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Διοίκηση στη Ναυτική Επιστήμη και Τεχνολογία

“ Methods of Monitoring and Diagnosis of Diesel Engine Failures ”

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Διπλωματική Εργασία

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"Ευχαριστίες"
Θα ήθελα να εκφράσω τις θερμές ευχαριστίες μου στον επιβλέποντα καθηγητή μου, κύριο Θεόδωρο Ζάννη, και τους κ.κ. Ιωάννη Κατσάνη και Ευθύμιο Παριώτη για την καθοδήγηση, τις συμβουλές τους και το χρόνο που μου παρείχαν για τη συγγραφή της παρούσης εργασίας. Επίσης σημαντική ήταν η βοήθεια του συναδέλφου κυρίου Σπυρίδωνα Κούστα για τις πληροφορίες που με βοήθησε να συλλέξω. Και φυσικά θα ήθελα να ευχαριστήσω την σύζυγό μου Ειρήνη και τον υίο μου Γρηγόρη για την αμέριστη στήριξη και συμπαράσταση κατά τη διάρκεια όλου του προγράμματος.

CONTENTS

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FIGURE CONTENTS

TABLE CONTENTS

SUMMARY

The main objective of this paper is to provide a literature review of the basic methods for monitoring and diagnosing diesel engine failures. The diesel engine is currently the thermal engine with the highest efficiency compared to all other known thermal engines and mainly for this reason it finds application in both the transport and power generation sectors. The continuous use of diesel engines requires high availability and very high reliability. For this reason, various systems have been developed in recent decades for monitoring operation and early diagnosis of possible faults. The aim of this paper is to analyse the different operation monitoring and fault diagnosis systems in order to understand the principle of operation, the advantages and disadvantages of each method and to finally propose the best - based on the literature - method. Based on the international literature, the most widely used diesel engine fault diagnosis methods and on which this study focuses are the following:

- Diagnostic method with measurement of cylinder pressure and thermodynamic simulation of the operation of each cylinder of a diesel engine.
- Diagnostic method with measurement and analysis of mechanical vibration.
- Diagnostic method with measurement and computational analysis of crankshaft torsional oscillations.

Chapter 1^o of this paper discusses the various methods of maintenance. Chapter 2^o describes in detail the diagnostic method based on cylinder pressure measurement and thermo-fluidic simulation of the operation of each cylinder of a diesel engine. For this method, a specific example of its application to a two-stroke main marine diesel engine and a four-stroke marine electric locomotive is discussed in detail. Chapter 3^o describes in detail the method of mechanical vibration measurement and analysis. For this method, international norms for its application to diesel engines based on standards of diesel engine manufacturers are given and an example of the application of the method for detecting incorrect adjustment of the exhaust valve of a diesel engine is discussed. Chapter 4^o presents the diagnostic method of the operating condition of a diesel engine which is based on the measurement and analysis of crankshaft torsional oscillations. For this method, too, an example of application to a six-cylinder four-stroke diesel engine is described in detail. Finally, chapter 5^o compares the degree of effectiveness of each diagnostic method and selects the best one based on existing literature data. In the same chapter, some suggestions for future extension and improvement of the present work are given.

Key words: Fault diagnosis, thermo-fluidic simulation, vibration

ΠΕΡΙΛΗΨΗ

Η παρούσα εργασία έχει βασικό στόχο την βιβλιογραφική επισκόπηση των βασικών μεθόδων παρακολούθησης λειτουργίας και διάγνωσης βλαβών κινητήρων diesel. Ο κινητήρας diesel είναι σήμερα η θερμική μηχανή με τον μεγαλύτερο βαθμό απόδοσης σε σχέση με όλες τις άλλες γνωστές θερμικές μηχανές και κυρίως για το λόγο αυτό βρίσκει εφαρμογή τόσο στον τομέα των μεταφορών όσο και στον τομέα της ηλεκτροπαραγωγής. Η συνεχής χρήση των κινητήρων diesel απαιτεί υψηλή διαθεσιμότητα και πολύ υψηλή αξιοπιστία. Για το λόγο αυτό τις τελευταίες δεκαετίες έχουν αναπτυχθεί διάφορα συστήματα παρακολούθησης λειτουργίας και έγκαιρης διάγνωσης πιθανών βλαβών. Αντικείμενο της παρούσας εργασίας είναι η ανάλυση των διαφόρων συστημάτων παρακολούθησης λειτουργίας και διάγνωσης βλαβών με σκοπό την κατανόηση της αρχής λειτουργίας τους, των πλεονεκτημάτων και των μειονεκτημάτων εκάστης μεθόδου και την τελική πρόταση της βέλτιστης – με βάση τα βιβλιογραφικά στοιχεία – μεθόδου. Με βάση την διεθνή βιβλιογραφία οι πιο ευρέως διαδεδομένες μέθοδοι διάγνωσης βλαβών κινητήρων diesel και στις οποίες επικεντρώνεται η παρούσα μελέτη είναι οι ακόλουθες:

- Διαγνωστική μέθοδος με μέτρηση της πίεσης κυλίνδρου και θερμοδυναμική προσομοίωση της λειτουργίας εκάστου κυλίνδρου ενός κινητήρα diesel.
- Διαγνωστική μέθοδος με μέτρηση και ανάλυση μηχανικών κραδασμών.
- Διαγνωστική μέθοδος με μέτρηση και υπολογιστική ανάλυση στρεπτικών ταλαντώσεων στροφαλοφόρου άξονα.

Στο 1^{ο} κεφάλαιο της παρούσας εργασίας εξετάζονται οι διάφορες μέθοδοι συντήρησης. Στο 2^{ο} κεφάλαιο περιγράφεται αναλυτικά η διαγνωστική μέθοδος που βασίζεται στην μέτρηση της πίεσης κυλίνδρου και στην θερμορευστομηχανική προσομοίωση της λειτουργίας κάθε κυλίνδρου ενός κινητήρα diesel. Για την συγκεκριμένη μέθοδο αναλύεται διεξοδικά συγκεκριμένο παράδειγμα εφαρμογής της σε δίχρονη κύρια ναυτική μηχανή diesel και σε τετράχρονη ηλεκτρομηχανή πλοίου. Στο 3^{ο} κεφάλαιο περιγράφεται αναλυτικά η μέθοδος μέτρησης και ανάλυσης μηχανικών κραδασμών. Για την συγκεκριμένη μέθοδο παρατίθενται οι διεθνείς νόρμες εφαρμογής της σε μηχανές diesel με βάση πρότυπα κατασκευαστικών οίκων κινητήρων diesel και αναλύεται ένα παράδειγμα εφαρμογής της μεθόδου για την διαπίστωση εσφαλμένης ρύθμισης βαλβίδας εξαγωγής κινητήρα diesel. Στο 4^{ο} κεφάλαιο παρουσιάζεται η διαγνωστική μέθοδος της κατάστασης λειτουργίας ενός κινητήρα diesel η οποία βασίζεται στην μέτρηση και στην ανάλυση στρεπτικών ταλαντώσεων στροφαλοφόρου άξονα. Τέλος στο 5^{ο} κεφάλαιο γίνεται σύγκριση του βαθμού αποτελεσματικότητας κάθε διαγνωστικής μεθόδου και προκρίνεται η βέλτιστη με βάση υφιστάμενα βιβλιογραφικά δεδομένα. Στο ίδιο κεφάλαιο παρατίθενται ορισμένες προτάσεις για μελλοντική επέκταση και βελτίωση της παρούσας εργασίας.

1 Introduction - Methods of Maintenance and Monitoring of Diesel Engines

1.1 Introduction

Piston engines are thermal engines that harness the chemical energy of the fuel to produce mechanical work, using the products of combustion as a working medium. Diesel engines are currently finding particularly wide application in both the transport and power generation sectors. For this reason, the demands placed on them have also increased. The combination of the needs for economical operation, environmental friendliness and personnel safety require the engine to operate reliably, continuously and efficiently and to minimise the intervals when it is out of service for maintenance. It is therefore clear that maintenance and monitoring of the engine are particularly important in order to prevent failures or, if not prevented, to deal with them more quickly. This will increase the useful life of the engine and reduce costs. In this chapter, the main methods of maintenance monitoring of diesel engine operation will be presented and explored [1,2].

1.2 Maintenance

Since the beginning of machine building, man has been trying to keep machines in the right working condition. To do this, he had to perform maintenance work. Initially, these maintenance processes were mainly concerned with repairing faults. However, a consequence of this type of maintenance was that between the breakdown and the completion of the repair, the machine was out of service. Moreover,when one component failed,other components also failed, requiring complex and time-consuming repairs. Later, machines became more and more complex, increasing in numberand replacing humans. In some applications,people became more dependent on machines, which required higher reliability. [1,2].

The need to reduce maintenance costs and downtime and increase reliability has forced manufacturers and users alike to resort to improved methods of monitoring operation and maintenance. The balance between these three factors (cost, downtime and reliability) varies depending on the maintenance methods used. The next chapter analyses the existing maintenance methods [1,2].

1.2.1 Maintenance Strategy (Maintenance Strategy)

Choosing the right maintenance strategy for a system, a machine or even a component is a very complex process. The main objective is to keep the machinery in optimal operating condition with the minimum of effort. But this goal requires the realization of several smaller goals, equally important, to the process.

The most important criteria for the selection of the maintenance plan will be presented in this section [1,2].

1.2.1.1 Maintenance Cost

The reduction in maintenance costs is not only due to the reduction in the cost of developing and performing inspections and processes, but also to another very important factor: The minimization of the time the machine is out of service due to a failure .Adopting the technique of no maintenance schedule and the final repair of the machine after failure and inability to operate may initially seem like a profitable solution, since it saves the costs of inspections and minor repairs. However, it turns out to be a loss-making solution as the cost of the time the machine is out of service is very high. Even in the case where a machine or component is not vital to a larger system this time will cause problems. A further disadvantage of this technique is that the failure that occurs can cause a chain of failures in neighbouring systems, so that the final cost of repair is significantly higher than the cost of maintenance. However, the maintenance of the machine is also a process that entails an economic loss since the time the machine is out of service cannot be avoided due to the replacement of various components. Inevitably, the operation of a machine entails a considerable cost to the user. For this reason, it is necessary to study and evaluate all the individual costs of running the machine in order to follow a final maintenance profile that will reduce the time the machine is out of service and reduce the overall cost [1,2].

1.2.1.2 Optimal Operational Behaviour

The optimum operating behaviour of a machine is to maintain it at the operating levels originally specified by the manufacturer. This includes not only the best possible performance of the machinery, but also its operation in accordance with international regulations. In order to maintain the engine at these levels of operation, proper and regular maintenance is required as frequent use of the engine leads to a drop in performance, resulting in a decrease in reliability and availability as well as in fuel consumption and exhaust emissions. In particular, in recent years, the regulations on exhaust emissions by international environmental organizations have become particularly stringent, which leads manufacturers to attach great importance to the optimal operating behaviour of their engines [1,2].

1.2.1.3 Reliability

The reliability of a machine is another factor that determines the choice of maintenance plan. Reliability is defined as the Mean Time To Failure (MTTF). It is important to note that the degree of reliability is determined by the role of the machine in the wider system. For example, the engine of a warship has a different degree of reliability compared to the engine of a yacht, since in the former case a potential failure can be fatal in times of operations. In cases where high reliability is required, the engine maintenance profile is based on the study of the engine's Reliability Centered Maintenance (RCM) rate. In even more demanding reliability cases, the costly method of having a back-up engine in case of failure is followed. The degree of reliability depends mainly on the needs of the whole and the purpose of the system in question and its determination contributes to the selection of the optimal maintenance method [1,2].

1.2.1.4Maintenance methods

According to Fagerland, Rothaug and Tokle [1], a maintenance method or strategy is defined as the combination of all techniques and manipulations performed on an object in order to restore or maintain it in a condition so that it can perform its function effectively. The available maintenance methods are as follows:

- Corrective maintenance.
- Maintenance Improvement.
- Preventive Maintenance
- Predictive Maintenance.

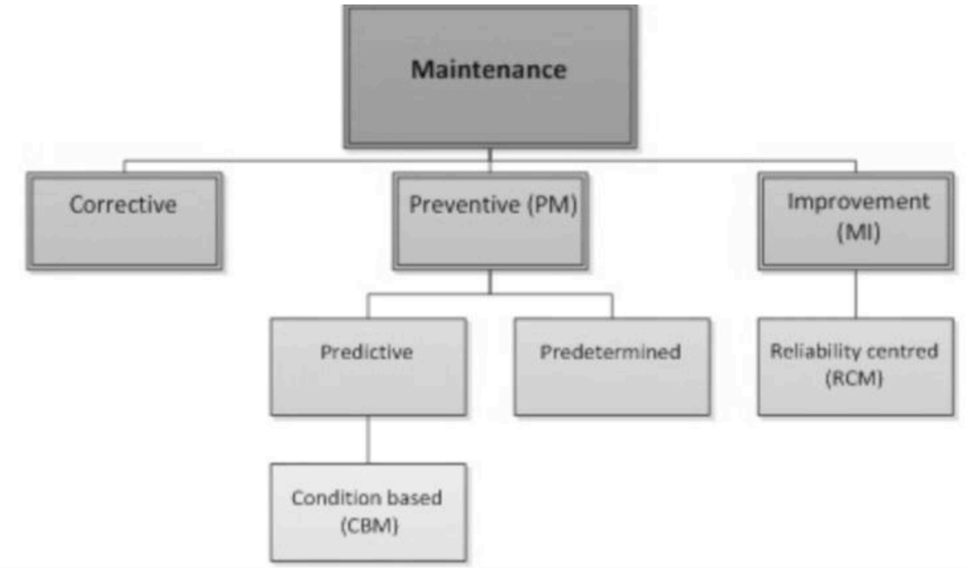

Figure 41: Engine maintenance strategies based on literature data [1]

1.2.1.4.1 Corrective Maintenance (Corrective Maintenance)

This maintenance does not consist of the planned actions and checks to be carried out during the operation of a machine but of its repair after a breakdown. This technique is based on the much lower cost but has the main disadvantage of reduced reliability. It concerns those components that are not of primary importance, are easy to repair and whose failure does not have a serious impact on the rest of the system. Corrective maintenance is also the back-up maintenance profile in case the primary one fails, either due to sudden failure or due to a selection error. These cases are particularly serious because there was usually a reason why a particular profile was selected [1,2].

1.2.1.4.2 Maintenance Improvement

Maintenance improvement is not a separate maintenance strategy but a perspective, the implementation of which is intended to reduce the need for maintenance as much as possible. It is applicable to all profiles but requires a very good knowledge of the system in order to find and reinforce those points that are most likely to fail. It is mainly based on the study of relevant statistics and probabilities in order to predict an upcoming failure due to the stress on the machine during operation. Of course, the success of the system requires the existence of statistical evidence for the gradual reduction of the optimal operation of the system [1,2].

1.2.1.4.3 Preventive Maintenance

Preventive maintenance aims to prevent the occurrence of failures and consequently the amount of time a machine is out of service. This prevention is achieved through the observance of appropriate schedules provided by the manufacturer, involving the performance of specific checks and actions at certain intervals or at certain operating hours. These intervals are determined by the importance of a component for the functionality of the machine and the required reliability. High or low reliability requires high or low inspection intervals respectively [1,2].

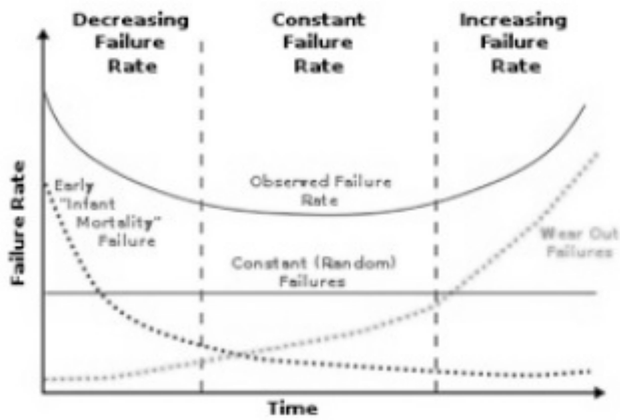

Figure 42. Preventive maintenance diagram [1]

The above diagram is the basis of predictive maintenance, since it can give the lifetime of a component if the current stress rate is known and always taking into account a certain safety margin. The main drawback, however, is that the result can often be fictitious. This is either because the manufacturer has relied on the wrong data or because the component in question does not behave in the expected way. The manufacturer also tests the behaviour and extracts the curve for operation under specific conditions. Therefore, operation under different conditions can affect the lifetime of the item and lead to either unnecessary early maintenance or late maintenance with all the risk this entails [1,2].

A second way of implementing the preventive maintenance profile is through the dynamic and regular study of the current stress state of the machinery and other indicators of the operating condition, in order to draw conclusions about the planned maintenance intervals. This study is usually carried out by monitoring the condition of the machinery. [1]

1.2.1.4.4 Predictive Maintenance

It is essentially a part of preventive maintenance based mainly on condition monitoring rather than on research and study of data and curves and aims through observation to plan the necessary maintenance. Like all the previous profiles, it has advantages and disadvantages. The main advantage is that through this observation an upcoming failure can be avoided but also the exact time and the required maintenance actions can be planned, thus improving the reliability of the system. However, this method of maintenance is particularly damaging economically for several reasons. Firstly, because the materials do not achieve their maximum useful life, since they are replaced regularly in case of failure. Above all, it is very costly because it requires a very good knowledge of the system, which means that studies must be carried out. For this reason, a cost-benefit analysis must be carried out so that the user can conclude that it is in his interest to adopt this plan [1,2].

In closing the chapter on maintenance strategy it is important to note that there is no better or worse maintenance plan. Each case requires its own plan, which can only be properly selected through a detailed study of the requirements and operating conditions of the system in question, without this of course prohibiting the adoption of another system or the replacement of the one already selected. However, although the choice of design is very complex, it is vital for reducing costs, downtime and quality. [1]

1.3 Condition-Based Maintenance of the machine (Condition-Based Maintenance)

As the choice of an appropriate maintenance strategy began to take on an increasingly important role among buyers and manufacturers, a method to aid this choice,condition monitoring, began to evolve. The method provides information about the health of the system from measured failures and canpredict future maintenance requirements based on this information. The aim is to detect and increase the frequencyoffailures,thusidentifyingfutureand potential failures. In this way, maintenance can be planned in atimelymannertopreventadditional failures and optimizeuptime. This method is quite reliable because current techniques allow for detailed monitoring and detection of even the slightest variations in normal operation such as the difference in vibration. However, not all faults can be detected, either because some occur unexpectedly or because they were not detected. This part, namely the non-detection of faults, is a subject of research to refine the condition monitoring [1-10].

There are two ways in which the method is carried out. By monitoring the voltage and checking the condition. Voltage monitoring requires that the condition is monitored continuously ,in order to detect the fault voltage and implement the maintenance plan appropriately. Conindition monitoring can only be determined if the condition of the system can be adequately estimated and sufficient data has been collected. In this way, the condition of a particular component can be automatically linked to an estimated maintenance interval. Condition checks are also useful when there are multiple systems of the same type. By comparing measurements from these different systems,conclusions can be drawn about the relevant elements causing the problems [1-10].

The most important part of the application of condition monitoring in relation to maintenance prediction is the optimal operation of the program, since only if the possible faults and their symptoms are identified can it be used. Such a program consists of various condition measurement techniques, which must be applied in the right way so that they can detect the possible symptoms. As far as these techniques are concerned, they will be discussed in the rest of the chapter as well as some of their applications and relevant examples. [1]

1.3.1 Status monitoring - Performance monitoring

The similarity between the two methods is that they condition an parameterize the system, but their objectives are quite different. It should be noted that conditioning measurements can use several methods,while performance measurements are single methods that address a set of parameters. Since these two methods are almost identical, they are often integrated into a single system. But what is fundamentally different is howeach is applied. Performance analysis,on the otherhand, is used to improve environmental and energy efficiency. However, the study of certain parameters during performance monitoring can be useful for assessing the situation. [1]

1.3.2 Monitoring methods

Predictive maintenance assesses the current status of investigations and therefore appropriate tools for monitoring the status are required. The necessary characteristics of these contracts are that they should be easy to implement, usefully represent the state of the system, be as cheap as possible and not affect the minimum performance of the system. Perhaps the most important requirement is a useful representation of the current state of the system. Often the best representation is achieved by reproducing potentially faulty components as faithfully as possible. This approach may seem simple once noise and interference have been removed, but this is not always possible.

The most common methods include vibration measurement, process parameter measurement, visual inspection, tribology and thermography. These techniques can be categorized as indirect and direct assessment methods. Indirect methods measure the process parameters of a system, while direct methods directly measure the strength of aparticular component using sensors to determineits strength. So, when it is decided to perform a "concentrationmeasurement", a choice needs to be made as to what type of "concentrationmeasurement" to perform. However,it turns out that in many cases it is only possible to verify all possible failures in an integrated system if more than one technology is selected. Combining more than one technology can detect more faults and summarize the symptoms,making diagnosis of faults easier. However, selecting and implementing the right technology can be challenging. The potential for system faults must be investigated and system knowledge must be as high as possible. It is also necessary to investigate where the sensors will be installed and the faults they may introduce. The initial inspectionis also very important in the selection. Each technology requires specialized equipment and the information provided by this equipment needs to be digitized, processed, analyzed and presented to the user. Only when implemented correctly can the initial costs be recovered. [1]

1.3.2.1Vibration monitoring

Discriminating harmonics in motor oscillations is the operating principle of diagnostics based on vibration analysis. This method uses vibration measurements taken from various points on the engine and usually from the crankshaft [11,12]. Through the spectral analysis (Fourier transform) of the measured vibrations, it is theoretically possible to detect and identify a malfunction or failure. This can be done either by comparison with the healthy state or by using a threshold of the intensity of some harmonics. The method has found application in large-scale diesel engines such as marine or power generation engines by estimating torsional oscillations in the engine crankshaft from instantaneous rotational speed measurements [11,12]. In many cases, manufacturers themselves pre-install such systems, but mainly for monitoring operation rather than for fault diagnosis. Through torsional oscillations it is possible to distinguish to some extent the contribution of each cylinder [11,12]. Also, in case of availability of real-time cylinder pressure measurements it is possible to determine the "healthy" spectrum giving more reliability to the method. Although the method is practically general in application, i.e. it does not require a large amount of historical data and is applicable regardless of the operating point at which the measurement is taken , it has limitations in the ability to identify the fault because the frequency spectrum is affected by all the engine subsystems with very difficult to distinguish the contribution of each component or subsystem. In addition, there are also oscillations of external origin from auxiliary systems that affect both measurement and analysis. [11,12]

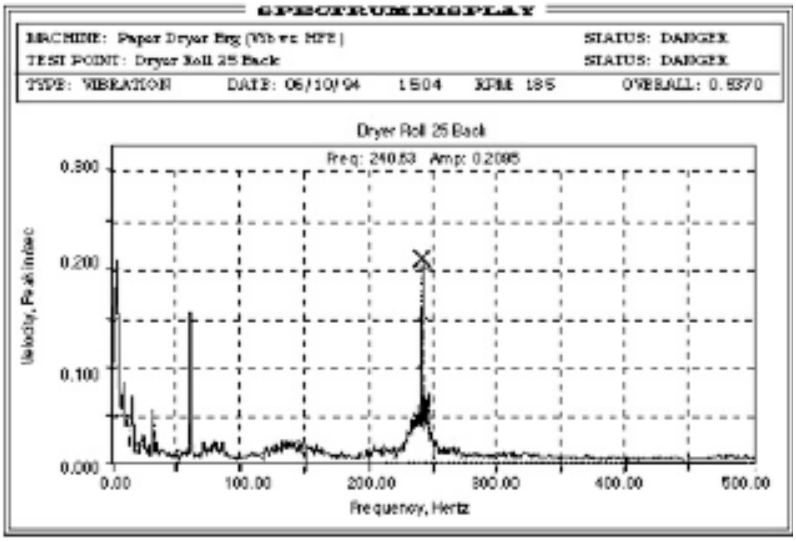

Figure 43. Vibration profile [11,12]

1.3.2.2Thermography

Vibration monitoring is not the most reliable method of monitoring a system and for this reason other methods such as thermography have been developed. This method measures the infrared radiation of a system. It is based on the fact that all objects with a temperature above absolute zero emit infrared radiation.If a system is monitored through thermography then an experienced analyst can detect thermal anomalies that will cause a failure. The types of faults that can be detected with this technology are mechanical defects, leadproblems and componentfailures. [1-12].

Infrared radiation can not be observed with the naked eye, so special equipment is needed that must be easy to use. The type of equipment suitable for the desired effect should also beconsidered. If the temperature of one or more specific points needs to be measured, an infrared thermometer [1-12] is the most suitable instrument. Infrared thermometers accurately indicate temperatures at relatively small points and are therefore suitable for measuring temperatures at critical points in the system, such as bearing caps. Less commonly used line scanners provide a one-dimensional comparative radiation line.Thermal imaging is an ideal application for components with in a system [1,2].

Unlike other devices,they can display the infrared profile of the entire system. There are different types of imaging equipment,ranging from relatively inexpensive monochrome versions without image storage or recall to more expensive infrared scanners with microprocessors. Figure 1 shows an example of an 'expensive' system where an industrial electrical fuse is assumed to have failed. [1,2].

Analyzing thermal images is the most difficult part of the technique. This is because heat and other external factors can render the work useless. In addition, transmitted and reflected radiation must be filtered out so that only the emitted light remains in the image. The interpretation of such images requires extensive training,so specialists need to be trained for this purpose or work with a specialized company. [1,2].

Figure 44. Infrared imaging of a fuse box [1,2]

1.3.2.3Tribology

Other tribology involves measuring the coefficient of friction, which involves unwanted friction between surfaces. This is primarily based on analyzing the lubricating oil and investigates the properties of the oil that may indicate wear problems in the system. By analyzing the amount and nature of residues in the lubricant, components in contact with the lubricant can be protected from wear [1-12]. The main application of tribology in predictive maintenance programs is to plan oil change intervals according to the strength of the oil. Although this may not seem crucial for the durability of the system, it is not [1-12]. Poor lubricant quality can lead to failure of the entire system and therefore lubricant quality is very important for the system. An example of a system where lubricant quality plays an important role is the hydraulic system. [1-12].

1.3.2.4Process parameters

This technology enables the measurement of process parameters using sensor technology to determine the state of the system. All process parameters that could indicate a fault can be measured to determine the current state. Examples include pressure, temperature, torque and strain. The most basic approach is to measure the useful output of the system and determine if it is within the desired range. If it is not within this range, there may be a problem with the system, indicating either a development issue or a need to change the configuration. Process parameter measurements have a wide range of applications and can not only reveal faults, but can also indicate the state of the system in terms of efficiency. Often, these two applications are used simultaneously as fault and performance diagnostics. [1,2].

Monitoring can be done at the system level, called machine monitoring, in which the presence of a fault can be detected, but the location is often difficult to pinpoint, and at the component level, called component monitoring, which makes it much more accurate to find the exact location of the fault. This monitoring has a highly increased cost compared to the first one and requires detailed knowledge of the system in order to position the sensors correctly [1,2].

When implementing a process parameter measurement system, the success rate is determined by the selection of appropriate parameter measurements, the selection of reliable sensors, the selection of a correct signal processing system and the implementation of the processing system outputs. [1,2].

How to choose process parameters depends on the faults you want to find. In many cases, no one parameter will reveal all possible faults. The operator therefore needs to recognize possible defects, group them into groups and select the appropriate parameter group. The selected parameters may reveal groups of defects and preferably each of these defects should be distinguished by the output of the parameter. When selecting parameters, it should be understood that some parameters are not possible to select due to cost, weakness of existing sensors, difficulty in physically positioning the sensors, etc [1,2].

With technological advances in detection and processing systems, experts' views on performance and understanding of the situation may differ from period to period. For example, a parameter that was considered a typical parameter decades ago and was not taken into account due to lack of sufficient technical capacity may be taken into account again due to technological advances. An example is the measurement of in-cylinder pressure. Around 1970, when in-cylinder pressure started to be measured, existing pressure sensors could not withstand the high temperatures inside the cylinder. Therefore, this seemingly important parameter was never directly used in knitted ring systems [1,2].

Once the correct parameters have been found and the sensors and processors installed, it is important for the performance of the measurement system that their outputs are implemented correctly. This part of the system is called 'knitting system diagnostics'. The outputs of the system must be displayed in such a way that it is possible to draw a conclusion about the current operating state of the system. The first aim of diagnostic maintenance is always to determine the state of the system and, if the state is abnormal, to identify the cause, so-called diagnostics [1,2].

There are two types of diagnostic methods: automatic and manual. In the case of automatic diagnostics, with the help of odel, the system can present possible faults to the operator in order of probability. The disadvantage of this automatic diagnostics is that certain modules and software are very complex and therefore time and labor intensive. On the other hand, it has the advantage that anyone can become a diagnostician, as they do not have to evaluate the measurements and only receive the answers provided by the system. This increase in automated diagnostics also introduces a disadvantage of manual diagnostics: the diagnostician must be trained to interpret and filter the information presented to them [1,2]. However, these diagnosticians are not always available and can be transferred from this position for any reason, so knowledge and experience must be transferred to the new diagnostician. They also have different diagnostic methods, so even if they have the same training, they may diagnose the same symptoms differently. For these reasons, there is a growing need for automatization and the removal of the human factor from diagnostic methods. In this case, the information displayed is a quantitative representation of sensory measurements, rather than identifying the most likely malfunction. Thus, if there are errors in the displayed values, the operator can be provided with standardized procedures to fill in the errors [1,2].

After diagnosis, a 'forecasting' phase begins, in which failures that will occur in the near future are predicted. The results of this prediction form the basis for planning maintenance work. Forecasting is not always considered as part of the maintenance work, but it is an important step if the maintenance work is to be carried out as maintenance work [1,2].

1.4 Diesel Engine Condition Monitoring

This chapter describes the issues involved in measuring engine speed in diesel engines. First, general methods applied to diesel engines are described, covering all aspects of engine control from parameter setting to diagnostics. The control systems available for diesel engines will also be explained with examples. [1-13].

1.4.1 Diesel Engine Monitoring Methods

Diesel engines use specific condition monitoring methods which are selected according to the part of the engine for which we want to have data. In this way, these methods can be separated, i.e. according to the object of monitoring, because each engine part or subsystem has its own parameters. The following methods for monitoring the subsystems of a diesel engine have thus been developed [1-13]:

- Fuel injection system
- In-cylinder combustion
- Lubrication system
- Mechanical Components
- Heat exchangers
- Air and exhaust network

1.4.1.1 Fuel injection

The fuel injection process is critical to engine performance, as incorrectly timed fuel injection or leaking injectors can cause thermal stress problems. Thermal stress can lead to reduced engine quality and increased exhaust emissions. In other words, thermal stress is not only responsible for improving fuel economy. The fuel injection process is a separate part in the fuel economy process as it involves sensitive components such as pumps and burners. The coordination of these parts is crucial because if they fail, other parts of the engine are also affected [1-13].

The monitoring of the fuel injection is of course particularly difficult due to the limited space where the sensors have to be placed and the adverse conditions due to the high pressure, temperature and frequency. For this reason, an indirect method of measurement is usually used, such as measuring the injection pressure using a piezoelectric sensor in the fuel injection pipe. The following figure shows a typical figure in which measured fuel injection pressure diagrams for different fuels are presented and the process of tracking the fuel injection duration is illustrated [12-15].

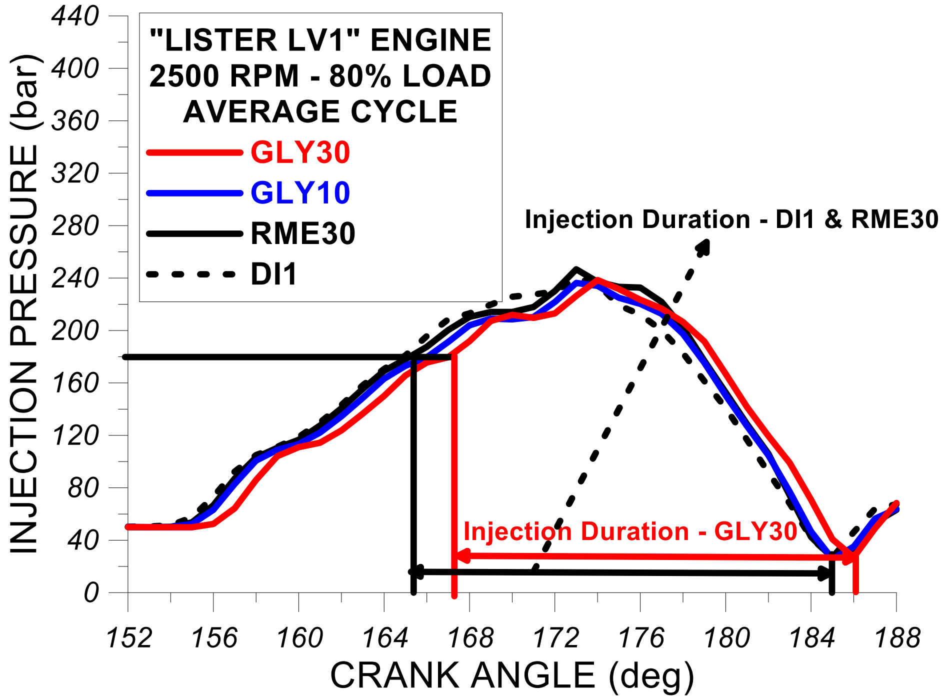

Figure 45. Variation of fuel injection pressure as a function of crank angle for various fuels. Experimental results are given for a single-cylinder 4X diesel engine at 2500 rpm and at 80% load.[12-15]

1.4.1.2Burning

This is the most critical process for the engine and will have a wider impact if a failure occurs. Failures in the cylinder can be detected by measuring the cylinder combustion process, but other failures that occur before air or fuel reaches the cylinder can also be detected as they affect the cylinder combustion process [12-15]. A lot of information can be measured from the combustion process, including temperature, pressure and time, angle and volume. The effective flow rate is also a useful indicator to determine cylinder durability [12-15].

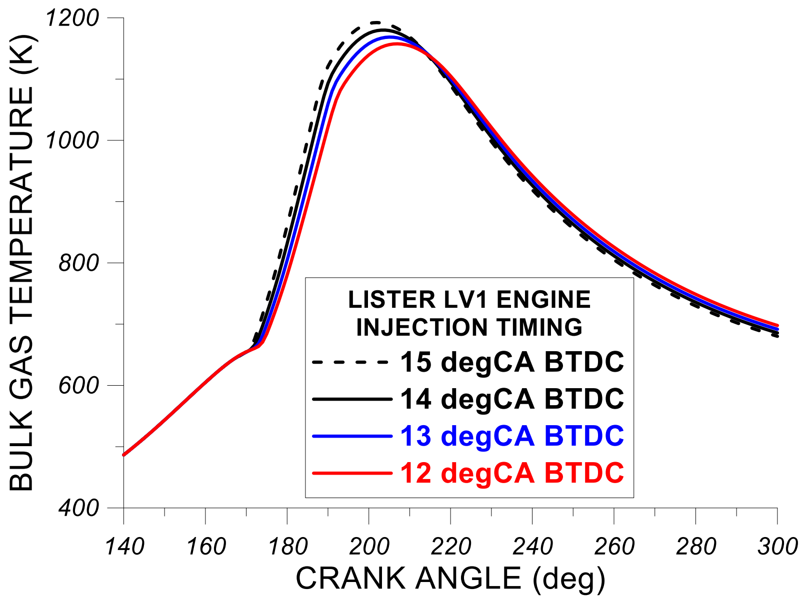

Figure 46. Effect of reducing the fuel injection advance on the average cylinder gas temperature.[12-15]

The combustion process is evaluated through dynamometer diagrams which give us information such as the maximum cylinder pressure, the compression pressure, the expansion pressure and the angles at which they occurred [12-15]. These diagrams are extracted through sensors which are mounted inside the cylinders which makes this study highly economically damaging since the harsh environment inside the cylinders requires highly robust sensors [1-10]. Considering the information extracted from the diagrams and the problems they can reveal, it becomes clearer that combustion process monitoring is not only a condition monitoring method but also a performance monitoring method [1-10].

1.4.1.3Lubrication system

The lubrication system is an important part of a diesel engine. Most moving parts inside the engine, as well as surrounding parts, require lubricant. The quality of the lubricant affects component wear and can lead to indirect damage. In addition, metal deposits and contaminants in the lubricant can circulate inside the engine and damage other parts. To check the quality of the lubricant, temperature and pressure measurements and friction tests are performed around the lubricant [1-10].

1.4.1.4Metal Engine Parts

Bearings and piston rings play an important role in the mechanical system of a diesel engine. Bearings are subjected to high loads and wear constantly. As a result, cracks form and eventually parts are damaged. Wear can be measured in various ways, including wear sensors, surface temperature sensors and tribology. Piston rings are an important part of the engine as they prevent oil, fuel and combustion gases from leaking out. Piston ring wear can be measured with proximity sensors, which show the distance between the ring and the compression surface [1-10].

1.4.1.5Heat Exchangers Heat Exchangers

Heat exchanger maintenance is more focused on improving performance than engine maintenance. A faulty heat exchanger reduces engine performance, but does not necessarily lead to a failure. The parameters to measure are the inlet and outlet air temperatures of the heat exchanger [1-10].

1.4.1.6Air intake and exhaust networks

When measuring the air and exhaust networks inside the engine, the aim is to detect possible malfunctions and optimize performance. In the first case, the main causes are air flow and air leaks; in the second, emissions and exhaust gas quality are monitored. Reduced airflow can cause problems such as metal stress due to increased temperatures. Pressure is highly variable and has very small values, so it makes little sense to install sensors in the network circuit. Temperatures are also very high, which creates a harsh environment for sensors. Therefore, some components, such as compressor filters and turbine blades, require more frequent periodic maintenance. From these components, information on turbine and compressor performance and changes in air pressure can be obtained. Exhaust gas analysis can also provide data on fuel timing and fuel quality [1-10].

1.5 Examples of Diesel Engine Operation Monitoring Systems

Over the years, different diesel engine control systems have been introduced by many companies. Although these systems look similar, many of them use different technologies, hardware and software [1]. These differences are mainly due to the technology available, but the underlying concepts can be very different. Parameters that are considered extremely important and representative by one company may not be considered important by another [1]. Therefore, the various systems are listed and discussed below.

Most of these schemes have been established for maritime employment. The reason why most systems are being developed in this sector is because high reliability is required to avoid loss of propulsion and carefully planned maintenance. These are the main reasons why such a monitoring system has such a high cost [1]. Some examples of monitoring systems are [1]:

- CYLDET-CM
- DEFD
- KBMED
- CPMPS

1.5.1 Operation Monitoring System CYLDET-CM

CYLDET-CM is a system developed for temperature measurement in marine diesel engines. It is designed according to the number of transducers and the degree of signal processing. The system can measure cylinder pressure, jacket temperature and jacket wear [1-10].

1.5.2 DEFD Operational Monitoring System

The Diesel Engine Fault Diagnosis method was invented by Lloyd's Register and its main functions are [1-10]:

1) Early identification of specific errors
2) Multiple error detection
3) Sensor fault detection
4) Provision for reporting unrecognised changes in engine operating parameters
5) Collaboration with other models to assess the future impact of identified errors

1.5.3 KBMED Operation Monitoring System

KBMED was developed by Huazhong University of Science and Technology in China. It is an integrated engine diagnostic and fault diagnosis system, especially for diesel engines. It uses information processing techniques based on the system's existing knowledge, measurement results, and user input to determine engine status and diagnose faults [1-10].KBMED includes a task management program, knowledge development and management program, diagnostic re-examination program, diagnostic process interpretation program, fault diagnosis program, fault diagnosis program, and fault diagnosis programs, control and signal measurement systems, intelligent signal analysis, and knowledge bases [1-10].

1.5.4 CPMPS Operational Monitoring System

CPMPS is based on existing techniques, but focuses more on fault diagnosis, performance optimization, and predictive maintenance. This method includes five main functions [1-10]:

- Status monitoring
- Performance monitoring
- Fault diagnosis
- Maintenance forecast
- Performance optimisation

1.6 Malfunctions - Diesel Engine Failures

As is obvious, efforts to prevent a diesel engine from failing are constant. However, although large sums of money and a lot of time are spent, the progressive deterioration of the engine's functions due to constant stress makes the occurrence of failures inevitable. The following are some of the most common faults that occur as a result of the degradation or malfunction of one or more diesel engine units [1-15].

- **Loss of power**: Power is a practical parameter for measuring engine performance. Severe loss of power can cause the engine to stall. The main causes of power loss are ignition failure, excessive exhaust leakage from the combustion chamber to the crankcase, failures related to the fuel injection system and failures related to the engine supercharging system [1,2,3-15].
- **Excessive emissions - Altered exhaust gas composition**: international organisations have set strict criteria on the emissions of pollutants from diesel engines as the emissions cause air pollution that is harmful to human health. In addition, the quantitative and qualitative analysis of exhaust gases is indicative of the quality of combustion and the change in their composition may be due to the following factors: fuel injector blockage, wrong timing of fuel injection, clogged intake air filter, loss of injection pressure, loss of compression pressure, excess fuel, turbocharger malfunction, clogged fuel filter [1,2,3-15].
- **Failure of the lubrication network**: The main lubrication system failures are incorrect oil pressure (usually pressure drop) and alteration of the lubricating oil composition. Oil pressure drop can result in friction from two moving parts of the machine without lubrication, which can lead to serious damage. Diesel engine oils may lose their property due to various factors. The most common causes are unburnt hydrocarbons from incomplete combustion of the fuel, oxidation products and ash from the lubricating oil and metal particles from wear of metal moving parts [1,2,3-15].
- **Noise and vibration**: The factors that cause engine noise can be categorised as follows [1,2,3-15]:
    - Mechanical noise caused by the interaction of two moving mechanical parts
    - Noise caused by combustion
    - Noise during air intake and exhaust

  In diesel engines, an important source of noise can be the injection system, in particular the position of the injector needle and the control valve. Vibration requires special attention since in some cases, such as torsional vibration, it can even cause shaft breakage. Most failures can be diagnosed by means of vibration signal analysis [1,2,3-15].
- **Wear and tear**: Corrosion and abrasion are the main causes of wear of a machine and the degree of their effect is proportional to the operating conditions. Wear can occur on any of the main moving parts of the machine such as the points associated with the pistons. They are mainly due to metal residues and particles entering from the inlets, which rub against each other [1,2,3-15].
- **Thermal overload**: Engine anomalies can significantly increase temperatures within the cylinders and thus lead to thermal overloading of the engine. Engine

thermal overload can be a single or combined result of several factors such as poor fuel quality, fuel injector leaks, low injection pressure, possible clogging of the turbocharger air cooler, possible coolant leakage, high oil temperature and incorrect timing of fuel injection. Engine overheating has the following consequences [1,2,3-15]:

- Higher temperatures in the combustion chamber walls will result in an increased corrosion rate at high temperatures.
- The thermal stresses on the upper head of the piston and the sleeve increase, making it more likely that thermal cracks will develop.
- High temperatures in the upper part of the cylinder sleeve can significantly degrade the lubricating capacity of the oil and thus cause significant wear on the piston springs and the piston body.

- **Leaks**: Leaks are a serious problem in diesel engines and occur in the fuel injection system, the oil network and the freshwater and marine networks [1,2,3-15].
- **Other damage**: They include knocks, clogging of fuel, oil and/or cooling water filters and degradation of fuel quality [1,2,3-15].

2 Operation Monitoring and Fault Diagnosis of Diesel Engines with Cylinder Pressure Measurement and Computational Simulation

2.1 Introduction

The role of the diesel engine in shipping is particularly important because it is one of the main ways of propelling ships. However, it is a highly complex system because it is the result of the cooperation of many subsystems. For this reason, the malfunctioning of even one of the subsystems is capable of bringing an entire ship to a standstill [4-10]. Several methods of fault monitoring and diagnosis have therefore been developed, one of which will be analysed in this chapter. This method concerns the measurement of cylinder pressure. A simulation model of a diesel engine and its use will also be described. The benefits of computational simulation are clearly greater than the experimental process but which makes the accuracy of the models the main concern [4-10].

2.2 The Logic of the Diagnostic Technique with Cylinder Pressure Measurement and Computational Simulation and its Advantages

The technique of diagnosing operating conditions and fault prognosis is primarily based on the measurement and processing of cylinder pressure measurement. This technology has been developed and evolved by various researchers but dominant in this field is the technology developed by Dr. Dimitrios Chountis, Professor of the H.M.P. [4-10]. This diagnostic technique is recommended because it has certain advantages compared to other methods [3-10]:

1) The technique is based on a thermodynamics-based simulation model of operation. This feature ensures the generality of application, since with the thermodynamic approach it is possible to describe any type of engine regardless of operating characteristics, dimensions, operating conditions, etc. [3-10]
2) For the application of the technique, a large amount of engine operating data, recorded during the engine's lifetime, in normal operation or under failure, is not necessary [3-10].
3) It provides results for the state of the engine and its subsystems, such as the fuel injection system and the supercharger [3-10].

4) It provides indications for engine tuning (e.g. injection timing, valve timing, etc.), which is not possible when using diagnostic methods such as vibration analysis [3-10].
5) With the present methodology (i.e. simulation model), in addition to identifying a failure, the cause of the failure can be determined [3-10].

2.3 Measurement of the Cylinder Pressure Dynamic Chart

The pressure in the cylinder is a valuable source of information about the processes taking place in the combustion chamber [3-10]. Processing the measured pressure can provide important information such as maximum pressure, indicated pressure and indicated average pressure. However, more complex calculations can be performed to predict the air mass velocity, heat release rate, ignition angle, combustion time and compression quality [3-10]. The accuracy of the measurement of the casing pressure is critical for the reliability of the results obtained with these techniques. However, despite their widespread use, their measurement is subject to technical difficulties and potential sources of error, such as the capacity and weight of the measuring instrument, electrical noise and external power supply [3-10].

2.3.1 Piezoelectric Sensor

Initially, cylinder pressure measurements were performed mechanically, providing a graphical representation of the cycle in the form of pressure - volume and pressure - crank angle [3-10]. The recording is based on primary mechanisms and therefore is relatively accurate and reliable and as a result is still in use. However, the potential for exploiting these diagrams is limited. Nowadays, modern measurement acquisition systems are based on the use of piezoelectric crystals and optical sensors [3-10]. Combustion pressure is measured using the sensor that converts the pressure into a measurable quantity, usually a potential difference [3-10].

Figure 47. Piezoelectric sensor [3]

This voltage is digitised and supplied to the computer. The processing of the measured voltage value sequence to produce a pressure value sequence with a correct crank angle reference is a very important issue. Equally critical is the measurement setup, which includes all the connections and components to enable the measurement to be carried out [3-10]. Figure 8 illustrates a typical set-up for combustion pressure measurement.

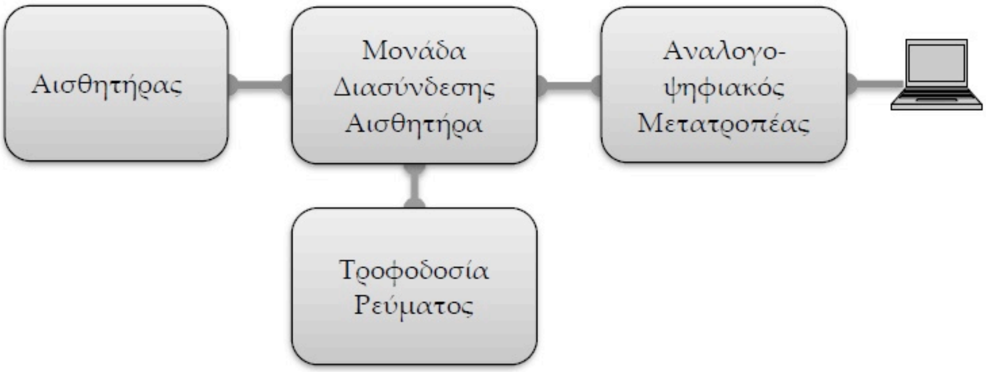

Figure 48. Schematic illustration of the device with combustion pressure measurement [1]

The almost exclusive use of piezoelectric sensors for combustion pressure measurement is a consequence of their superiority over other sensors in terms of technical characteristics [3-10]. Piezoelectric sensors exhibit very good accuracy, wide measuring range, good thermal characteristics, robustness and small size [3-10]. In addition, they combine high sensitivity due to the wide output signal range (electrical voltage or load, depending on the device) with respect to the measurement scale. The modulus of elasticity of the sensing elements is very high resulting in zero strain under pressure [3-10].

As a result, the sensors are very robust, have a very high natural frequency and particularly good linearity over a wide operating range. They are unaffected by electromagnetic fields and radiation. An important advantage of piezoelectric sensors over resistance sensors - whose resistance varies as a function of pressure - is the operating temperature range, which for the former reaches up to 367^{o} C and for the latter up to 157^{o} C [3-10]. Piezoelectric sensors are superior to optical sensors because of their higher accuracy [3-10].

The main disadvantage of piezoelectric sensors is their inability to measure pressure absolutely, which creates the need to reference the pressure to a known value [3-10]. At the same time, a second problem is the pressure thermal drift, which occurs during the measurement and is due to the change in the temperature of the sensor. This problem is addressed during the processing of the measured signal [3-10].

The measurement by means of the piezoelectric sensor is based on the solution of the following differential equation of 2^{ou} degree [3-10]:

$$\frac{a_o}{a_b} = \frac{1}{\sqrt{\left[1 - \left(\frac{f}{f_n}\right)^2\right]^2 + \frac{1}{Q^2}\left(\frac{f}{f_n}\right)^2}} \quad (1)$$

Where:

- fn: natural frequency of the sensing element (Hz)
- f: frequency (Hz)
- ao: sensor output
- αb: reference output at resonant frequency (for f= fn)
- Q: amplitude increase factor at resonant frequency (10~40) [1]

2.3.2 Sensor Interface Unit

The sensor interface module is the necessary circuitry to convert the pressure into a measurable electrical voltage. The high resistance of the piezoelectric sensor is an unfavourable factor for signal transmission, mainly due to losses in the conductors and the introduction of noise [3-10]. For this reason, most manufacturers place, in industrial sensors, a buffer very close to or inside the sensor housing [3]. This buffer converts the intensity load generated by the excitation of the sensor crystal, which is proportional to the pressure. Also to reduce the wiring, it is chosen to carry the signal and power from the same pair of conductors [3-10]. The interface circuit provides a constant current supply to the sensor using a current source, filters the DC component at the output of the circuit using a capacitor, and protects the inputs and output from overload using capacitors of special specifications [3].

2.4 Overfill Pressure Measurement - Scan Pressure

The measurement of sweep pressure has many applications in diagnosis. Among other things, it is used to estimate the mass of air trapped in the cylinder to be used in combustion. It is also necessary, together with the rotational speed, to diagnose the operation of the supercharging couple [3-10].

Piezoelectric sensors used in the measurement of combustion chamber pressure have very good accuracy in measuring pressure changes. In contrast, they do not provide the same accuracy in measuring the absolute value of pressure [3-10]. For this reason, various methods are applied to measure or estimate the pressure at some crank angle so that with this reference the pressure values in the measured diagram correspond to the absolute values [3-10]. The measurement of the sweep pressure gives this possibility, as we can equate with it the values of the pressure diagram at specific intervals of gas exchange (during the scrubbing) [3-10].

From the above, the usefulness of measuring sweep pressure for the diagnostic method becomes clear. The device by which the sweep pressure is measured is approximately the same as the device used to measure the combustion pressure [3-10]. That is, it consists of a sweep pressure sensor, the sensor interface module and a signal digitising module [3].

2.5 Description of Integrated Combustion Pressure Measurement Device for Industrial Application

In order to carry out an investigation in either a laboratory or industrial environment, it is necessary to develop the appropriate measurement chain, which consists of the sensors, the corresponding interface units, the power supply units, the analogue-digital converter and the computer for the acquisition, processing and storage of the measurements. Key requirements for application in both a naval and commercial ship environment, in addition to reliability, are portability, usability and robustness [3-10]. Figure 9 shows an integrated portable measurement acquisition system consisting of the measurement acquisition unit, computer, sensor and cables (in corresponding storage space).

Figure 49. Portable system for taking pressure measurements of a cylinder pressure measurement E.M.P. [3]

Figure 10 shows the analog-to-digital converter and the interface and power supply circuit. This device constitutes the measurement acquisition unit, which, together with the sensor and the connection cable, is sufficient for making measurements [3].

Figure 50. Portable system measurement acquisition unit [3]

2.6 Processing of Primary Measurements Relevant to Diagnosis

This chapter systematizes and develops methods for processing the primary measurement data in order to express them in terms of the measured quantity as a function of crank angle [3].The processing of this data involves the measurement of many quantities whose variation within the cycle is important such as combustion pressure, instantaneous rotational speed, injection pressure, sweep pressure and temperatures [3].

2.6.1 Setting digitisation parameters

The digitisation of a sensor signal is done by a digital-to-analogue converter, which is clocked by an internal counter. The measurements obtained are determined by a crank angle encoder, but in practical applications, without an encoder, a fixed time-frequency sampling rate is used [3-10].

There are two cases of sampling, with a fixed time step and a fixed angle step. The determination of the number of samples is calculated as a function of the desired number of cycles to be recorded and the necessary number of cycles depends on the application [3-10]. After recording a number of cycles, an average cycle of these successive cycles is estimated. The use of the average cycle reduces variation due to noise or sampling error [3-10].

2.6.2 Determination of upper dead centre

The reporting of measured points versus crank angle is one of the most critical procedures, as small errors give very large deviations in the estimation of various quantities such as indicated power etc. [3-10]. There are various methods of determining the upper dead centre (ABM) both metrological, using sensors, and computational, by solving a system of equations [3-10]. The measurement processing procedure varies depending on the method. The objective of this measurement processing step is to produce a set of position values (digitized sample serial number) corresponding to the piston passes through the ANS [3-10].

2.6.3 Noise removal

In field measurements there are many possible sources of parasitic noise, mainly electrical [3]. Proper design of the measurement setup with proper application of grounding significantly reduces parasitic noise, but it is not possible to remove it completely without the application of filters [3]. The application of filters can be done either digitally, i.e. by appropriate computer processing of the measurement, or analogously by processing the analog signal before digitizing it. Digital filters are superior to analog filters because they are more flexible in terms of tuning, more reliable because they do not rely on components that wear out and of course do not add any additional cost [3].

The usual source of noise in combustion chamber pressure measurement is electrical in nature and comes from the potential difference at different points considered to be ground loop, namely the difference between the potential of the grounding of the power supply of the measuring device and the potential of the metallic parts of the engine [3]. In the case where the computer operates with an accumulator the noise is much lower and is due to the variation of the reference voltage (ground) during differential measurement. The proper design of the measurement setup and in particular the sensor power supply circuit can eliminate it [3]. A source of noise other than the grounding level may be electromagnetic interference from a source close to the measuring device (electric motor, transformer) [3].

In addition to purely electrical noise, other sources of pressure signal anomalies include digitization errors and dynamic effects near the sensor position [3]. For example, there are several techniques for noise removal and smoothing of the combustion pressure signal such as using the mean cycle, applying filters and analyzing the signal in the frequency domain using the discrete point Fourier transform [3].

2.6.4 Matching measured pressure values to crank angle

Using the calculated ANS positions, each sample is referenced to a crank angle. For this purpose, the following steps are followed [3]:

1) isolation of the usable part of the measurement (in this step the initial and final part of the measurement is removed to isolate integer cycles)
2) development of the time series of the measured quantity
3) estimation of the average rotational speed per revolution
4) linear reduction of time to crank angle

2.6.5 Sensor Thermal Current Correction

The correction of the sensor heat flux is discussed in this section for the case of combustion pressure [3]. Piezoelectric sensors are highly responsive to rapid pressure changes. The signal they produce contains a constant component (DC), which does not correspond to the measured pressure, i.e. they show weakness in measuring the absolute value of the pressure, which must be reported in another way [3]. The constant component, however, varies during the measurement as a result of sensor temperature and electrical phenomena (e.g. capacitor charge levels of the sensor interface). The movement depends on the time during which the sensor is attached to the vent [3]. This phenomenon, which is mainly due to the heating of the sensor, is called thermal drift [3].

The correction of the thermal displacement is done in the dynamometer diagrams. To apply it, the positions on the dynamometer diagram at which the intake and exhaust valves (or ports) are open, i.e. the scavenging period, must be isolated. This is easily done by knowing the timing of the port valves and having a "rough" estimate of the crank angle during the measurement [3]. A line is drawn through the position and pressure value pairs whose equation is determined by the least squares method. The vertical correction of each measured value V, and hence the measured correction of each measured value, is calculated from the directivity of the line, α (slope) [3].

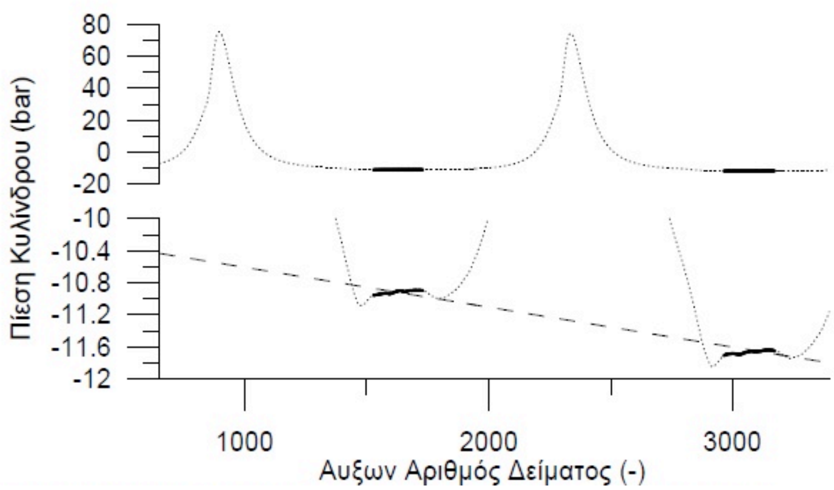

Figure 51. Thermal displacement correction methodology based on the calculation of the inclination of a line passing through points of the dynamometer [3]

2.6.6 Conversion of Measured Voltage Values to Pressure Values.

The measured voltage values must be converted into pressure values. The sensors shall be accompanied by a calibration certificate stating the conversion factor of the measured voltage into pressure values [3]. However, after the conversion the absolute value of the pressures shall be referenced to a known reference value. This reference refers to the vertical movement of the dynamometer diagram and is referred to in the literature as 'pegging' [3]. A simplistic and accurate approach is to move the dynamometer so that the measured value of the sweep pressure, or atmospheric pressure in the case of a naturally aspirated engine, is identical to the value of the dynamometer pressure in the phases when the intake and exhaust valves are simultaneously open (purging) (Figure 11) [3]. Several reference methods have been proposed and evaluated. Some require absolute pressure reference while others use the multimodal compression curve [3].

Finally, the pressure P of the random sample I is given by the equation:

$$p_i = cV_i \quad \left(cV_{peg} \quad p_{peg} \right) \tag{2}$$

where C is the sensor constant for voltage to pressure conversion, $V_{P,peg}$ is the average of the digitised voltage at the points used for reference and Ppeg is the reference pressure corresponding to the previous points.

2.6.7 Calculating the average circle and referencing it to a crank angle

To limit the non-deterministic components of the measured pressure, it is common to derive results using the mean cycle. Having assigned crank angle values to the pressure values, this is easily done as follows [3]:

1) Development of series "P - θ" with fixed crank angle step Δθ (By linear interpolation a new series "P' - θ'" with θ' multiples of Δθ is constructed from the series "P - θ").
2) Average cycle calculation [3]

2.7 Diesel Engine Simulation Model Summary Description

2.7.1 General description

At the heart of the diagnostic method is a multi-zone combustion model based on the fact that thermodynamic simulation can describe the combustion processes occurring in the cylinders of different types of diesel engines [5]. The multi-zone module has replaced the simpler two-zone module previously used for diagnostics in the diagnostic process as it more accurately describes the mechanism by which fuel mixes with air in the cylinder and takes into account the effects of engine geometry and fuel injection characteristics [5]. This made it possible to apply this module to different types of diesel engines without changing the engine speed, thus allowing "constant tuning" according to the engine speed. Since the diagnostic technique is based on evaluating the value of the Modell coefficient, it is important that the Modell coefficient is "fixed tuned" according to the operating conditions [5].

2.7.2 Computer simulation of the processes taking place inside the cylinders

2.7.2.1 Gas Cylinder Heat Loss Model

The instantaneous rate of heat loss of the gas to the cylinder walls is calculated from the following relation [4-10]:

$$\dot{Q} = A \;\; h_c\left(T_g \quad T_w\right) + c_r\left(T_g^4 \quad T_w^4\right) \tag{3}$$

where the heat transfer coefficient h_c is calculated from the following relationship [4-10]:

$$h_c = c\,Re^{0.8}\,Pr^{0.33}\,\frac{\lambda}{l_{car}} \tag{4}$$

The average temperature of the gas inside the cylinder at each degree of crank angle is calculated from the following relationship [4-10]:

$$T_g = \frac{\sum_{i=1}^{n_z} m_i c_{vi} T_i}{\sum_{i=1}^{n_z} m_i c_{vi}} \quad (5)$$

where 'i' indicates the respective fuel bundle band number. The total number of fuel bundle zones is n_z [4-10].

2.7.2.2Gas Leaks to the Cervical Chamber

An important parameter for the diagnosis of diesel engines is the calculation of the gas leakage from the combustion chamber to the crankcase because it affects both the compression quality and the quality of the exhaust. In the present analysis a simplified procedure is used in which an equivalent leakage area (A_{eq}) is used which is equal to [4-10]:

$$A_{eq} = \pi D \delta r \quad (6)$$

Furthermore, the equations of isentropic compressible flow are used to estimate the flow of gas leaks to the crankcase. In the previous expression δr is the equivalent cylinder-spring tolerance which defines the wear level of the piston-spring interface of the piston-sleeve cylinder [4-10].

As long as combustion has not started, leaks are only from the cylinder air. When combustion has started and the flame has spread throughout the cylinder then the contribution of each fuel zone to the configuration of the exhaust gas leaks to the crankcase is as follows [4-10]:

$$dm_{bl,i} = dm_{bl,tot} x \frac{m_i}{m_{tot}} \quad (7)$$

where $dm_{bl,tot}$ is the total exhaust gas leakage rate, m_i is the mass of each zone and m_{tot} is the total instantaneous cylinder gas mass [4-10].

2.7.2.3Fuel Batch Simulation - A Polysonic Approach

The multi-zone model is based on the assumption that each fuel bundle exiting the injection nozzle is divided after injection into separate control volumes called 'zones' in three dimensions relative to the injector [4-10]. The number of zones into which each fuel bundle is axially divided depends on the injection time and the calculation step chosen each time in the omputational model [4-10]. In general, the fuel beam is divided into five bands in the radial direction and eight bands in the circumferential direction for an integration step with a crank angle of 0.5° [4-10]. A schematic diagram of the fuel beam splitting into bands is shown in Figure 12 [4-10].

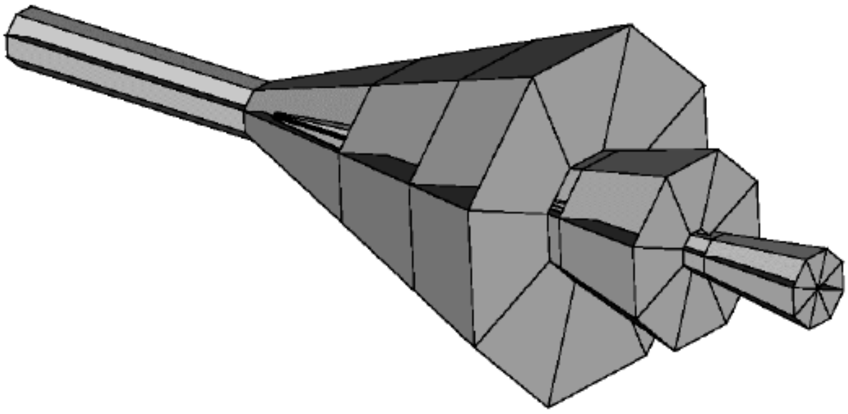

Figure 52. Schematic illustration of the three-dimensional segmentation of the fuel beam into zones (multi-zone view) [4-10]

The fuel beam penetration is calculated from empirical correlations that give the expansion velocity of each fuel beam in its axial direction as follows [4-10]:

$$u = u_{inj} = c_d \; \frac{2\Delta p}{\rho_l}^{0.5} \quad \gamma\iota\alpha \; x<L$$
$$u=u_{inj} \; \frac{L}{x}^{n} \quad \gamma\iota\alpha \; x \quad L \qquad (8)$$

where the fuel beam decay length L is given by the following expression [4-10]:

$$L = u_{inj} t_{br} = c_1 \; \frac{\rho_l}{\rho_a}^{0.5} \; d_{inj} \qquad (9)$$

where c_1 is a constant and ρ_α and ρ_l are the densities of the cylinder air and fuel [4-10].

2.7.2.4Calculation of Air Entrainment Rate from Fuel Bands

The air entrainment rate of each fuel bundle is calculated from the following formulation of the momentum conservation principle [4-10]:

$$m_f u_{inj} = \left(m_a + m_f\right) u_p \qquad m_a = \; m_f \frac{u_{inj}}{u_p} \quad m_f \qquad (10)$$
$$m_{a,cor} = c_a m_a$$

The air entrainment rate constant "c_a " is used to adjust the overall entrainment rate taking into account the geometry of the combustion chamber and especially the quality of fuel droplet dispersion within the cylinder air [4-10].

2.7.2.5Fuel vaporisation

The injected fuel is distributed in zones containing a certain number of fuel droplets whose mean Sauter diameter (SMD) is calculated from empirical correlations based on the cylinder air density, fuel characteristics and injector nozzle geometry [4-10]. The fuel vaporization rate is calculated from the Borman and Johnson model considering both the sensible heating of each fuel droplet and the latent heat of vaporization of the fuel [4-10].

2.7.2.6Cylinder Scanning Process

The scavenging process is very important for two-cylinder inverted diesel engines [4-10]. Therefore, the diagnostic method applies the 2o-zone cylinder scavenging model where the cylinder pit is divided into two parts; one part receives only fresh air and the other part receives exhaust gases and fresh air from the previous cycle [4-10]. In this new approach, part of the intake air escapes directly into the exhaust manifold (shaltcirculation), which directly affects the exhaust gas temperature [4-10]. dma,inl is the volume of air entering the cylinder, part of which enters the new cylinder and the rest into the exhaust cylinder. These masses are given by the following relationship [4-10]:

$$\begin{aligned} dm_{a,fz} &= dm_{a,inl}\left(1 \quad C_{1scav}\right) \\ dm_{a,cz} &= dm_{a,inl} x C_{1scav} \end{aligned} \tag{11}$$

If $dm_{g,exh}$ is the total amount of exhaust gas in the exhaust manifold this is taken partly from the fresh air zone and from the combustion products. These masses are given by the following relationships [4-10]:

$$\begin{aligned} dm_{g,fz} &= dm_{g,exh} \cdot \ C_{2scav} \\ dm_{g,cz} &= dm_{g,exh}\left(1 \quad C_{2scav}\right) \end{aligned} \tag{12}$$

where C_{1scav} and C_{2scav} are constants of the scan model. At the end of the sweep process (start of the substantial compression path) perfect mixing of the two zones is assumed leading to the formation of a single gas zone [4-10]. The constants are calculated using an iterative procedure to verify the measured exhaust gas temperature before the supercharging turbine [4-10].

2.7.2.7Stroviloubereputy

It is usually very difficult to obtain operating maps of compressors and turbines because these data are not provided by engine manufacturers or turbocharger system manufacturers [4-10]. Therefore, in this diagnostic method, the compressor and turbine maps are reproduced by a similarity estimation method using experimental data from acceptance tests of each diesel engine [4-10]. The method gives satisfactory results for engine loads from 40% to 100% and the ovomial coefficients are calculated using the least squares method to match the ovowing values [4-10]:

$$\begin{aligned} n_{Cis} &= f_1\left(\varphi\right) \\ n_{Tis} &= f_2\left(\varphi\right) \\ k_{is} &= f_3\left(\varphi\right) \quad \Delta h_{is}/U^2 \end{aligned} \tag{13}$$

where: $\varphi = \dot{m}/\rho AU$ is the reduced flow coefficient. The data required are as follows [4-10]:

- Pressure before and after the overfill compressor.
- Pressure before and after the turbocharger turbine.

- Air temperature before and after the compressor.
- Exhaust gas temperature before and after the turbocharger turbine.
- Speed of rotation of the supercharger shaft.
- Air and exhaust gas mass flow rate calculated from the computer simulation using the previous data.

2.7.2.8Air Overfill Refrigerator

The pressure drop Δp_{ac} and the degree of utilization "e" of the supercharged air cooler are calculated as functions of the air mass flow rate from the following relationships [4-10]:

$$\varepsilon = 1 \quad b\dot{m}^2 \tag{14}$$

$$\Delta p_{ac} = a_c \dot{m}^2 \tag{15}$$

where the degree of utilisation of the air cooler is defined as:

$$\varepsilon = \frac{T_{air,in} \quad T_{air,out}}{T_{air,in} \quad T_{w,in}} \tag{16}$$

The constants "a_c " and "b" are calculated from the simulation model constant determination procedure based on the data of the acceptance test of each engine [4-10].

2.7.3 Description of the Diagnostic Method with Cylinder Pressure Measurement and Thermodynamic Simulation

Data from measured engine parameters such as pressure, temperature, etc. are often used for diagnostics, but since these parameters are affected by many subsystems, it is difficult to determine the actual cause of an engine malfunction or failure [4-10]. For example, low peak canboost pressure can be a result of reduced fuel supply, faulty injectors, incorrect injection advance, low overfill pressure or increased gas leakage [4-10]. Therefore, methods should be developed to identify the real causes of failures and malfunctions [4-10]. For this purpose, a step-by-step method is applied to isolate the parameters affecting compression, compression, venting and gas exchange. The method is based on known multi-zone combustion modeling [4-10]. First, this odel is calibrated to predict the data obtained from the engine during the acceptance test (see Table 1). During this process the reference onstants of the odel are calculated and the resulting simulation is the 'new engine' [4-10]. Table 2 shows the odel constants calculated in the calibration procedure based on acceptance test data [4-10].

A schematic of the calibration process is shown in Figure 13, where Xj is the input data of the simulation model, e.g. engine speed, Ycal,j is the result of the simulation model and Yexp,j is the measured value. The variable βj is the simulation model constant calculated by the calibration procedure [4-10]. This procedure is repeated for the current operating environment and new simulation engine constants are calculated [4-10]. Such a simulator is called a "Current Machine". A failure or a malfunction exists in the machine when the following criterion [4-10] is satisfied:

$$\left|\frac{\beta \quad \beta_0}{\beta_0}\right| \cdot 100\% \quad 3\% \tag{17}$$

where β is each constant of the "Present Machine" and β_0 is the corresponding constant of the "New Machine" simulator. The threshold of 3% is used to account for measurement errors present during field measurements e.g. on the ship. The state of an engine system or an engine part is given by the relation [4-10]:

$$\frac{\beta}{\beta_0} \cdot 100\% \tag{18}$$

Table 15. Receipt test data used to calibrate the diagnostic simulation model [5]

Basic engine data	Import/export system data
Engine Revolutions	Gas pressure before/after the turbocharger turbine
Fuel supply	Overdrive shaft speeds
Actual power	Air pressure before/after compressor
Maximum combustion pressure	Exhaust gas temperature before/after the turbine
Maximum compression pressure	Air inlet temperature in the compressor
Fuel pump rule position settings	Air temperature before/after the overfill air cooler
Variable Injection Timing (VIT) indicator settings	Water temperature before/after the overfill air cooler
	Air pressure drop in the overfill cooler

Table 16. Constants determined by the calibration procedure [5]

Fixed	Description
CR	Degree of compression
c	Constant calculation of the heat transfer coefficient
Δr	Equivalent piston spring - cylinder sleeve spring clearance
Tw	Cylinder wall temperature
Adel	Calculation constant of ignition delay
Ca	Constant calculation of the air entrainment rate of the fuel bundles
Kb	Constant calculation of the combustion rate
Aeff	Equivalent active cross section of turbine nozzle of turbine supercharger
Cp	Fuel pump condition assessment station
Aac	Constant calculation of air pressure drop in the overfill cooler
b	Constant calculation of the degree of utilization of the overfill air cooler

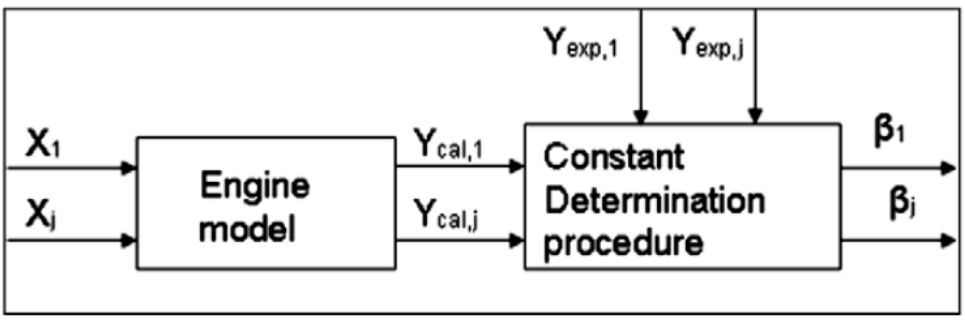

Figure 53. Philosophy of determining the constants of the simulation model [5]

The process of determining the constants for the simulation engine is divided into two parts. The first part is related to the crank cycle and the second part to the engine gas exchange and related subsystems [4-10]. Therefore, it is not easy to determine the ANS speed using a shaft encoder, especially for electric vehicles, unless there are significant production changes in a given machine [4-10]. At the same time, accurate determination of the ANS position is particularly important because a 1 degree crank angle error in determining the ANS position leads to a corresponding error of 8 - 10% in the calculation of the indicated cylinder power [4-10]. In this diagnostic method, the position of the ANS is determined using a thermodynamic method developed, evaluated and published by Dr. Dimitrios Chountis, Professor of the H.M.P. [4-10], which has an accuracy of 0.1 - 0.2 degrees of crank angle. The reliability of this method is attested in the present analysis by the accurate prediction of the power of each cylinder of both the main 2X engine and the electric motor [4-10].

The reference values of the reference constants of the simulation model "b_0 " are calculated using the values of the maximum combustion pressure and the maximum compression pressure of each cylinder obtained during the acceptance tests of diesel engines [4-10]. This is a standard procedure due to the fact that no cylinder pressure potential diagrams are obtained during the acceptance tests of main and auxiliary marine diesel engines. However, this is not a major problem because the engine is in excellent condition during the acceptance tests, e.g. it has minimal leakage and the compression ratio is known [4-10]. This is demonstrated by Figure 14 where a comparison is given between the acceptance measurements and the corresponding calculated values from the simulation model for the main 2-H marine diesel engine. As can be seen from the results in Figure 14, there is a very good agreement between the theoretical results of the simulation model and the acceptance tests [4-10].

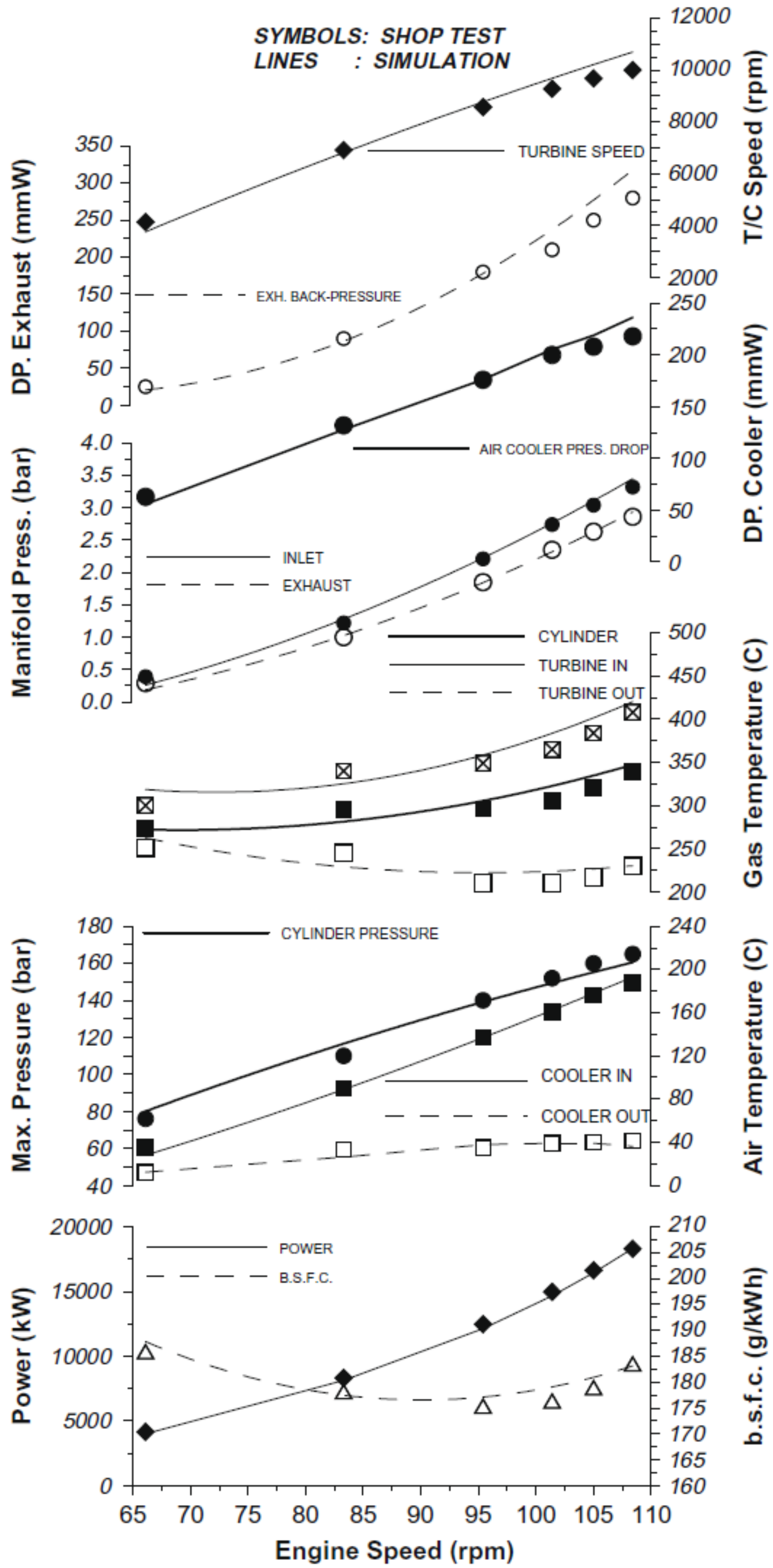

Figure 54. Comparison between receipt tests and results of the diagnostic simulation model [5]

On the other hand, for the present machine condition, the complete pressure dynamometer diagram for each cylinder is used [4-10]. In Figures 15a and 15b, comparisons of theoretical and experimental pressure values are shown for cylinder 1 of the main 2X marine diesel engine (Figure 15$^{\alpha}$) and cylinder 5 of the electric locomotive (Figure 15b) [4-10]. The very good agreement of theoretical and experimental pressure values for cylinder 1 of the main engine and cylinder 5 of the electric locomotive reveals the success of the process of determining the constants of the simulation model described previously [4-10].

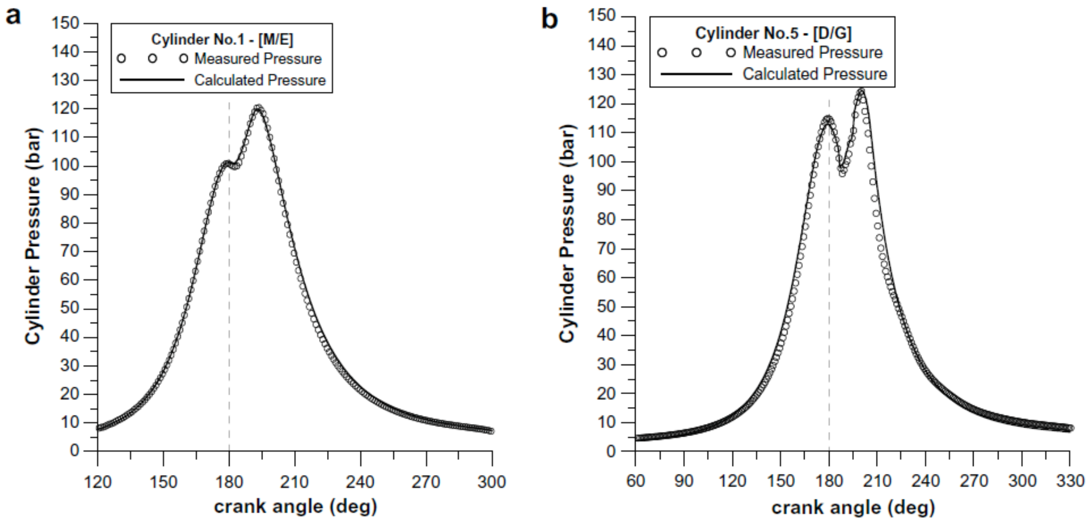

Figure 55.(a) Comparison between calculated and measured values of cylinder pressure for cylinder 1 of the main two-stroke marine engine after calculation of model constants from acceptance tests and (b) Comparison between calculated and measured values of cylinder pressure for cylinder 5 of the electric locomotive after calculation of model constants from acceptance tests [5]

2.7.3.1 Correlation between Engine Operating Parameters and Simulation Model Constants

The simpler diagnostic method presented here is based on the correlation between machine parts and simulated mockdel moments. This correlation has been previously validated with reference to the biconcave computational model with sensitivity analysis presented in detail in [4-10]. Here, the procedure is repeated to obtain similar results. The main difference concerns the constants associated with the ombustion mechanism, i.e. the use of multiple zone instead of one.From this procedure the main findings are as follows [4-10]:

2.7.3.1.1 Simulation Model Constants Primarily Related to the Compression Mechanism

The compression ratio CR has the greatest effect on the initial stage of the compression stroke while the constant δr, which provides the degree of wear of the sleeve-piston spring interface, has the greatest effect on the part of the duty cycle around the TDC having the most significant effect on the angle of occurrence of the peak compression pressure shifting its value to the left of the TDC [4-10].

The cylinder wall temperature has the same qualitative effect as δr but less quantitative effect [4-10].

The constant c mainly affects the last part of the compression and is obtained from the acceptance tests because it is characteristic for a specific type of machine [4-10].

Finally, the TDC pressure is thermodynamically estimated by the simulator's model constant determination procedure. The previous constants are estimated in such a way that the calculated cylinder pressures exactly match the measured values at all operating conditions of the studied engine. [4-10].

2.7.3.1.2 Simulation Model Constants Primarily Related to Combustion and Explosion

The constant a_{del} is related to the ignition delay and the constant K_b is related to the cylinder pressure gradient after ignition [4-10].

2.7.3.1.3 Simulation Model Constants Primarily Related to Fuel Injection System

The constant ca is related to the maximum injection pressure and is used to estimate the state of fuel distribution from the injectors. On the other hand, the oil rule o is used to determine the state of the fuel pump [4-10]. This is the standard method when the fuel injection pressure cannot be measured. Therefore, the fuel mass flow rate is related to the fuel pump index "yp" as follows [4-10]:

$$\dot{m}_f = c_p \rho_f y_p \tag{19}$$

where the constant c_p is related to the overall condition of the fuel pump.

2.7.3.1.4 Simulation Model Constants Primarily Related to the Import/Export System

The constant a_{ac} determines the degree of coalescence of the supercharged air conditioner, while the b constant is related to the efficiency of the air conditioner and is used to describe its performance [4-10]. The constant A_{eff} s related to the average pressure before the turbocharger turbine and represents the condition of the inlet nozzle of the turbocharger turbine (increased nozzle blackening or nozzle distorted cross section). Finally, the state of the compressor and turbine is estimated by the corresponding isentropic efficiency compared to the efficiency corresponding to the reference case using the coefficient in equation (13). are used to estimate the state of the compressor and turbine through the respective isentropic efficiencies compared to those corresponding to the reference state [4-10].

2.7.3.1.5 Estimation of Cylinder Fuel Consumption Rate

From the analysis of the combustion heat release rate, an estimate of the total amount of fuel burned during an operating cycle can be derived based on the following relationship [4-10]:

$$m_{finj} = \frac{Q_{g,cum}}{LHV} \tag{20}$$

where LHV is the lower heating value of the fuel and $Q_{g,cum}$ is the final sum of the cumulative combustion heat release rate, obtained by integrating the instantaneous gross combustion heat release rate given by the following relationship [4-10]:

$$\frac{dQ_{gross}}{d\varphi} = \frac{dQ_{net}}{d\varphi} + \frac{dQ_{loss}}{d\varphi} \tag{21}$$

2.7.4 Experimental Procedure for Taking Measurements on Marine Diesel Engines on Board

As has been shown above, the measurement of cylinder pressure is the most important part of this diagnostic method [4-10]. For this purpose, in this analysis, measurements of the pressure of all cylinders of both the main 2X marine diesel engine and the electric locomotive were performed with an air-cooled piezoelectric sensor (piezotron) mounted on the power valve of each cylinder of both engines [4-10]. Measurements of the pressure of each cylinder were taken at a sampling rate of 0.5 degrees crank angle and still a number of duty cycles were taken which ranged from 20 to 100 cycles depending on the speed of each engine [4-10]. From these cylinder pressure measurements for each duty cycle, the mean duty cycle dynamometer diagram was obtained which was used as input data in the diagnostic process [4-10]. In addition, the data given in the following table are necessary for the diagnosis and are obtained from conventional experimental equipment [4-10].

Table 17. Measurements obtained from local sensors mounted at various points on marine diesel engines [5]

No	Measured Parameter	Type of measuring instrument	Accuracy
1	Pressure manifold input	Industrial type absolute pressure transducer (piezoresistive)	± 0.5%
2	Multi-exhaust pressure	Industrial type absolute pressure transducer (piezoresistive)	± 0.5%
3	Exhaust gas outlet temperature from cylinder	Type K (Class 1) thermocouples	± 1.5^{o} C
4	Exhaust gas inlet temperature in the turbocharger turbine	Type K (Class 1) thermocouples	± 1.5^{o} C
5	Exhaust gas outlet temperature from the turbocharger turbine	Type K (Class 1) thermocouples	± 1.5^{o} C
6	Overdrive shaft speeds	Magnetic sensor	± 1%
7	Compressor inlet temperature	Type J (Class 1) thermocouples	± 1.5^{o} C
8	Compressor outlet temperature	Type J (Class 1) thermocouples	± 1.5^{o} C
9	Compressor inlet pressure	U-type tube (H_2 O)	± 1 mm
10	Overfill air cooler outlet temperature	Type J (Class 1) thermocouples	± 1.5^{o} C
11	Water temperature before overfill air cooler	Type J (Class 1) thermocouples	± 1.5^{o} C
12	Water temperature after the overfill air cooler	Type J (Class 1) thermocouples	± 1.5^{o} C
13	Air pressure drop in the air cooler overfill air cooler	U-type tube (H_2 O)	± 1 mm
14	Fuel pump indicator	Fuel pump mechanical rule position	± 0.2 mm

2.7.5 Application of the Diagnostic Method with Cylinder Pressure Measurement and Thermodynamic Simulation to a 2X Main Marine Engine and a Diesel Electric Engine

The basic construction and operational characteristics of both the main 2X marine diesel engine and the electric locomotive considered in this analysis are given in the following table. Table 5 shows the operating conditions of the two diesel engines under which measurements were made at sea [4-10]. The actual power of the main 2X marine diesel engine was calculated from an installed rpm meter and the corresponding actual power of the electric locomotive was calculated from an electrical power meter using the electrical efficiency of the generator determined during the acceptance tests of this engine [4-10].

Table 18. Basic construction and operational data of the 2X main marine engine and the 4X electric locomotive to which the diagnostic method with cylinder pressure measurement was applied [5]

	2-X Main Naval Diesel engine	4-X Electric machine Diesel
Number of cylinders	7	7
Cylinder diameter (mm)	600	210
Piston travel (mm)	2400	320
Degree of compression	19:1	17:1
Maximum power (kW)	15785	1120
bmep (bar)	19.0	24.1
Maximum combustion pressure (bar)	160	200

2.7.5.1 Application Results of the Diagnostic Method of Cylinder Pressure Measurement and Thermodynamic Simulation in a 2X Main Marine Engine and a Diesel Engine

2.7.5.1.1 Cylinder Compression Quality

An important parameter of the overall performance of an engine is the quality of the pressure in each cylinder [4-10]. Based on this parameter, faults and malfunctions related to the sleeve-piston-spring mechanism and valve leaks can be detected [4-10]. Valve leaks can be ignored if the exhaust gas temperature is correct. Figure 16 shows the compression quality of each cylinder of the main engine, the 2X marine diesel engine and the electric Roxitan. The compression quality is defined by the following relationship [4-10]:

$$n_{CQ} = \frac{CR_{cur}}{CR_{ref}} \cdot 100\% \tag{22}$$

For the compression quality n_{CQ} a value above 95% is ideal while a value up to 90% is acceptable. A value of compression quality less than 90% indicates the need for inspection of the corresponding cylinder [4-10]. These values are determined by the limits of variation of compression pressure values given in the manufacturer's manuals [4-10]. Figure 16 shows that all cylinders of the main 2X marine engine have a compression quality above 95% [4-10]. Cylinders No. 3, 5 and 7 have the highest compression quality. The situation is similar for the electric engine with the exception of cylinder No. 4 which has a compression quality of less than 95% [4-10]. Roller No. 7 has the best compression quality. For this reason, it is recommended to inspect cylinder No. 4 of the electric machine because its condition may deteriorate in the near future and this is expected to have a negative effect on the efficiency of this machine [4-10].

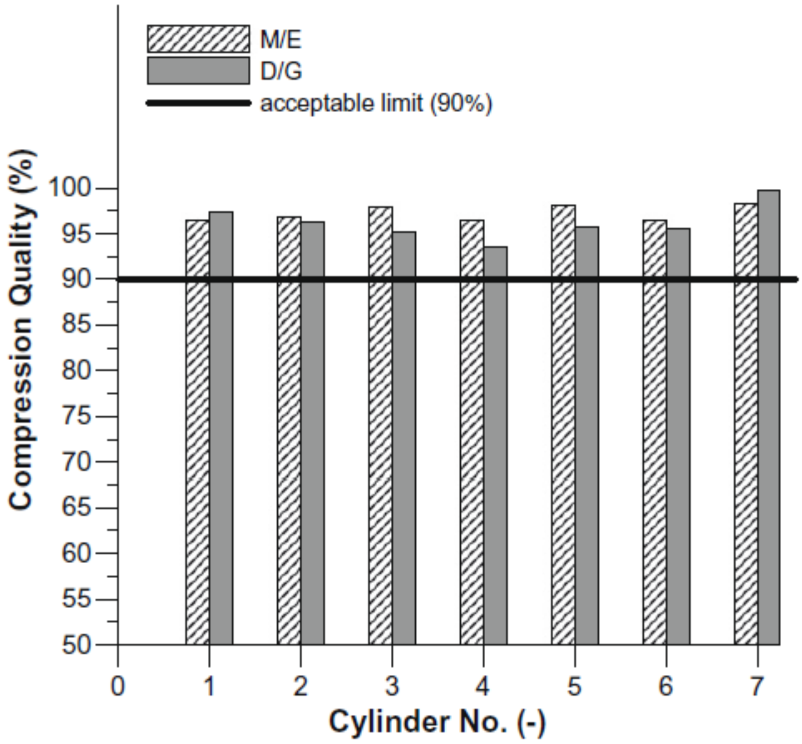

Figure 56. Compression quality of the main 2X marine diesel engine and electric locomotive [5]

2.7.5.1.2 Cylinder Fuel Consumption

As mentioned earlier, the fuel consumption of each cylinder is calculated from equation (20). Figure 17 shows the calculated fuel consumption of each cylinder for the main engine and the electric motor [4-10]. It can be seen that there is a small error in the case of the main engine. No. 5 cylinder has relatively high fuel consumption, while No. 6 and No. 7 cylinders have slightly lower fuel consumption [4-10]. In the case of electric systems, No. 3 and No. 6 cylinders have significantly higher fuel consumption rates than the other cylinders of the same engine [4-10]. This result is generally consistent with the fuel consumption of each cylinder, indicating that fuel consumption is the main reason for the difference in fuel consumption between cylinders [4-10]. Therefore, the fuel consumption of the two cylinders no. 3 and no. 6 cylinders of the two cylinders are proposed to be reduced. This is expected to have a positive impact on the smooth performance of the cylinder of the electro-cleaner [4-10].

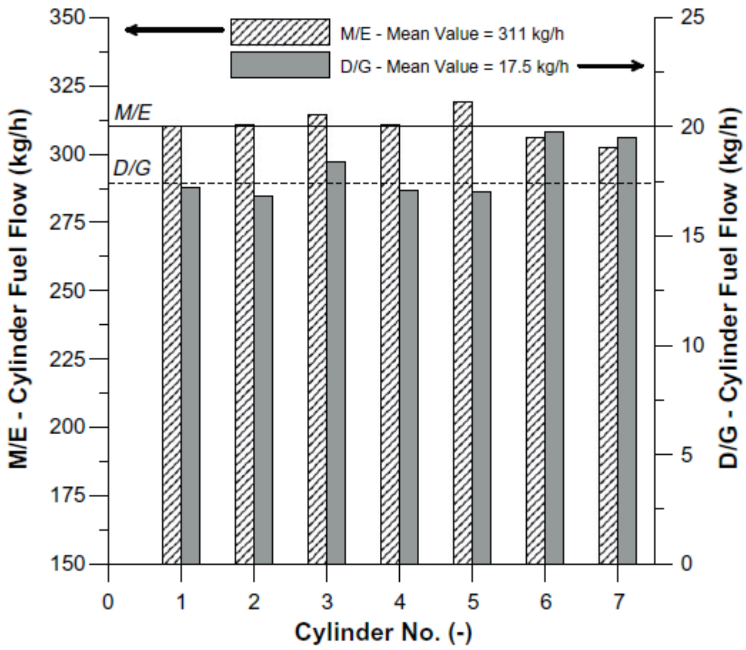

Figure 57. Estimated fuel consumption for all cylinders of both the main 2X marine diesel engine and the electric locomotive [5]

2.7.5.1.3 Angle of ignition and injection propulsion

The theoretical results for the ignition angle are given in Figure 18 for each cylinder of both the main 2X marine diesel and electric locomotive [4-10]. Having determined the ignition angle the simulation model is used to calculate the injection (advance) angle of each cylinder of each engine. An iterative procedure is used to estimate the injection advance so that ignition occurs at the corresponding angle given in Figure 18 [4-10]. The iterative procedure uses the average per step integration cylinder gas temperature as obtained by applying the final gas constitutive equation [4-10].

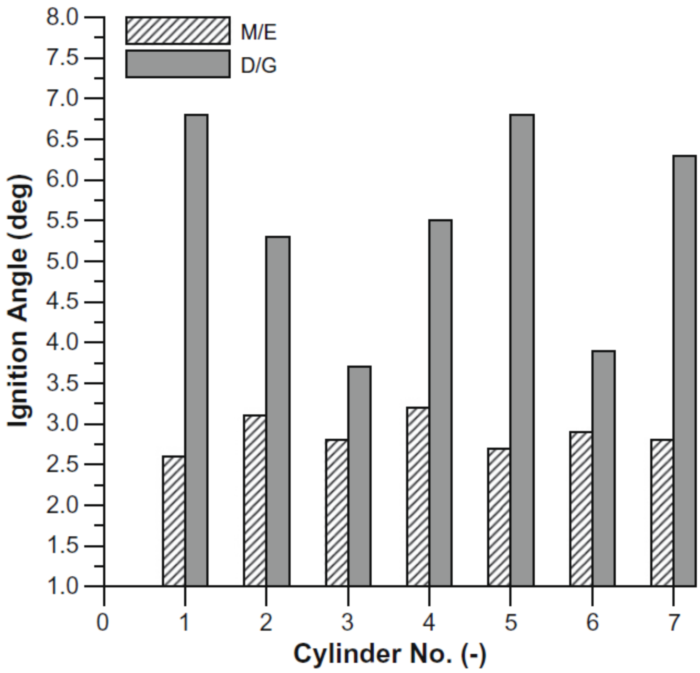

Figure 58. Estimated ignition angle of all cylinders of both the main 2X marine diesel engine and the electric locomotive [5]

Figure 19 shows theoretical results for the injection advance of each cylinder of both the main engine and the electric machine as well as the corresponding reference values as obtained from the acceptance tests [4-10]. For the calculated injection advance, a deviation of ±0.5 degrees of crank angle for the main engine and ±1.0 degree of crank angle is considered acceptable [4-10]. For the main engine no significant deviations are observed for the injection advance between cylinders revealing proper tuning of the fuel burners of all cylinders. On the other hand, they are observed in the inter-cylinder injection advance for the prime mover [4-10]. In particular, cylinders No. 1 and No. 5 have significantly higher injection advance values compared to the corresponding reference value and the other cylinders, while cylinders No. 3 and No. 6 have lower injection advance values again compared to the corresponding reference value and the other cylinders [4-10]. However, the previous differences cannot explain the variation of the maximum combustion pressure observed between cylinders because the latter is affected by the compression quality, injection rate and the condition of the fuel burner [4-10]. Therefore, it is proposed to adjust the fuel injection advance rate for the aforementioned cylinders of the electric motor to achieve a uniform injection start in all the cylinders of the electric motor [4-10].

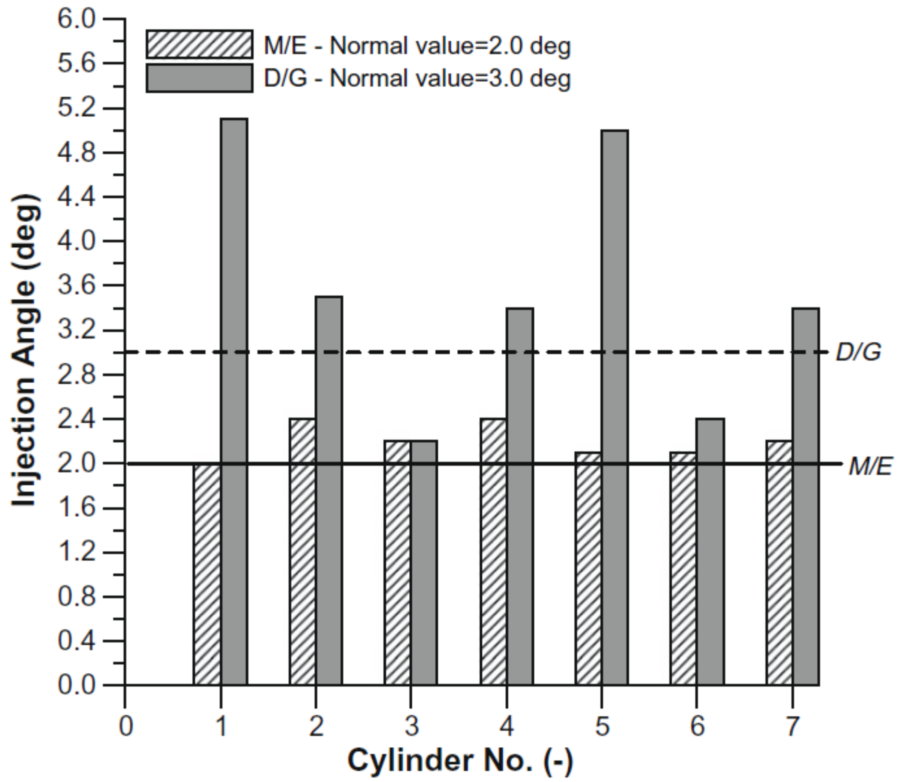

Figure 59. Estimated injection advance of all cylinders of both the main 2X marine diesel engine and the electric locomotive [5]

2.7.5.1.4 Fuel burner nozzle condition

Burner quality is an overall parameter that describes the condition of the fuel burner. The assessment of injector or burner quality is based on the following relationship [4-10]:

$$n_{IQ} = \frac{c_{a,cur}}{c_{a,ref}} \cdot 100\% \tag{23}$$

For the injector condition (n_{IQ}), a value above 95% is considered ideal, while a value in the range of 90 to 95% is acceptable [4-10]. Injector quality less than 90% indicates the need for injector inspection. These limits have been derived from computer simulation with the deterioration of specific fuel consumption bsfc and exhaust gas temperature as key criteria [4-10]. The results for injector quality are given in Figure 20 for each cylinder of both the main engine and the electric motor [4-10]. As observed, all injectors of the main engine have injector quality close to 100%. A similar situation is observed for the injectors of the electric machine with the exception of the injector of cylinder No. 3 which is slightly below the limit and the injector of cylinder No. 6 which has injector quality close to the limit [4-10]. For this reason it is recommended that the fuel injectors of cylinders No. 3 and No. 6 be inspected [4-10].

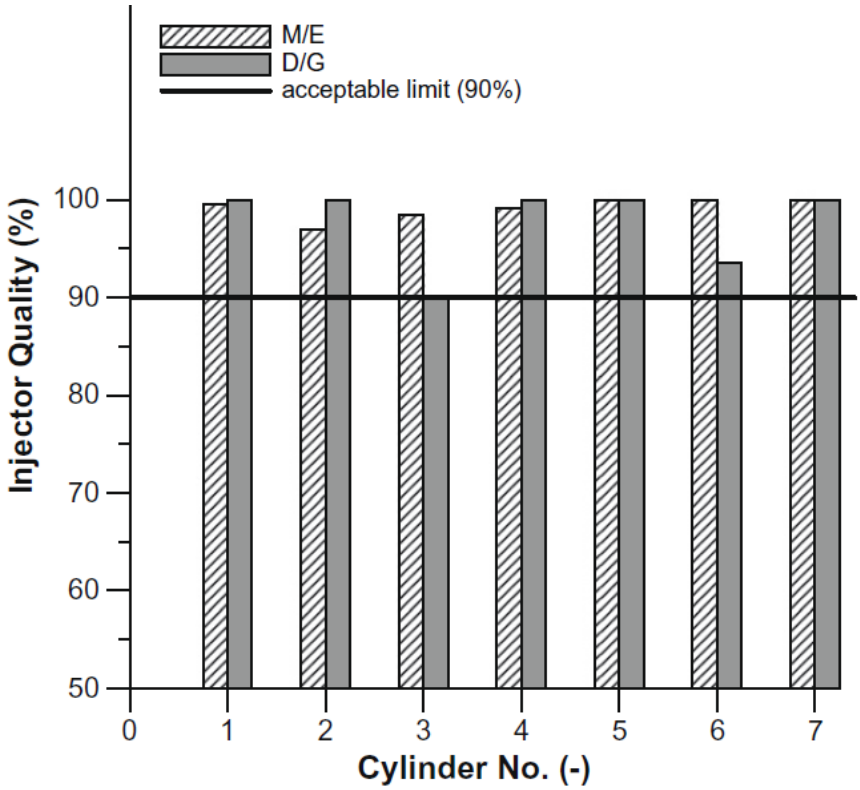

Figure 60. Estimated injector quality of all cylinders of both the main 2X marine diesel engine and the electric locomotive [5]

2.7.5.1.5 Overall Condition of the Main 2X Engine and Electric Machine

Based on the results of the diagnostic method based on the cylinder pressure measurement and the thermodynamic simulation, Table 5 showing the overall condition of the main engine and Table 6 showing the overall condition of the electric machine are formed [4-10]. The ideal values of each parameter are in green, the acceptable values of each parameter are in blue and the problematic values are in red [4-10].

A parameter often used for diagnosis is the exhaust gas temperature. Its value can indicate the presence of a problem, but not the actual cause of the phenomenon [4-10]. Therefore, using diagnostic results [4-10], an attempt was made to explain the deviation in exhaust gas temperature between the cylinders of the two engines, as shown in Figure 21. The figure shows that cylinder 7 of the main engine has a higher exhaust gas temperature compared to the other cylinder, although it has a slightly lower exhaust gas temperature [4-10]. The diagnostics show that this is not due to the cylinder structure or the injection system, but to the characteristics of the cylinder close to the turbocharger turbine, where the exhaust gases from all cylinders are concentrated [4-10].

The general conditions are shown in Table 6. No. 1 and No. 5 cylinders, the average exhaust gas temperature is lower than the thermal conductivity, this is because the injection advance of these cylinders is higher than the thermal conductivity values of the acceptance tests [4-10]. No. 3 and No. 6 cylinders is due to the injection advance in these cylinders and the quality of the injected material [4-10]. Finally, in the No. 7 cylinder has higher exhaust gas temperature due to higher fuel consumption and turbocharged turbine compared to other cylinders [4-10]. Therefore, using this method, actions based on exhaust gas temperature measurements can often be implemented, which may ultimately lead to incorrect cylinder tuning [4-10].

Table 19. Overall condition assessment of 2-X diesel main engine as derived from the diagnostic procedure [5]

	Main engine						
Cylinder number	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Compression quality	Ideal	Ideal	Ideal	Ideal	Ideal	Ideal	Ideal
Forward infusion	Ideal	Accepted	Ideal	Accepted	Ideal	Ideal	Ideal
Burner quality	Ideal	Ideal	Ideal	Ideal	Ideal	Ideal	Ideal
Fuel pump status	Ideal	Ideal	Ideal	Ideal	Ideal	Ideal	Ideal

Table 20. Overall assessment of the electromechanical condition as obtained from the diagnostic procedure [5]

	Electric machine						
Cylinder number	**1**	**2**	**3**	**4**	**5**	**6**	**7**
Compression quality	Ideal	Ideal	Accepted	Accepted	Ideal	Ideal	Ideal
Forward infusion	Problematic	Accepted	Accepted	Accepted	Problematic	Accepted	Accepted
Burner quality	Ideal	Ideal	Problematic	Ideal	Ideal	Accepted	Ideal
Fuel pump status	Ideal	Accepted	Accepted	Problematic	Ideal	Ideal	Ideal

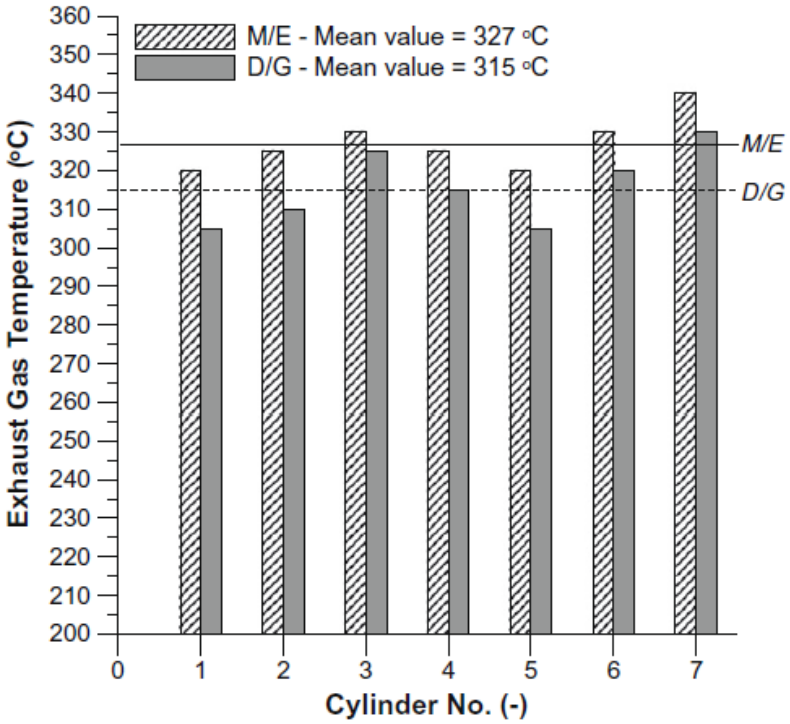

Figure 61. Measured exhaust gas temperatures of each cylinder of the main 2X marine diesel engine and the electric locomotive [5]

2.7.5.1.6 Condition of Turbine Excess Pressure and Air Cooler of 2X Main Marine Diesel Engine

The determination of the compressor and turbine condition is based on the comparison of the isentropic efficiency of the current engine condition with the corresponding reference values obtained from the analysis of the acceptance tests as follows [4-10]:

$$n_{C,cond} = \frac{n_{Cis,cur}}{n_{Cis,ref}} \cdot 100 \qquad (24)$$

$$n_{T,cond} = \frac{n_{Tis,cur}}{n_{Tis,ref}} \cdot 100 \qquad (25)$$

The results for the main engine are given in Table 7. No results are available for the electric machine due to the lack of acceptance tests which are necessary to estimate the reference values [4-10]. As shown in Table 7, the quality index of the compressor was calculated equal to 94%. The reduced compressor quality index compared to the reference value is an indication of wear or changes in compressor blade geometry [4-10]. A 6% reduction in the compressor quality index is considered to be acceptable considering the uncertainties of the respective measurements. On the other hand, the condition of the turbine was found to be excellent (turbine quality index equal to 100%) [4-10]. Finally, a small increase in the active flow surface of the turbine nozzle of the turbine overfill was detected. The increase in the cross-sectional area of the turbocharger turbine was equal to 3% and probably reveals the onset of exhaust gas leakage from the turbocharger turbine [4-10]. From the measured values of the inlet and outlet temperatures of the turbocharging air and water in the air cooler, the degree of utilization is calculated [4-10]. This is compared with that for a new engine operating in its present condition and thus a slight reduction of 5% in the quality index of the main engine's turbocharging air cooler is observed. This demonstrates little fouling of the surface of the turbocharging air exchanger (cooler) (Table 7) [4-10].

Table 21. Condition of supercharger and supercharging air cooler of 2X main marine diesel engine [5]

Estimated parameter (%)	**2-X Main Diesel Marine Engine**
Compressor quality indicator	94.0
Turbine quality indicator	100.0
Turbine nozzle surface area index	103.0
Overfill air cooler quality indicator	95.0

3 Operational monitoring and fault diagnosis of diesel engines with vibration measurement and analysis

3.1 Introduction

Vibration is an everyday phenomenon that is found in homes, in the workplace, during the operation of public transport and in many other situations. Usually, vibration is treated as an undesirable, annoying, destructive effect of a useful process, such as, for example, when a vehicle is moving on a road surface [11-13]. However, there are cases where vibration is useful for a specific purpose, such as when using an impact drill. For mechanical systems in particular, vibration occurs as a result of forces developed on the moving parts of a machine and on components connected to it. This chapter will present the method of vibration analysis as a technique for monitoring operation and fault diagnosis in diesel engines [11-13].

3.2 Literature review

The development of microcomputers and related software has completely changed the possibilities of taking measurements from diesel engines to monitor their condition and operational performance. Monitoring the operating condition of marine diesel engines is a high priority because the failure or catastrophic failure of a main diesel engine can lead to the loss of a ship [11-13]. Different failures that occur within a diesel engine and associated equipment that works with the diesel engine are usually associated with increased vibration levels [11-13]. Such failures include abnormal combustion resulting from a faulty valve or incorrect fuel injection process, wear or partial thermal stress on engine bushings, deformation of support bearings or the crankshaft-camshaft drive system, destruction of gearbox teeth, wear of bearings, failure or blockage of the supercharger, etc. [11-13].

Vibration measuring accelerometers can be placed at strategic points on a diesel engine. However, analyzing vibration signals is not an easy process especially with engines such as diesel engines, which can have high levels of vibration substrate [11-13]. Haddad et al. [11-13] analyzed and calculated the force received by the piston from combustion considering all possible piston positions using engine body vibration measurement and analysis. Nurhadi et al [11-13] investigated the correlation between vibrations measured by accelerometers installed in the body of a spark-ignition engine and the availability of engine components affected by the excitation of the measured vibrations. In their experiments, Nurhadi et al [11-13] used a spark-ignition motor with an electric motor via a V-belt to generate heterogeneous vibration and equalized compression and compression by scintillator motion. They also stated that analysis of the vibration signal can identify the cause of the vibration, such as gear tightness, timing chain tightness, and abnormal thermal expansion, etc. Gu et al [11-13] showed that defects in the injector change the vibration energy of the injection pulse, and based on this, the vibration energy of the injector pulse was changed, Thomas et al [11-13] also use pattern readentification techniques to detect tibial damage based on the vibration signal. Debotton et al [11-13] applied vibration signal analysis techniques to determine the operating condition of internal combustion engines. Accelerometers installed in spark-ignition engines were used. They also used a Fast Fourier Transformation (FFT) analyzer to convert the measurements from the time axis to the frequency axis and used this procedure to analyze the vibration signals of the engine during thermal and abnormal operation Grimmelius and Meiler [11-13] used a crankshaft shear strain signal by applying baseline signal analysis and maximum shear strain value analysis, and developed a pattern recognition algorithm to detect in-service failures of diesel engine cylinders. Teraguchi et al [11-13] investigated the effect of lubricant thickness between the piston spring and cylinder wall on induced vibration. Alhussain et al [11-13] analyzed the measured vibration and noise generated by piston motion and further investigated the effects of load, speed, and temperature on the measured vibration and noise on the measured values of vibration and noise, they obtained useful information on the quality of the lubricant and the stability of the diesel engine. Alhussain et al [11-13] described the noise of a diesel engine and investigated its basic characteristics on the time and frequency axes to determine the mean, curvature, and frequency of the noise. and also investigated its mean, curvature, and RMS values. They also identified and diagnosed incorrect exhaust valve and cam spacing by measuring the maximum RMS value of the noise signal for each cylinder [11-13].

3.3 Measuring chain

The development of the measurement chain is the most important process, and it is responsible for measuring, recording and storing the measured physical quantities on the device, because without measurements no diagnostic methods can be applied [11-13]

3.4 Vibration meters

A typical arrangement for measuring vibration starts with measuring transducers - vibration meters. In order to select the appropriate vibration transducers it is necessary to determine the frequency range from which the measurements are expected to be taken. This is determined by the rotational speed of the shaft. In addition, the dimensions of the mechanical system play an important role [11-13]. The environmental conditions (temperature, dust, noise, etc.) where the sensor will be used are equally important because in unfavourable environments, such as the engine room, instruments with specific properties are required. Finally, a factor to be taken into account is the need for portability. By portability we mean the need for easy transportation of the measuring chain in order to take measurements with the same equipment on different machines [11-13].

3.4.1 Preamp

The vibration signals require amplification in order for even the slightest changes to be discernible. This is the basic function of preamplifiers. Another important function is the integration of the amplifiers, so that the measurement of a single quantity can be used to derive the others and provide a complete picture of the state of the machine [11-13]. It is also important that the output impedance of each preceding stage is equal to the input impedance of each subsequent stage along the measurement chain in order to limit signal power losses. This is one of the functions performed by preamplifiers [11-13]. The output impedance of vibration meters in particular is relatively large, so it is necessary for the signal to pass through preamplifiers in order to reduce or input impedance for the subsequent low impedance stages that follow. The signals contain the information mixed with noise [11-13]. The information signal and the noise do not share common frequencies so that noise removal is possible using simple linear, time-constant, systems. These systems are called filters and are either built into the preamplifiers or form a separate link in the measurement chain. The vibration signals are analog, so analog filters are used. Therefore, the preamplifiers adjust the signal in a way that makes it possible and easier to analyse and follow the vibration meters in series in the measurement chain [11-13].

3.4.2 Analogue to digital converter

The analogue form of the signals requires an additional layer in the measurement chain, the Analog to Digital Converter (ADC) card [11-13]. The purpose of the analog-to-digital conversion card is to convert analog signals into digital information that can be managed by the computer. Its basic functional characteristics are [11-13]:

1) Number of input channels: Expresses the maximum number of input sizes on the card
2) Discretion: Determines the number of binary digits used to digitize the input voltage and is related to the accuracy of the measurements
3) The sampling frequency: Expresses the amount of time required to perform an analog-to-digital conversion (or sampling frequency fs). The selection of the appropriate sampling frequency is made according to how slowly or rapidly the measured quantity is changing. In practice, the sampling frequency is chosen to be at least 10 times the maximum frequency of the signal
4) The operating range: Determines the maximum and minimum voltage that can be given as an input to the card and can be handled by the analog-to-

digital converter. The operating range of the card should be selected based on the range of the input signal. In this way the available discriminating capability can reproduce the input signal as accurately as possible.

5) The minimum discrete signal variation: this value expresses the smallest signal variation that the card can handle/recognise. Obviously a smaller value of the minimum discrete signal variation leads to a better representation of the original signal. Note that the measurement cannot be more accurate than the sensitivity of the instrument/sensor.

Depending on the characteristics of the quantities (frequency, size, number, accuracy) and the number of quantities (channels) that need to be measured, the appropriate type of card is selected [11-13].

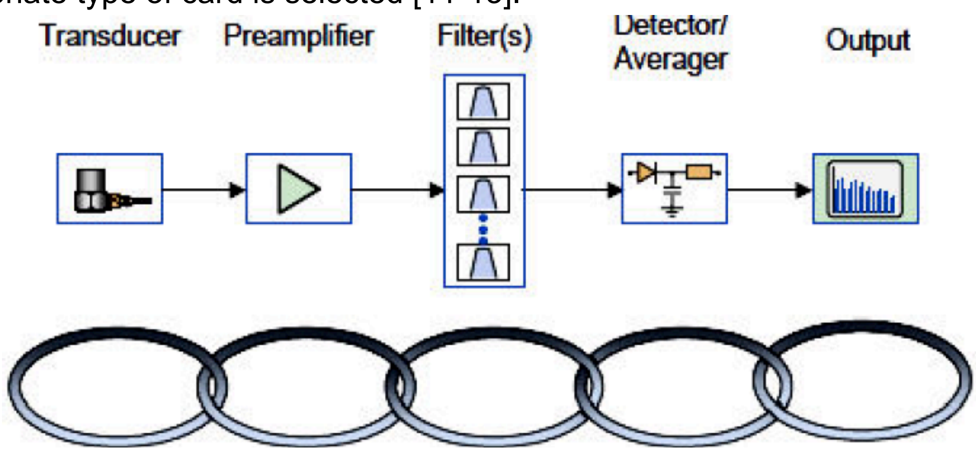

Figure 62. Schematic illustration of vibration measurement chain [11-13]

3.5 Diagnostics with Vibration Measurement and Analysis

In this section, dynamic faults, their basic signatures and the key quantities used to detect them through vibration analysis will be presented [11-13].

3.5.1 Typical Cases of Mechanical Failures Detected by Vibration Measurements

The mechanical failures that can be detected by vibration measurements, as represented in Figure 23, are [11-13]:

- Abdominal atrophy
- Bending axis
- Vibration of electromagnetic nature
- Instability, laxity
- Poor alignment
- Bearing wear
- Aerodynamic vibration
- Gear wear

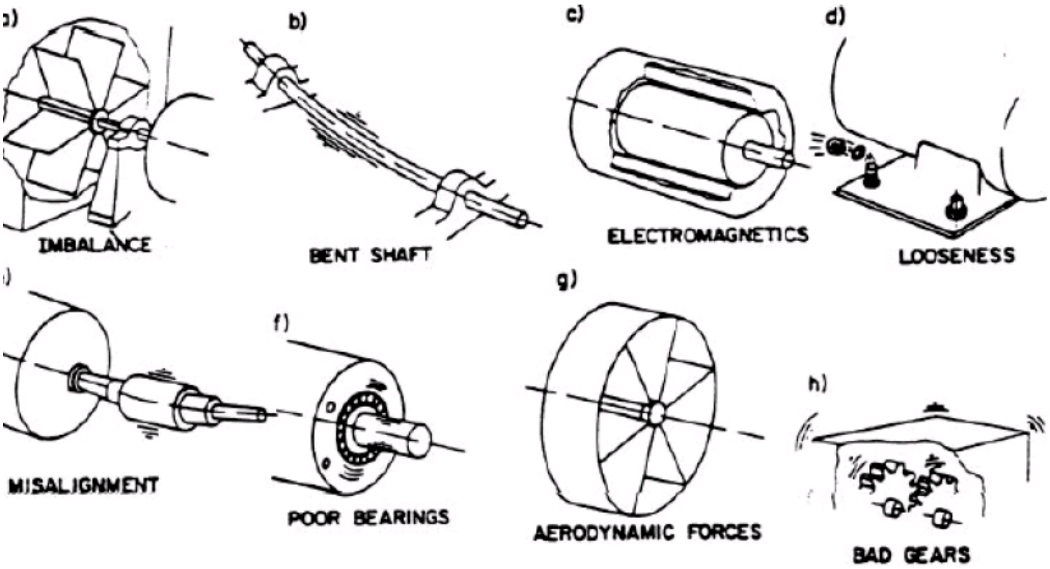

Figure 63. Mechanical failures that can be detected by vibration measurements [11-13]

3.5.2 Description of the Vibration Use Process

First, we place the vibrometer at the point where we want to take our measurement. Then, by means of suitable software, we are instructed to take measurements, the data of which are stored. This data expresses the acceleration of the vibration and using it we generate the power spectrum through which we can detect if there is any damage to the machine [11-13].

Figure 64. Mounting a vibration meter on a network section [11-13]

3.5.3 Measurement Chain Calibration

Calibration of the instruments is necessary before any measurement is made [11-13]. The aim of calibration is to verify the correct functioning of the measuring instruments in order to detect possible deviations a priori. In this way the results are reliable and the possibility of drawing incorrect conclusions is minimised due to the elimination of errors [11-13].

3.6 International Vibration Standards

The best known standards used for vibration analysis are [11-13]:

- **ISO 8528-9**: provides basic information on how measurements should be taken (appropriate temperature and frequency) and at which points exactly (motor shafts, bearings, etc.)
- **ISO 10816-6**: applies to engines above 100KW, but not to engines in road vehicles. It specifies the points, conditions and frequency range where the measurement must be taken.
- **ISO6954-2000**: deals with the effects of vibration on ships on people.

International vibration standards in general provide information on the exact locations where sensors should be placed, the frequency range and the temperature at which measurements should be taken [11-13].

3.7 Internal combustion engine vibration measurements

Measurements are taken at different positions on the engine. The engine block, generator, and engine frame have specific measurement points. Engine auxiliaries such as pumps and filters are also measured. When measuring an engine, the crankshaft of the engine must be on the vertical axis of the engine, and the same is true when measuring the shaft of a turbocharger [11-13].

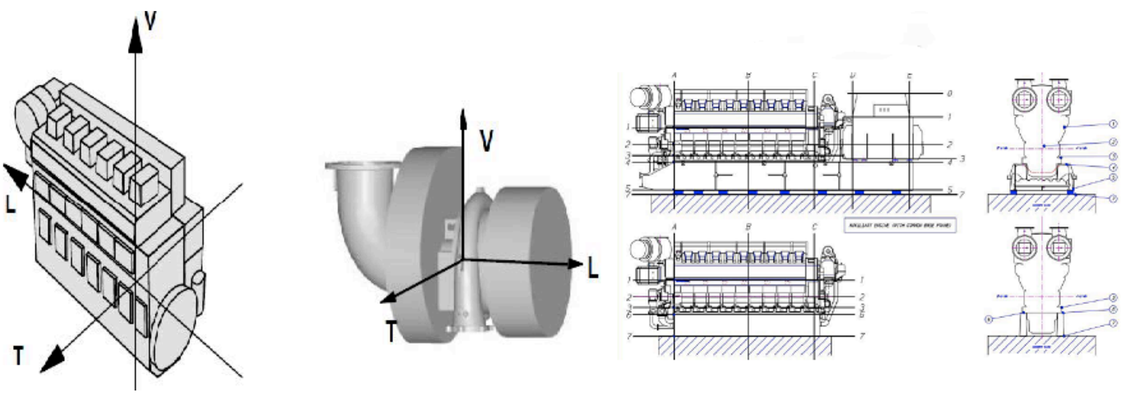

Figure 65. Shaft positions and vibration measurement points in an internal combustion engine [13]

3.7.1 Accelerometer

Accelerometers are used to detect vibrations of the object being measured. Accelerometers can be simple manual accelerometers that measure only one direction, or advanced accelerometers that measure all three directions [11-13]. The frequency at which the vibration is detected depends on how the vibration is applied to the object [11-13]. Accelerometers are attached to the object with pins when detecting high frequencies (up to 50 kHz) and with adhesives or magnets when detecting low frequencies (2 to 5 kHz and 2 to 3 kHz) [11-13].

Figure 66. Accelerometers [11-13]

3.7.2 Analyst

The analyser is the device used to analyse the time signal from the measurements. It records data that can later be analysed, and some devices can analyse in real time the measurements made. There are portable analyzers suitable for simple and quick vibration measurements [11-13].

Figure 67. Portable analyzer [11-13]

3.7.3 Vibration spectrum

The spectrum is a time signal analyzed at the frequency level. Values are displayed as RMS average or peak values. In the spectrum, you can see at which frequencies the amplitude is higher. Each peak in the spectrum represents a joint excitation [11-13]. If the frequency is low, it means that the damage-prone parts of the engine are re-excited at that frequency. If the vibration level of the engine is high, it is very important to know the frequency, because this provides important information and helps to understand the damage [11-13].

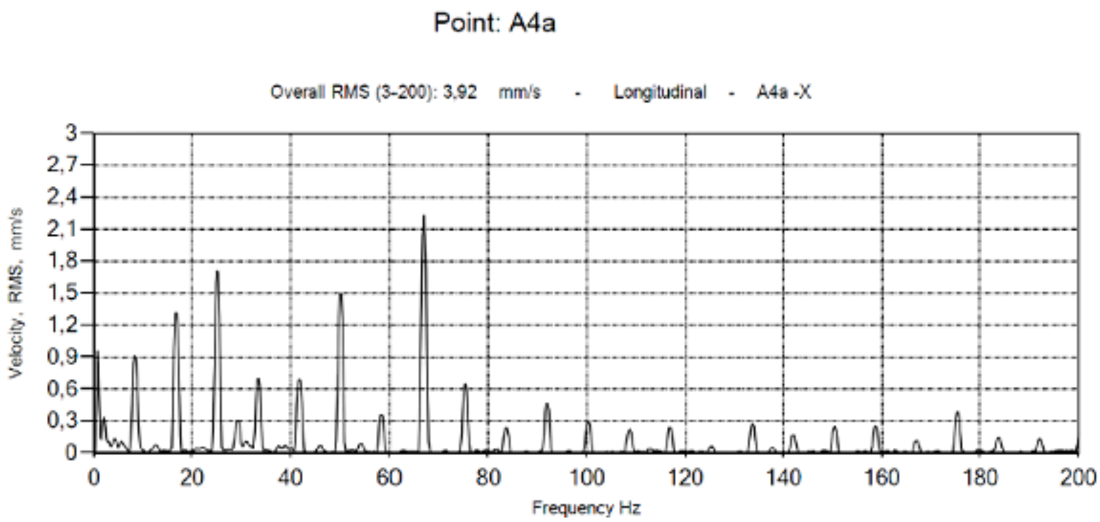

Figure 68. Typical vibration spectrum [11-13]

3.7.4 Scan

Scanning is done to identify low and high frequencies at various speeds and different engine loads. The scan helps to ensure that there are no natural frequencies that are contrary to the requirements of different motoring [11-13]. Different vibrations may have different peaks that are out of phase with each other (the spectrum shows only the smallest range). A scan allows one to see all peaks [11-13]. The sweep is done by gradually increasing or decreasing the speed and size of the mortar. This should be done to find all critical frequencies in the system. Once all critical frequencies are found, they should be checked to see if they are not the same as or identical to other engine excitation frequencies (e.g., fuel start command) [11-13].

3.7.5 Functional Deformation Diagram

It is the way in which a machine component or individual subcomponents move together during operation [11-13]. Knowledge of the vibration of the component during normal operation makes it easier to resolve a resulting failure. Measurements from various points on the engine can be used to model the vibration of the engine during operation [11-13].

3.7.6 Oscillator

Vibration measurements are intended to determine the natural frequency of a component or an entire machine. Smaller parts can be placed on a vibration table [11-13]. To perform a vibration test on an entire machine, a hydraulic cylinder is attached to the machine and subjected to random vibrations. This analysis is similar to a vibration spectrum, but yields better results [11-13].

3.8 Wartsila Internal Combustion Engine Vibration Measurement Guidelines

This paragraph lists some acceptable vibration levels established by Wartsila taking into account ISO standards [13].

3.8.1 Acceptable vibration levels according to Wartsila.

The maximum permissible vibration limit on Wartsila engines is in accordance with ISO 10816-6 standard of severity grade 18. According to this the total vibration should have maximum values of displacement, velocity and acceleration below 183 μm (RMS), 17.8 mm/s and 27.9m/s^2 respectively. These limits should be used in the frequency range 2Hz - 1000Hz [13].

3.8.2 Mechanical subsystems of NDEs subjected to vibration

The ISO 10816-6 standard specifies the allowable vibration levels for some components on the engine, but for components not listed in the standard, Wartsila provides standard total vibration levels for various mechanical and electrical components on the engine [13].

3.8.2.1Turbocharger and filter/suction

Acceleration and velocity of the turbocharger and filter are measured, with acceleration ranging from 3 to 1000 Hz and velocity ranging from 3 to 200 Hz The allowable vibration levels of the turbocharger filter for the W20 and W32/34 engines are 75 mm/s (RMS) velocity and 4 g acceleration The W20, Typical vibration levels for W20 and W32/34 engines are much less than these values [11-13].

3.8.2.2Other mechanical subsystems

The speed (mm/s) in the range 3-200Hz is measured in the subsystems of the machine. The acceptable vibration level in the lubricating oil module (LOM) and air cooler housing (ACH) on W20 and W32/34 engines is 35 mm/s (RMS) [11-13]. The acceptable vibration level of pumps, filters and low pressure pipes for W20 and W32/34 engines is 55 mm/s (RMS) [11-13]. The acceptable vibration level on high pressure pipes in W20 and W32/34 engines is 80 mm/s (RMS). For non-specified components in the engines the acceptable vibration level is 80 mm/s (RMS) [11-13].

3.8.2.3Electrical systems

The various electrical components in an engine also experience vibrations during operation, but at different amplitudes and frequencies [11-13].The vibration levels set by Wartsila for the electrical components of an engine are in accordance with ISO 10816-6. According to this standard, the displacement in the frequency range 2-10 Hz for IEC (Isolated Electrical Component) is 0.5 mm (RMS) and 1.1 mm (RMS) for non-IEC. the maximum velocity in the frequency range up to 200 Hz is 30 mm/s for IEC (RMS) and 80 mm/s (RMS) for on-IEC. Acceleration in the frequency range from 200 to 1000 Hz can average 3 g (RMS) for IEC and 10 g (RMS) for on-IEC [11-13].

3.9 Example of Vibration Measurement Application on Diesel Engine Exhaust Valve

In this paragraph an experiment of measuring vibrations on the exhaust valve of a Diesel engine is presented in order to understand both the application and the analysis of the information obtained by this method [12].

3.9.1 Experimental Installation for Vibration Measurement in Diesel Engine

Experimental vibration measurements were carried out on a four-stroke Ford FSD 425 diesel engine which has four cylinders, is direct injection and has a camshaft head for the exhaust valves [12]. This engine is widely used in generators and commercial vehicles. In order to reduce the exhaust noise, two silencers were installed in the exhaust duct. Table 8 shows the basic technical characteristics of the considered engine and Figure 29 shows the experimental test bed [12].

Table 22. Basic dimensions and operating parameters of the engine on which measurements were carried out [12]

Cylinder diameter	93.67 mm
Piston path	90.54 mm
Engine capacity	2496 mm^3
Maximum power	52 kW@2700 rpm
Maximum Torque	145 Nm@2700 rpm
Cylinder ignition series	1,2,4,3
Compression ratio	19:1
Intake valve timing	Opens 13 degrees before the ANS
Exhaust valve timing	It opens 51 degrees before the CNS and closes 13 degrees after the ANS
Maximum compression pressure	3.38 MPa @ rotational speed of the starter motor

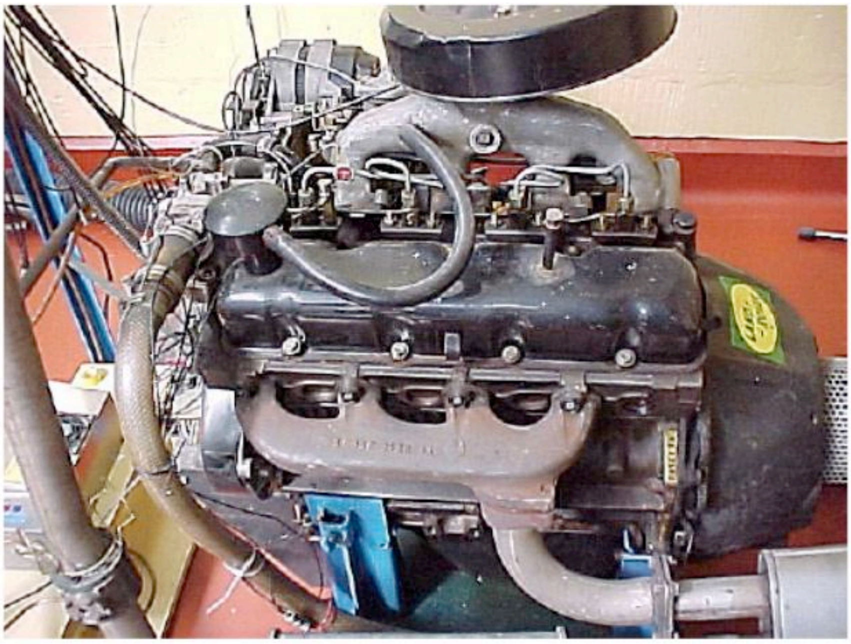

Figure 69. Photographic illustration of the diesel engine on which vibration measurements were made [12]

Conventional and advanced vibration signal processing techniques obtained from a specific engine body position were used to identify and diagnose certain combustion-related faults. The analysis performed was based on the following steps [12]:

- Installation of a modern vibration measurement system.
- Use of the time domain to visualise the vibration signals and their correlation with what is happening inside the cylinders of the engine under consideration.
- Examination of the effect of the load and speed of the diesel engine on the vibration signals. In particular, vibration measurements and analyses were performed at the following loads.
- Configuration of a specific fault in the engine such as reducing the distance between the exhaust valve and cam in cylinder 1 in order to measure and analyse vibrations from it.

The sampling system used to acquire and process the vibration signals is shown in the following figure (Figure 30).

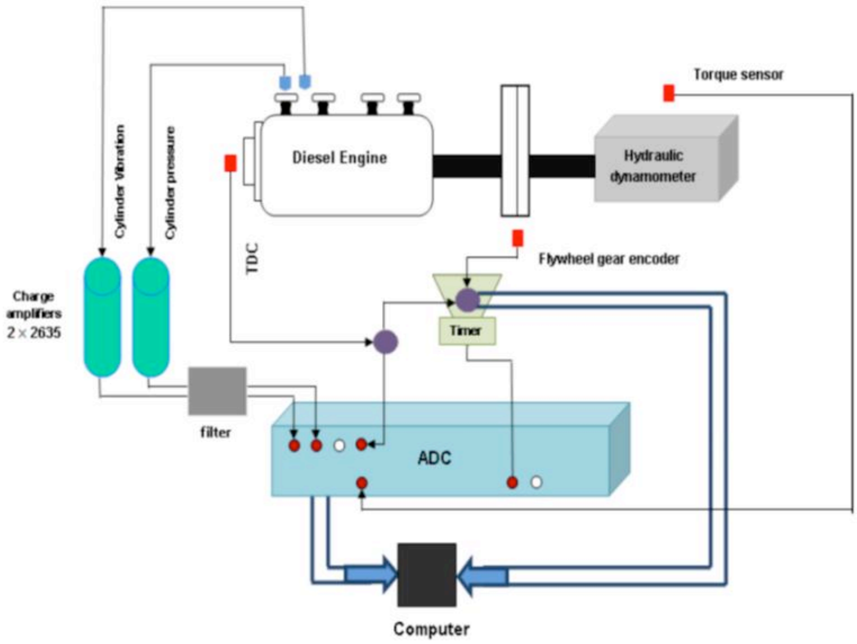

Figure 70. Schematic illustration of the experimental test bed and the measurement acquisition and processing system [12].

An accelerometer 4368 was placed close to the exhaust valve of cylinder 1, specifically between cylinders 1 and 2. The following figure shows the position of the accelerometer to measure the vibration generated by the incorrect distance between the exhaust valve and cam of cylinder 1 [12].

Figure 71: Photographic illustration of the position of the vibration accelerometer on the cylinder head between cylinders #1 and #2 [12]

The main objective of this analysis is to obtain real data from a test facility of a diesel engine operating under normal and abnormal operating conditions and to apply vibration signal analysis techniques to diagnose the condition of the engine. For this purpose, experimental vibration measurements were carried out at four loads: 0, 20, 40 and 60 Nm and at two rotational speeds of 1000 and 1500 sal [12]. Only the vibration signals generated by the incorrect distance between the exhaust valve and the cylinder 1 cam were examined. As shown in the following figure, the distance between the exhaust valve and the cylinder 1 cam was changed from 0.4 mm (healthy distance) to 0 mm, 0.25 mm and 0.6 mm. These incorrect distances simulate increased leakage at a rate, incorrect timing of the exhaust valve and incorrect cylinder pressure [12].

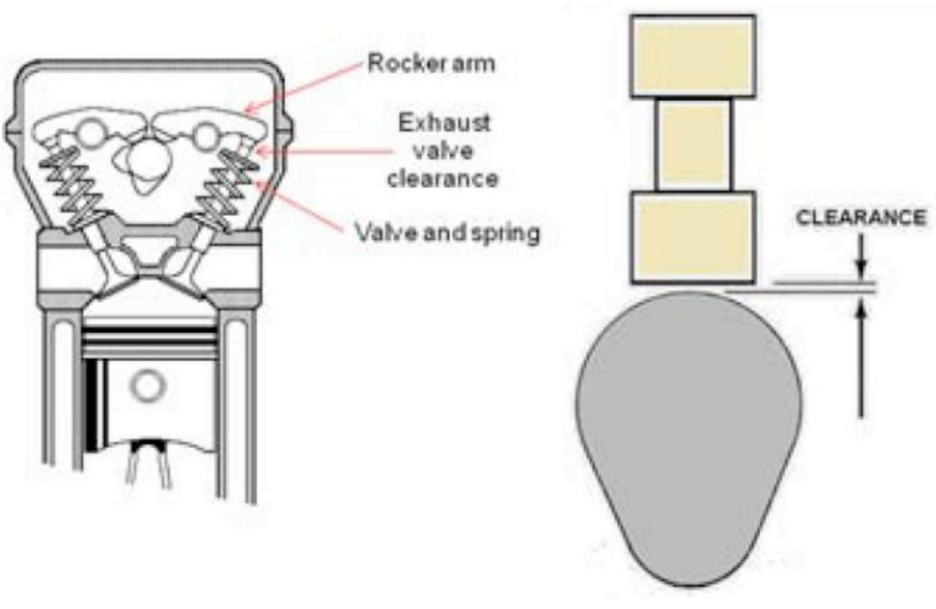

Figure 72. Schematic illustration of the variation of the cam - exhaust valve distance in order to record the vibration signal and detect the fault [12].

3.9.2 Vibration analysis

For piston engines, such as diesel engines, the method of time-major vibration signal analysis is useful for understanding specific time events, such as the ignition sequence of cylinders. However, time-axis vibration analysis does not provide excellent information for diagnosing the operating condition of engines and mechanical components [12]. For this reason, frequency-axis analysis of vibration signals is used. Therefore, in this investigation, the analysis of the vibration signal in frequency λ was based on vibration data measured with an accelerometer installed near the exhaust valve of the first cylinder at four loads: 0, 20, 40, and 60 Nm [12]. At these engine loads, the engine was operated steadily at 1000rpm and the distance between the tail of the exhaust valve and the cam was varied from 0.4 mm (Healthy condition) to 0 mm (Failure condition) [12]. A diagram of the fuselage main vibration signal during healthy condition and failure condition when a 60 Nm load was applied to the exhaust valve of cylinder 1 is shown. In Figure 33, the ignition system (33.3 Hz), the first harmonic (66.6 Hz), and the second harmonic (99.9 Hz) can be seen [12].

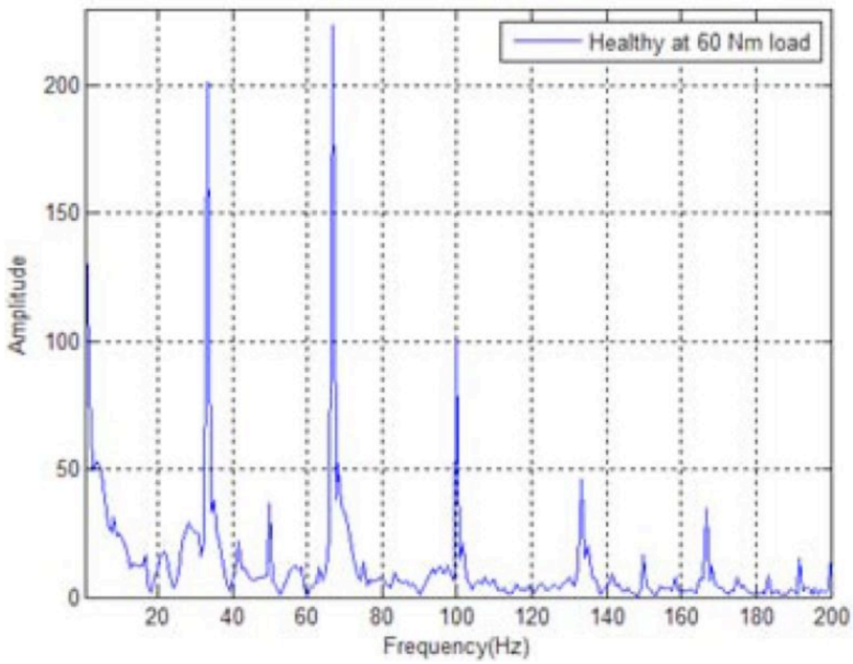

Figure 73. Frequency domain vibration signal analysis for sound/right distance between exhaust valve and cam in cylinder 1 of a four-cylinder diesel engine. The measurement was made at 1000 rpm and at an engine load of 60 Nm [12].

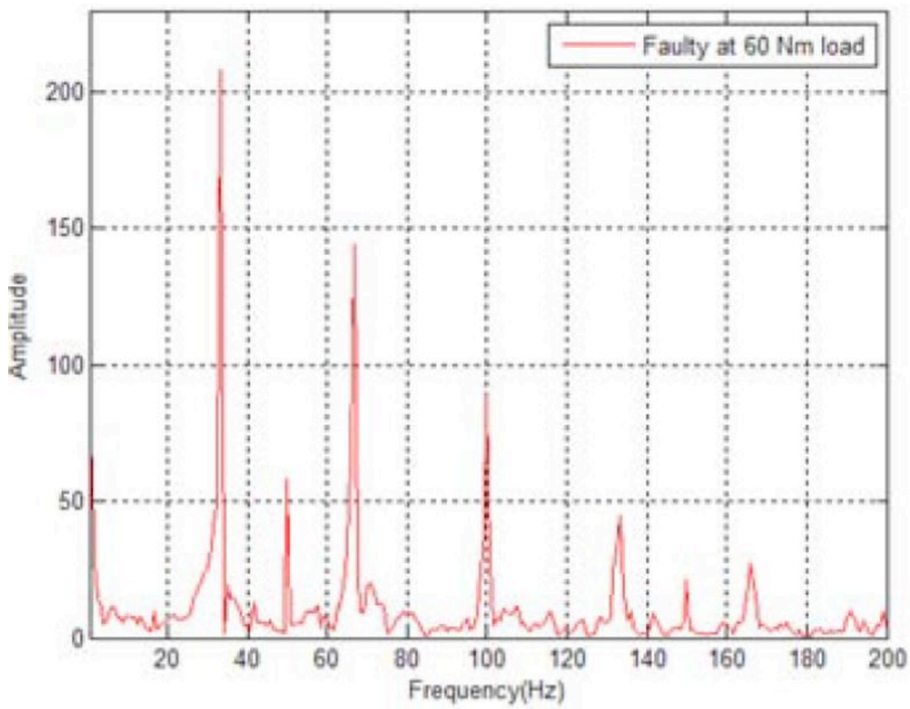

Figure 74. Frequency domain vibration signal analysis for incorrect exhaust valve to cam clearance in cylinder 1 of a four-cylinder diesel engine. The measurement was made at 1000 rpm and at an engine load of 60 Nm [12].

The measured error is the incorrect distance between the exhaust valve and the cam in cylinder 1, changed from 0.4 mm (sound distance) to 0 mm (incorrect distance/uncondition), which simulates some increase in leakage and also affects the pressure in the cylinder [12]. The results show that this error affects the opening and closing time of the exhaust valve. In the frequency domain, there was a clear difference between the signals of healthy and faulty conditions [12], indicating that a fault at a frequency of 50 Hz produces high amplitude transverse oscillations. Large amplitude oscillations also occur at frequencies of 130 and 150 Hz [12]. The occurrence of these additional large amplitude vibrations is due to the improper distance between the exhaust valves of cylinder 1 and the cam, which caused the exhaust valves of cylinder 1 to open earlier and later, resulting in abnormal heat shrinkage [12].

3.9.3 Conclusions of Experimental Study of Vibration Measurement on Diesel Engine Exhaust Valves

Using the technique of time-axis vibration analysis, which extracts information from vibration signals, we measure the vibration of diesel engines at different speeds and loads [12].

The time domain was used to relate the vibration signals to the crank angle of the engine. The engine was tested at different loads of 0, 20, 40 and 60 Nm and speeds of 1000 and 1500 raft [12]. The exhaust valve clearance was changed from 0.4 mm (healthy case) to 0.0 mm (faulty case). Analysis of the time and frequency axis data showed a small difference in the rate of change of the vibration signal due to the change of the exhaust valve clearance. This difference in the rate of change of the vibration signal was due to changes in cylinder pressure and vibration due to changes in the timing of the opening and closing of the exhaust valve [12].

The exhaust valve clearance changed from 0.4 mm (healthy clearance) to 0 mm (faulty) and the fault in cylinder 1 reproduced the exhaust leakage to some extent and affected the cylinder pressure. As can be seen from the results, the fault is affected by the opening and closing time of the exhaust valve. In the frequency domain, there was a clear difference between the healthy and faulty signals [12].

3.10 Diagnosis of Diesel Engines with Vibration Measurement - General Conclusions

In this chapter the diagnostics of diesel engine failures with vibration measurement was presented. Their nature, specific standards set by world organizations, and the layout of a measuring chain and the role of the individual instruments that make up the chain were discussed [12].

Vibration is an important and useful tool for diagnosing faults in rotating machinery such as pumps and turbochargers. By using the root mean square (RMS) of the vibration signals it is possible to assess the condition of the machine in conjunction with appropriate, available vibration standards and limits or manufacturer data. In addition, the Vibration Power Spectrum allows the type of fault to be determined according to the existing fault and symptom library of each machine. Some particular importance is the preservation and storage of the power spectrum of the healthy machine for direct comparison with the current spectrum [12].

4 Operation Monitoring and Fault Diagnosis of Diesel Engines with Torsional Vibration Analysis

4.1 Introduction

With the development of modern machinery industry, the application of internal combustion engine becomes more and more widely, and researches are focused on achieving high power, high speed and strong loads. Thus the issue of engine torsional oscillations is becoming more and more prominent. Engine operating conditions can have a major impact on the shaft, leading to torsional vibrations and resonance, vibrations which lead to failure-accidents. As the problem of motor torsional vibration becomes more and more apparent, research is intensifying on this phenomenon [15].

4.2 Torsional Oscillation Analysis Methods for Crankshaft Torsional Oscillations of Piston Engines

Based on the above axis simulation models, the solution of torsional oscillations for multi-freedom free oscillation calculation includes the Holzer method, the system matrix method and the transfer matrix method. The methods for multi-freedom computation of forced oscillation include energy methods, the amplification factor method and the system matrix method The development of computers replaced handwritten calculations with digital ones, and some common methods for computing torsional oscillations emerged such as the transfer mode method and the finite element method [15].

4.2.1.1 Holzer method

The Holzer method is a numerical method widely used today, especially in engineering. The basic idea is that the sum of the moments of inertia of any mass is zero when the shaft is subjected to free vibration without water. This method is useful for estimating low-order torsional frequencies in the early stages of design, but is less accurate and time-consuming for higher-order calculations [15].

4.3 Experimental Crankshaft Oscillation Studies

4.3.1 Current Torsional Vibration Measurement Methods

Vibration torsional measurements play an important part in the study of crankshaft vibrations. There are two types of vibration measurements: contact and nano-contact. In the contact method, a sensor is mounted on the shaft and the measured signal is sent to the instrument via a ring collector or radio signal. The non-contact method measures the magnitude of the angular velocity along the shaft or gear. If the Doppler wave is measured correctly, it is also possible to measure the vibrations with a laser beam [15].

4.3.2 Mechanical Measurement

The Geiger probe is a typical mechanical instrument for measuring torsional oscillations. It is designed so that the reception and recording of the signal can be read by mechanical devices. It is simple and practical and is widely used. The DVL is also an instrument of the same type as the Geiger but in which the torsional oscillations are sensed by means of a rubber belt. The bandwidth of mechanical calculation systems is quite limited, so that low-frequency torsional oscillations may not be detected. Also the measurements cannot be directly analyzed by modern instruments, so this measurement method is less and less used [15].

4.3.3 Contact measurement

Contact measurement consists of placing a sensor directly on the crankshaft. The measured signals are transmitted to the appropriate instrument via a ring collector or radio frequency [15]. To monitor the dynamic response of the shaft or parts of it, the strain gauge device should eliminate the interference of transverse oscillations and be able to recognize the influence of temperature. The torsional oscillation meters used in this method include vibration-torsion meters, piezoelectric sensors and those of the shock-induction type [15]. Contact measurement is widely used in internal combustion engine vibration testing due to its high sensitivity, wide frequency range, ease for recording the measured signal and analysis. However, this measurement system itself has a rotational inertia, which has an impact on the measurements. Also in all kinds of contact measurements the measuring devices must be placed on the shaft, which thus loses its original structure, which is often prohibited [15].

4.3.4 Contactless measurement

The measuring device of the non-contact torsional vibration calculation method is not installed directly on the crankshaft, but collects signals through photoelectric and magnetoelectric conversion from different parts of the crankshaft [15]. When the shaft rotates the tooth structure installed on the shaft can cause ringing by forming pulse modulation signal sequences in the sensor, whose amplitude and phase could provide information about axial torsional oscillations [15]. The non-contact measurement method does not need to install special devices on the shaft, but uses the repeatability of specific parts of the shaft without interfering with its normal operation and is the main method of measuring torsional oscillations [15].

4.3.5 Laser measurement

The technique of measuring torsional oscillations with Doppler beams was developed after the measurement of fluid velocity by this method. When the laser beam is irradiated on the surface of the shaft the linear velocity of its surface causes the scattered light to change the Doppler frequency [15]. The instantaneous angular velocity of the axis represents the instantaneous change of its frequency [15]. All that is required to set the measurement point is a smooth surface on the axis. However, its transverse vibrations affect the accuracy of the measurement [15].

4.4 Example of Torsional Vibration Analysis Application for Fault Detection in a Diesel Engine with Cylinders in Series

4.4.1 Introduction

In recent years, engine manufacturers and research centres have focused on the field of fault detection through the study of torsional oscillations of the crankshaft and the analysis of its instantaneous rotational speed [15]. The occurrence of a fault causes an excitation in the crankshaft, which results in a change in its rotational speed and the generation of oscillations with a characteristic amplitude and frequency related to the torsional behaviour of the engine load system. In this section, a method of fault location in a medium-speed diesel engine will be presented through the analysis of the crankshaft torsional oscillation amplitudes [15].

4.4.2 Problem definition

The problem to be studied is the fault location in a cylinder using the amplitudes of the torsional oscillations and the lower harmonic orders (0.5, 1 and 1.5) of the inertial masses and the DFT of the measured engine speed signal based on the excitation caused by the forces from the exhaust gases inside the cylinders. Particular reference will be made to the seriousness of the use of torsional oscillation amplitudes in monitoring the operation of an internal combustion engine [15].

4.4.3 Engine model

To solve the problem, we must first transfer the engine to a model, i.e. replace the continuous and continuous system, which is the crankshaft together with all the masses on it and moved by it, by a discrete system consisting of discrete masses, i.e. in essence moments of inertia I concentrated at various points along the axis and connected together by various elastic members having only torsional rigidity K and not mass and damping C [15]. The continuous and synoptic three-dimensional crankshaft assembly, with its continuously distributed masses and elastic connecting members, is reduced to an equivalent in terms of elastic torsional behavior to a torsionally discrete system. This equivalence obviously consists of an equality in terms of I $\kappa\alpha\iota$ K Of course, even today, the above reduction is not entirely mathematically feasible, but nevertheless this reduction is usually carried out quite accurately in practice based on various simplifying assumptions. The basic assumption is that all masses are reduced to certain planes perpendicular to the longitudinal axis of the spindle, i.e. we assume that we have discs - flywheels of negligible thickness with an inertia moment I corresponding to the various masses which each disc represents. The discs are considered to be connected to each other by an elastic member with a stiffness of K equal to that of the real spindle between the considered points. As a spindle we consider the whole crankshaft system with all the masses of its moving mechanism [15].

The engine to be studied is a four-stroke six-cylinder in-line diesel engine model SL 90 of Kirloskar Oil Engine Pune-90. Figure 35 shows the crankshaft of the engine consisting of the starting gear at position 1, the six cylinders at positions 2,3,4,5,6,7 and finally the flywheel-bolt at position 8. These elements are the basic masses of the moving mechanism of the crankshaft system that will form the rotating discs in the corresponding equivalent discrete crankshaft system [15].

The equivalent discrete system (Figure 35) consists of eight rotating discs of negligible thickness, each characterised by its moment of inertia I which is corresponding and proportional to the mass represented by each disc [15]. Thus we will have I_7 for the starting gear, I_1 έως I_6 for the six cylinders and I_8 for the flywheel. Also defined for the specific crankshaft system are seven distinct values of the magnitude of the stiffness K where each of these values corresponds to a specific portion of the crankshaft bounded between two consecutive rotating discs as shown in Figure 35 [15]. Thus we have K_6 the value of the torsional stiffness of the part of the shaft between the rotating discs 6 and 7, i.e. the starting gear and the first cylinder [15]. Then, using exactly the same logic, the remaining values of the torsional rigidity are defined K_1, K_2, K_3, K_4, K_5 defined between the successive rollers and K_6 $\kappa\alpha\iota$ $K7$ the value of the torsional stiffness between the sixth cylinder and the flywheel-volute. C_1 to C_6 is defined as the damping of the cylinders [35].

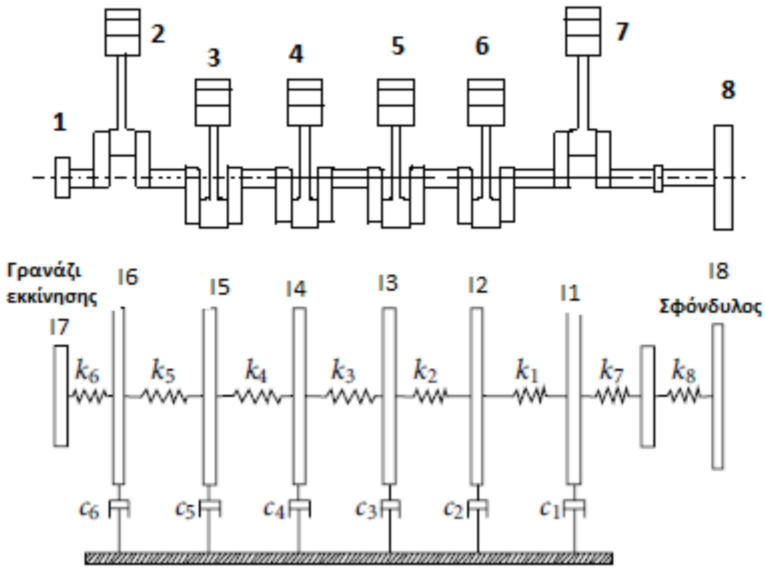

Figure 75. Diesel engine crankshaft mass model [15]

Table 23. Basic construction and operational characteristics of the diesel engine in which torsional oscillations of the crankshaft were analyzed [15]

Engine type	In series, four-stroke, oil
Number of cylinders	6
Rotation wear	Anti-meteorological
Ignition Series	1-5-3-6-2-4
Power	309HP
Cylinder diameter	118mm
Piston Route	135mm
Compression Ratio	15.5

4.4.4 Interpretation and Calculation of the Residual Rotating Vector due to Torsional Oscillation of Torsional Masses

For the engine considered, which is used as an example, the crankshaft cranks are arranged at 120 degrees crank angle. The firing order of the cylinders is 1-5-3-6-2-4 [15]. Considering that the order of the rotating vectors corresponds to the number of complete operating cycles during a complete rotation of the crankshaft, then the first order rotating vectors will be spaced at an angle equal to 120 degrees and will be arranged in a circular manner according to the ignition order starting from cylinder 1 and ending with cylinder 4 [15]. The second order rotating vectors rotate at twice the speed i.e. twice the crank angle than the first order vectors [15]. This results in each second order rotating vector being separated from the other by an angle equal to 2 X 120 = 240 degrees of crank angle. Even the second order vectors will be arranged circularly based on the order of ignition [15].

4.4.5 Calculation of the Critical Resonant Frequencies due to Torsional Oscillation of the Crankshaft based on the Holzer Pinnacle Method

Figure 36 shows the circular rotating vector diagram for torsional oscillation of the crankshaft at 1387.22 rpm of the diesel engine under consideration with the aim of analyzing the minor and major critical harmonic orders [15]. As can be seen from the figure, the third order is the main critical excitation order for the calculated torsional oscillation. The corresponding critical rotational velocities for the occurrence of resonance due to torsional oscillation of the crankshaft are [15]:

$$
\begin{aligned}
&= 1148vpm / 3 = 191.330rpm \\
&= 2379vpm / 3 = 396.500rpm \\
&= 4162vpm / 3 = 1387.33rpm
\end{aligned}
\tag{26}
$$

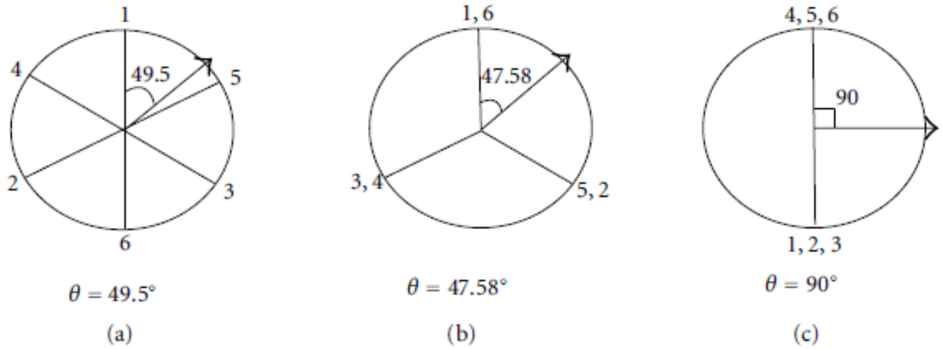

Figure 76. Diagram of rotated vectors of torsional oscillations of order 0.5, 1 and 1.5 [15]

The 3^n order excitation of the three-node oscillation falls within the operating range of the motor from 750 to 2200 sal. A circular rotating vector diagram is thus formed as shown in Figure 36, which corresponds to a rotational speed equal to 1387.33 sal or 4162 vibrations per minute (vpm). Figures 36(a) and 36(b) show the phase diagram of the resulting rotating vectors for oscillation order 0.5 and order 1 with phase angles between the vectors of 60 and 120 degrees crank angle. The number of the cylinder and the amplitudes of the vectors corresponding to it are shown in Figure 36 using data presented in Table 10 [15]

Table 24. Summary of torsional vibration amplitude values from Tables 2, 3 and 4 of the Holzer method [15]

Ignition range	Values of torsional amplitudes from Tables 2, 3 and 4 of the Holzer method		
Frequency	**1148 vpm**	**2379 vpm**	**4162 vpm**
Cylinder 1	0.68	-0.36	-3.16
Cylinder 5	0.14	-0.38	3.02
Cylinder 3	0.46	-0.63	0.64
Cylinder 6	-0.03	-0.11	1.29
Cylinder 2	0.59	-0.56	-1.89
Cylinder 4	0.31	-0.57	2.75

It was found that more than one vector has the same angular position so they are added numerically. Then taking the components on the vertical axis (cosine components) and adding them together gives the total component of the vectors on the vertical axis (vertical component). Similarly, adding the components of the vectors on the horizontal axis (sine components) and adding them gives the corresponding total component of the vectors on the horizontal axis.[4] The resulting total of all the rotated vectors will be given by the following relation:

$$[Resultant\ Vector]^2 = [Sine\ Component]^2 + [Cosine\ Component]^2 \qquad (27)$$

It was found that the successive vectors of order $4^{ης}$ are each 4 x 120 = 480 degrees of crank angle or (360 + 120 degrees) from each other and therefore have the same problem as the rotating vectors of order $1^{ης}$ and the corresponding vectors of order $7^{ης}$ and $10^{ης}$ [15]

Figure 9(a) shows the orders 0.5, 2.5, 3.5, 5.5 and 6.5; the vectors of orders 1.5, 4.5 and 7.5 are shown in Figure 9(c) and as can be seen they all act up and down as designed and have a relatively large recommended magnitude. In the case of vector order numbers which are multiples of half the number of cylinders i.e. orders 3, 6, 9 and so on for the six-cylinder diesel engine under consideration all vectors act in the same direction (not shown in Figure 9) and are therefore called "Principal Orders"[15]

For order 0.5: The phase angle between the rotated vectors is: $\theta = \tan^{-1}$ (sum of all vertical components of the rotated vectors/sum of all horizontal components of the rotated vectors) $\theta = -49.50$ degrees [15]

For class 1: The phase angle between the rotated vectors is: $\theta = \tan^{-1}$ (sum of all vertical components of the rotated vectors/sum of all horizontal components of the rotated vectors) $\theta = -47.58$ degrees [15]

For order 1.5: The phase angle between the rotated vectors is: $\theta = \tan^{-1}$ (sum of all vertical components of the rotated vectors

vectors/sum of all horizontal components of the rotated vectors) $\theta = 90$ degrees [15]

For a given engine the resulting recommended rotating vector can be calculated as mentioned above and this recommended rotating vector can be represented (by a vector with a width equal to the torsional oscillation amplitude) as shown in Figures 9(a), 9(b) and 9(c) [15].

4.4.6 Interpretation and Calculation of the Reluctant Rotating Vector due to Torsional Oscillation by Gas Cylinder Forces

Extensive experimental measurements were carried out on a four-stroke six-cylinder direct-injection diesel engine (Kirloskar SL90-SL8800TA) [15]. This engine was operated at constant rotational speed and at full loads. To simulate a cylinder with abnormal operation (defective cylinder), the engine was operated under normal operating conditions. The pressures inside all cylinders were measured with piezoelectric transducers. The average indicated pressure and torque due to cylinder gas pressures were calculated from the measured cylinder pressure diagrams [15]. The calculated torque and measured rotational speed were subjected to Discrete Fourier Transform (DFT) in order to calculate the amplitudes and phases of the harmonic components of the shaft oscillation. Figure 37 shows the actual cylinder gas pressure curves generated from the piezoelectric transducer measurements on all 6 cylinders when the engine was operated under normal operating conditions [15].

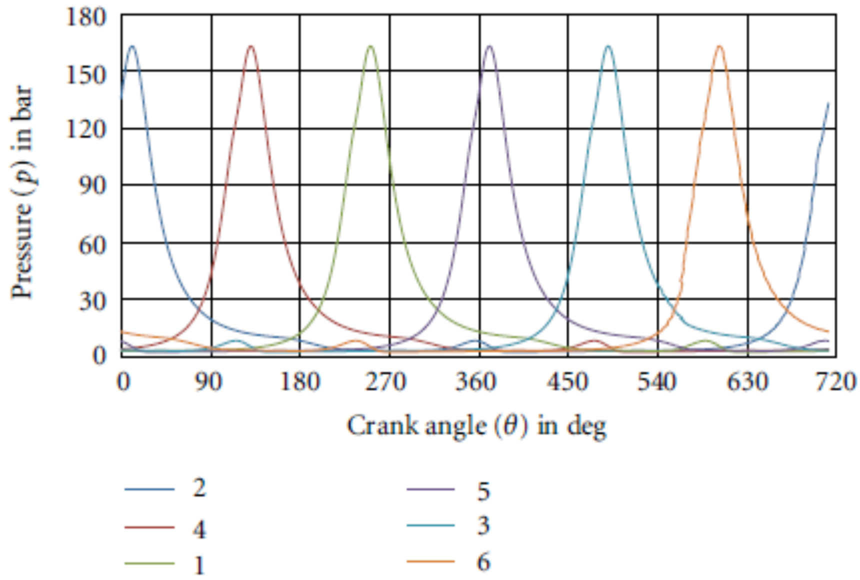

Figure 77. Gas pressure - Crank angle [15]

Table 25. Natural system frequencies for an oscillation frequency of 1148 vpm [15]

	Number of mass	Moment of inertia J (kgm^2)	Moment of inertia per unit of axis deformation $J\omega^2$ (MNm)	Deformation in the plane of mass θ (± rad)	Torque in the plane of mass $J\omega^2 \theta$ (MNm)	Total torque $\Sigma J\omega^2 \theta$ (MNm)	C axis torsional stiffness (MN m/rad)	Change in deformation Δθ (rad)
Camshaft glazing	1	0.094	0.124	1	0.124	0.124	0.391	0.317
Cylinder 1	2	0.075	0.099	0.683	0.068	0.191	0.195	0.098
Cylinder 2	3	0.075	0.099	0.585	0.058	0.249	0.195	0.128
Cylinder 3	4	0.075	0.099	0.457	0.045	0.295	0.195	0.151
Cylinder 4	5	0.075	0.099	0.307	0.030	0.325	0.195	0.167
Cylinder 5	6	0.075	0.099	0.140	0.014	0.339	0.195	0.174
Cylinder 6	7	0.075	0.099	-0.033	-0.004	0.336	3.520	0.095
Slinger	8	1.97	2.596	-0.129	-0.334	0.002		
F=1148 npm (oscillation per minute)								

Table 26. System frequencies for an oscillation frequency of 2379 vpm [15]

	Number of mass	Moment of inertia J (kgm^2)	Moment of inertia per unit of axis deformation $J\omega^2$ (MNm)	Deformation in the plane of mass θ (± rad)	Torque in the plane of mass $J\omega^2\theta$ (MNm)	Total torque $\Sigma J\omega^2\theta$ (MNm)	C axis torsional stiffness (MN m/rad)	Change of deformation Δθ (rad)
Camshaft glazing	1	0.094	0.053	1	0.53	0.532	0.391	1.361
Cylinder 1	2	0.075	0.424	-0.361	-0.153	0.379	1.952	0.194
Cylinder 2	3	0.075	0.424	-0.555	-0.235	0.144	1.952	0.0735
Cylinder 3	4	0.075	0.424	-0.628	-0.267	-0.123	1.952	-0.063
Cylinder 4	5	0.075	0.424	-0.565	-0.240	-0.363	1.952	-0.186
Cylinder 5	6	0.075	0.424	-0.379	-0.161	-0.524	1.952	-0.269
Cylinder 6	7	0.075	0.424	-0.111	-0.047	-0.571	3.52	-0.162
Slinger	8	1.97	11.149	0.052	0.575	0.004		
F=2379 npm (oscillation per minute)								

The results from the application of this technique for the diesel engine under consideration are presented graphically Figure 38. It is observed that when the cylinders are operating uniformly contributing equally to the total engine torque, the first three orders of harmonics (K = 0.5, 1 and 1.5) play a significant role in the frequency spectrum of the total engine torque due to cylinder gas forces and, therefore, appear with a very low contribution to the frequency spectrum of the crankshaft rotational speed.[15]

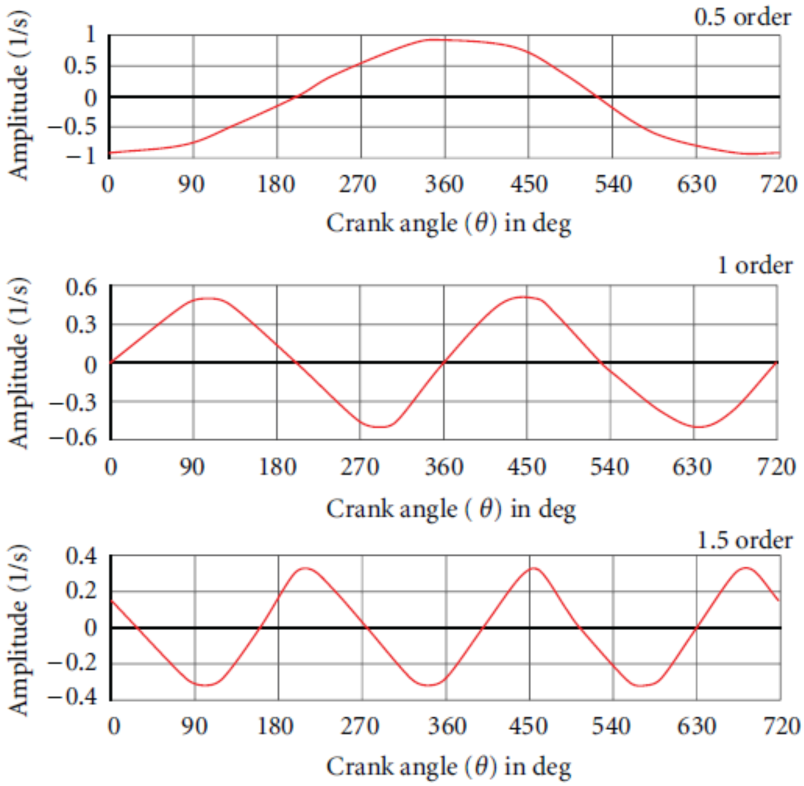

Figure 78. Detection of a defective cylinder by the phases of the three lowest harmonics [15]

Comparing the frequency spectrum of the crankshaft speed when the cylinders are combined with the spectrum when the cylinders are defective, it can be seen that there are significant differences in the amplitudes of the first three positions of the shaft orsional vibration damages [15]. As long as the cylinders of the machine are working properly, the amplitudes of these orsional osilation harmonics will remain within certain limits. As the cylinder starts to reduce the contribution of the orrsional oscillation harmonics to the frequency spectrum, the amplitudes of the first three harmonics start to increase. These amplitudes can be used to determine the cylinder's contribution to the engine torque produced by the gas pressure in the cylinder [15]. The identification of defective cylinders can be carried out by analyzing the vibration phases, i.e. the rotation vectors of the three axis harmorder. Identification of the defective cylinder can be achieved by analyzing the phases i.e. the rotating vectors of the three lowest harmonic orders of the oscillation [15].

Figure 39 was plotted by reconstructing the pressure measurements from the six cylinders in series corresponding to the cylinder firing order (1-5-3-6-2-4) when the engine was operated under normal operating conditions. Figure 40 shows the three lower harmonic orders of the measured rotational speed while maintaining the proportions in terms of amplitudes and phase differences of the vectors [15].

In the rotating vector diagrams of Figure 12 of the first three harmonic orders of the measured rotational velocity, the recommended vector in each harmonic order is also shown in Figure 39 by an arrow with a width equal to the amplitude of the oscillation. The calculation of the constituted vector in each harmonic order in Figure 39 was performed using data from Figure 39. One can observe that for each of the three harmonic classes the vectors point to the group of cylinders that produce less work. The cylinder that is detected three times (in each harmonic order) to be among the least productive cylinders is the defective one as shown in Figure 40 [15].

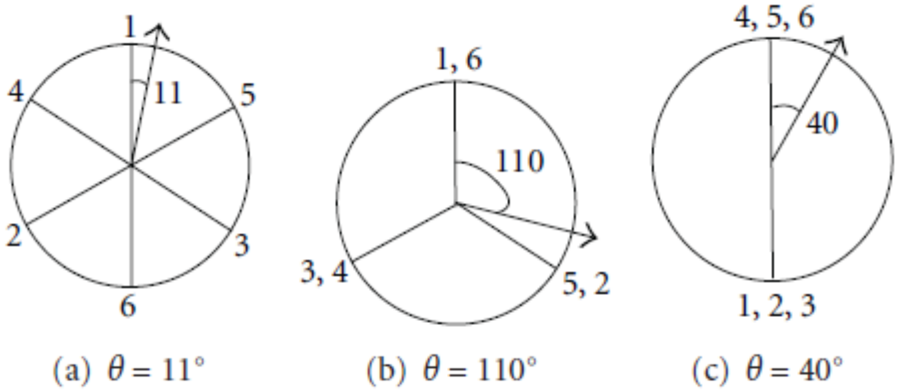

Figure 79. Rotational vector diagram of the three lowest harmonic orders (0.5, 1 and 1.5) of the rotational speed of the shaft [15]

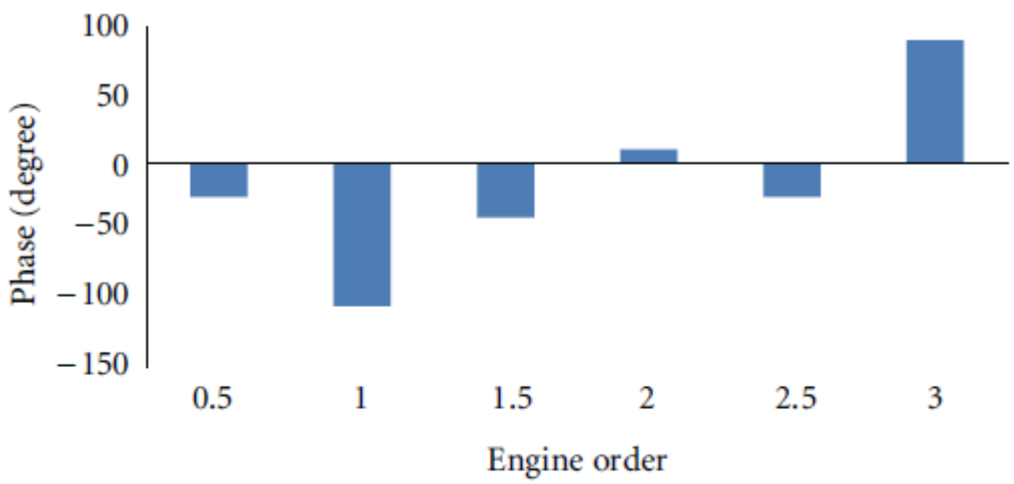

Figure 80. Phase versus cylinder ignition order [4]

4.4.7 Diesel Engine Fault Detection Method with Torsional Vibration Analysis

Based on the above schemes, the following method is developed [15]:

1) The rotated vector diagrams are drawn based on the cylinder firing order for three lower harmonic orders (0.5, 1 and 1.5) by placing the cylinder that fires at 0 degrees in the assumed duty cycle at the Top Dead Center (TDC).
2) In these rotation vector diagrams the corresponding vectors of the measured rotation speed are represented in a system of coordinate axes.
3) The cylinders to which the vectors point are the ones that contribute the least and get a "-" sign. If there are cylinders that receive a '-' sign for all three classes of harmonics, these are clearly identified as the cylinders that contribute the least to the total engine power.
4) This method can detect a defective cylinder at a very early stage i.e. before the engine is started (Table 11) and detection can be done once the contribution of this defective cylinder is reduced from the nominal value to zero with reference to the contribution of the other cylinders (Table 12).

Table 27. Identification of the defective cylinder by its position in the rotated vector diagrams of Figure 39 [15]

	Cylinders					
K	**1**	**5**	**3**	**6**	**2**	**4**
0.5	-	-				
1	-	-		-	-	
1.5		-		-		
I_i	**0**	**-1**	**0**	**0**	**0**	**0**

Table 28. Identification of the defective cylinder by its position in the rotated vector diagrams of Figure 40 [15]

	Cylinders					
K	**1**	**5**	**3**	**6**	**2**	**4**
0.5	-	-				
1	-	-		-	-	
1.5		-		-		-
I_i	**0**	**-1**	**0**	**0**	**0**	**0**

4.5 Conclusions of Technical Monitoring of Operation and Fault Diagnosis of Diesel Engines with Torsional Oscillation Analysis

Tables 14 and 15 evaluate the ability of this method, consisting of cylinder pressure measurement and shaft vibration analysis, to detect defective cylinders in diesel engines. Thus, the shaft torsional vibration amplitudes shown in Tables 11 and 12 are used for the early detection of defective cylinders in diesel engines: from the analysis of the osiological osilation for a six-cylinder diesel engine, it was found that the Fourier analysis of the looorder harmonic amplitudes of the inertial mass (0.5, 1 and 1.5) and the measured velocities based on the of-gas forces of the cylinder are effective in detecting defective cylinders. The method can predict the failed cylinder (cylinder 5 in this paper) in time based on the phase diagram of the three harmonics as soon as the inertial trajectory of the failed cylinder starts to decrease from its initial value to zero compared to the inertial trajectories of the other cylinders [15].

5 Conclusions of the Diploma Thesis - Suggestions for Future Work

5.1 Conclusions of the Diploma Thesis

Diesel engines are, today, a cornerstone of both warships and the entire society in terms of production, propulsion and even entertainment. Their extensive use results in wear and tear and consequent damage. In this paper, the necessity of their maintenance and operation monitoring was initially touched upon in order to prevent the occurrence of failure. Subsequently, since despite all maintenance, failures are inevitable, some methods of diagnosis were presented, each with its advantages and disadvantages.

In the vibration diagnostic method the basic conclusion is that the measured frequency spectrum and RMS (i.e. the amplitude of the vibration) should be compared with corresponding standard/healthy vibration spectra to see if and how big a problem exists in a part of the machine. That is, in the application with the exhaust valve that was not properly adjusted we knew beforehand that it was not properly adjusted or even if we didn't know we would have to go purposefully close to the valve to measure and find the problem plus we knew the problem was the valve misadjustment. If we didn't know beforehand the valve may have been wobbling or not seated well on the seat due to corrosion from the exhaust. If we didn't know this we probably couldn't detect it from vibration analysis. Vibration measurement seems to be suitable for detecting mechanical faults e.g. turbo shaft misalignment. For issues within the cylinders of an engine where we have complex thermo-fluid and mechanical issues it is difficult to identify the cause of an unsound vibration in a part of the engine.

Torsional vibration analysis can identify which cylinder is not working properly by judging its contribution to the torsional vibration of the shaft relative to the other cylinders. However, this methodology cannot identify the reason why the faulty cylinder is not operating correctly, i.e. it does not operate as uniformly as the other cylinders. This torsional vibration analysis method identifies the cylinder that is not operating correctly but does not provide information as to what the cause(s) of the non-uniform operation of the defective cylinder is i.e. the non-uniform operation is due to a problem with the fuel injection system e.g.e.g. incorrect injection advance or reduced fuel injection pressure or a problem with the turbocharger system or a problem with the piston - pusher - crank mechanism e.g. particularly worn compression springs leading to a loss of pressure within the cylinder during air compression.

Therefore, the best diagnostic method of all seems to be the one with the measurement of the cylinder pressure and the thermodynamic simulation of the operation of each cylinder because, compared to the others, it can more easily identify the cause of a problem and propose corresponding improvement or remedial measures.

5.2 Suggestions for Future Work

In order to extend and improve this work in the future, the following future work is proposed:

- Measurements on a diesel engine power couple using the vibration measuring device of the Department of Naval Engineering and Marine Mechanics in order to investigate the smooth operation of the crankshaft, the camshaft and the supercharger shaft.
- Investigate other minimally intrusive methods of monitoring operation and identifying potential faults in diesel engines such as acoustic methods.

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