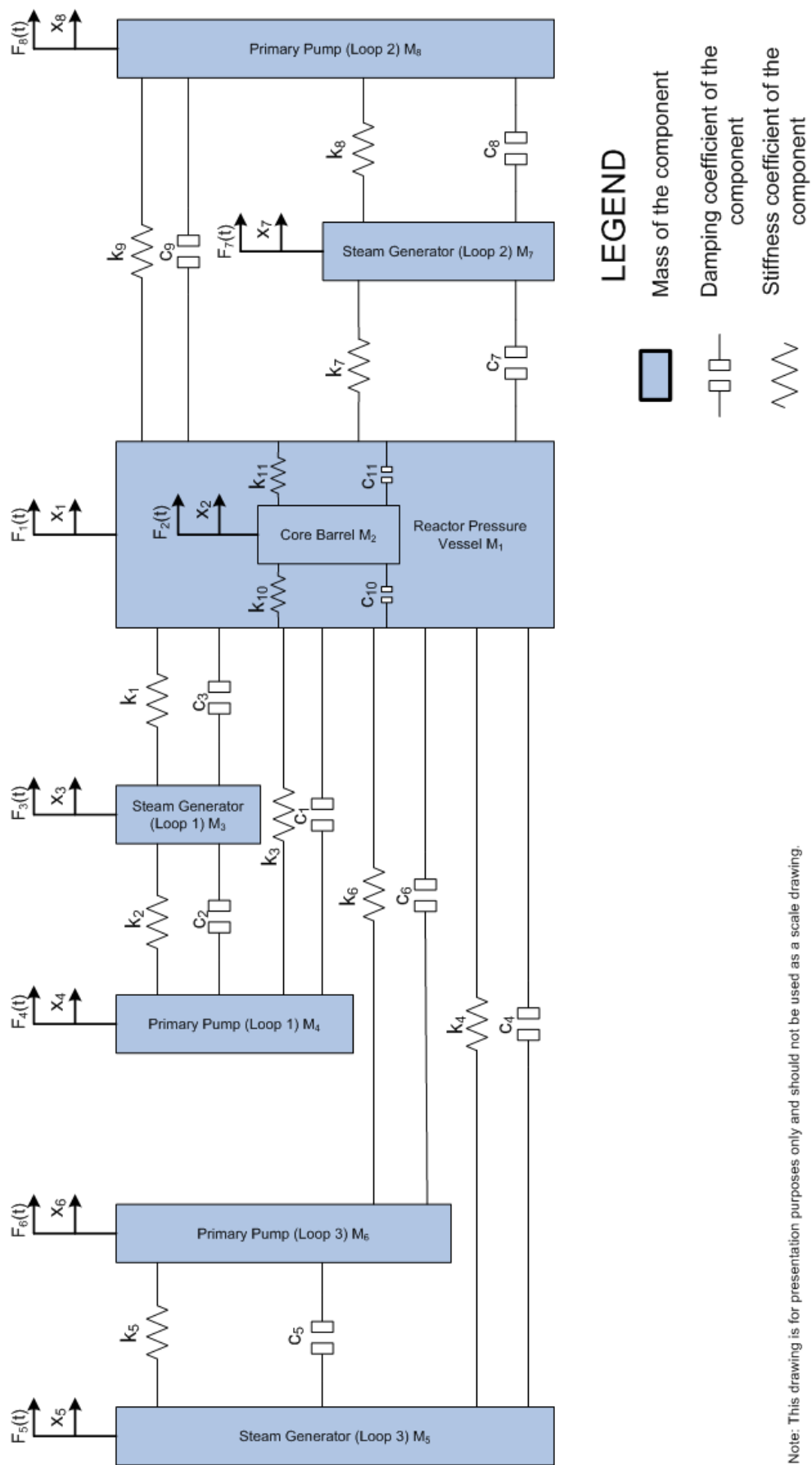


Figure 37: Mass, damping and stiffness simulation of the KNPS primary system
(Source: Figure by the author)

(c_1 - c_{11}) and stiffness values (k_1 - k_{11}). All the forces (F_1 - F_8) acting on these components were chosen to be in the same direction, which is in the positive x direction. The changes in displacement due to these forces were named x_1 to x_8 .

The model of the primary system was simplified by considering the layout in terms of its x and y component. With the seven accelerometers mounted on the RPV measuring acceleration in three directions only the x component is used in this model. All three loops of the KNPS were designed at different angles and these angles are divided into its x and y component where this model only makes use of the x components. Modelling the vibration into the y component can easily be done.

In the mathematical model the following assumptions are made:

- Any vibration in the z direction (vertical) is ignored.
- The angle at the steam generators, between the RPV and MCP, is assumed to be 90 degrees.
- The angles at the three inlet channels (24°36') are ignored and the pipe entries from the MCP are assumed to be perpendicular to the RPV.

It is understood that this crude estimation will yield solutions which leaves a lot to be desired, however it does not eliminate the fact that this model will help simulate an understanding of the measurement chain as well as the basic flow path from the movement of components to the final analysis on the frequency spectra.

For populating the mass matrix, the rules described in section 7.1 were applied to populate the 8 degrees of freedom system. As can be seen in the matrix the diagonal is filled with the masses M_1 to M_8 . The remaining terms were filled up with zeros.

$$\underline{\underline{\text{Mass (M)}}} := \begin{pmatrix} M_1 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & M_2 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & M_3 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & M_4 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & M_5 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & M_6 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & M_7 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & M_8 \end{pmatrix}$$

Again the stiffness matrix was completed using the guidelines specified above.

The stiffness matrix is illustrated below.

$$\underline{\underline{\text{Stiffness (K)}}} := \begin{pmatrix} k_1 + k_3 + k_4 + k_6 + k_{10} + k_9 + k_7 + k_{11} & -k_{10} - k_{11} & -k_1 & -k_3 & -k_4 & -k_6 & -k_7 & -k_9 \\ & -k_{10} - k_{11} & k_{10} + k_{11} & 0 & 0 & 0 & 0 & 0 \\ & -k_1 & 0 & k_1 + k_2 & -k_2 & 0 & 0 & 0 \\ & -k_3 & 0 & -k_2 & k_2 + k_3 & 0 & 0 & 0 \\ & -k_4 & 0 & 0 & 0 & k_4 + k_5 & -k_5 & 0 \\ & -k_6 & 0 & 0 & 0 & -k_5 & k_6 + k_5 & 0 \\ & -k_7 & 0 & 0 & 0 & 0 & 0 & k_7 + k_8 \\ & -k_9 & 0 & 0 & 0 & 0 & 0 & -k_8 & k_8 + k_9 \end{pmatrix}$$

In a similar manner the damping matrix can be simplified and is shown below.

$$\underline{\underline{\text{Damping (C)}}} := \begin{pmatrix} c_1 + c_3 + c_4 + c_6 + c_{10} + c_9 + c_7 + c_{11} & -c_{10} - c_{11} & -c_1 & -c_3 & -c_4 & -c_6 & -c_7 & -c_9 \\ & -c_{10} - c_{11} & c_{10} + c_{11} & 0 & 0 & 0 & 0 & 0 \\ & -c_1 & 0 & c_1 + c_2 & -c_2 & 0 & 0 & 0 \\ & -c_3 & 0 & -c_2 & c_2 + c_3 & 0 & 0 & 0 \\ & -c_4 & 0 & 0 & 0 & c_4 + c_5 & -c_5 & 0 \\ & -c_6 & 0 & 0 & 0 & -c_5 & c_6 + c_5 & 0 \\ & -c_7 & 0 & 0 & 0 & 0 & 0 & c_7 + c_8 \\ & -c_9 & 0 & 0 & 0 & 0 & 0 & -c_8 & c_8 + c_9 \end{pmatrix}$$

The generated time series contains 1024 points for every 8 seconds' sample for each component of x. This results in a Δt of 7.813×10^{-3} seconds which is the time between

data points (i.e. between x_i and x_{i+1}). According to Levy and Kroll (1951:1), for a single mass and damping system to obtain stability during numerical integration it is a prerequisite that the Δt value does not exceed the root square of the mass divided by the stiffness of the component. This theory can also be applied to a multi degree of freedom system with multiple mass and springs present. For the latter the following equation applies:

$$\Delta t < \frac{0.31}{\omega} \quad (7.6)$$

where ω is defined as the highest natural frequency in rad/sec in the system. In this model the value of Δt is well below the set point stated by equation 7.6. This resulted in convergence and rendering results as expected which will be discussed later in this chapter.

The excitation force acting on the system was simulated and is indicated below. At first resonance frequencies were determined using a random excitation force. The frequency spectrum of a random excitation force is typically flat. Hence, all resonances in the system will be excited. The simulated force vector consists of the sum of a 25 Hz component (which was inserted to simulate the primary coolant pump shaft rotation) and a random force. The random force was added to illustrate the flow induced vibration caused by the flow of the water coolant inside the primary system. The frequency spectrum of a random time series is typically flat. Using the random force vector would ensure the excitation of all frequencies present in the system. The measurement is simulated for one dimension only (x-direction); the forces acting on the 3-loop system in the x-y plane are assumed to be symmetric. The unit of the force vector is Newton and is displayed below:

$$\underline{\text{Force}}(t) := \begin{pmatrix} 2 \\ 1 \\ 3 \\ 4 \\ 3 \\ 4 \\ 3 \\ 4 \end{pmatrix} \cdot 10^8 \sin(2 \cdot \pi \cdot 25t) + \begin{pmatrix} 2 \\ 4 \\ 3 \\ 1 \\ 3 \\ 1 \\ 3 \\ 1 \end{pmatrix} \cdot 10^8 (-1 + \text{rnd}(1.2))$$

The two terms described above are indicated: The first term indicates the 25 Hz MCP shaft rotation speed. The different amplitudes of the sinusoidal excitation term were determined with the following logic:

Constants one to four (i.e. the constants used in first 4 components of the first vector term above) were allocated to the primary components. The component which is highly influenced by the 25 Hz external force was given a constant of four. This value is chosen based on the relative peak values of the measured signatures. From there as the influence on the components decreased the number is also decreased:

- The constants corresponding to the MCPs were chosen as four,
- For the constants corresponding to the steam generators a constant of three was chosen,
- For the constants corresponding to the RPV a constant of two was chosen,
- Finally the constants corresponding to the core barrel were given a constant of one.

The sinusoidal excitation components are thus chosen somewhat arbitrarily, albeit with an underlying logic based on the distance of the sensors to the source of the vibration.

The second term contained the random force simulating flow induced vibration due to the water coolant forces acting in the primary system. The fluctuation in forces differs between -1 and 1. It was reasoned that the highest vibrations were caused inside the core barrel followed by the steam generator, RPV and MCPs. These values were chosen arbitrarily. Since the mechanical degradation is primarily dependent upon stiffness changes, choosing arbitrary amplitude values for the excitation force components is justified since the amplitude values are not expected to change the frequency values of the resonance frequencies.

As previously mentioned equation 7.5 can be used to determine all the displacements in the series if the first two displacements are given. The initial displacements for all eight components at time equal to zero seconds were chosen to be zero since it is assumed that at that time no forces act on the system, starting from equilibrium. It was therefore only necessary to calculate the physical positions of the eight components on the x axis. The centre point of the RPV and therefore also the core barrel was chosen to be the zero point on the x axis. Components to the left of the zero point were given negative

values whilst components to the right were positive. The results are shown in the $\underline{x}_1$ vector. Very small additional displacement values in the direction of the force were added to the displacement value in $\underline{x}_1$. Both vectors $\underline{x}_1$ and $\underline{x}_2$ are measured in meters. Vector $\underline{x}_2$ is also shown below.

$$\underline{x}_1 := \begin{pmatrix} 0 \\ 0 \\ -0.091 \cdot 10^4 \\ -1.338 \cdot 10^4 \\ -0.860 \cdot 10^4 \\ -0.142 \cdot 10^4 \\ 0.951 \cdot 10^4 \\ 1.48 \cdot 10^4 \end{pmatrix} \quad \underline{x}_2 := \begin{pmatrix} 1.0 \cdot 10^{-4} \\ 3.3 \cdot 10^{-4} \\ -0.091 \cdot 10^4 + 4.5 \cdot 10^{-4} \\ -1.338 \cdot 10^4 + 1.3 \cdot 10^{-4} \\ -0.860 \cdot 10^4 + 4.5 \cdot 10^{-4} \\ -0.142 \cdot 10^4 + 1.3 \cdot 10^{-4} \\ 0.951 \cdot 10^4 + 4.5 \cdot 10^{-4} \\ 1.48 \cdot 10^4 + 1.3 \cdot 10^{-4} \end{pmatrix}$$

Once values in the matrices and vectors in equation 7.5 were determined, the time domain series of the displacement of the eight components were calculated. The time domain graphs for the three steam generators and three MCPs are shown in Figure 38 and Figure 39 respectively. The RPV and core barrel time domain series will have a similar appearance than those displayed in Figure 38 and Figure 39.

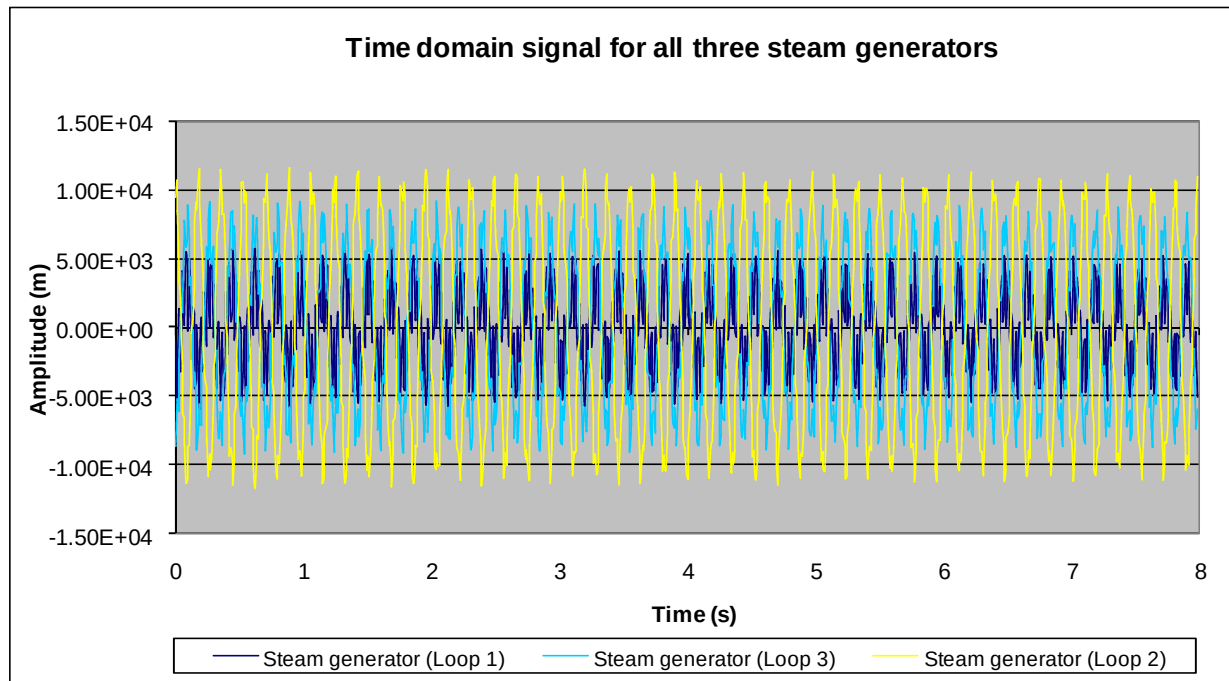


Figure 38: Displacement signal in the time domain for the three steam generators
(Source: Figure by the author)

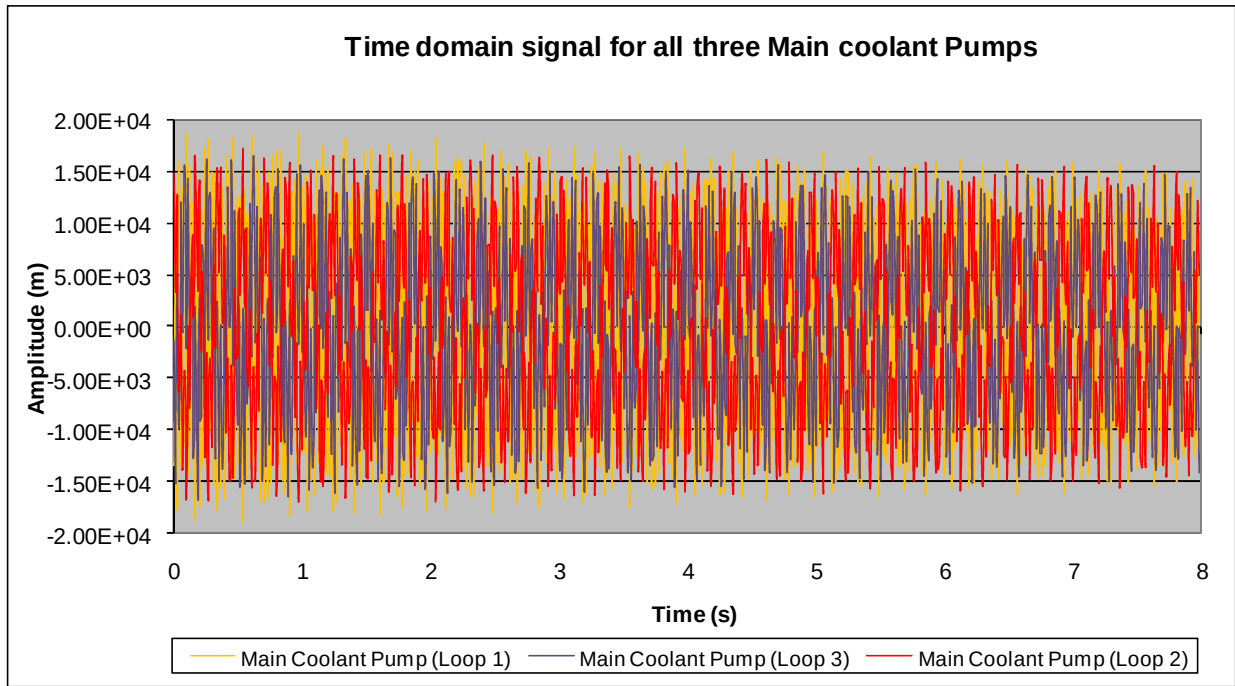


Figure 39: Displacement signal in the time domain for the three MCPs (Source: Figure by the author)

To correctly simulate the calculation process of the vibration monitoring system at the KNPS the time series were subjected to a Hanning window. This is done to minimise the effect of discontinuity of data points at the beginning and end of the sample. The Hanning window is defined as follows (Stremmer, 1992:140):

$$Hanning = 0.5 \left[1 - \cos \frac{2\pi(n-1)}{n} \right]$$

The Hanning window is illustrated in Figure 40.

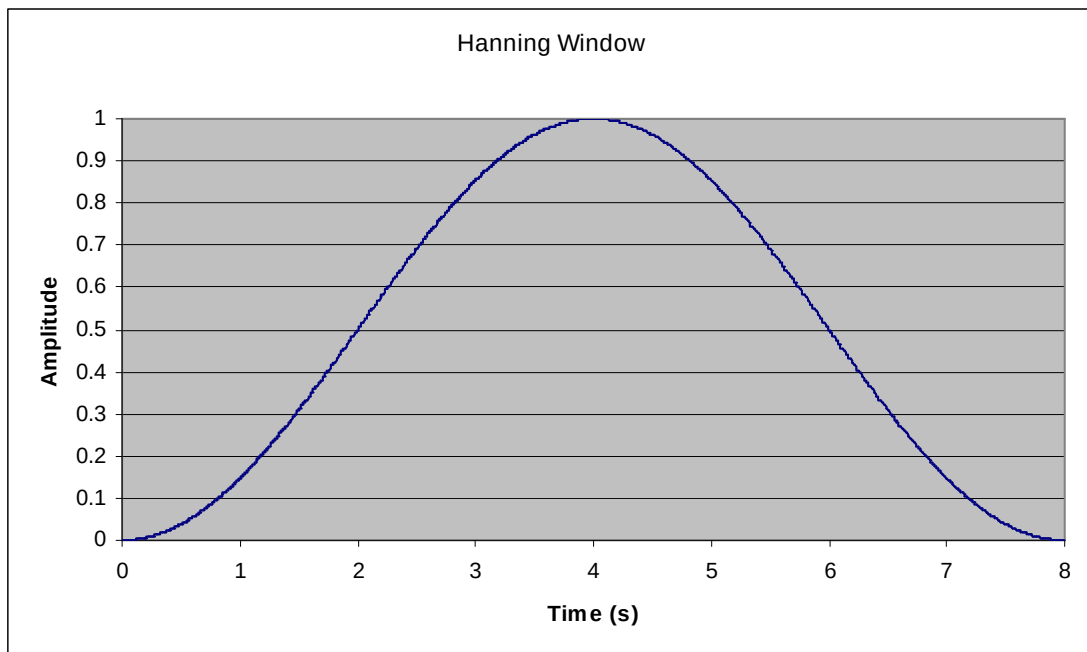


Figure 40: Hanning filter window (Source: Figure by the author)

Multiplying the time domain data displayed in Figure 38 and Figure 39 with the Hanning window results in the figure shown in Figure 41. Only the processed signal for the steam generator (loop 3) and the MCP (loop 2) is shown. Due to the multiplication, data outside the Hanning window are neglected. This is also consistent with the Fourier transformation, since the complex integral runs from minus to plus infinity.

A random external noise factor was also added to the time domain data in order to simulate signal losses within the system. This is explained in more detail in section 7.6.

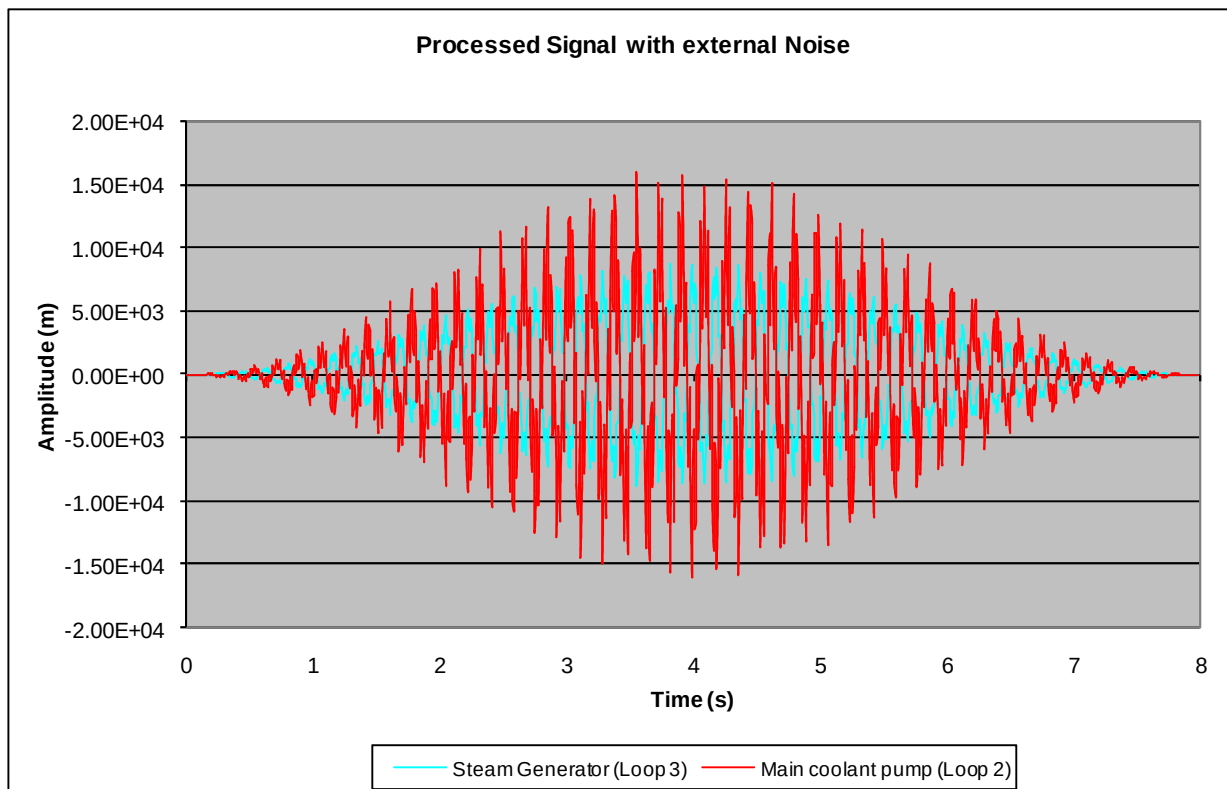


Figure 41: Processed signal with external noise (Source: Figure by the author)

Fourier analysis is performed on the processed time domain signal of the eight primary components. From these analyses the operating frequencies of the components are determined. By determining the absolute values of the real and imaginary values, the resonances are determined in the frequency domain as can be seen in Figure 42. The following resonances were identified in the frequency spectrum:

- MCP / Steam generator at 5.625 Hz,
- Core barrel beam mode at 8.125 Hz,
- Vessel rocking mode (0- 180 degrees) at 17 Hz and
- MCP / RPV at 35.5 Hz.

The peak at 25 Hz is not caused by a component's resonance frequency. As previously mentioned this peak (not a resonance) is caused by the simulated 25 Hz external force which was added to the system.

The frequency spectra are displayed on a logarithmic scale in order to clearly distinguish large and small peaks in the spectra. The model was simulated a total of 10

times from which the APSD were calculated. This is done to minimise the effect of external noise.

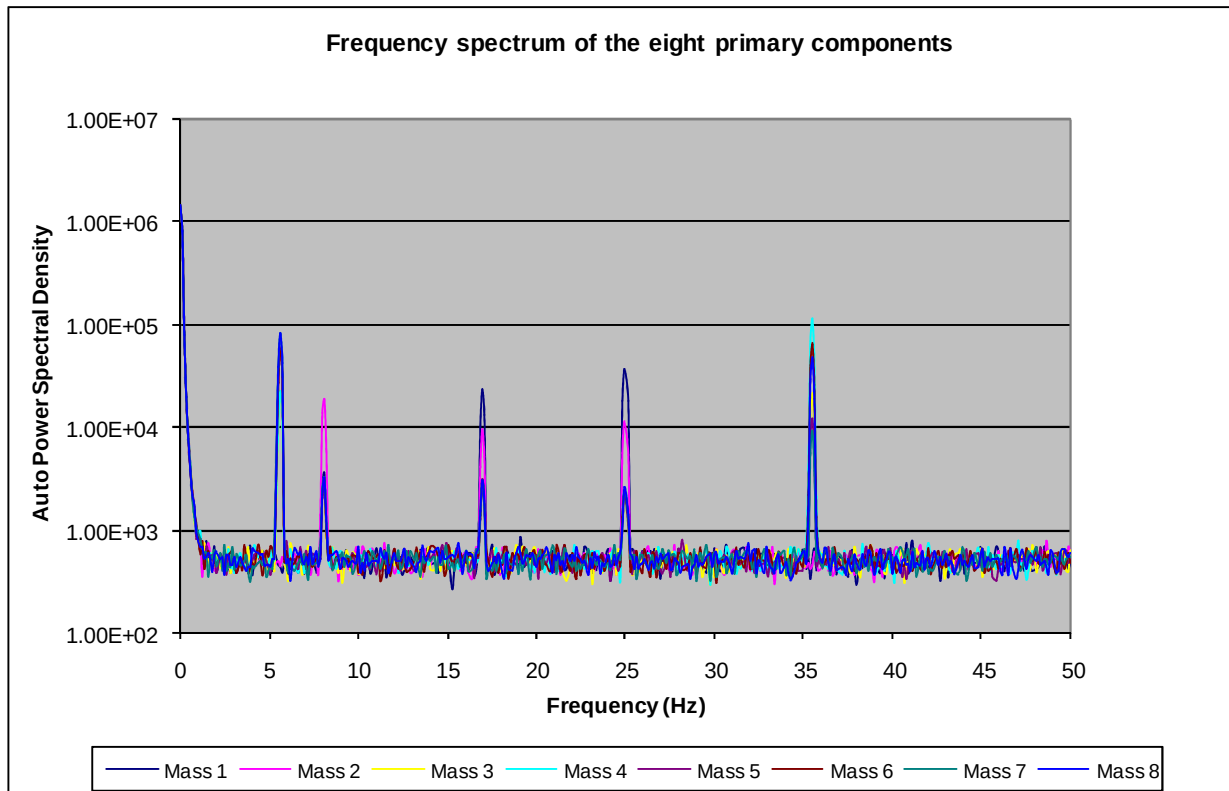


Figure 42: Frequency spectrum for the eight primary components (Source: Figure by the author)

7.3. Calculated results

As discussed in the introduction to this chapter, the masses of the eight chosen components were obtained from plant personnel at the KNPS. It must be mentioned that due to the nature of the components (long rigid cylindrical structures), only a small percentage of the component's mass will contribute to the kinetic energy or rocking of the component. However due to the simplicity of this model the latter was not considered. The stiffness and the damping characteristics of the pipe work connecting the components were not known. The masses of the components are specified in Table 7.

Table 7: Masses of primary system components

Primary system component	Mass (Kg)
RPV	329 000
Core barrel	244 000
Steam Generator	494 000
MCP	93 100

All the masses specified in Table 7 refer to the operating masses of the components. This means that the RPV, core barrel, steam generator and MCP are filled with water coolant. The mass of the MCP also include oil for lubrication.

The resonance frequencies of the component were obtained from van Niekerk (2000:4). The natural frequency of a component can be calculated using the following equation:

$$f = \frac{\sqrt{\frac{k_{eff}}{m}}}{2\pi} \quad (7.7)$$

where the symbols are similar to those defined in section 2.1.

Due to the complexity of the eight degree of freedom model the calculation of the stiffness parameters is not trivial and k_{eff} were calculated iteratively. The reason for this is that all the components being connected have in some way an influence on the other. These values were therefore determined once the model had been completed using iteration. The assumption was made that the stiffness coefficient in a component is similar in all three loops. The following stiffness values were calculated using several iterations:

Table 8: Stiffness coefficients of the primary components

Description	Stiffness coefficient
Stiffness between the RPV and the steam generator (k_1 , k_4 and k_7).	6.2×10^8 N/m
Stiffness between the MCP and the steam generator (k_2 , k_5 and k_8).	2.91×10^9 N/m

Stiffness between the RPV and MCP (k_3 , k_6 and k_9).	1.1×10^8 N/m
Stiffness between the RPV and the core barrel (k_{10} and k_{11}).	3.8×10^8 N/m

The damping parameters do not have any impact on the operating frequency except for influencing the form of the resonances. The assumption was made that the damping parameters of the components in all three loops were similar. Values were chosen between 1000 N.s/m and 3700 N.s/m.

7.4. Vessel rocking mode

The vessel rocking mode denotes the movement or vibration of the RPV during operation. Three types of vessel rocking modes are recorded at the KNPS (see Figure 19):

- Movement in the 0 – 180 degree direction,
- Movement in the 90 – 270 degree direction and
- Vertical direction.

Due to the simplicity of the model used to simulate the primary system, vibration measurements are restricted to one direction only. When simulating the vessel rocking mode only one direction will be visible in the frequency domain. Therefore, to illustrate both the movements in the horizontal direction, a simulation was done of the normal system. A slightly different stiffness was measured in the y-direction. In order to simulate this difference, and hence the two measured stiffness values, the stiffness between the RPV and the core barrel was changed in a separate simulation run. The stiffness of the latter was changed because the value of k_{10} and k_{11} directly influence the vessel rocking mode. The results of the simulation for the two runs are displayed in Figure 43. Vibration in the vertical direction is not included in this analysis.

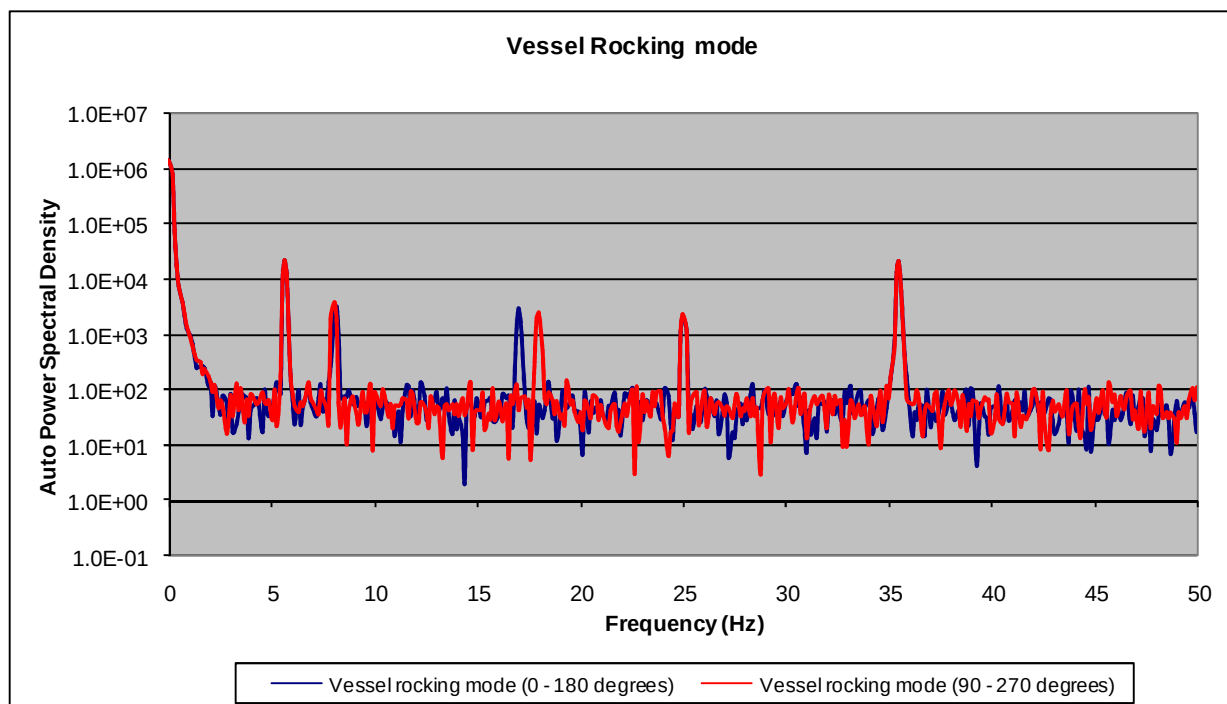


Figure 43: Vessel rocking mode (Source: Figure by the author)

The peak associated with the vessel rocking mode vibration in the 0 – 180 degree direction is distinguishable in blue at 17 Hz while the 90 – 270 degree vibration is displayed in red at approximately 18 Hz.

7.5. Mechanical degradation

Mechanical degradation in this system would be associated with changes in the stiffness of the system; this would be identified through the deviation of the resonance in the frequency domain.

To illustrate this concept, time domain measurements were recorded from which the corresponding FFT and APSD spectra were computed. Adequate measurements are essential in developing a frequency trend in order to simulate the deviation in the frequency resonances.

In Figure 44, the frequency signatures of ten simulations are displayed. After each measurement the spring hold down stiffness (k_{10} and k_{11}) of the core barrel inside the RPV were decreased to simulate the effect of mechanical degradation within the component. A reference line (black) is drawn at 8.125 Hz. The resonance of the core

barrel shifts to lower frequencies as the stiffness in the component decreases as can be seen in Figure 44. This is also shown in Table 9. The difference in frequency (Δf) after each stiffness change is indicated in the last column.

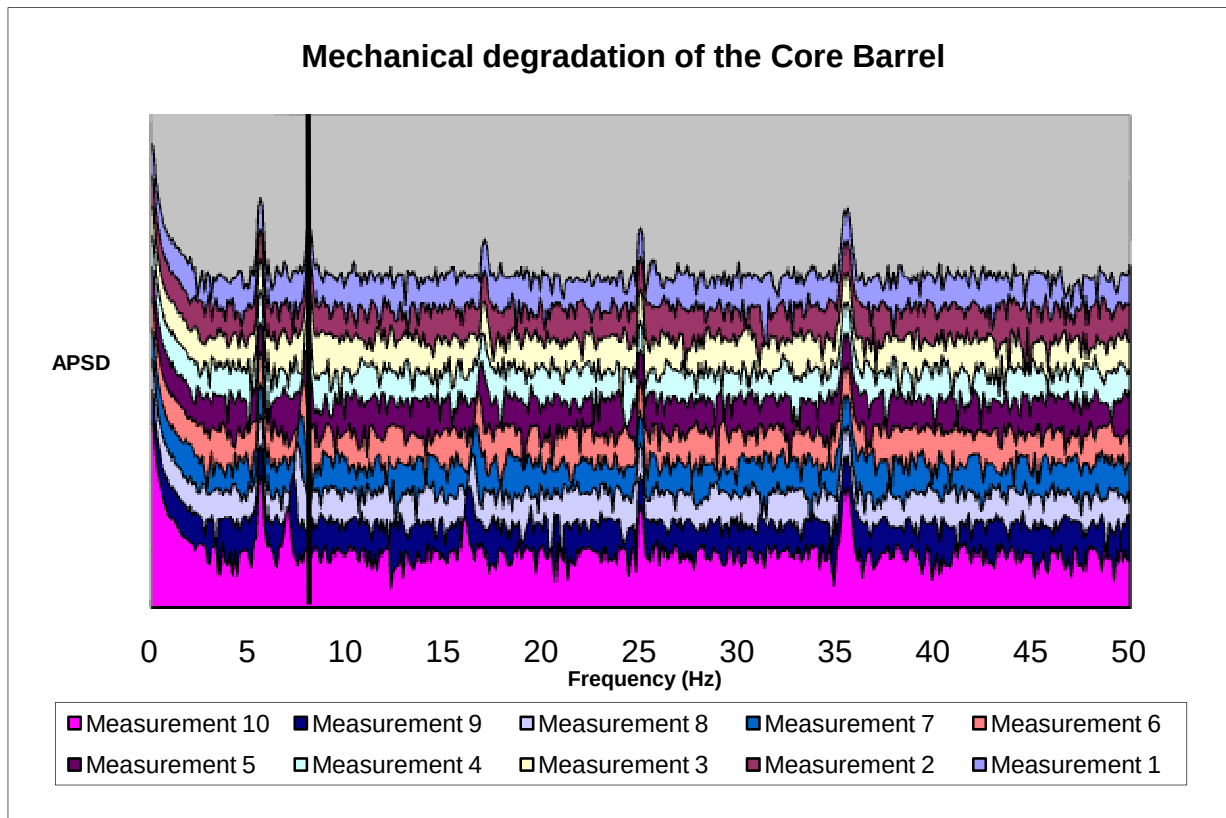


Figure 44: Simulation of core barrel degradation (Source: Figure by the author)

Table 9: Stiffness / Frequency change

Measurement	Core Barrel beam mode (Hz)	Stiffness Coefficient (N/m)	Δf (Hz)
1	8.125	3.8×10^8	-
2	8.125	3.79×10^8	0
3	8	3.75×10^8	0.125
4	8	3.7×10^8	0
5	7.95	3.6×10^8	0.05
6	7.875	3.42×10^8	0.075
7	7.625	3.25×10^8	0.25
8	7.5	3.05×10^8	0.125
9	7.25	2.8×10^8	0.25
10	7	2.5×10^8	0.25

7.6. Signal degradation

Signal degradation refers to any effect on the signal compromising signal output. In this section this phenomenon is illustrated using the same KNPS primary system model described above.

In order to illustrate this, five measurements were recorded and on each subsequent measurement the amplitude of the random white noise component added to the time domain data was increased. The effect is displayed in Figure 45. The first measurement (blue) was recorded with a very small random white noise component; hence all four operating resonances and the 25 Hz external force peak are visible. As the measurements increased the resonances became less distinguishable. This is partially due to the wide band spectrum, associated with the white noise, covering the resonances and decreasing the signal dynamic range. This degradation in the signal can be caused by the following:

- Loss of voltage or current in the signal. This is due to the long cable lengths employed,
- Damage to signal converter / connectors,
- Sensor degradation due to the high radiation level within the primary system etc.

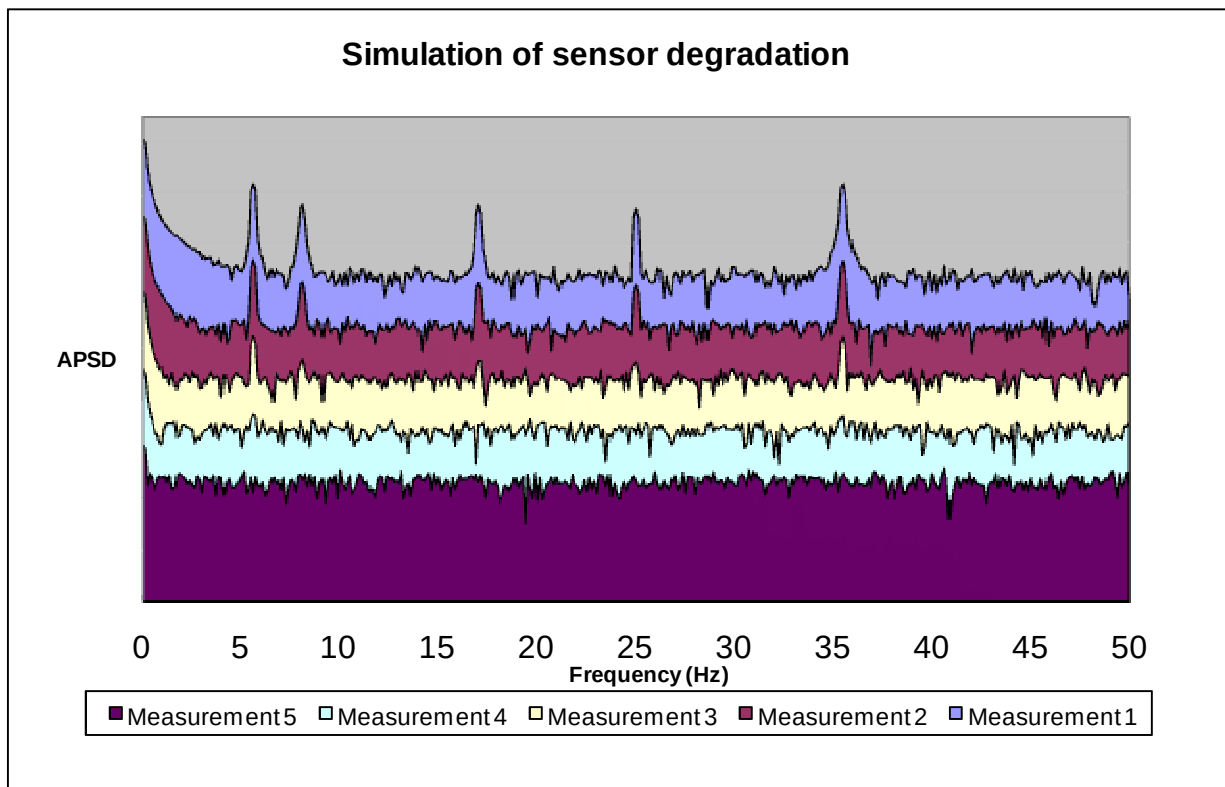


Figure 45: Simulation of sensor degradation (Source: Figure by the author)

8. SUMMARY AND CONCLUSIONS

Vibration monitoring of active and passive components can provide invaluable information regarding the mechanical integrity of the primary components. Due to the stringent safety criteria associated with nuclear plants, it is important to detect mechanical degradation before component failure occurs. Installing permanent vibration sensors on the primary components to serve as a continuous online vibration monitoring system allow plant personnel to detect vibration problems early, hence enhancing plant operator confidence and the general safety of the plant.

The following conclusion has been drawn from the measurement results obtained in this study:

1. The primary components of the KNPS were found to be without any mechanical degradation. This is supported by the fact that the respective resonances of the components have not deviated from the reference signatures.
2. Three vibration signals were found to have deteriorated significantly. In order to monitor the components with confidence it is recommended that the sensors be replaced at the next outage.

The mathematical model developed to simulate the vibration as well as the vibration monitoring signals provides a deeper understanding of the vibration of the primary system components. The model was used to simulate mechanical degradation in the primary system as well as degradation in the vibration signals.

The model can be used in parametric studies to simulate mechanical degradation of any of the components. As a simulator the system can also be utilised for training of operators.

Recommendation for future studies

It is recommended that the correct values for the stiffness and damping matrices be obtained and / or deduced from data supplied by the original equipment manufacturer and used in the model developed in this study. It is furthermore recommended that

information regarding the primary component support mechanisms be obtained and incorporated in the model.

The primary components of the KNPS were drawn in the 3D software package, Solidworks™ 2010. The components included the RPV, the core barrel, the three steam generators and the three MCPs.

The primary components were imported into an assembly mode together with the pipe work connecting the components. From here all the corresponding components were connected and the final assembly is illustrated in Figure 46.

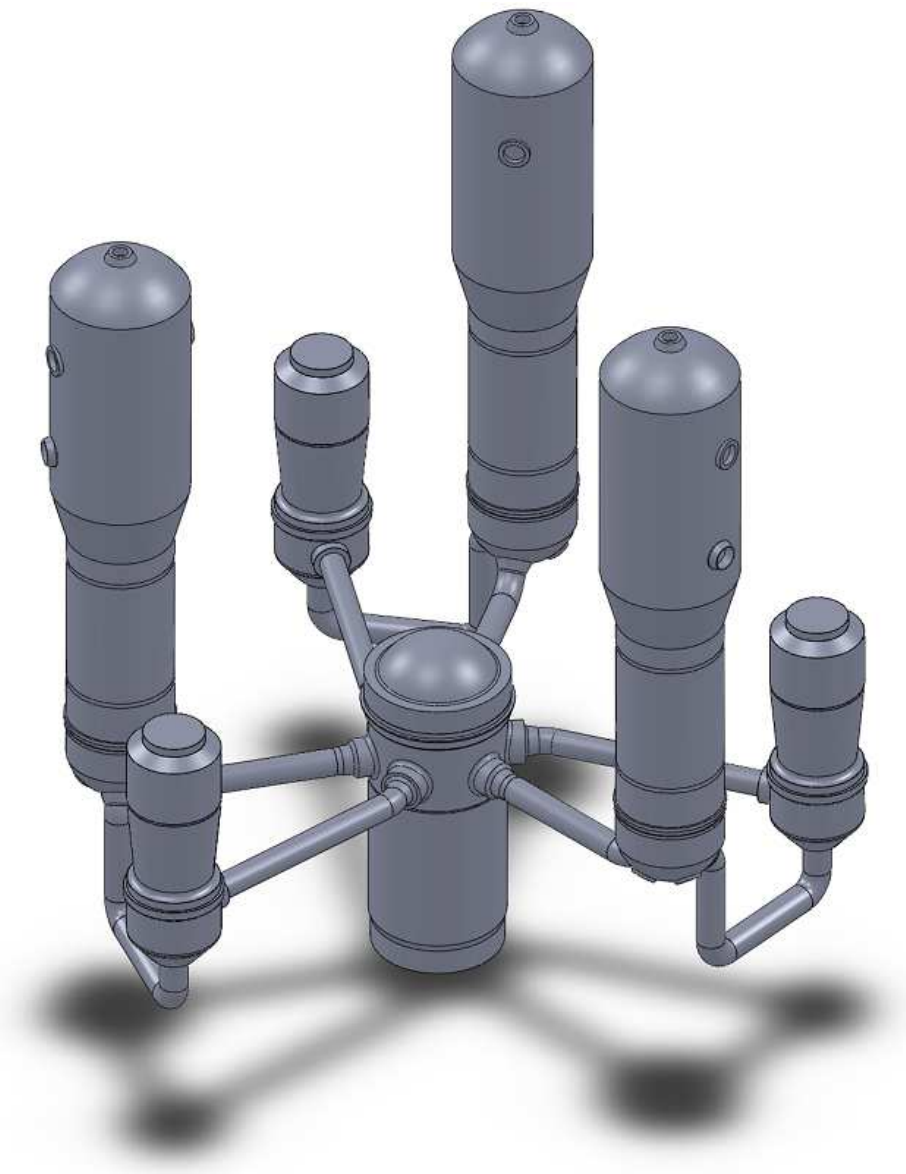


Figure 46: 3D drawing of the primary system of the KNPS (Source: Figure by the author)

It is recommended that a Finite Element Analysis (FEA) be done on the primary system of the KNPS. According to Park *et al.* (2003:186) it is possible to obtain the operating frequencies with the use of a FEM. This can be done by importing a simplified model of the primary system similar to that of Figure 46 into a program like ANSYSTM or NastranTM. It will also be possible to simulate mechanical degradation in the primary system (Park *et al.*, 2003:186).

It will also be beneficial to extend the mathematical model by adding the internals of the RPV and core barrel. This will broaden the scope of resonances visible in the frequency domain. Some of the resonances include:

- Fuel assemblies beam mode (fundamental -, 2nd - and 3rd modes)
- Control rod drive mechanism etc.

The dimensions of the model should also be extended to either 2D or 3D. This will increase the accuracy of the model in terms of correctly simulating the KNPS primary system. These improvements will result in e.g. the vessel rocking mode in all three directions being simulated and will hence appear in the frequency spectrum.

The multi-channel simulation outputs can also be used to calculate the CPSD, coherence and phase relationships between any two signals for comparison with measurements.

9. APPENDIX A

9.1. A brief overview of the history of nuclear reactor development

The first nuclear reactor was built as part of the Manhattan Project to develop a nuclear deterrent capability during World War II. According to the World Nuclear Association (2010) this project was directed by the 1938 Nobel Prize winner for his work in Physics, Enrico Fermi. The reactor was constructed in a room underneath squash courts at the University of Chicago (Department of Energy, 1982:10).

After World War II nuclear reactors were primarily used for peaceful purposes, i.e. the production of electricity. The first electricity produced from a nuclear reactor occurred on December 20th 1951 at the Experimental Breeder Reactor I in Idaho, United States of America (Department of Energy, 2003:8). The reactor contained electrical power output of 200 kW with an efficiency of approximately 15%.



Figure 47: Experimental Breeder Reactor near Idaho, USA (Source: www.inl.gov)

It was only 6 years later when the first commercial electricity producing nuclear power plant went into operation. Situated in Shippingport Pennsylvania, this PWR contributed 60 MW_e to the national grid. With a total cost of \$72.5 million, this reactor was constructed to serve dual purposes namely (Department of Energy, 2003:9):

- Power supply to the national grid as mentioned above and
- Plutonium production for military purposes.



Figure 48: Shippingport nuclear power plant (Source: <http://thephoenixsun.com>)

On the 1st of October 1982 the plant was shut down after 25 years of service. The decommissioning of the plant was completed in 1989.

After the Shippingport nuclear power plant successfully went critical and produced electricity to the grid, the pathway to peaceful nuclear energy was set and public acceptance blossomed resulting in many nuclear reactors being constructed in the 1960s and 1970s.

However on the 28 March 1979 the nuclear industry faced the worst accident in the United States of America (USA) in nuclear power production history. A combination of equipment malfunction and operator error led to the partial core meltdown of the reactor core of unit 2 of the Three Mile Island (TMI) reactor (United States nuclear regulatory commission, 2009). The reactor primary circuit was contained inside a containment which served as additional safety barrier. Although only a small amount of radiation was released to the environment, the financial impact was substantial. This forced the United States National Regulator Committee to increase plant safety acceptance criteria.



Figure 49: Three Mile Island nuclear power plant (Source: <http://thenewschronicle.com>)

Although unit 2 of the TMI reactor (right hand side of Figure 49) is not in service after the accident, Unit 1 is still operating and had a lifetime expansion of 20 years until 2034.

On 26 April 1986 the core of the Chernobyl nuclear power plant unit 4 in the former Union of Soviet Socialist Republics (USSR) melted due to a sudden power increase. As this reactor did not have a containment, high radiation release was inevitable. The result was high radiation leaks resulting in the death of thousands of people due to high radiation exposure. Once again this forced safety criteria to be re-evaluated and increased to prevent the re-occurrence of an accident of this magnitude. The International Nuclear Event Scale (INES) categorized this accident as a level 7, which is the maximum level allocated to an accident (International Atomic Energy Agency, 2008:2).

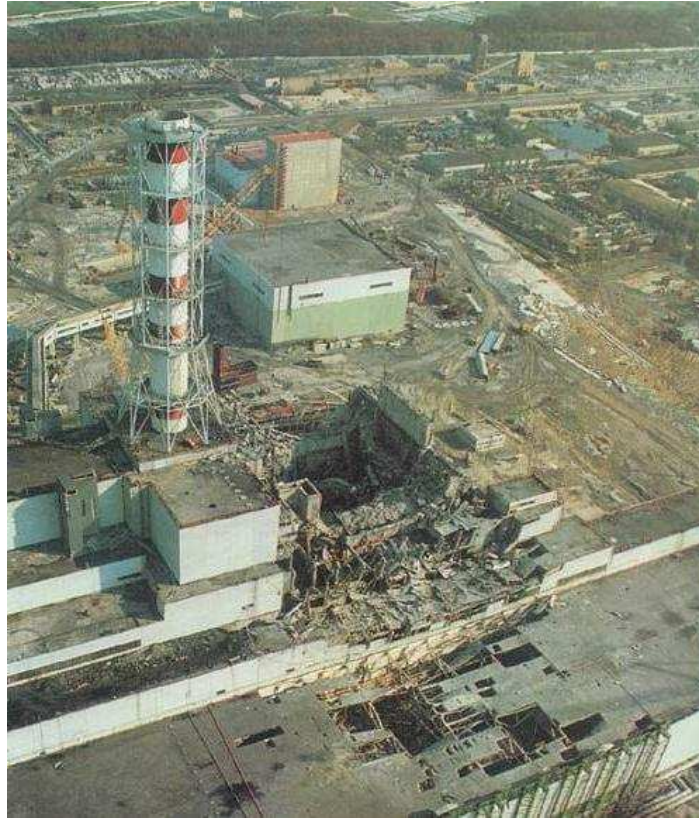


Figure 50: Chernobyl nuclear power plant unit 4 accident (Source: www.toptenz.net)

Following the TMI and Chernobyl accidents, enthusiasm to build and commission new nuclear plants dropped quite considerably. However, nuclear power has gained new ground as a sustainable safe and environmentally friendly source of energy due to growing international concerns about global warming. The carbon footprint of nuclear power compares well with that of renewable energy and has shown to be among the cheapest forms of energy.

It is expected that nuclear power will increase its share in the energy mix of countries. However safety criteria in nuclear power plants needs to be continuously evaluated as to prevent similar nuclear disasters as this will result in a major decline in financial, political and public support. Simultaneously, many reactors are entering a regime of plant life extension, with obvious commercial benefits.

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