Diesel Engine Analysis Guide

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R E P O R T   S U M M A R Y

Diesel Engine Analysis Guide

This guide provides a throrough background on diesel engine analysis including combustion, vibration, and ultrasonic analysis theory. Interpretation of results is also provided. This guide is intended to enable nuclear utility personnel to make informed decisions regarding the nature and use of diesel engine analysis, including how to set up an effective program, how to establish analysis guidelines, how to make use of the resulting data to plan maintenance, determine the causes of off-design operating conditions, and warn of potential component failures.

INTEREST CATEGORIES

Nuclear plant operations and maintenance
Plant electrical systems and equipment
Engineering and technical support
Maintenance practices

KEYWORDS

Monitoring
Diagnostics
Predictive maintenance
Maintenance
Diesel Engines

BACKGROUND Diesel engine analysis has not seen consistent widespread use in the nuclear power industry, in part because utility personnel interested in establishing an engine analysis program have not had a single comprehensive source for technical guidance to establish and evaluate it. At many of the locations where engine analysis has been used in the nuclear power industry, it has been successful in identifying engine problems before these problems progressed to catastrophic failures. For example, engine analysis has successfully identified bent connecting rods, cracked cylinder heads, scuffed cylinder liners, leaking fuel injectors and valves, and collapsed valve lifters in nuclear standby EDGs. In addition, engine analysis can and has been used successfully to identify problems in diesel engines in other industries such as detonation, cylinder load imbalance, faulty fuel injection, leaking valves and piston rings, worn or scored cylinder liners, worn rocker arms, defective valve lifters, worn cams, and damaged wrist pins.

Engine analysis techniques were first used to support engine design and development efforts. Diesel engine analysis has been used in the marine, rail, and pipeline industries for many years and has proven cost-effective and successful. It has been used to tune engines, to improve fuel efficiency, to support predictive maintenance methods, to assist with problem diagnosis, and to support longer maintenance intervals.

OBJECTIVES

- To provide a thorough discussion on the theory and practical aspects of implementing combustion, vibration, and ultrasonic analyses on diesel engines at nuclear power plants.
- To enable nuclear utility personnel to make informed decisions regarding combustion, vibration, and ultrasonic analysis of diesel engines, including how to set up an effective program, how to establish analysis guidelines, how to make use of the resulting data to plan maintenance, determine the causes of

off-design operating conditions, and warn of potential component failures.

APPROACH The project team gathered and reviewed pertinent technical literature regarding the subject of diesel engine analysis. Information and experience from utility personnel, consultants, and manufacturers was condensed. The resulting document was reviewed by a technical advisory group composed of experienced diesel engineers. This guide is applicable to selected makes and models of diesel engines in nuclear standby service in the United States and around the world.

RESULTS This guide includes a thorough background on diesel engine combustion, vibration, ultrasonic analysis, theory, interpretation of results, guidance on identifying specific modes of degradation, and examples of specific engine combustion, vibration, and ultrasonic analyses for diesel engines in nuclear standby service. In addition, troubleshooting guidance and a discussion of acceptance guidelines are reviewed. Also, engine analysis programs, equipment costs, equipment selections, and setups are covered.

EPRI PERSPECTIVE Development of this guide was sponsored by the Electric Power Research Institute Nuclear Maintenance Applications Center (EPRI/NMAC) to provide consistent, comprehensive, and detailed guidance for EDG system engineers on diesel engine analysis. The guide was developed in cooperation with the various EDG owners groups, engine and analysis equipment vendors, and a utility Technical Advisory Group (TAG).

PROJECT

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Diesel Engine Analysis Guide

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Important Notice Regarding This Document

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ALCO Owners Group

Cooper Bessemer Owners Group

ESI-EMD Owners Group

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INTRODUCTION

1.1 Purpose

The purpose of this guide is to provide a thorough discussion on the theory and practical aspects of implementing combustion, vibration, and ultrasonic analyses on diesel engines at nuclear power plants. This guide is intended to enable nuclear utility personnel to make informed decisions regarding combustion, vibration, and ultrasonic analysis of diesel engines, including how to set up an effective program, how to establish analysis guidelines, and how to make use of the resulting data to plan maintenance, determine the causes of off-design operating conditions, and warn of potential component failures.

1.2 Background

To date, diesel engine analysis has not seen consistent widespread use in the nuclear power industry, in part because utility personnel interested in establishing an engine analysis program have not had a single comprehensive source for technical guidance to establish and evaluate it. At many of the locations where engine analysis has been used in the nuclear power industry, it has been successful in identifying engine problems before these problems progressed to catastrophic failures. For example, engine analysis has successfully identified bent connecting rods, cracked cylinder heads, scuffed cylinder liners, leaking fuel injectors, collapsed valve lifters, and leaking valves in nuclear standby emergency diesel generators (EDGs). In addition, engine analysis can and has been used successfully to identify the following problems in diesel engines in other industries:

- Detonation
- Cylinder load imbalance
- Faulty fuel injection
- Leaking valves and piston rings
- Worn or scored cylinder liners
- Worn rocker arms
- Defective valve lifters
- Worn cams
- Damaged wrist pins

Engine analysis techniques were first used to support engine design and development efforts. Diesel engine analysis has been used in the marine, rail, and pipeline industries for many years and has proven cost-effective and successful in these industries. It has been used to tune engines to improve fuel efficiency, support predictive maintenance methods, assist with problem diagnosis, and support longer maintenance intervals.

1.3 Scope

This guide includes a thorough background on diesel engine combustion, vibration, and ultrasonic analysis, theory, and interpretation of results, guidance on identifying specific modes of degradation, examples of specific engine combustion, vibration, and ultrasonic analyses for EDGs in nuclear standby service, troubleshooting guidance, and a discussion of acceptance guidelines. In addition, engine analysis program and equipment costs and equipment selection and setup are addressed.

This guide is applicable to selected makes and models of diesel engines in nuclear standby service in the United States and around the world. These include:

- ALCO model 251
- Colt-Pielstick models PC2, PC2.3, and PC2.5
- Cooper-Bessemer model KSV
- Fairbanks Morse Opposed-Piston (FMOP) model 38TD 8-1/8
- General Motors EMD model 645
- Nordberg model FS-1316-HSC
- SACM model UD 45
- Cooper Enterprise (formerly TDI) models DSR and DSRV
- Worthington model SWB12

Table 1-1 summarizes important characteristics of these engines.

1.4 Development

Development of this guide was sponsored by the Electric Power Research Institute's Nuclear Maintenance Applications Center (EPRI/NMAC) in order to provide consistent, comprehensive, and detailed guidance for EDG system engineers on diesel engine analysis. The guide was developed in cooperation with the various EDG owners groups, engine and analysis equipment vendors, and a utility Technical Advisory Group (TAG).

Table 1-1
Characteristics of Engines in U.S. Nuclear Service

Make	ALCO	Colt-Pielstick	Cooper-Bessemer	Cooper-Enterprise	EMD	Fairbanks Morse	Nordberg	SACM	Worthington
Model	**251**	**PC2, PC2.3, PC2.5**	**KSV**	**DSR DSRV**	**645**	**Opposed Piston 38TD8-1/8**	**FS-1316-HSC**	**UD45**	**SWB12**
Continuous Power Rating (kW)	1750–2600		4000–5500	3500–8800	2100–2850	2700–3000	3500 - 4000	2700	
BMEP	201–265		194–214	225			199 - 227		
Working Cycle	4 stroke	4 stroke	4 stroke	4 stroke	2 stroke	2 stroke	4 stroke	4 stroke	4 stroke
# of Cylinders	16/18	12/14/16	16/20	8/16/20	16/20	12	16	16	12?
Bore (in)	9	15.74	13.5	17	9-1/16	8-1/8	13.5	9.45	
Stroke (in)	10.5	18.11	16.5 (master) 17.1 (link)	21	10	10 (each piston)	16.5	8.66	
Aspiration	Turbocharged	Turbocharged	Turbocharged	Turbocharged	Gear-Driven Turbocharged	Gear-Driven Blower / Turbocharged	Turbocharged	Turbocharged	Turbocharged
Piston Action	Single-Acting	Single-Acting	Single-Acting	Single-Acting	Single-Acting	Opposed Piston	Single-Acting	Single-Acting	Single-Acting
Cylinder Arrangement	45° V	45° V	45° V	Inline (DSR) 45° V (DSRV)	45° V	Inline	45°V	50°V	V?
Compression Ratio	11.5	13 (PC2) 11.5 (PC2.5)	10.6		16	13.5			
# of Intake Valves	2	2	2	2	0	0	1	2	2
# of Exhaust Valve	2	2	2	2	4	0	1	2	2
Connecting Rod Type	Master	Master	Articulated Master-Link	Articulated Master-Link	Master (Fork & Blade)	Master	Master	Articulated Master-Link	Master
Connecting Rod Length (in)	21		Main 41 Articulated 40.938						
Timing: IVO/IPO IVC/IPC EVO/EPO EVC/EPC Fuel	80.0° BTDC 37.5° ABDC 57.5° BBDC 60.0° ATDC 25-27.5 BTDC		80 BTDC 40 ABDC 45 BBDC 50 ATDC 27-29 BTDC	75 BTDC 25 ABDC 60 BBDC 40 ATDC 22 BTDC	45 BBDC 45 ABDC 77 BBDC 119 BTDC		50-110 BTDC 40 BBDC to 20 ABDC 50 BBDC EVC 65 ATDC		
# of Nuclear Power Plants	9	9	16	10	31	22	3	2	2
Number of Engines in U.S. Nuclear Service	26	19	36	20	82	47	8	2	4
Speed (rpm)	900	514	600	450	900	900	514	1200	
Cylinder Displaced Volume (in^3)	668	3528	2362 (master) 2443 (link)	4767	645	1037	2362	607 (master)	

2

BACKGROUND

2.1 Traditional Approach to Engine Maintenance

Emergency diesel generators in nuclear standby service utilize large, medium-speed diesel engines driving 1750–8800 kW generators. Many of these diesel engines were designed in the 1950s for service in marine propulsion, natural gas transmission, and railroad locomotive applications. The base-loaded applications for which the engines were designed involve long periods, as long as several weeks or months, of continuous operation. EDGs in nuclear standby service, however, usually operate no more than 150–200 hours per year. These hours of operation are usually accumulated during periodic surveillance testing required by the U.S. Nuclear Regulatory Commission (U.S. NRC) to demonstrate that the engines will start in the event of an emergency.

The engines are normally maintained in a standby condition with jacket water and lubricating oil systems maintained at keep-warm temperatures of 100°F–170°F to aid in starting and to reduce wear during starting. Since surveillance tests last only about 1–4 hours each, the ratio of run time to number of starts for engines in nuclear standby service is usually much higher than that for engines in commercial service (approximately 1 hour of operation per start for nuclear standby engines versus hundreds or thousands of hours per start for commercial engines).

Commercial operators of diesel engines in marine propulsion, natural gas transmission, railroads, and other base-loaded applications perform engine maintenance at scheduled intervals since their diesel engines are base-loaded. This maintenance usually consists of a series of engine teardowns and overhauls during which the engine and subsystems are refurbished with new components after 8,000–18,000 hours of operation or after fixed intervals of time. It is difficult or impossible to predict wear out and failures, and a relatively large amount of degradation is allowed to occur before maintenance actions are required. In order to ensure that the statistical distribution of failures is addressed, components are replaced early if a high confidence level is required. Only the complete failure of a component would cause operators to deviate from the scheduled maintenance approach.

When engines were sold for use at nuclear power plants, the engine vendors developed maintenance recommendations for nuclear standby engines based on their experience with engines in base-loaded operation. When these maintenance intervals were developed, the fixed-time intervals were typically adopted without regard to the number of engine operating hours anticipated during the intervals.

The result in some cases was that annual engine teardown requirements resulted in component replacement after only 150–200 hours of operation. Thus, components were regularly replaced just following or during break-in. Reliability was therefore challenged by maintenance-induced and infant mortality failures. Most U. S. nuclear power plants have Technical Specification or license commitments to follow vendor-recommended maintenance requirements for their EDGs. As a result, many utilities have had little or no flexibility to create a maintenance program that reflects their experience, needs, and operating histories. At the same time, nuclear utilities have been under tremendous pressure to control maintenance costs while improving maintainability and reliability.

2.2 The Integrated Maintenance Approach

Condition-based, predictive, and performance-based maintenance methods have been developed as an alternative to calendar-based maintenance. Condition-based maintenance methods include **visual** inspections or **dimensional** measurements of components to ascertain if wear or physical conditions are within the allowable limits. However, the degree of engine disassembly required to take adequate, accurate measurements presents the same maintenance-induced problems, downtime, and labor costs as a full engine rebuild. Non-intrusive visual inspections of some components are possible. In any case, condition-based methods cannot detect all failures or wear on-line before the failure occurs.

The premise of performance-based maintenance is that engine and generator degradation will be reflected in the performance of the EDG. Therefore, any wear or degradation that does not result in decreased performance is considered acceptable. Performance-based maintenance is very effective at eliminating what could be called unnecessary maintenance. However, engine performance degradation must proceed to where it is noticeable before any corrective action is identified. Thus, maintenance activities cannot effectively be planned without experiencing periods of degradation or unscheduled unavailability.

Both condition-based and performance-based methods use single snapshots of EDG condition. Experience demonstrates that a single snapshot of performance does not produce a complete assessment of EDG condition. Predictive maintenance methods use essentially the same data obtained from condition- and performance-based maintenance methods, except that trending of analysis data is added. Data from inspections, performance monitoring, and testing are trended over time.

Change analysis is used to detect differences in the data. It is normally assumed that changes are a result of wear or degradation. If sufficient historical experience is available, the rate of any downward trends can be estimated, and the point in time at which maintenance will be required can be predicted.

By looking forward, utilities can more readily plan maintenance activities. This can have significant benefits where replacement parts have a long lead time or when minimal EDG downtime is available to complete the maintenance. However, two potential downsides are possible with predictive maintenance. First, non-fleet operators (those with one or only a few engines) rarely have experience with all possible degradation modes. Even fleet operators may not combine experience from multiple engines to base maintenance predictions upon. Therefore, the nature of trends may be difficult to determine (that is, the data and trends are apparent, but the meaning is not). Second, even small changes in trends are typically evident. This can result in unwarranted conclusions that any changes in trends are causes for remedial actions (that is, any unknown data pattern is taken as a cause for concern).

The idea of integrated maintenance was developed to capitalize on the advantages of each method while minimizing the downsides. Specifically, integrated maintenance presumes that effective condition-based, performance-based, and predictive-based maintenance tools (test equipment, procedures, manuals) and data (historical experience, trends, acceptance criteria) are in place.

Given this, conditions (wear or degradation) that are indicated by one test, inspection, performance, or monitoring method are verified by one or more additional techniques. Additional techniques or data can be used to further refine the diagnosis. In this way, more precise and focused maintenance decisions can be made. Predictions regarding degradation rate and EDG operability can be more accurate. This is largely due to the fact that degraded machinery exhibits symptoms of component degradation rather than component degradation itself. Another advantage of this method is that it forces the cause of a failure to be assessed and recorded for future reference.

Some of the motivations for implementing an integrated maintenance program for nuclear standby EDGs are to:

- Increase EDG reliability by identifying and correcting degradation before failures occur
- Increase availability and reliability of EDGs through reduction of unnecessary or intrusive maintenance actions
- Reduce EDG maintenance costs by focusing maintenance efforts on known or suspected problems

Most nuclear power plants already perform one or more elements of integrated maintenance programs on EDGs, although few capitalize on the benefits of integration. For example, many stations already analyze engine lube oil to determine both the condition of the oil and the condition of the engine, and many plants regularly perform internal borescopic inspections of EDGs.

2.3 Integrated Maintenance Programs

In addition to engine combustion, vibration, and ultrasonic analyses, several other advanced techniques for evaluating engine condition and performance have been developed and implemented by a wide variety of diesel engine operators. Some of these techniques include:

- Non-intrusive inspections of engine internal components using borescopes, video probes, or thermographs
- Exhaust gas emissions analysis

- Trend line analysis of engine operating parameters
- Periodic testing of engines or components
- Jacket water chemical analysis
- Lube oil chemical, ferrographic, and spectrographic analysis
- Fuel oil chemical analysis
- Fuel oil consumption measurement

Engine analysis is most effective when used in combination with one or more of these additional techniques in an integrated maintenance program. The premise of such a program is that these individual methods are useful in providing insight into particular degradation mechanisms, but in only a few cases will individual methods positively identify the specific component prior to a failure.

With an integrated maintenance program in place, conditions indicated by one test, inspection, performance, or monitoring method are investigated using one or more additional techniques (see Figure 2-1). In this way, more precise and focused maintenance decisions can be reached. As many of the potential causes as possible are eliminated using the additional non-intrusive methods, thereby reducing the amount of engine disassembly necessary to find the cause of the problem. The symptoms of the degradation are used to find the cause of the problem rather than searching for the degraded component. This method forces the root cause of a failure to be determined and recorded for future reference.

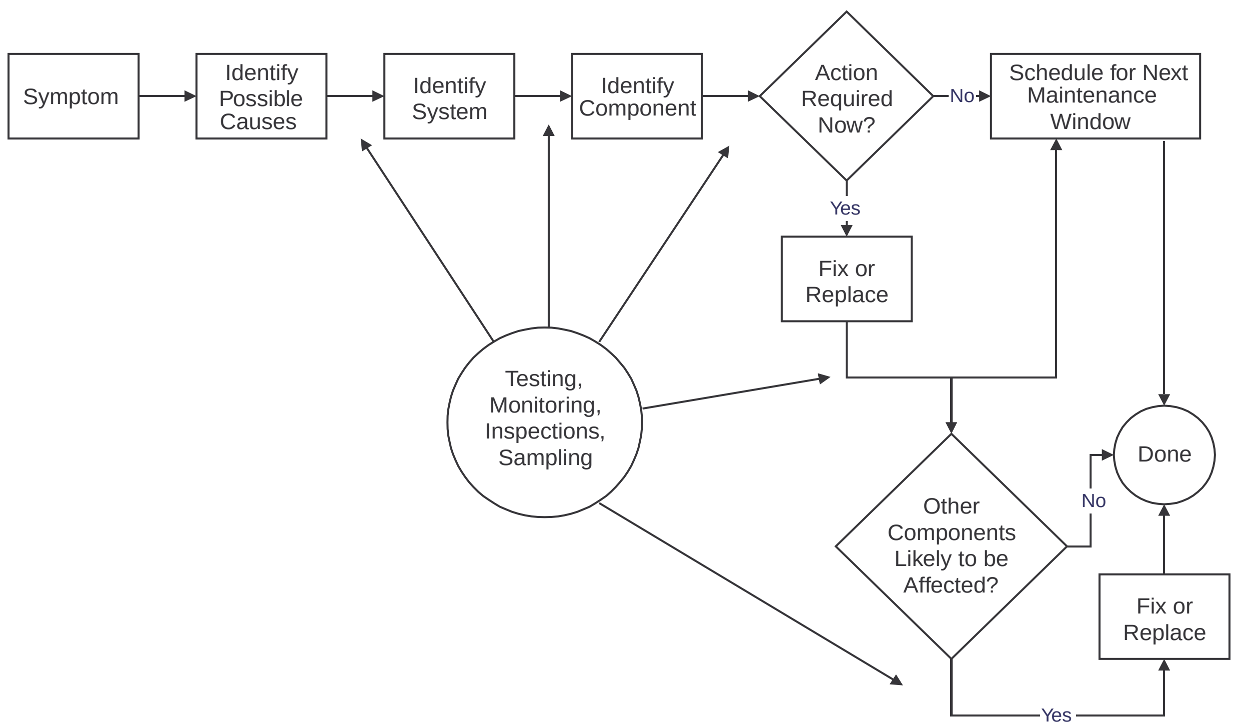

Figure 2-1
Integrated Maintenance Approach

For example if operating logs indicate low exhaust temperatures for one cylinder of an engine, some or all of the following actions might be taken:

- Check for a faulty or out-of-calibration thermocouple.
- Measure cylinder peak firing pressure and peak firing pressure angle to check for proper combustion.
- Perform ultrasonic and vibration analyses of the cylinder to check fuel injection and valve timing.
- Perform a borescopic inspection of the cylinder to identify damaged or leaking valves or plugged fuel injector nozzle holes (check the pattern on the piston crown).
- Check the fuel injector nozzle opening (pop) pressure.
- Monitor the crankcase vacuum to check for piston ring blowby.

Each of these additional techniques can be used to investigate potential causes of the low exhaust temperature.

2.4 Engine Operability

For operators of commercial diesel engines, engine analysis results are simply a maintenance tool used to help determine when engine maintenance is required and to troubleshoot potential problems. However, nuclear power plants are required to periodically demonstrate the operability of EDGs and to maintain them in a state of readiness. As a result, questions may arise regarding engine operability when engine analysis results indicate that engine degradation has occurred or might have occurred.

Some plant personnel have expressed concerns that if operating limits or guidelines are established, regulators will use them to question EDG operability or to force plants to declare EDGs inoperable whenever the limits or guidelines are exceeded. It appears that some plants have delayed or avoided the use of engine analysis due in part to this concern.

Some plant personnel advocate the use of engine analysis as an additional maintenance tool to be used to enhance and focus existing maintenance capabilities and believe that analysis results should not be used for operability decisions.

The Cooper-Bessemer Owners Group (CBOG) implemented a revised maintenance program for Cooper-Bessemer Model KSV engines in nuclear standby service in 1992 that relies on engine analysis and performance monitoring. The revised maintenance program was endorsed by Cooper-Bessemer and was issued as an update to the vendor recommended maintenance program. An engine analysis guide has been developed and published by CBOG as part of the revised maintenance program.

CBOG utilities gather engine analysis data for engineering and maintenance information purposes. When a parameter does not meet the agreed-upon guideline (see Section 6), plant personnel take appropriate maintenance action, which may include scheduling specific maintenance actions immediately or during the next scheduled outage, or increasing monitoring until the problem can be fixed.

To date, none of the CBOG stations have declared an engine inoperable or entered an EDG limiting condition of operation (LCO) based solely on engine analysis results. Conditions that could affect reliability of engines are generally scheduled for corrective maintenance. Several off-design conditions have been found through the use of engine analysis, allowing utilities ample time to prepare for the necessary maintenance actions during the next scheduled outage.

Most vendors for engines in nuclear standby service have not established limits for many of the parameters recorded during engine analysis. Typically, the only limits that exist are for engine balance parameters, such as cylinder peak firing pressures and exhaust gas temperatures. Reasons for this include:

- Engine balance criteria are typically based on an engine structural limitations, such as cylinder head or cylinder head stud loads, turbocharger or piston crown melting temperatures, and other structural limitations.
- Engines used in nuclear standby service were designed before the availability or widespread use of modern engine analysis equipment, and it was not anticipated that operators would have the need for detailed engine performance criteria.
- Interpretation of some engine analysis data may be considered qualitative due to the nature of the data and how it is recorded.
- Engine manufacturers might not be familiar with engine analysis techniques.

With these considerations in mind, one possible use of engine analysis results for operability considerations is as follows:

- Engine analysis guidelines that are based on engine manufacturer's engineering design bases could be considered operating limits that should not be exceeded and corrective actions should be taken if they are exceeded. For example, engine firing pressure limits are typically based on the mechanical limits of the power assembly and should not be exceeded. The limits on the balance of the firing pressures between cylinders are usually based on limiting excessive engine vibration and also should not be exceeded.
- Engine combustion, vibration, and ultrasonic analyses produce some results that are qualitative in nature and that must be interpreted to reach any conclusions regarding engine condition. Unlike engine balance, which is accomplished to ensure that certain criteria are met, combustion, vibration, and ultrasonic analyses are used to identify, localize, or troubleshoot known or suspected problems that might have been identified during engine balancing. In addition, combustion, vibration, and ultrasonic analyses can be used to focus intrusive maintenance activities. For example, the results could be used to identify the worst power cylinders for additional inspections during normal planned maintenance activities.

2.5 Degradation and Failure

All components have finite lives. In addition, most adjustments tend to drift somewhat over time for a variety of reasons. One of the challenges of engine analysis is the ability to distinguish between normal (anticipated) and abnormal (unanticipated) degradation, as well as between acceptable and unacceptable degradation. To support these concepts, methods have been developed for formulating acceptance guidelines, as well as methods of analysis for parameters such as cylinder-to-cylinder variation, cycle-to-cycle variation, and test-to-test variation. These are discussed in more detail in Sections 4 and 5.

3

ELEMENTS OF AN ENGINE ANALYSIS PROGRAM

3.1 Infrastructure

3.1.1 Management Support

As with any new program at a nuclear power plant, the first and most important element necessary for the establishment of a successful engine analysis program is the support of plant management. This support involves the commitment of utility resources, personnel, time, and funding to the development and maintenance of the program. The program will likely require the cooperation of several plant departments to run the engine and collect, analyze, trend, and interpret the data. It will likely take several engine analysis runs before personnel acquire the skills, techniques, and experience necessary to take full advantage of engine analysis as a maintenance tool.

Plant management should also recognize that it might be some time before any payoff is seen from engine analysis in terms of a reduction in intrusive maintenance or the discovery of an impending failure, although several utilities have had very good early successes that largely paid for the entire engine analysis program investment.

3.1.2 Engine Analysis Equipment

In order to perform engine analysis of a diesel engine, an engine analyzer is needed. The capabilities, features, and cost of engine analyzers vary from manufacturer to manufacturer and from model to model, just as they do with most equipment. Since engine analyzer vendors generally do not rent or lease engine analysis equipment, two options are available for nuclear power stations wanting to establish an engine analysis program.

The first option is to establish an in-house engine analysis program, where utility personnel collect, trend, and analyze the engine analysis data. In this case, purchase of engine analysis equipment is necessary. The cost of portable engine analyzer equipment, capable of measuring and recording cylinder pressure, vibration signatures, and ultrasonic signatures as a functions of crankshaft angle, currently ranges from approximately $25,000–$70,000.

Table 5-1 includes a summary of available engine analysis equipment, including cost and capabilities of the various analyzers. In addition, continued funding will be required for equipment repairs and upgrades, training classes, software, and any necessary calibration equipment.

The second approach available is to contract for engine analysis service. Many of the engine analysis equipment vendors and several consulting companies offer contract engine analysis services for fees ranging from $4,000–$10,000 per engine. This cost typically includes data analysis and a final report that documents engine maintenance recommendations. One caution to this option is that it is important to determine where and how often a contractor has successfully performed engine analysis on the particular engine make and model installed at the nuclear power plant. A contractor's lack of familiarity with off-design operating conditions could result in a great deal of wasted time and effort.

Many of the engine analysis equipment vendors will provide demonstrations of their equipment for station personnel. The demonstration sometimes includes analysis of one EDG, thus providing an introduction to engine analysis

At first glance, a quick comparison of the costs associated with each of these options might indicate that contracting for engine analysis services is the cheaper option. However, the number of engines to be analyzed and the frequency of the analysis must also be considered in the financial assessment. For utilities with a small number of tests to be performed, contracting for engine analysis services might be cost-effective, although the ability to quickly address emergent problems may be lost. For utilities with more tests to be performed, it is likely that purchase of engine analysis equipment will be more cost-effective.

If contract services are chosen, it is recommended that a long-term relationship be established with one company due to the emphasis on trending and specific engine experience necessary for a successful engine analysis program.

If the decision is made to purchase engine analysis equipment, the station personnel responsible for developing the engine analysis program are faced with selecting the optimal equipment from the wide variety of systems offered. However, with the ongoing efforts to limit nuclear power plant operating and maintenance costs, financial limitations heavily influence equipment purchases. The types of equipment available range from portable engine analyzers, which can be easily carried and operated by one person, to large systems of permanently mounted test instrumentation capable of recording dozens of parameters. Ideally, the engine analyzer chosen should have the following characteristics:

- Portability
- Digital storage of data
- Data export capability
- Good human factors
- Support calibration
- Built-in trending and analysis of stored data
- Low maintenance
- Measurement or calculation of all parameters of interest

3.1.3 Personnel

Whether engine analysis is performed by utility personnel or by contract personnel, it is recommended that the utility dedicate personnel to the engine analysis program, on either a full-time or part-time basis, and allow these personnel the time needed to develop the data collection and analysis skills required for successful engine analysis. Station management will endorse the engine analysis program and station personnel will be in control of the program only if station personnel are knowledgeable in engine analysis.

It will likely take several engine analysis runs and training sessions for these personnel to become comfortable gathering data and performing analyses, and initial program implementation will likely require a significant amount of engineering effort. As personnel become familiar with engine analysis techniques, the time required for data collection and analysis can be expected to decrease.

The first time engine analysis is performed on an EDG at a nuclear power plant, it can be expected to take up to 8 person-hours to complete the data collection and 20 person-hours to analyze the data, assuming no major anomalies are found in the engine. After personnel are experienced with the techniques, these times can be expected to decrease to approximately 6 and 8 person-hours respectively.

3.1.4 Equipment Calibration and Maintenance Program

Engine analysis equipment must be calibrated and maintained in order to ensure the accuracy of the data collected. Because the data will be used to support EDG maintenance and operating decisions, it is critical that data collection errors be minimized. An effective calibration program identifies equipment problems before the equipment is used to collect data, helping to avoid misdiagnosis of EDG problems. Both the data collection unit and the pressure transducers and accelerometers require calibration.

In addition to regular periodic calibration of engine analysis equipment, calibration checks should be performed following data collection to ensure that instruments have not drifted.

Maintenance of engine analysis equipment is crucial in order to ensure that scheduled engine analysis runs are not delayed by equipment failures. Many of the engine analysis equipment vendors offer extended warranties on their equipment. In addition to covering necessary repairs or replacement, these warranties may include hardware and software upgrades for a specified period of time.

Obviously, if engine analysis services are contracted, equipment calibration and maintenance are the responsibility of the contractor. However, station personnel should ensure that the contractor arrives at the plant with calibrated and operable equipment. There have been instances where engine analysis tests have been delayed due to missing, failed, or out-of-calibration equipment.

3.1.5 Training

Most engine analysis equipment vendors and many independent contractors offer engine analysis training. Two types of training are typically given. The first is training on the use of the particular engine analysis equipment, including how to set up, install, and use the equipment and its associated software. This training is generally given by the equipment vendor and much of it is specific to the make and model of the engine analysis equipment used. Actual engine analysis, is typically not addressed. The second type of training available is training on the interpretation and analysis of the data collected. This type of training is typically available from both engine analysis equipment vendors and independent contractors.

It is important to obtain training specific to the type of engine to be analyzed. Engine analysis methods vary significantly between natural gas and diesel engines and between two-stroke and four-stroke engines.

3.2 Programmatic Issues

3.2.1 Integration with Other Maintenance Programs

As discussed in Section 2, engine analysis is most effective when it is used in combination with other non-intrusive maintenance or inspection techniques, such as lube oil analysis. However, it takes a combined effort by several departments at a nuclear power plant to utilize the inspection techniques and draw conclusions from the data. In order for an integrated maintenance program to be successful, it must have the backing of plant management to allocate the resources and ensure cooperation among the various departments.

3.2.2 Parameters to Monitor

With modern equipment, it is possible to monitor hundreds or even thousands of parameters. However, just because it is possible to do something does not mean that it is appropriate to do so. Diesel engine performance parameters that should be monitored include those that:

- Are relatively easy to monitor
- Can have potential responses developed
- Can indicate and warn of anticipated failures
- Can be used to develop or calculate additional performance parameters that are candidates for monitoring

Suggested parameters for monitoring in a diesel engine analysis program include:

- Cylinder pressures (compression and firing) versus crankshaft position
- Vibration signatures versus crankshaft position
- Ultrasonic signatures versus crankshaft position
- Engine data such as auxiliary system temperatures and pressures

Most modern digital engine analyzers are capable of recording cylinder pressure and vibration and ultrasonic signatures as functions of crankshaft position. Auxiliary system temperatures and pressures are typically recorded for EDGs at nuclear power stations as part of the EDG operating logs. However, several stations now have the capability to monitor and record these parameters on installed computer-based monitoring systems.

3.2.3 Monitoring Frequency

In addition to being capable of monitoring thousands of parameters, modern equipment is also capable of monitoring parameters on a near-continuous basis. Once again, however, just because something is possible does not mean that it is appropriate. Instead, the monitoring frequency for both engine analysis and other maintenance methods should be determined by the lead time required to evaluate the information and respond to any potential incipient failures with sufficient time to prevent equipment damage. As a result, the periodicity will vary from site to site, depending on the maintenance methods used, the expertise of station personnel, and the manpower available.

Stations that have established engine analysis programs, typically perform engine analysis two or three times per fuel cycle. Engine analysis is performed following outage work to confirm EDG performance at the beginning of the cycle. In plants with an 18-month or 24-month fuel cycle, engine analysis is often performed again at the midpoint of the cycle. Finally, engine analysis is performed a few months before each refueling outage both to confirm engine condition and to identify any components requiring maintenance or inspection during the upcoming outage.

The post-maintenance engine analysis should be completed and problems corrected prior to performing surveillance tests that could lead to valid start or load failures. In this case, engine analysis is used to show that outage maintenance was performed properly and that the engine was reassembled correctly. In addition, engine analysis can be used to investigate or diagnose specific problems that occur at other times during the operating cycle.

3.2.4 Standardized Data Collection Methods

Since much of the analysis of engine analysis data consists of trending and comparing the data, it is critical that variations in measured engine parameters due to external factors be minimized. External factors include:

- Equipment calibration
- Data collection techniques
- Differences in equipment
- Changes in engine load
- Environmental factors.

Equipment calibration was discussed in Section 3.1.4 and is addressed in detail later in this guide.

Differences in analysts' techniques have the potential to affect engine analysis results and, possibly, the conclusions drawn from the results. For example, each analyst might take vibration and ultrasonic signatures at a different location on the cylinders. As a result, the vibration and ultrasonic signatures likely are not directly comparable, and incorrect conclusions might be drawn regarding the condition of one or more cylinders. Therefore, it is recommended that vibration and ultrasonic signatures be taken in the same location on each cylinder and during each engine analysis run.

To date, there is no industry standard for data collection, processing, and display. Some engine analyzers may process the recorded data differently than others do. Many of the analyzers filter and process cylinder pressure signals and vibration and ultrasonic signatures before recording or displaying the data. If the equipment used on a particular engine is changed from one analysis run to the next, the changes in the data might be masked by changes in the data processing performed by the engine analyzer. Should it be necessary to change the type of analyzer used from one engine analysis run to the next, the filtering and processing performed by both analyzers should be well understood.

Variations in engine load directly affect individual cylinder loads. As a result, any change in engine load during the engine analysis run affects the data collected. In order for cylinder-to-cylinder trending to be effective, the engine load during data collection for each cylinder must held constant. In addition, for trending to be effective from one engine analysis run to the next, total engine load must be the same for each run.

One method to ensure that engine load is constant is to lock the governor. When the governor is locked, the load limit knob on the governor actuator is used to minimize load swings. Use of the load limit knob prevents the fuel racks from moving unless the EDG undergoes a complete load rejection or overspeeds; it does not disable any of the safety features of the EDG. However, it might not be appropriate to lock the governor at all sites.

If the governor cannot be locked, operators must pay close attention to generator load during engine analysis testing to minimize load swings. If the governor is locked during engine analysis, the load limit knob must be returned to its original setting at the conclusion of the data collection.

Variations in engine inlet air temperature from season to season, or even from day to night, affect engine analysis data. Peak firing pressures and exhaust temperatures are higher when ambient air temperatures are higher. These changes can complicate data analysis and, in some cases, make a well-balanced engine appear to be imbalanced. Therefore, it can be useful for stations to establish methods to adjust engine analysis results for varying environmental conditions.

One method of adjustment is through the use of engine performance maps. A thorough discussion of engine performance maps is outside the scope of this guide; however, more details can be found in References 1 and 2.

3.3 Analysis

3.3.1 Trending

Much of the analysis of engine analysis data consists of trending results from one cylinder or engine to another or from one test to another. As a result, one of the necessary skills of an engine analyst is that of interpreting changes. Engine analysts spend a good portion of their time comparing newly recorded engine analysis data to previously recorded data in an effort to identify signs of engine degradation or faults.

This is particularly true for analysis of vibration and ultrasonic signatures. Vibration and ultrasonic signature analyses are qualitative processes where comparisons are made between different signatures. Changes in timing and amplitude of engine events are possible indications of changes in the condition of engine components. In addition, the appearance of new events or the disappearance of previously seen events are also possible indications of changes in engine material condition.

3.3.2 Baseline Data

Most of the manufacturers of diesel engines used in U.S. nuclear service have not published or released typical or expected values for the parameters measured during engine analysis. As a result, individual nuclear power stations or diesel owners groups have to determine what the baseline or normal values for these parameters are. This can be done by performing engine analysis on an engine that is known to be well-balanced and in good mechanical condition.

Baseline engine analysis data is necessary as the starting point for parameters that are trended and for vibration and ultrasonic analysis that is comparison based. If data for a well-running engine are not available, it might not be possible to identify a poorly running engine.

It should be noted that baseline data collected on one engine might not be applicable to another engine of the same make and model due to configuration differences or differences in rated load. Name plate data and operating data for both engines should be compared prior to using baseline engine analysis data from one engine on another engine.

3.3.3 First Time Engine Setup

Before data that can be considered baseline data can be collected, the engine must be in good operating condition and well-balanced. As a result, it might be necessary to balance or tune the engine prior to running it for the purpose of collecting baseline data. The state of engine balance can be checked by performing engine analysis.

If the analysis shows that the engine is well-balanced, this data can be considered baseline data. If it turns out that the engine is not well-balanced, it must be balanced prior to the collection of baseline data. Section 5 of this guide discusses engine balancing.

3.3.4 Acceptance Guidelines

For some performance parameters monitored during engine analysis, numerical limits are appropriate and are either available or can be developed. These numerical acceptance guidelines are used as performance criteria to determine when engine maintenance or adjustments are needed. Parameters for which numerical acceptance guidelines are appropriate include:

- Engine average peak firing pressure
- Maximum range of cylinder average peak firing pressures
- Engine average peak firing pressure angle
- Maximum range of peak firing pressure angles
- Engine average fuel injection timing
- Maximum range of engine fuel injection timing
- Maximum range of fuel rack settings
- Fuel injector nozzle pop pressures
- Maximum cylinder exhaust gas temperature
- Maximum range of exhaust gas temperatures

While several of the engine vendors have published desired or expected values for some of these parameters, only Cooper-Bessemer has established guidelines for all of them. For other engines, analysis guidelines must be developed. Potential sources of analysis guidelines include:

- Engine manufacturer design data
- Baseline data collected on a well-running (balanced) engine
- Diesel engine owners group development
- Outside consultant development
- Nuclear power station development

More details on the development of engine analysis guidelines are contained in Section 4 of this guide.

3.3.5 Case Histories

A library of case histories is a particularly useful tool for the engine analyst. Case histories are examples of data taken during engine analysis runs of engines with known faults. The purpose of the data is to illustrate analysis results that are indicative of specific modes of engine degradation or poor engine analyzer setup. While none of the case histories exactly match results for a given engine, the characteristics of the curves due to the specific mode of degradation are identified in the case histories and can be compared to data collected during engine analysis runs to identify similarities to assist in the identification of the specific engine fault.

If engine analysis identifies an engine fault that is subsequently corrected, the engine analysis data that identified the fault should be added to the library of case histories. If a fault occurs that was not identified beforehand by engine analysis, the engine analyst should review previously collected engine analysis data to determine if the fault provided indications of its presence, but was not detected. Again, any data that are found to provide warning of the fault should be added to the library of case histories.

Sharing case histories among stations with the same make of diesel engine is an easy and cost-effective way to gain the benefits of experience without suffering the associated failures directly. If one station suffers an engine failure that provided warning of its occurrence through engine analysis, other stations with similar engines need not suffer the same failure if the case history is made available to them.

In addition to their benefit for future fault identification, case histories provide a written record of the successes and capabilities of an engine analysis program.

3.3.6 Analysis Flow Charts

Analysis flow charts consist of step-by-step instructions on how to perform engine analysis. Generally, they are specific to each make and model of diesel engine. The analysis flow charts are not intended to take the place of station procedures, but are intended to assist operators and system engineers by ensuring that the analysis and evaluation of engine data is thorough and repeatable and that any troubleshooting steps are accomplished in an efficient and logical order.

It is recommended that the individual diesel owners groups develop or fund development of analysis flow charts for their specific engine make and model to ensure that the stations take the same general approach to engine analysis. This will assist in making engine analysis results transferable from one station to another.

3.3.7 Feedback from Maintenance Personnel

When engine analysis is performed, the analyst should make an effort to be present for the data collection to be aware of any anomalies or difficulties experienced in the data collection effort. If the analyst is aware of conditions during the engine analysis run, it can eliminate later questions regarding the data collected.

In addition to witnessing the data collection, the analyst should make an effort to get feedback from station maintenance personnel regarding problems experienced with the engine, both during engine analysis and during normal standby or operating conditions. When the experience of the maintenance personnel is combined with the engine analysis data, it can provide insight to engine faults that would otherwise be missed.

4

THEORY

4.1 Combustion

The diesel engine is a reciprocating internal combustion engine. Each mechanical cycle of a diesel engine involves intake, compression, expansion, and exhaust of a fuel-air mixture. Engines are classified by the type of mechanical working cycle used. Two working cycles are used in diesel engines for U.S. nuclear standby service: the two-stroke cycle and the four-stroke cycle. Engines that use a two-stroke cycle are called two-stroke engines. Engines that use a four-stroke cycle are called four-stroke engines.

A stroke is defined as the motion of the piston between the points of minimum and maximum cylinder volume, referred to as top dead center (TDC) and bottom dead center (BDC), respectively (see Figure 4-1). Each stroke consists of 180° of crankshaft rotation. At TDC and BDC, the piston reverses direction and is, for an instant, motionless.

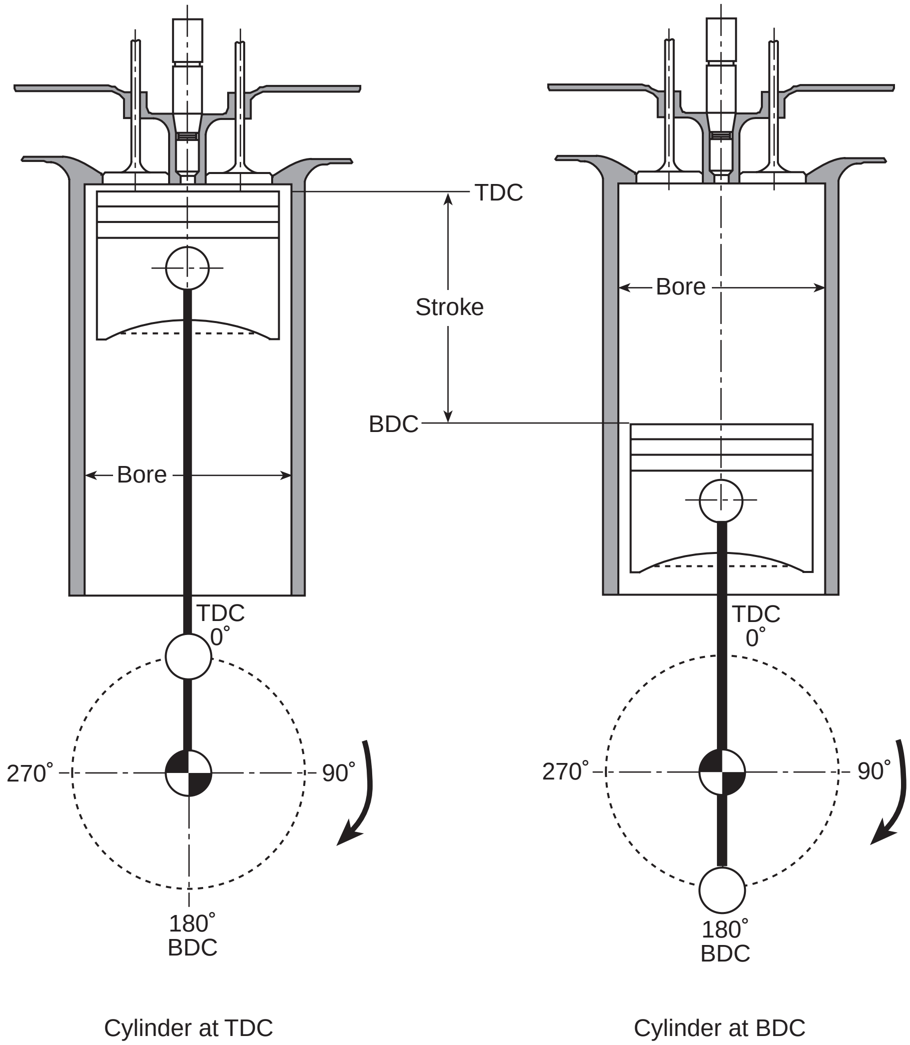

Figure 4-1
Piston Stroke: Top and Bottom Dead Center

The Fairbanks Morse Opposed-Piston (FMOP) engine has two crankshafts that are geared together so that the lower piston reaches inner dead center (IDC) or outer dead center (ODC) several degrees before the upper piston. This difference is referred to as the *crank lead*. As a result of the crank lead, the positions of minimum and maximum cylinder volume in a FMOP engine occur just after the lower piston has passed through IDC or ODC, but before the upper piston has passed through IDC or ODC (see Figure 4-2).

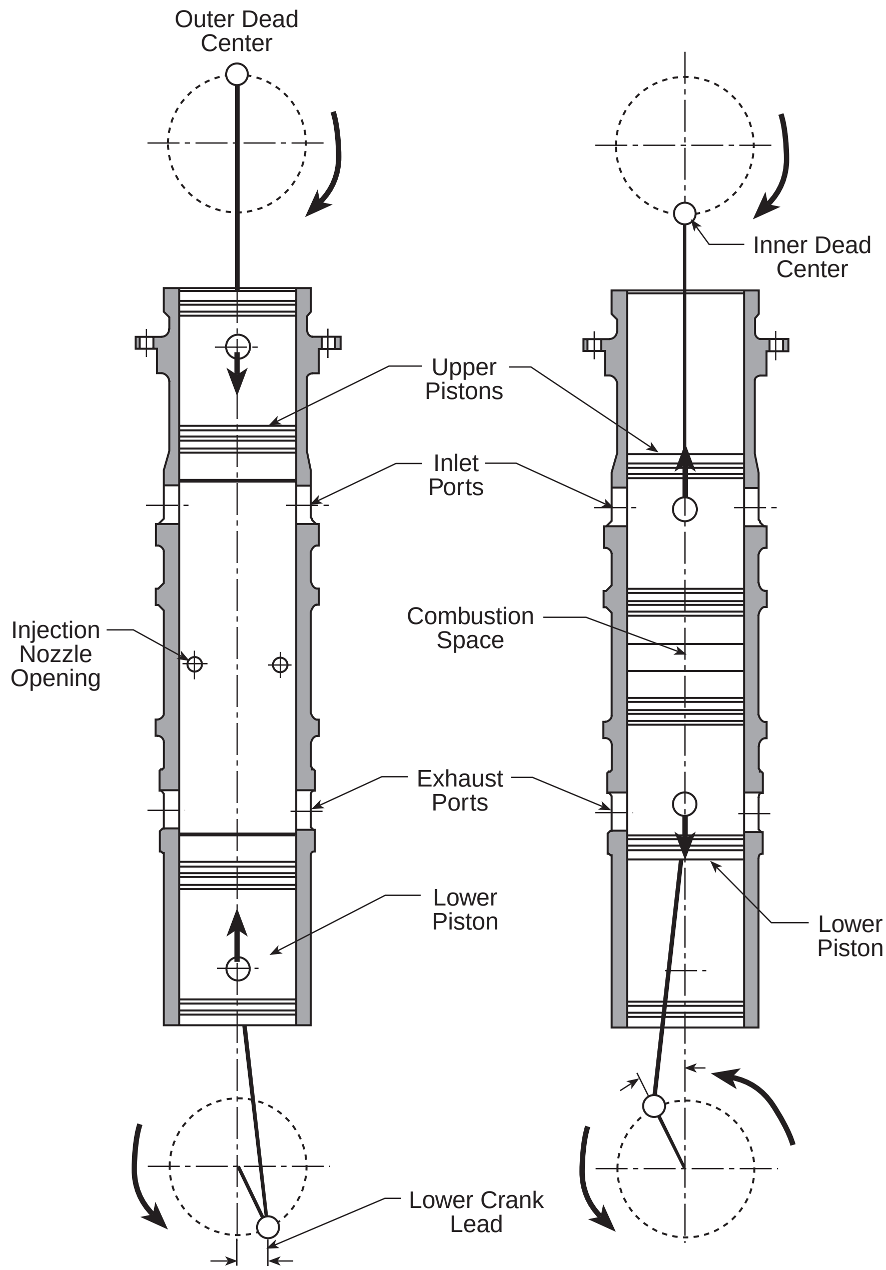

Figure 4-2
FMOP Engine: Inner and Outer Dead Center Arrangement

4.1.1 Two-Stroke Engines

In a two-stroke engine, the mechanical cycle is completed in two strokes of the piston or one revolution (360°) of the crankshaft. The two strokes are (see Figure 4-3):

- A compression stroke occurs when the piston compresses the cylinder contents as it moves from BDC to TDC (or from ODC to IDC for an FMOP engine), during which the inlet ports and exhaust ports / valves are closed. Shortly before the end of the stroke, fuel is injected and combustion begins.
- A power or expansion stroke occurs when the high-pressure products of combustion drive the piston from TDC to BDC (or IDC to ODC for an FMOP engine). Before the piston reaches BDC, the inlet ports and exhaust ports / valves are opened to exhaust the combustion products and take in a new charge of fresh air. The cycle then repeats.

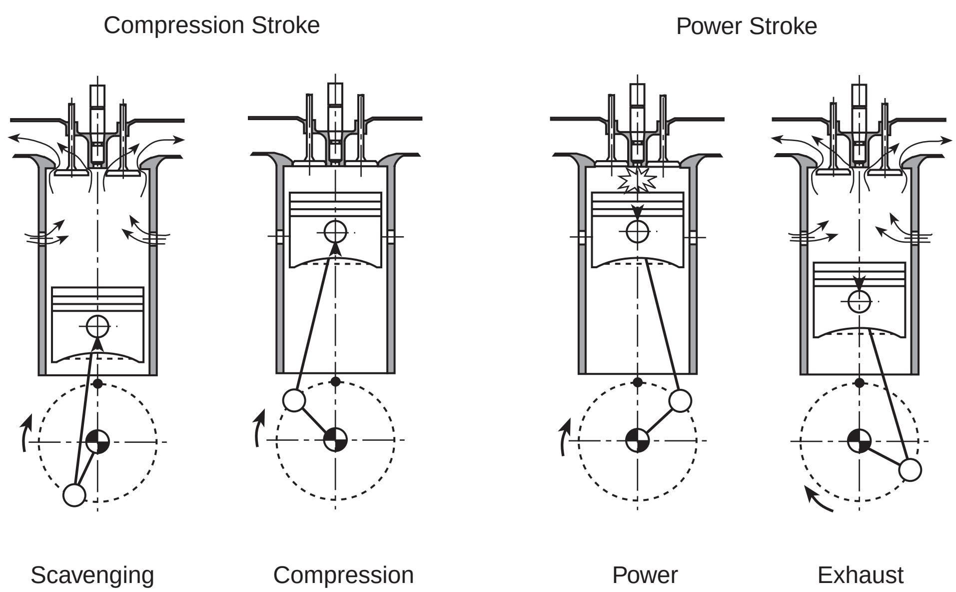

Figure 4-3
Two-Stroke Engine Cycle

Because combustion occurs once per crankshaft revolution in a two-stroke engine, air intake and exhaust flow must also be completed during each revolution. The exchange of exhaust gas from one combustion event for a charge of fresh air for the next combustion event must take place during a short time period. As a result, two-stroke engines normally use some type of forced induction in order to sweep out or *scavenge* the exhaust gases from the combustion chamber. Several gas exchange, or scavenging processes are used in two-stroke diesel engines. These include:

- Loop Scavenging (see Figure 4-4). In a loop-scavenged engine, the inlet and exhaust processes are controlled by ports in the cylinder wall. A single piston covers and uncovers the ports, allowing intake and exhaust to occur. While loop scavenging is the simplest scavenging method, it has several disadvantages. The symmetrical timing leads to mixing of the inlet air and combustion products, reducing the purity of the air charge. None of the engines used in nuclear standby service in the U.S. use loop scavenging.

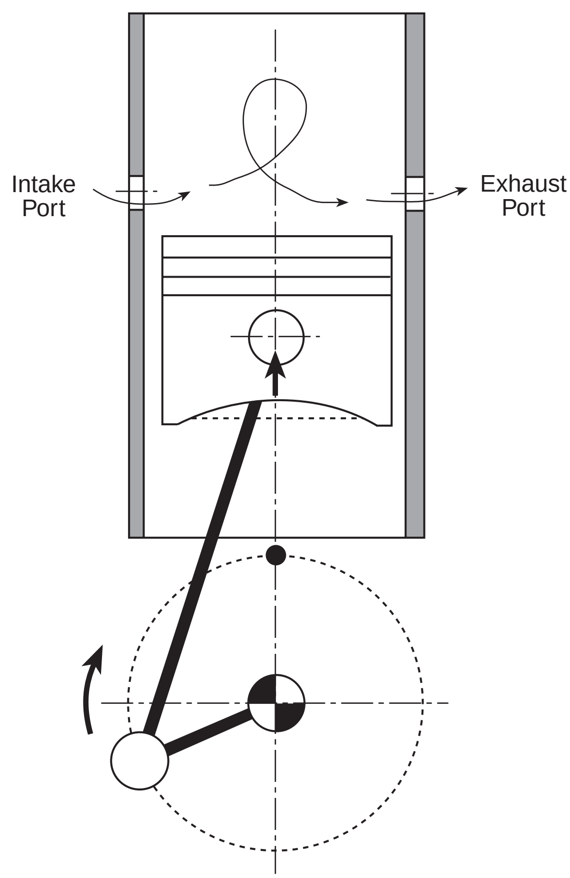

Figure 4-4
Loop Scavenging

- Uniflow Scavenging with Single Piston (see Figure 4-5). In the uniflow-scavenged, single-piston engine, air intake is controlled by ports in the cylinder wall, and the exhaust is controlled by camshaft-operated exhaust valves. The advantages of this system are that the inlet air tends to displace, rather than mix with, the combustion products, and the timing of the inlet and exhaust processes can be adjusted to prevent loss of the fresh air charge to the exhaust. The EMD 645 is a uniflow-scavenged, single-piston engine.

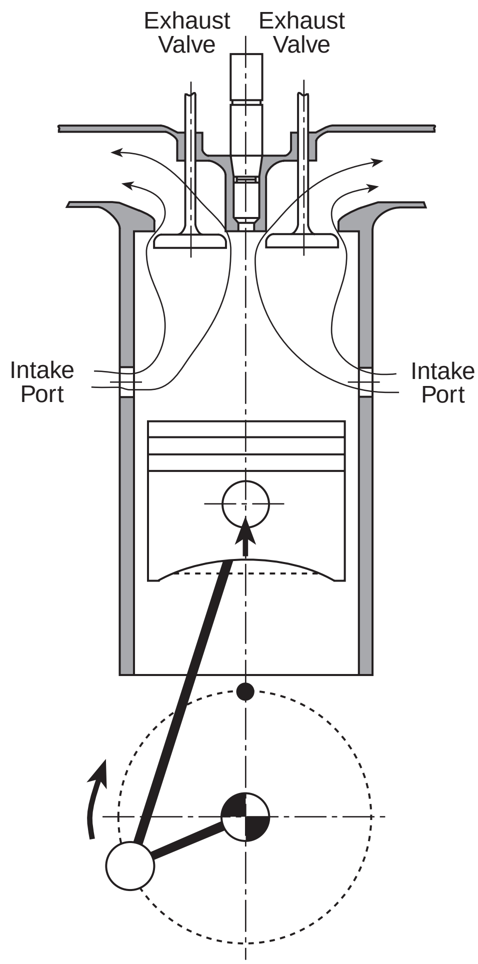

Figure 4-5
Uniflow Scavenging with Single Piston

- Uniflow Scavenging with Opposed Pistons (see Figure 4-6). In a uniflow-scavenged, opposed-piston engine, the inlet ports are controlled by one piston and the exhaust ports are controlled by the other piston. This system has the same advantages as the uniflow-scavenged, single-piston engine but is more complicated mechanically. Its advantage is the high specific power output that can be achieved from each cylinder with two pistons. The FMOP engine is a uniflow-scavenged, opposed-piston engine.

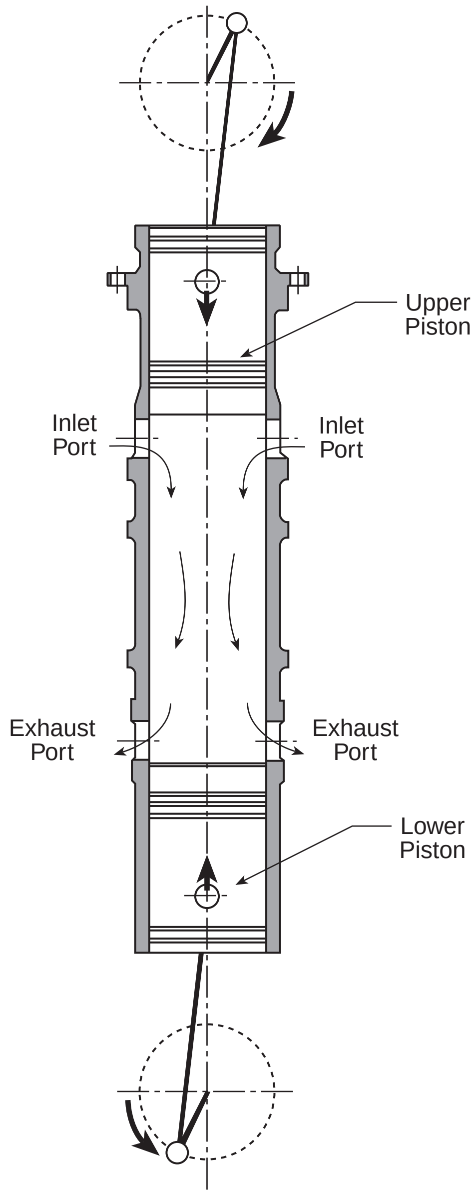

Figure 4-6
Uniflow Scavenging with Opposed Pistons

4.1.2 Four-Stroke Engines

In a four-stroke engine, the mechanical cycle is completed in four strokes of the piston or two revolutions (720°) of the crankshaft. These four strokes are (see Figure 4-7):

- An intake stroke occurs when a charge of fresh air is forced into the cylinder as the piston moves from TDC to BDC. The inlet valves open shortly before the intake stroke begins and close shortly after it ends.
- A compression stroke compresses the air charge as the piston moves from BDC to TDC. Toward the end of the compression stroke, fuel is injected into the cylinder and combustion begins.
- A power or expansion stroke occurs when the high-pressure products from combustion drive the piston from TDC to BDC. The exhaust valves open before the end of the power stroke.
- An exhaust stroke occurs when the piston moves from BDC to TDC. There is typically some residual pressure remaining in the cylinder at the end of the power stroke, so the exhaust stroke begins with an initial rush of combustion products past the exhaust valves. The inlet valve opens toward the end of the exhaust stroke, and the exhaust valves close shortly after the exhaust stroke ends. The cycle then repeats.

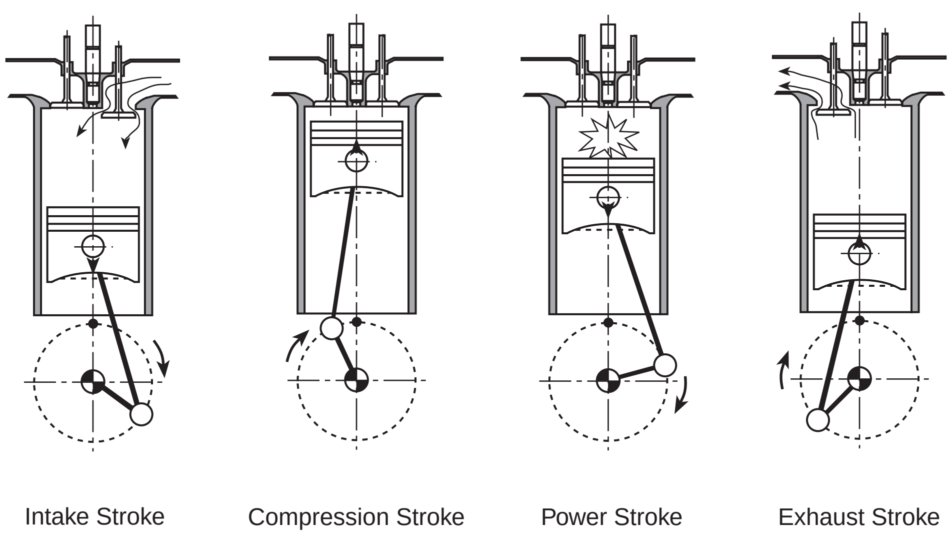

Figure 4-7
Four-Stroke Cycle

Because combustion occurs every other revolution of the crankshaft in a four-stroke engine, the remaining revolution can be used for gas exchange. The ALCO 251, Colt-Pielstick PC2 series, Cooper-Bessemer KSV, Cooper-Enterprise DSR and DSRV, SACM UD45, and the Worthington SWB12 are four-stroke engines.

4.1.3 Four Phases of Combustion

For discussion purposes, the combustion process in a diesel engine is typically divided into four phases [3]. The four phases are described below and shown in Figures 4-8 and 4-9, which are plots of the cylinder pressure and rate of heat release as functions of crank angle. The plots shown are typical of both two- and four-stroke engines.

- The ignition delay phase is the time between the start of fuel injection and the start of combustion. Fuel is injected into the cylinder at a temperature that is necessarily below its auto-ignition temperature of approximately 950°F [4]. During the ignition delay phase, the injected fuel is atomized, vaporized, mixed with air, and raised to its auto-ignition temperature.
- When the fuel has reached its auto-ignition temperature, combustion occurs and the rapid combustion phase begins. This phase is characterized by a rapid rise in pressure due to the rapid combustion of the fuel that mixed with air and reached auto-ignition conditions during the ignition delay phase. In addition, some of the fuel that enters the cylinder during this phase reaches ignition conditions and burns during this phase. Heat released by the burning fuel produces increased cylinder pressure. This pressure acts on the surface of the piston, which creates torque on the crankshaft. The rapid combustion phase ends when the peak firing pressure is reached.
- The third stage is the mixing-controlled combustion phase. The premixed fuel supply has been consumed during the rapid combustion phase, and the rate of combustion during the mixing-controlled combustion phase is controlled by the rate of fuel injection and fuel-air mixing. This phase ends when fuel injection ends.
- The final stage is the late combustion stage. In this stage, heat release continues into the expansion stroke as unburned and partially burned fuel burn when they come into contact with oxygen. However, cylinder pressure declines rapidly as piston motion increases the cylinder volume.

In a typical diesel engine, approximately 80% of the heat release occurs during the rapid and mixing-controlled combustion phases [3].

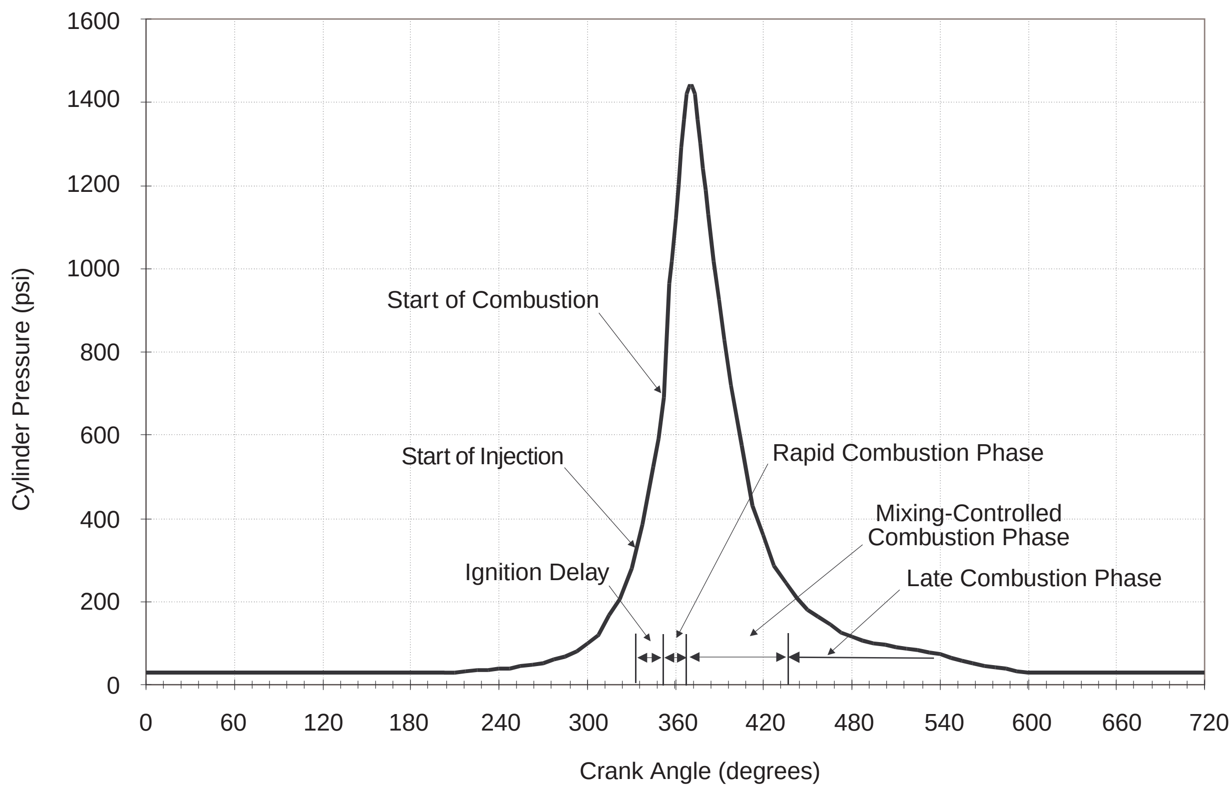

Figure 4-8
Cylinder Pressure Versus Crankshaft Angle

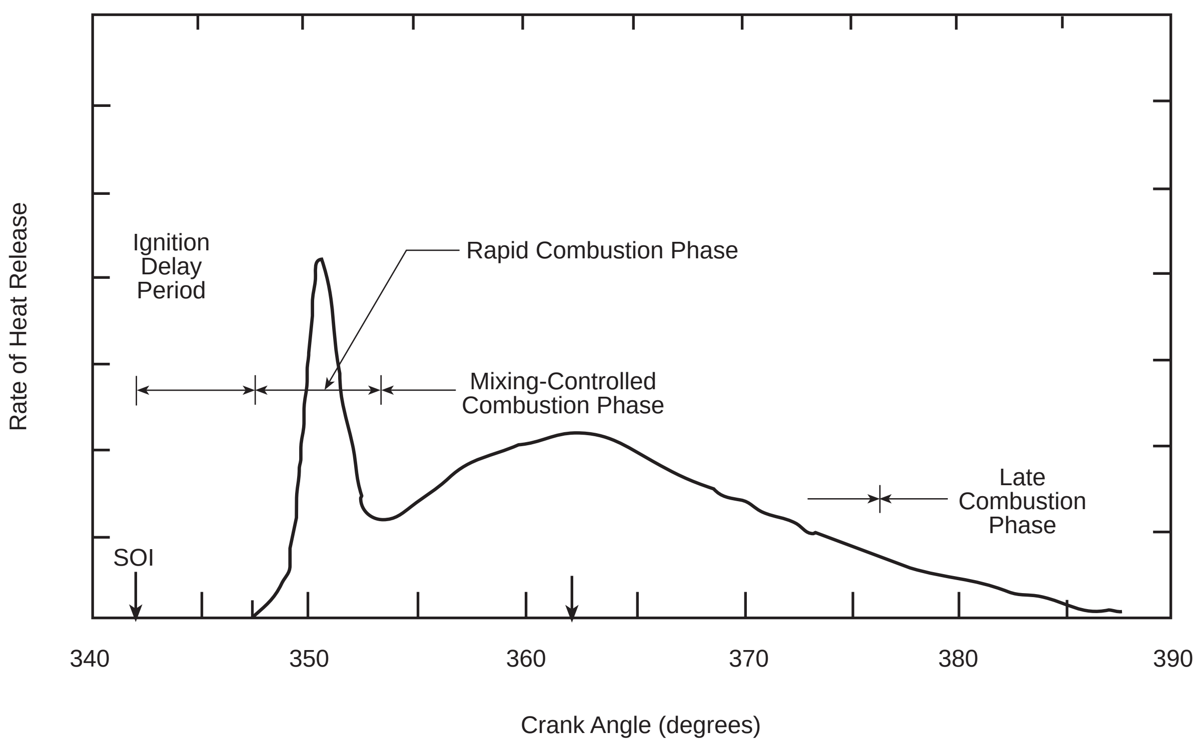

Figure 4-9
Rate of Heat Release Versus Crankshaft Angle

4.1.4 Engine Parameters

The engine output torque drives the generator. Engine torque is normally measured with a dynamometer, which is a mechanical, electromagnetic, or hydraulic load attached to the engine crankshaft. The power delivered by the engine to the dynamometer can be measured. This power is known as the *brake power* (P_b) because the dynamometer acts as a brake during the test. The brake power can be divided by the engine speed (for two-stroke engines) or by one-half the engine speed (for four-stroke engines) to calculate the brake work per cycle.

Cylinder pressure data can be used to calculate the power transfer from the cylinder gas to the piston. The energy released by the combustion of the fuel in the cylinder is converted to work through the action of gas pressure on the pistons. By definition, work is produced when a force (F) acts through a distance (x).

$$W = \int F dx$$

The force (F) is produced by cylinder pressure acting on the piston surface. Substituting pressure (P) times piston surface area (A) for the force (F) and the change in cylinder volume (dV) divided by piston area (A) for the distance traveled (dx) yields

$$W = \int F dx = \int PA \frac{dV}{A} = \int P dV$$

Using the above equation, cylinder pressure data can be used to calculate the work performed by the gas on the piston. For a complete mechanical cycle, the work performed in the cylinder is given by

$$W_i = \oint P dV \qquad \textit{Eq. 4-1}$$

If cylinder pressure is plotted as a function of cylinder volume, the area inside the curve represents the indicated work. For a two-stroke engine, where the pressure versus volume curve is a single loop (see Figure 4-10), it is quite easy to determine the *indicated work* (W_i). For a four-stroke engine, where the diagram consists of two loops (see Figure 4-11), the area in the upper loop is the positive work from compression and combustion while the lower loop represents the pumping work to exhaust the combusted gases and draw in a fresh charge of air. The pumping work is always negative on normally aspirated engines, but may be zero or positive on highly turbocharged diesel engines if the inlet manifold pressure is greater than the exhaust manifold pressure.

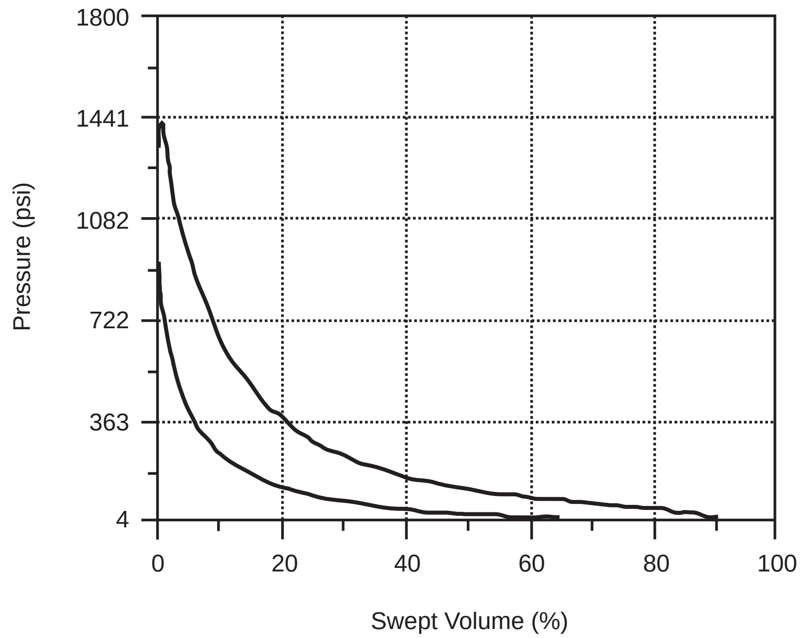

Figure 4-10
Two-Stroke Pressure Versus Volume Curve

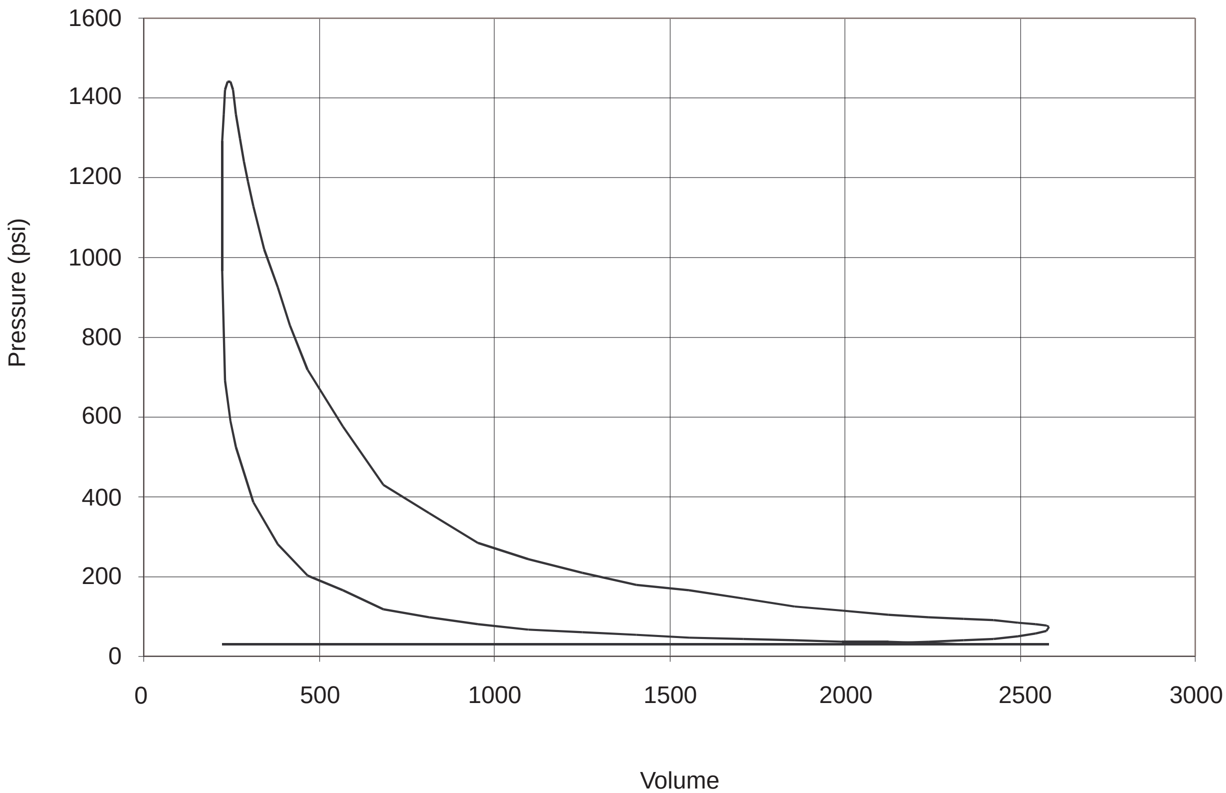

Figure 4-11
Four-Stroke Pressure Versus Volume Curve

The work performed by the gas on the piston is known as the *indicated work* (W_i). The rate at which all indicated work is done is known as the *indicated power* (P_i), and is given by

$$P_i = \frac{WN}{n}$$

where

W is the indicated work calculated from Equation 4-1
N is the engine speed
n is the number of crankshaft revolutions per power cycle
(2 for 4-stroke engines and 1 for 2-stroke engines).

Indicated power is usually given in units of horsepower and referred to as indicated horsepower (IHP).

The difference between the brake power and the indicated power represents the *friction power*. The friction power includes the power necessary for gas exchange, power used in overcoming friction between engine components, and the power necessary to drive engine auxiliaries such as pumps. The ratio of the brake power to the indicated power is called the mechanical efficiency of the engine:

$$\eta = \frac{P_b}{P_i}$$

The terms *brake* and *indicated* are used to describe many other engine performance parameters. When discussing engine performance parameters, it must always be understood whether a given parameter is a brake or indicated value.

The *indicated mean effective pressure* (IMEP) is defined as the indicated work divided by the cylinder volume displaced by the piston during each cycle. It is a measure of torque per unit displaced volume. IMEP is also equal to the constant pressure that, if applied only during the entire power stroke, would yield work equal to the work of the complete cycle (see Figure 4-12). It is calculated as

$$IMEP = \frac{W_i}{V_2 - V_1} = \frac{W_i}{\Delta V}$$

Delta V is the cylinder volume swept by the piston (equal to the piston stroke times cylinder cross-sectional area). V_2 is the maximum cylinder volume (at BDC). V_1 is the minimum cylinder volume (at TDC).

Rearranging and substituting Equation 4-1,

$$IMEP = \frac{P_i n}{N \Delta V} \qquad \textit{Eq. 4-2}$$

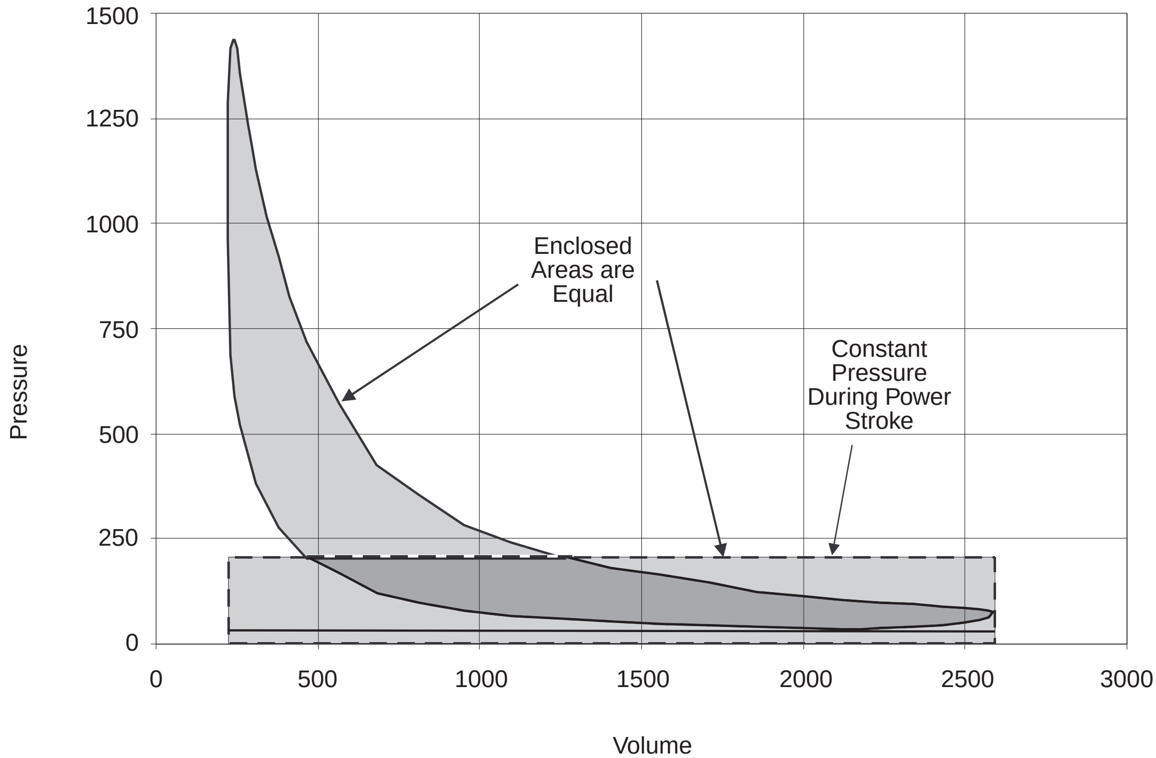

Figure 4-12
Indicated Mean Effective Pressure

Brake mean effective pressure (BMEP) can be calculated similarly using brake power.

4.1.5 Measurement Methods

Historically, engine manufacturers have used cylinder pressure and volume data in engine research and design, in particular to determine the work done in the cylinder per cycle using Equation 4-2. Peak pressures in large, slow-speed engines could be measured using pressure gages and check valves. Engine indicators were used to plot cylinder pressure versus cylinder volume curves for one mechanical cycle. These plots became known as indicator diagrams or indicator cards. An indicator used a piston and cylinder connected to the engine so that the engine cylinder pressure acted on the indicator piston. The indicator piston was connected to a pencil that recorded the cylinder pressure on a card that was given a reciprocal motion proportional to the movement of the engine piston. However, both peak pressure gages and engine indicators are only useful for slow speed engines.

Instruments that measure the pressure at only one point during the mechanical cycle are also available. By determining the pressure at many points throughout the cycle, a complete history of the cylinder pressure during the cycle can be created. However, because it takes several hundred cycles to create a single pressure history, this method was less accurate because the instantaneous cylinder pressure varies from one cycle to the next. As reviewed in Section 6, variations in peak cylinder pressures of up to 2% from one cycle to the next are not unusual in diesel engines.

Fortunately, the development of fast acting pressure transducers has allowed combustion analysis techniques to be improved and used on faster running engines. These transducers rely on materials that convert mechanical deformation into electrical signals. The most common types in use for engine combustion analysis are resistive (strain-gage), capacitive, and piezoelectic transducers. On most engines, the transducer is connected to an *indicator valve*, which is connected to the cylinder by an *indicator passage* machined through the cylinder head. A detailed discussion of pressure transducer characteristics and use is included in Section 5.

Cylinder volume is a function of engine geometry and the position of the piston in the cylinder. Piston position can be calculated using engine geometry data and crankshaft position or angle. The crankshaft position information comes from a reference signal derived from the crankshaft. There are two methods commonly used to obtain the crank angle position. The first method is to use a once-per-revolution pulse that is taken from an interruption device. An interruption device uses some anomaly that is produced once with each revolution of the shaft. This interruption is usually positioned at the TDC of the number one piston for reference purposes. The anomaly can be anything from a reflective piece of tape to a high or low spot on the flywheel. This once-per-rotation interruption is used to create an electrical pulse to indicate each time that point passes a pickup or sensor.

The once-per-turn method has one potentially significant disadvantage. With only one trigger per revolution, the intermediate angles are interpolated assuming constant rotational velocity. Therefore, if the engine does not turn a consistent angular velocity, P-θ data accuracy will be affected. To overcome this uncertainty, it is preferable to use a shaft encoder. This type of crankshaft position device provides a pulse train at fixed angular increments in addition to the once-per-turn pulse required for absolute positioning. Commonly there is one pulse per degree, resulting in 360 reference points per revolution.

The most common method of measuring crank angle is through the use of optical or magnetic encoders. Many engine analyzers are supplied with an encoder tailored to the analyzer.

4.2 Combustion Analysis

4.2.1 Combustion Curves

Several different combustion curves can be created from the cylinder pressure versus crankshaft angle data. The characteristics and features of these curves can be used to evaluate engine condition.

4.2.1.1 Cylinder Pressure Versus Crank Angle Curve (P-θ)

The starting point of diesel engine analysis is the cylinder pressure versus crank angle, or P-θ, curve. The P-θ graph displays the pressure in the combustion chamber as the crank shaft rotates through one complete mechanical cycle of 360° (2-stroke engine) or 720° (4-stroke engine).

For a given engine, the nominal pressure versus crank angle curve can be established. To assess engine performance, new cylinder patterns are evaluated and compared with the nominal patterns. From the P-θ curves, information such as the start of combustion, rate of pressure rise, peak firing pressure, expansion pressure, and terminal pressure can be baselined, evaluated, and trended.

Figure 4-13 shows a typical P-θ curve for a two-stroke turbocharged diesel cylinder. At the left edge of the plot, the piston is at bottom dead center (BDC) with both intake and exhaust ports / valves open. At this point, the exhaust blowdown is nearing completion, and a fresh charge of air is being drawn into the cylinder. As the piston moves toward top dead center, first the intake ports and then the exhaust ports / valves are closed. Once both the inlet and exhaust are closed, pressure in the cylinder begins to rise as the piston motion compresses the air trapped in the cylinder.

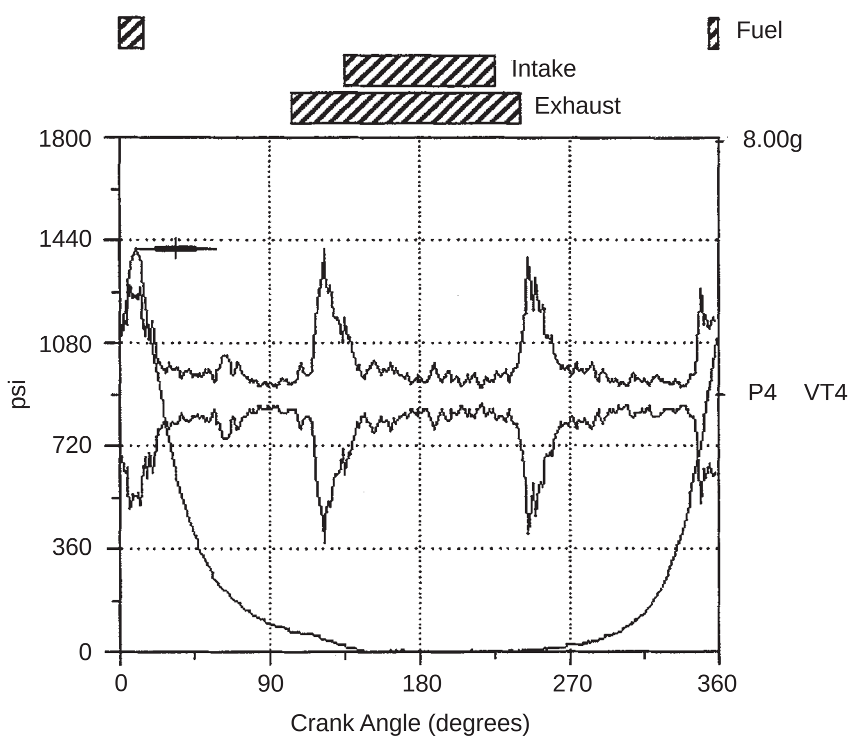

Figure 4-13
Two-Stroke Engine Pressure Versus Crank Angle Curve

At approximately 20°–30° before top dead center (TDC), fuel injection begins. Following the ignition delay, combustion begins followed by a rapid increase in cylinder pressure to the peak firing pressure (P_{max}). Between the start of combustion and the point at which P_{max} occurs, the piston reaches TDC and changes direction to begin the power stroke. The pressure produced by combustion drives the piston toward BDC, and at approximately 100° after TDC (ATDC), the exhaust valves / ports open and exhaust blowdown begins to remove the combustion products from the cylinder.

At approximately 140° ATDC, the inlet ports are opened to draw a fresh charge of air into the cylinder. Typically, the ports and / or valves are arranged such that when both the inlet and exhaust are open at the same time, the inlet air flow assists in the removal of the combustion products by pushing them towards the exhaust ports / valves.

Figure 4-14 shows a P-θ curve for a typical four-stroke cylinder. Starting at the left edge of this plot, the piston is at BDC with the intake valves open. The intake stroke has just completed, but the intake valves remain open for a time as the piston moves toward TDC. When the intake valves close, compression begins and cylinder pressure begins to rise. At approximately 20°–30° before TDC, fuel injection begins. After the ignition delay, combustion begins, followed by a rapid pressure rise to P_{max} at an angle of θ_{Pmax}. The cylinder reaches TDC and reverses direction during this pressure rise. This begins the power or expansion stroke.

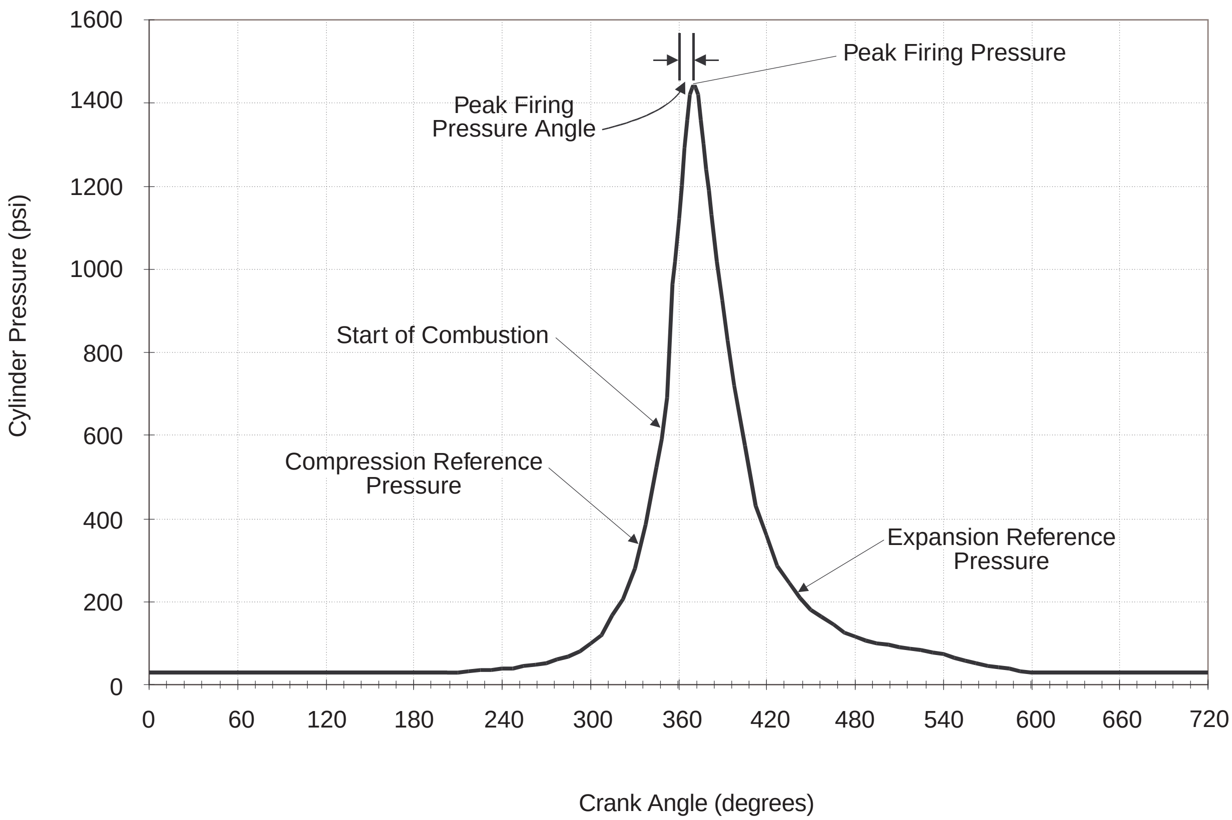

Figure 4-14
Four-Stroke Engine Pressure Versus Crank Angle Curve

Near the end of the power stroke, the exhaust valves open, and the piston moves from BDC to TDC during the exhaust stroke to remove the combustion products from the cylinder. The inlet valves open toward the end of the exhaust stroke so that the incoming inlet air can assist in the removal of the combustion products. During the intake stroke, the piston moves from TDC to BDC, drawing in a fresh charge of air. During the time that the inlet valves are open, cylinder pressure remains constant at the air inlet manifold pressure.

4.2.1.2 Cylinder Pressure Versus Volume Curve (P-V)

The pressure versus volume curve, or P-V curve, is another form of presenting measured cylinder pressure. In this case, the horizontal axis is the volume in the cylinder. It can be plotted as either total volume or swept volume and in absolute volume or volume relative to BDC volume. The piston is at TDC at the left of the plot and BDC at the right (see Figures 4-10 and 4-11).

Pressure-volume curves are derived from P-θ curves by converting crank angle to cylinder volume using engine geometry information. Most engine analyzers make this calculation automatically using the engine geometry data entered into the analyzer.

From Reference 3, for a simple piston, cylinder, and connecting rod system, (see Figure 4-1):

$$V(\theta) = V_c[1 + \frac{1}{2}(r_c - 1)(R + 1 - \cos\theta - \sqrt{R^2 - \sin^2\theta})] \qquad \text{Eq. 4-3}$$

where

V_c	=	cylinder clearance volume
r_c	=	compression ratio
R	=	ratio of connecting rod length to crank radius
θ	=	crank angle (θ = 0˚ at TDC, 180˚ at BDC)

P-V curves for two- and four-stroke engines are significantly different. As shown in Figure 4-10, two-stroke P-V curves consists of a single loop since the cycle is completed in one revolution of the crankshaft. The lower portion of the loop is the compression stroke and the upper portion is the power stroke. The maximum pressure reached on the P-V curve is the same as the maximum pressure shown on the P-θ curve for the same cycle. As shown in Figure 4-11, four-stroke P-V curves contain two loops since it takes two revolutions of the crankshaft to complete the cycle. During each cycle, the piston passes through TDC and BDC twice. The upper loop consists of the compression and power strokes while the lower loop includes the exhaust and intake strokes.

As discussed in Section 4.1.4, the work produced in the cylinder is equal to the area enclosed by the P-V curve.

4.2.1.3 Log Cylinder Pressure Versus Log Volume Curve (Log P-Log V)

Log P versus log V curves are sometimes plotted using cylinder pressure versus crank angle data, respectively. An example of a log P versus log V curve for a four-stroke engine is shown in Figure 4-15. Studies have shown that the compression and expansion processes are closely approximated by a polytropic process where:

$$pV^n = constant$$

where

$n = 1.25^{en}1.35$ [3]

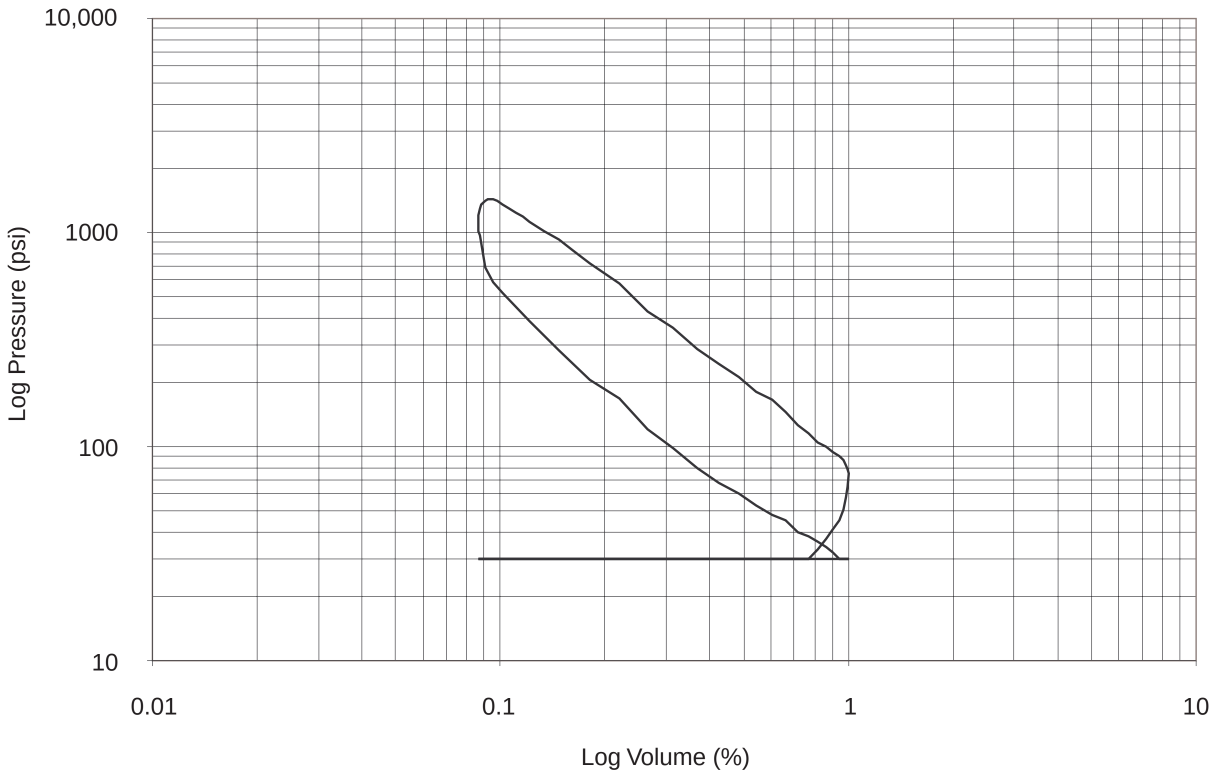

Figure 4-15
Four-Stroke Log Pressure Versus Log Volume Curve

On a log P versus log V curve, the compression and expansion processes plot as straight lines with slope equal to -n (see Figure 4-16). The start of combustion occurs at the point where the compression curve departs from the straight line. Other deviations from the straight line slope can be an indication of leakage during the compression or exhaust processes.

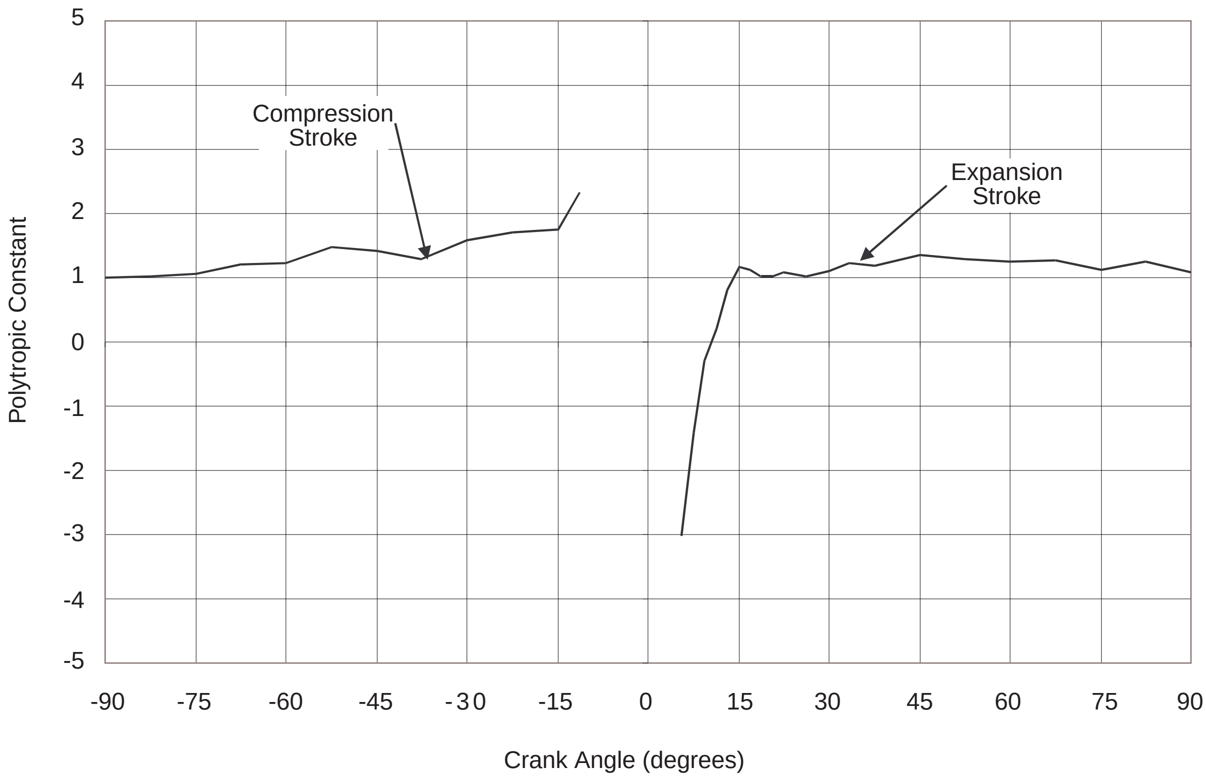

Figure 4-16
Polytropic Constant (n)

4.2.1.4 First Derivative Curve (dP-dθ)

The first derivative curve is a plot of the rate of pressure change versus crankshaft angle (dP-dθ). Figure 4-17 shows a first derivative curve for a typical diesel engine. The main characteristic of interest is the maximum pressure rise rate (MPRR), which occurs during the rapid combustion phase. Ideally, the rate of heat release, and therefore the MPRR, should be as rapid as possible without causing uncontrolled combustion (detonation or knock) that can damage the engine. For any set of circumstances, there will be some maximum pressure rise rate that is optimum. High rates of pressure rise can lead to increased engine stresses and vibration, including crankshaft vibration and bending [5].

The first part of the first derivative curve represents the compression stroke. Cylinder pressure is rising during the compression stroke due to the piston motion compressing the cylinder contents. When fuel is injected into the cylinder, the rate of pressure rise decreases slightly due to the heat transfer to the injected fuel. After a short delay (the ignition delay), the fuel ignites causing the cylinder pressure to rise rapidly. Following the rapid combustion phase, the rate of pressure rise decreases, cylinder pressure reaches a peak, and then starts to decline as piston motion continues to increase the cylinder volume. Note that in some cases, it can be difficult to determine the start of injection and start of combustion accurately from first derivative curves. In these cases, the second derivative curve might be useful (see Section 4.2.1.5).

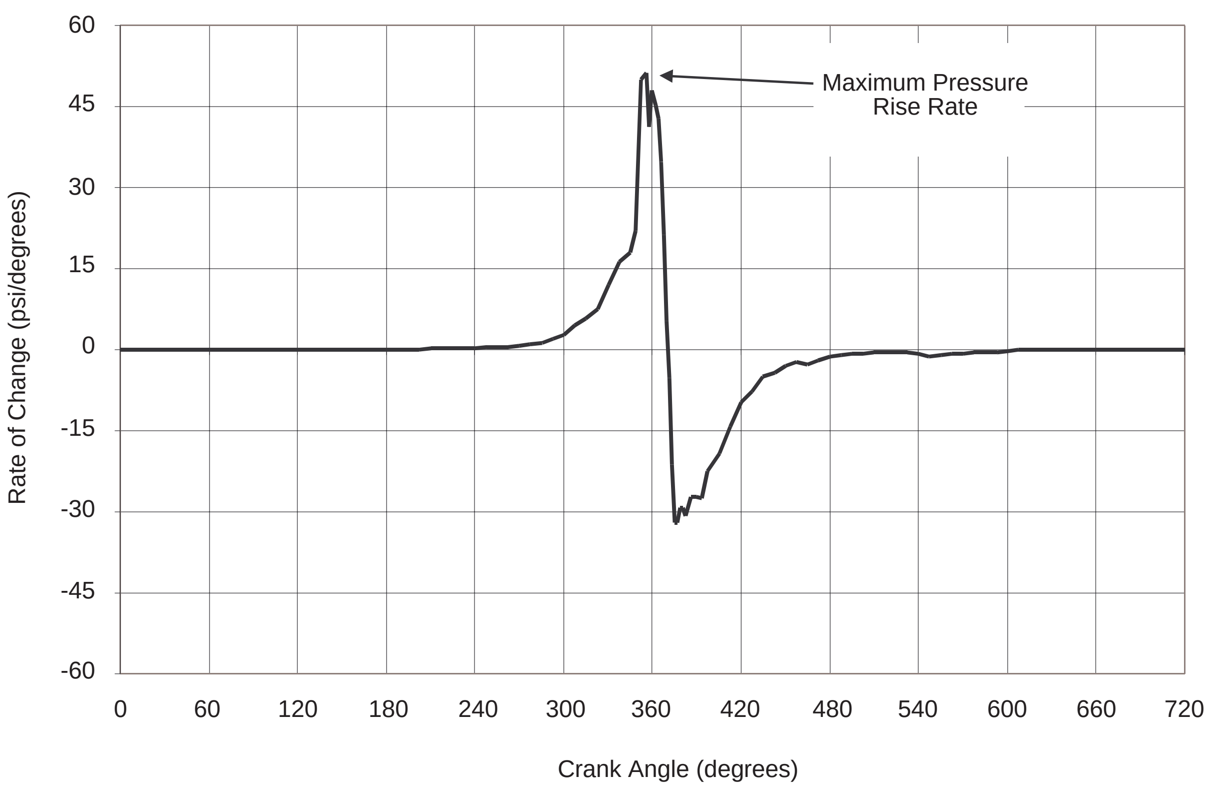

Figure 4-17
First Derivative Curve

4.2.1.5 Second Derivative Curve (dP^2-$d\theta^2$)

Another type of combustion curve that can be plotted is the second derivative curve (dP^2-$d\theta^2$). This curve represents the rate of curvature of the pressure versus crank angle curve. The point at which combustion begins can be determined from this curve (see Figure 4-18) [6].

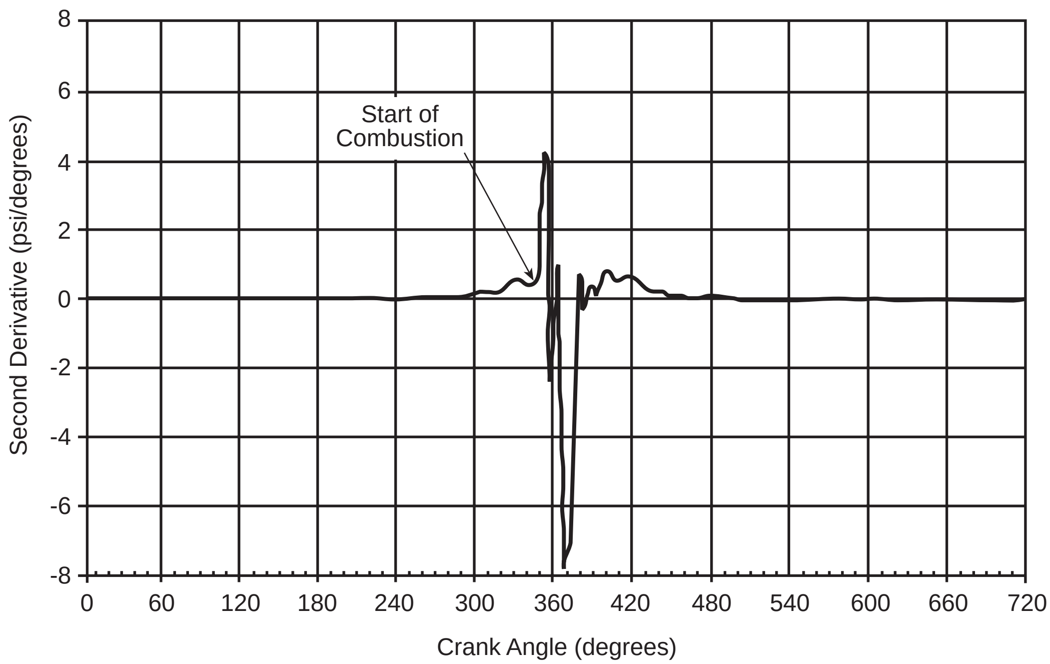

Figure 4-18
Second Derivative Curve (dP^2-$d\theta^2$)

4.2.2 Combustion Characteristics

An experienced analyst can draw conclusions regarding engine condition comparing the combustion curves of the cylinders to one another. Prior to the advent of digital engine analyzers, this is how engine analysis was done. However, digital engine analyzers permit a large number of data to be reduced to a number of key characteristics that can be extracted or calculated from the collected data. By evaluating and comparing these key characteristics and engine performance, the mechanical condition can be assessed to a significant degree.

4.2.2.1 Cylinder Peak Firing Pressure (P_{max})

Cylinder peak firing pressure is the maximum pressure in the cylinder during a mechanical cycle (see Figures 4-13 and 4-14). Peak firing pressure is a function of a large number of variables including:

- Compression ratio
- Engine speed
- Cylinder leakage and heat transfer
- Fuel injection timing, quantity of fuel injected
- Intake air temperature and pressure
- Fuel properties

Because of the potential for cycle-to-cycle variation in peak firing pressure in a cylinder, P_{max} should be evaluated on a statistical basis from a number of cycles (200 cycles is considered a valid sample) in each cylinder and across a number of cylinders by deriving parameters such as:

- The average peak firing pressure in each cylinder over a number of cycles
- The maximum peak firing pressure which occurred in each cylinder over a number of cycles
- The minimum peak firing pressure which occurred in each cylinder over a number of cycles
- The standard deviation of peak firing pressures over a number of cycles
- The difference between the overall engine average peak firing pressure and the average peak firing pressure for each cylinder
- The maximum pressure rise rate that occurred in each cylinder over a numberof cycles

4.2.2.2 Cylinder Peak Firing Pressure Angle (θ_{Pmax})

Peak firing pressure angle, θ_{Pmax}, is the angle at which the peak firing pressure occurs (see Figures 4-13 and 4-14). It is usually reported as the number of degrees after top dead center (ATDC) at which the peak occurs. If the peak firing pressure occurs too early, excessive loads, stresses, and temperatures can result. If the peak firing pressure occurs too late, combustion energy will be lost and mechanical efficiency will be low. This angle must be correct to attain the best performance from a cylinder. The peak firing pressure angle for each cylinder should be compared to the engine average angle. If a cylinder θ_{Pmax} varies significantly from the engine average, an off-design condition is indicated and should be investigated. Peak firing pressure angle is strongly dependent on fuel injection timing.

4.2.2.3 Combustion Start Angle

Combustion start angle is the angle at which pressure starts to rise measurably due to heat release from combustion of the fuel (see Figures 4-13 and 4-14). At this point, pressure in the cylinder increases beyond that which would occur due to compression only. The combustion start angle can be determined from first or second derivative curves. The second derivative curves have been reported to give a more accurate determination [6].

4.2.2.4 Expansion Reference Pressure

Some engine analyzers use the power stroke to evaluate cylinder condition. Expansion reference pressure is the cylinder pressure at a pre-determined crank angle during the power expansion stroke (see Figures 4-13 and 4-14). This allows cylinder pressure comparisons from one cylinder to the others. This measurement must be used with some discretion. Expansion reference pressure is affected by engine load, speed, and other operating variables. When any of these factors change, the expansion reference pressure will also change. Expansion reference pressure should primarily be used to evaluate the spread in expansion reference pressures across the entire engine. Comparison of the expansion reference data from measurement sets having different operating conditions is not recommended.

4.2.2.5 Terminal Pressure

Terminal pressure is the residual pressure remaining in the cylinder at BDC at the end of the power stroke (see Figures 4-13 and 4-14). In the case of a two-stroke scavenged engine, it is equal to the scavenge air pressure. In this case, the terminal pressure provides information to evaluate the scavenging blower and the force of the air purging the cylinder. In a four-stroke engine it is equal to the pressure in the exhaust port. If the cylinder pressure does not drop properly, it indicate exhaust restriction that prevents all the exhaust gas from leaving the cylinder. Exhaust restriction prevents a correct fuel/air mixture for the next cycle. Alternately, the cause could be delayed exhaust valve or port opening.

4.2.2.6 Start of Fuel Injection

Because it has no immediate affect on the cylinder pressure, the start of fuel injection cannot be detected from the P-θ curve. However, it can normally be detected from the ultrasonic signal trace. Injection timing will affect the start of combustion, MPRR, peak firing pressure, and peak firing pressure angle.

4.2.2.7 Compression Reference Pressure

Some engine analyzers use the initial rising portion of the compression stroke to evaluate cylinder compression performance. Compression reference pressure refers to the cylinder pressure at a pre-determined angle during the compression stroke before combustion occurs. The point where compression reference pressure should be measured varies from engine to engine due to differences in engine design. Engine analysis equipment vendors can recommend suitable crank angle positions. Note that the compression reference pressure is not the highest compression pressure produced by the cylinder and should not be compared with the true compression pressure specified by the engine manufacturer.

Almost any angle prior to the start of combustion can be selected as the reference point. Usually, an angle just before the start of injection is used. The extracted pressure at this angle can be used for trending and comparison purposes. It should be noted, however, that differences in engine speed, intake air temperature, and intake air pressure will affect the compression reference pressure. If a problem affects cylinder compression, the change can be detected.

4.2.2.8 Peak Compression Pressure

Cutting off the fuel to one cylinder and recording the cylinder pressure during the mechanical cycle can provide indications of several engine problems. A cylinder that has been cut off from fuel is said to be *motoring*. Similar to the peak firing pressure in a firing cylinder, the *peak compression pressure* is the maximum pressure reached during a mechanical cycle in a motoring cylinder. The magnitude of the peak compression pressure will vary with inlet manifold temperature and pressure, and it will depend on the mechanical condition of the cylinder components. Figure 4-19 shows a pressure versus crank angle curve for a motoring four-stroke cylinder.

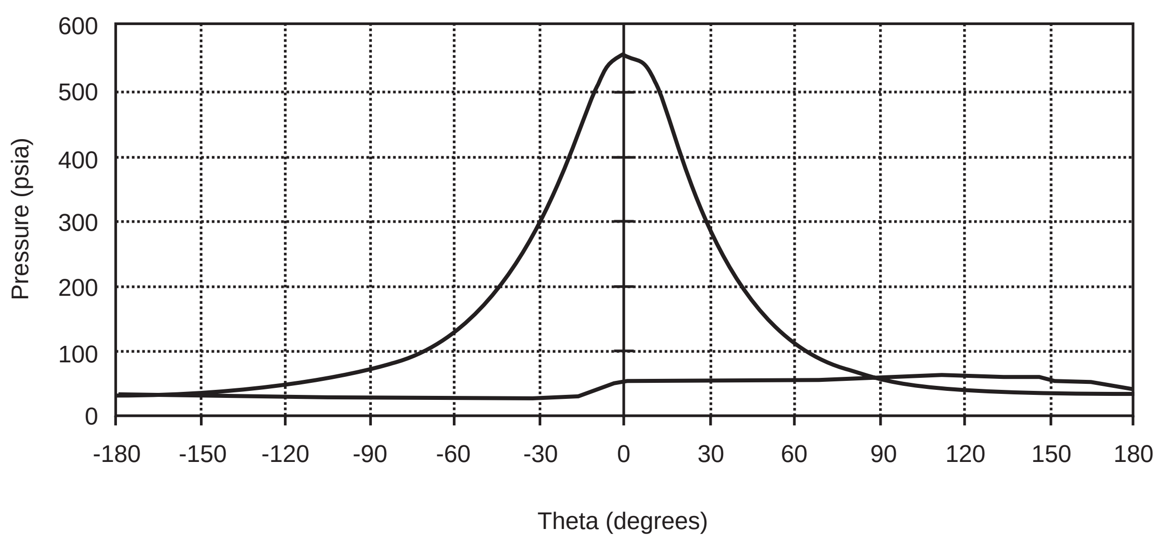

Figure 4-19
Four-Stroke Motoring Cylinder Pressure Versus Crank Angle

Note that this curve is nearly symmetrical around TDC. However, the peak cylinder pressure actually occurs slightly before TDC. The angle between TDC and the peak compression pressure is known as the thermodynamic loss angle (TLA). The pressure in a motoring cylinder is affected by the following:

- Intake air temperature and pressure
- Volume change caused by piston motion
- Heat transfer losses from the compressed air to the engine
- Gas leakage past piston rings and ports / valves

During most of the motoring cycle, the effects of piston motion dominate. However, as the piston nears TDC it slows, and at TDC it is motionless. During the compression stroke just prior to TDC, the pressure and temperature of the cylinder contents are reaching their highest values. As a result, heat transfer and leakage, both of which reduce cylinder pressure, are at maximum. The rise in pressure due to piston motion is less than the heat transfer and leakage as the piston approaches TDC, causing the pressure to begin to decrease before the piston reaches TDC. Peak compression pressure would only occur at TDC if there were no heat transfer or leakage from the cylinder. All engines experience some heat transfer and leakage loss and therefore have peak compression pressures located slightly before TDC.

The peak compression pressure measured during motoring will depend on:

- Heat loss from the cylinder
- Initial cylinder temperature and pressure
- Engine speed
- Valve timing
- Piston ring blowby
- Port / valve leakage

In addition, on turbocharged engines, load has a significant affect on indicated peak compression pressure, mainly due to the higher initial cylinder pressure.

The combustion characteristics and parameters discussed in this section can be determined directly from engine analysis data. Other characteristics, such as indicated horsepower and indicated mean effective pressure (IMEP), can be derived from the combustion curves.

In addition to quantifying combustion characteristics from each cylinder, these characteristics should be evaluated across all cylinders. The engine average and spread for each should be determined. Spread can be used to determine if a cylinder abnormality exists. If there is a consistent problem in a cylinder, it will affect every stroke of the piston. If only data from that cylinder were evaluated, problems could be masked. Comparison to other cylinders can help identify a problem. When a cylinder parameter differs substantially from the engine average, it is a strong indication of a potential problem.

4.2.3 Variability

Engine analysis data will vary from cycle to cycle, from cylinder to cylinder, and from test to test. There are a number of reasons for these variations and a large part of engine analysis is determining when this variation is due to real degradation of the engine and when it is due to other factors.

4.2.3.1 Cycle-to-Cycle Variability

The mixing of fuel and air in the cylinder prior to and during combustion is a random process. As a result, the combustion process and the parameters that are measured during engine analysis vary from cycle to cycle. The variation is random [7]. The cycle-to-cycle variability of peak firing pressure in a well-tuned diesel engine typically has a standard deviation on the order of 1%–4% [8]. Thus, some variation is normal and expected. However, if the variability of one cylinder is much greater than the others, it could be an indication of a problem.

To more accurately describe the condition of any given cylinder, multiple measurements of combustion parameters should be made. The average of a number of measurements is a better estimate of the true average than is any single measurement, since the standard deviation of a series of measurements decreases as the number of measurements increases.

The potential for substantial cycle-to-cycle variability complicates analysis. Simply analyzing one P-θ curve could provide incorrect indications regarding engine condition. Instead, engine analysis should be based on several samples and statistical measures. Most engine analyzers collect data from many combustion events and use these data to determine the average peak firing pressure, the average deviation (a quantitative measure of cyclic dispersion), the maximum peak firing pressure, and the minimum peak firing pressure. Other attributes can be derived similarly.

Whenever there is a fuel injection, air or mixing problem, cycle to cycle comparison is an excellent anomaly detection tool. While comparing many P-θ curves from a cylinder manually is tedious and time consuming, many analyzers are capable of recording several cycles worth of data, selecting key pressure points and generating statistics such as average, maximum, minimum, and standard deviation of the peak firing pressures, and generating a typical P-θ trace to display based on the statistical data. These statistics are a powerful analysis tool.

4.2.3.2 Cylinder-to-Cylinder Variability

In an ideal world, the performance of all of the cylinders of a given engine would be the same and cylinder-to-cylinder variation in combustion parameters would be zero. However, some cylinder-to-cylinder variation will exist due to variations in fuel rack settings, fuel injection timing, mechanical variations in engine parts, and other effects. Any outliers in comparisons of data from cylinder to cylinder suggest the need for further action to tune the cylinder or to take corrective actions to fix degraded parts.

4.2.3.3 Test-to-Test Variability

Changes in atmospheric conditions will affect engine analysis data. Because it is unlikely that atmospheric conditions will be the same from one test to the next, it must be recognized that the data can vary for reasons that have nothing to do with engine condition. However, many characteristics of the engine analysis curves and data should remain the same. Changes in test conditions should not affect the relative magnitudes of engine analysis data from cylinder to cylinder. For example, if one cylinder of engine has always had the highest peak firing pressure, changes in atmospheric conditions will not be responsible for it suddenly having the lowest peak firing pressure in a test. In addition, the general shapes of the various combustion, vibration, and ultrasonic curves should not vary from test to test unless engine degradation has occurred.

4.2.4 Interpretation of Combustion Signatures

There are two concerns when interpreting the combustion curves and parameters described above:

1. Have any engine design limits been exceeded?
2. Are there any indications of problems?

Assuming that engine design limits exist, the first concern is relatively straight forward to evaluate. The measured data can be directly compared to the limits. If any limits are violated, corrective actions can be taken. Addressing the second concern requires judgment and experience. Where fixed limits do not exist, the data from each cylinder can be compared to other cylinders and against previously measured baseline data. Differences found in these comparisons may provide indications of adjustments which need to be made or corrective actions which need to be taken.

If a problem is found or suspected, the data and equipment set up should be examined to ensure that the problem is real and not the result of false data.

4.2.5 Analysis Guidelines

Most engine vendors do not have bases or guidelines for most of the data recorded during engine analysis. Peak firing pressures and exhaust gas temperatures are typically the only guidelines related to engine analysis used by vendors. Table 4-1 summarizes the available analysis guidelines for the engines covered by this guide. These values are taken from engine vendor documents, EDG owners group publications and individual nuclear power plant manuals. As can be seen, guidelines for most of the parameters of interest in engine analysis do not exist. As a result, individual nuclear power plants or EDG owners groups will likely have to develop guidelines using an engineered approach.

In general, two types of analysis guidelines are used:

- Guidelines that are quantitative and are based on the engine manufacturer's design basis or on previously recorded operating data

- Guidelines that are based on data that is qualitative in nature and must be interpreted to make conclusions regarding engine condition

The engineered approach to developing analysis guidelines involves a combination of the following methods or sources of information:

- Operating limits published by the engine vendor
- Baseline data collected on a properly tuned engine in good operating condition
- Engine design basis, if available
- Industry good-practice
- Engineering judgment

Appropriate parameters for numerical analysis guidelines include:

- Maximum engine average peak firing pressure
- Maximum cylinder peak firing pressure
- Variation of cylinder peak firing pressure over a number of cycles
- Maximum cylinder-to-cylinder variation in peak firing pressures
- Engine average corrected peak firing pressure angle
- Maximum range of peak firing pressure angles across all cylinders
- Engine average fuel injection timing
- Maximum range of fuel injection timing across all cylinders
- Maximum range of fuel rack settings across all cylinders at full load
- Allowable range of fuel injection nozzle pop pressures for new fuel injection nozzles
- Allowable range of fuel injection nozzle pop pressures for used fuel injection nozzles
- Maximum cylinder exhaust gas temperature
- Maximum range of exhaust gas temperatures across all cylinders

A detailed discussion on the development of analysis guidelines for these parameters is contained in Section 5 of this guide.

Guidelines for vibration and ultrasonic signatures are extremely difficult to quantify. Rather than being compared to numerical limits, vibration and ultrasonic data are best used by comparing them to previously recorded signatures for the same cylinder or from other cylinders. However, it must be stressed that these comparisons are qualitative in nature. As a result, vibration and ultrasonic analyses should only be used to direct or guide further inspections, analyses, or testing. It is recommended that EDG operability or maintenance decisions not be based on vibration or ultrasonic analyses alone.

Table 4-1
Existing Published Engine Analysis Guidelines

Category	Type	Engine Make/Model								
		ALCO 251	Colt-Pielstick PC2, PC2.3, PC2.5	Fairbanks Morse Opposed Piston 38TD-8-1/8	Cooper-Bessemer KSV [1]	TDI (Enterprise) DSR and DSRV	EMD 645	Nordberg HS-FS-1316	SACM UD45	Worthington SWB12
Fuel injection timing	Engine Average	25° BTDC			29°–31° BTDC	22° BTDC				
	Maximum range of all cylinders	NR			± 1°					
Fuel rack setting	Range (mm)				± 0.5 mm of engine avg.					
Fuel injector pop pressure	Per cylinder [new] (psig)	3,700–4,050	3,450–3,550	1,800–1,900	3,700–3,800					
	Per Cylinder [used] (psig)		3,200–3,550		3,400–3,700					
Cylinder peak firing pressure	Maximum engine average (psig)			1,270 (900 RPM)	1,690		1,400			
	Maximum range of all cylinders (psig)	150	70 alert 100 action		160		80–110			
	Standard deviation over many cycles (psig)									
Corrected firing pressure angle	Engine average				10°–14° ATDC					
	Maximum range of all cylinders				± 2° of engine avg.					
Exhaust gas temperature	Maximum single cylinder (°F)			850	1,100		1,100			
	Maximum range of all cylinders (°F)	150 (16 cylinder) 400 (18 cylinder multi-pipe)	126 alert 150 action		< 200					
Ultrasonic and vibration signatures	Numerical vibration and ultrasonic signature acceptance criteria have not been established or identified for any model of EDG in nuclear standby service. However, comparative guidelines are available. When this data is collected, it is used for trending individual cylinders and for comparing cylinders to one another.									

NR = Not reported

1. From Cooper-Bessemer Model KSV Emergency Diesel Generator Engine Analysis Guidelines, Revision 1, published by the Cooper-Bessemer Owners Group.

4.2.6 Trending

The fact that EDGs are usually only run for a few hours per month does not mean that trending cannot be performed. In fact, it means that trending must be performed in order to identify potential problems. However, the engine analyst must be aware that outside factors can affect engine analysis data. If these factors are not recognized and their effects are not considered or corrected, erroneous conclusions could be made regarding engine condition. The two factors that will vary between engine analysis runs and that can affect results are engine load and environmental conditions.

Individual cylinder loads are directly proportional to total engine load. If engine load varies from one engine analysis run to another, parameters such as peak firing pressure, IMEP, and fuel rack settings will vary. As a result, trending one run against another could indicate that the material condition of the engine has changed when, in fact, only the load carried by the engine has changed. If engine load varies during one engine analysis run, the individual cylinder data will be difficult to compare to one another. The effects of engine load changes must be accounted for before data trending will provide reliable information.

Additionally, changes in ambient air conditions have resulted in changes in engine balance and analysis data at several stations. Changes in temperature, pressures, and humidity can change combustion parameters sufficiently to make an engine balanced under different environmental conditions appear unbalanced. References 1 and 2 provide guidance on development of engine performance maps and variations in engine performance due to ambient environment changes. Individual nuclear power plants can generate engine performance maps and use them for trending purposes.

In addition, effects such as equipment calibration, accelerometer location and mounting method, and changes in signal processing can affect the usefulness of data for trending purposes.

Any of the combustion parameters discussed in Section 4.2.2 can be trended over time. However, the decision of which parameters to trend should be based on the potential benefits of trending each of the parameters, the potential for each of the parameters to drift over time, and the capabilities of the engine analyzer used. Recommended parameters for trending include:

- Cylinder peak firing pressures
- Cylinder peak compression pressures
- Terminal pressures
- Expansion reference pressures
- Compression reference pressures
- Cylinder power (IMEP or IHP) spread
- Exhaust temperature spread

4.3 Vibration and Ultrasonic Analyses

In traditional vibration analysis of rotating equipment, such as pumps, turbines, and generators, vibration amplitude is measured as a function of frequency. This type of analysis is referred to as frequency-domain analysis. Many EDG components can also be analyzed using frequency-based analysis. A large number of references cover vibration analysis in the frequency domain and, as a result, it will not be discussed further in this guide. This section is devoted to vibration analysis in the time domain. In time-domain analysis, the vibration amplitude is plotted as a function of time (or crankshaft angle). Many of the most common diesel engine problems can be detected by time-domain analysis, including leaking valves, valve timing errors, cylinder liner scuffing, and fuel injection problems. It is important to note that frequency-domain vibration analysis has not been demonstrated to be useful for detecting these problems.

Nuclear standby EDGs in the U.S. operate at 60 Hz generator output frequency. The large, medium-speed diesel engines that are prime movers for the generators operate at speeds of 450 RPM to 1,200 RPM with combustion events occurring at 3.75 Hz to 10 Hz in four-stroke engines and 7.5 Hz to 20 Hz in two-stroke engines. Vibrations caused by combustion-related events are typically at much higher frequencies than engine operating speed and are broad band in frequency. Therefore, combustion vibration sources are best analyzed by filtering out the low frequencies corresponding to engine speed and performing the analysis in the time domain. Using this technique, the timing of different combustion events in the mechanical cycle can be compared in each cylinder or from cylinder to cylinder. The vibration signal is measured and plotted as a function of time (or crank angle). This type of plot provides indications of the timing of combustion events (crank angle) and the relative strength of events (vibration amplitude).

Ultrasonics refer to relatively high frequency vibrational energy that is present in operating machinery. There is no precise definition of “ultrasonics.” In this guide, frequencies above 15 kHz are considered to be in the ultrasonic range. The principles of ultrasonic analysis are the same as those of vibration analysis.

4.3.1 Sources of Engine Vibration and Ultrasonic Noise

There are many sources of vibration and ultrasonic noise in a reciprocating engine. The reciprocating motion inherently introduces vibrational forces due to the inertia of reciprocating parts. Rotating components also vibrate. In addition, mechanical impacts (for example, valves landing on seats) and high velocity gas flows (for example, exhaust blowdown) produce high frequency signals. Signals from these sources can be measured and used to diagnose engine problems by comparing the measured signatures to expected or previously measured baseline signatures of a healthy or normal engine.

The ultrasonic or high frequency vibration energy of interest originates from several sources:

- Mechanical impacts, such as valve closure or piston slap
- High velocity gas flow, such as exhaust blowdown or piston ring blowby

- Mechanical roughness or friction
- Direct combustion-induced energy

All these events are expected in a new engine.

4.3.2 Characteristics of Engine Vibration and Ultrasonic Signals

Three general categories of vibration and ultrasonic noise are of interest in time-domain analysis: gas flow, mechanical impacts, and rubs. Conditions that cause vibration or ultrasonic signals produce one or more of these types of signals.

Gas flow vibration or ultrasonics usually occurs at high frequencies. As gas flows, it creates a high frequency whistle effect. It is normal for this to occur in all gas flows. The presence of this gas flow during times when no gas should be flowing is an indication of a leak. A high gas vibration or ultrasonic amplitude during times gas should be flowing freely is an indication of a possible obstruction.

Reciprocating mechanical components such as pistons, connecting rods, valves, fuel linkages, and piston bushings produce mechanical impacts. A knock occurs when mechanical clearance increases in a bushing pin or bearing. Reciprocating actions create forces that switch from one direction to the other, resulting in impact or knock.

Rubs occur when there is inconsistent (non-linear) contact throughout the movement of a reciprocating or rotating component. Piston liner rub is the most common form of this type of vibration in reciprocating machinery. When a cylinder liner has a rough or bad spot, the vibration increases each time the piston passes that point.

Engine events that produce vibration or ultrasonic signals typically occur once per mechanical cycle and are characterized by certain shapes or patterns when plotted versus time or crankshaft angle. Envelopes of the characteristic shapes are shown in Figure 4-20.

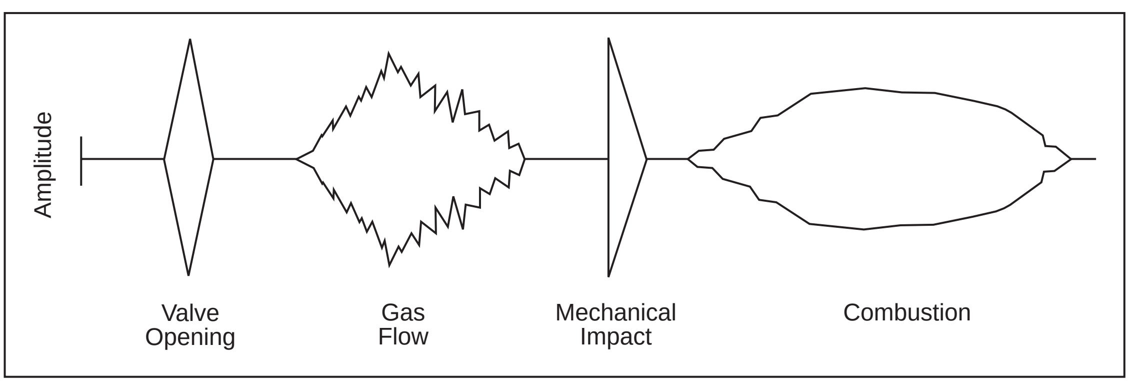

Figure 4-20
Envelopes of Typical Vibration and Ultrasonic Patterns

4.3.3 Measurement Methods

Engine and component vibration and ultrasonic signatures are typically measured using accelerometers attached to the component of interest. For measurement of combustion-related vibration and ultrasonic signals, the accelerometer is normally mounted on the cylinder head. On the FMOP engine, which does not have cylinder heads, the accelerometer is typically attached to the cylinder liner. Piezoelectric transducers are typically used because they are capable of measuring the high frequencies associated with combustion events. The selection and mounting of accelerometers are two of the most important decisions in vibration and ultrasonic analyses. See Section 5 for more detail.

4.3.4 Interpretation of Vibration and Ultrasonic Signatures

Vibratory motion can be described in terms of displacement, velocity, or acceleration. Displacement is the distance traveled by the point of interest in a specific direction. If displacement in the vertical direction is measured, the vibrating point will move a peak distance down and a peak distance up. This total distance traveled from top to bottom is referred to as peak-to-peak displacement.

Velocity is another characterization of vibratory motion. As the vibrating point reaches the upper peak of the displacement, it stops, reverses direction, and begins the downward motion. As it accelerates downward, it picks up speed, until it reaches the point of zero displacement. At this point, it begins to gradually slow down until it stops at the bottom peak of the displacement. Now this process repeats in the upward direction. The peak speed or velocity happens at the mid-point of displacement.

Acceleration is another way of describing vibratory motion. Maximum or peak acceleration occurs at the points of peak displacement. Vibratory acceleration is commonly expressed as a ratio to gravitational acceleration. For example, 2.0 g acceleration means twice gravitational acceleration.

When a vibration measurement point is set up for an engine, the decision must be made to measure displacement, velocity, or acceleration. Each has advantages. It is desirable to select the amplitude unit that will provide the best tracking of component condition. Generally, displacement provides the best amplitudes at low frequencies from 1 Hz–10 Hz. Velocity provides good condition information for general machinery producing frequencies from 10 Hz–100 Hz. Acceleration amplitudes are most sensitive to changes in high frequencies from 100 Hz and higher. Most engine analyzers measure vibration and ultrasonic signals using acceleration signals only; therefore, changing to velocity or displacement units is not a supported option.

Vibration transducers are sensors that convert motion into proportional electrical signals. Selection of these sensors requires proper frequency range, temperature range, and mounting considerations. The most common transducer used in the collection of vibration and ultrasonic signals from machinery is the accelerometer.

Proper attachment of the transducer is very important. The function of transducer is to the turn the mechanical motion into an electrical signal. To do this, the transducer must be attached so that it moves with the vibrating surface.

Selecting the point to attach the transducer for the vibration measurement is critical. It is desirable to select a location that will provide the most reliable and consistent engine condition information. Measurements should be made at the same point each time to take advantage of amplitude trending.

Nominal vibration and ultrasonic patterns from a four-stroke engine are shown in Figure 4-21. The pattern can best be interpreted by examining angular ranges.

- From TDC to exhaust valve opening, direct combustion energy is usually visible. Problems such as piston slap, piston ring blowby, and cylinder liner scoring can be detected in this region.
- From exhaust valve opening to TDC, only the exhaust blowdown can usually be seen. Opening of the exhaust valve is normally not visible. A mechanical impact ear TDC could be due to excessive wrist pin or connecting rod bearing clearance.
- Between TDC and BDC, a mechanical impact is usually visible when the exhaust valves close. Problems with the exhaust valve train could lead to early or late closing or high or low amplitude of the closing signal.
- Between BDC and TDC, a mechanical impact is usually visible when the intake valves close. Again, problems with the valve train could lead to early or late closing or high or low signal amplitude. Usually, the start of fuel injection is visible shortly before TDC. A gas flow pattern in this region is an indication of piston ring blowby during the compression stroke.

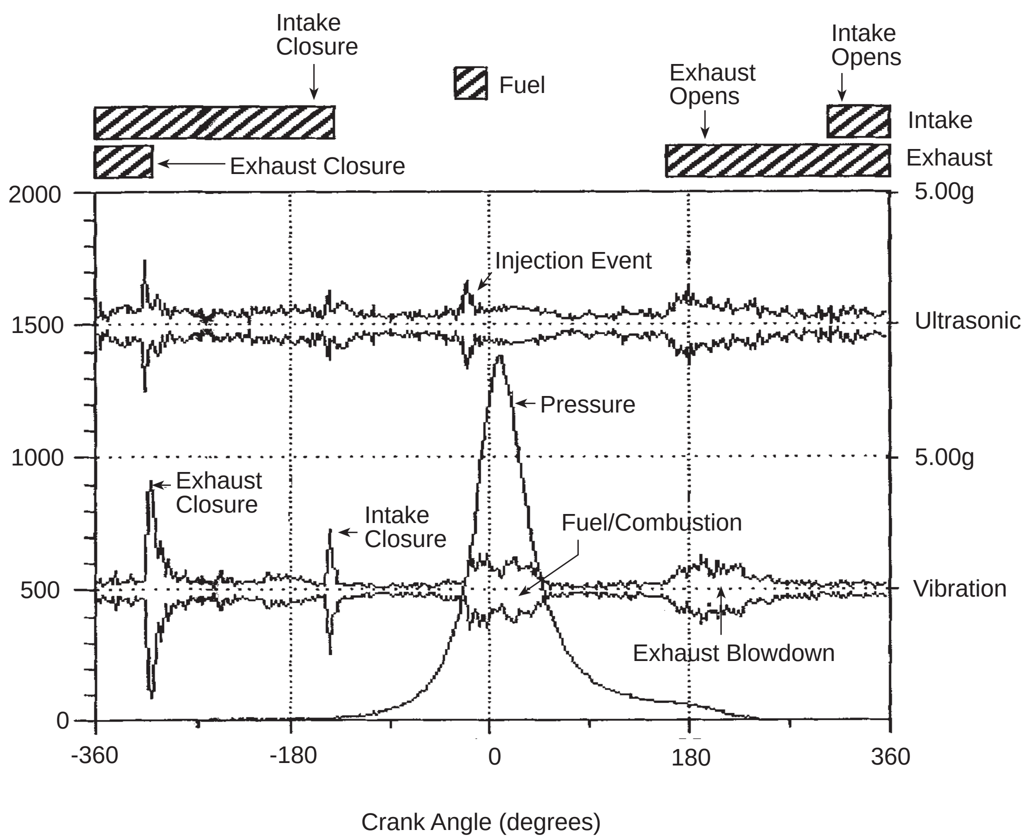

Figure 4-21
Nominal Four-Stroke Vibration and Ultrasonic Signatures

Nominal vibration and ultrasonic patterns from a two-stroke engine is shown in Figure 4-22.

- From TDC to exhaust valve/port opening, direct combustion energy should be visible. Problems such as piston ring slap, piston ring blowby, and cylinder liner scoring may be seen in this region.
- Normally, between exhaust valve/port opening and BDC only exhaust blowdown can be seen. If there are intake or exhaust ports, ring clip can be seen as the compression rings enter the port areas.
- Between BDC and exhaust valve/port closure, exhaust valve closure will be detected as a mechanical impact if the engine is equipped with exhaust valves. Again, problems with the valve train can lead to early or late closure, or high or low signal amplitude. Exhaust port closure will not be visible.
- Between exhaust valve/port closure and TDC, the start of fuel injection can usually be seen. A gas flow pattern in this region is an indication of piston ring blowby during compression.

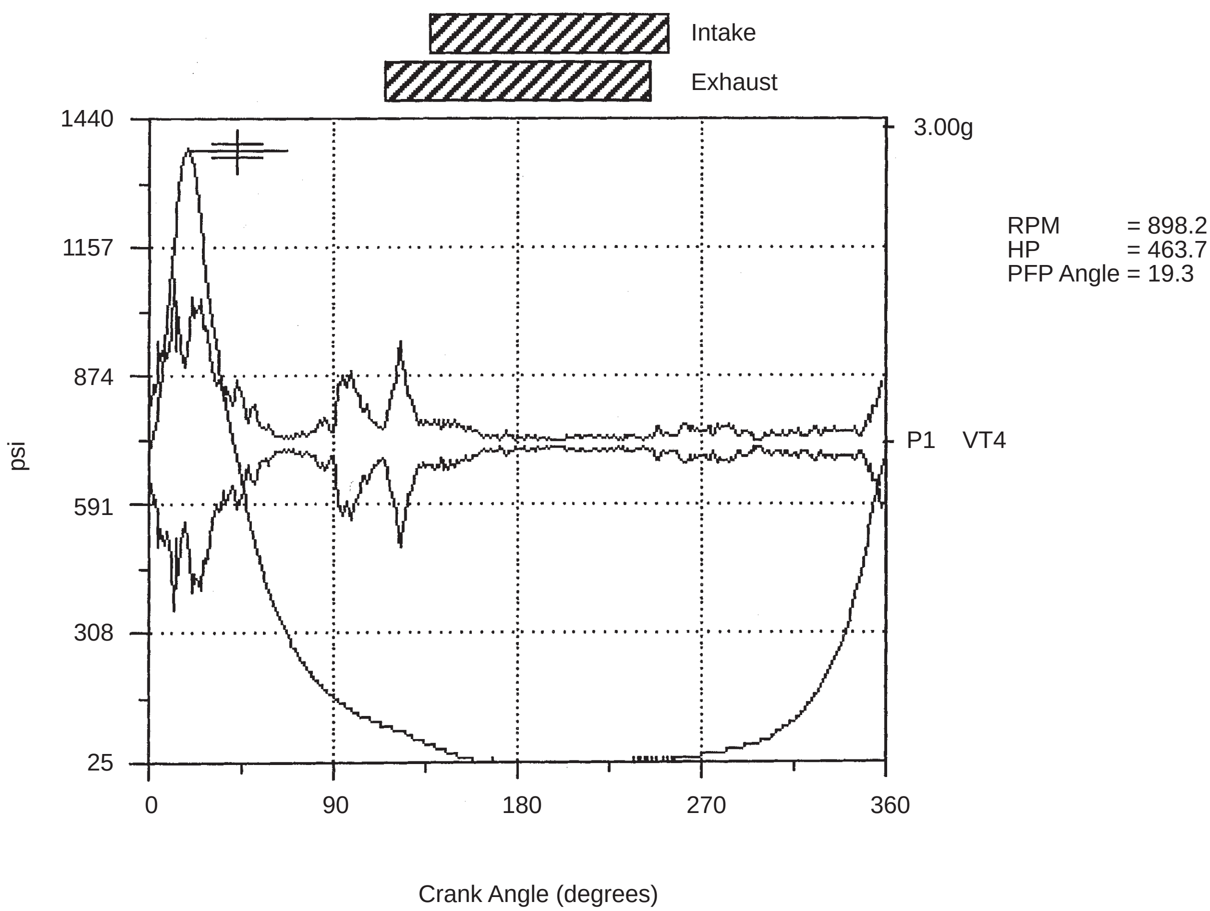

Figure 4-22
Nominal Two-Stroke Vibration and Ultrasonic Signatures

4.3.5 Analysis Guidelines

As stated earlier, all engines vibrate regardless of how well they are designed, assembled, and maintained. Threshold vibration levels above which an engine can be considered to be in need of corrective actions (bad condition) have been recommended by various sources. For example, Reference 9 provides recommended vibration limits and suggested actions for diesel generators. However, these threshold levels are usually not based on scientific analysis and prediction.

Interpretation involves of vibration and ultrasonic signatures involves pattern recognition (verifying the presence of expected events at the correct time and with acceptable amplitude) and checking for the absence of expected events. The issue of amplitude is problematic; it depends not only on the energy in the event sensed, but also on the transmission path, sensor characteristics, and mounting method. Obtaining consistent and meaningful amplitudes in ultrasonic analysis is more difficult than for conventional low frequency vibration analysis. Fortunately, engine faults usually produce large amplitude changes and minor changes due to technique are less significant.

Ultrasonic signals will provide indications of such events as valve closings, fuel injection, exhaust blow down, piston rings passing over inlet and exhaust ports, and direct combustion noise. In addition, similar events in adjacent cylinders can be picked up by the accelerometer. This interference from other cylinders is known as *cross talk*. Any other significant components in the pattern indicate potential defects.

The most commonly used method for monitoring engine vibration is to measure or determine baseline vibration signatures for an engine known to be in good condition and to monitor changes in these signatures over time. Changes in these signatures are an indication of changes in the material condition of the engine. Example of changes include:

- Increase or decrease in vibration amplitude
- Appearance of new vibrations or disappearance of previously seen vibrations
- Changes in vibration frequency (seen on amplitude versus frequency plots)
- Changes in timing of events (seen on amplitude versus crank angle plots)

Numerical analysis alert guidelines are usually not used for vibration analysis due to the difficulty in determining meaningful numerical limits. Instead, trending and comparison are used to monitor changes in vibration patterns over time.

It is recommended that, due to the subjective nature of the data, maintenance decisions not be made based on vibration or ultrasonic data alone. Instead, vibration and ultrasonic data should be used in combination with other analysis and monitoring data.

Ultrasonic signatures are usually band filtered. Higher frequency bands have higher sensitivity to gas flows, such as piston ring blowby. Lower frequency bands have more sensitivity to mechanical impacts, such as valve closing and piston slap. This function is performed by most engine analyzers and is not a user changeable feature.

4.3.6 Trending

Vibration patterns should be compared to previously measured vibration signatures for the same cylinder, other cylinders of the same engine, and from other engines of the same type. Significant variations in the amplitude or timing of the signatures can provide indications of problems. For example, an increase in the amplitude of the vibration for a valve closure event indicates that the valve is impacting the valve seat harder than it was previously. This can indicate that the valve seat is damaged or that the valve lifter has collapsed. A decrease in the amplitude of this event could indicate that the valve seat is damaged or the valve clearance needs to be adjusted. The timing of the valve closure event could also indicate a problem with the valve train. For example, if the valve closes earlier than expected, the valve clearance could be out of adjustment or the valve lifter could be collapsed.

In addition, the appearance of new vibration patterns or the disappearance of previously seen patterns can also provide indications of problems. Piston slap will typically show up a vibration spike in the area around TDC and could be an indication of excessive bearing clearances, improper piston ring loading, detonation, or uneven combustion.

5

EQUIPMENT SELECTION, SETUP, AND USE

Like most other analysis methods, the usefulness and accuracy of the results are largely dependent upon the measurement equipment and its use. This section provides guidance regarding equipment selection and use.

5.1 Key Analyzer Characteristics

There are currently many companies manufacturing engine analysis equipment. Features and capabilities of these systems vary greatly. We recommend that the end user review the available equipment and request an on-site demonstration of one or more systems before purchasing. In several cases, utilities have purchased engine analyzers but have stopped using them due to dissatisfaction with features or ease-of- use. It should be noted that should you choose not to purchase an analyzer, some consideration should still be given to the analysis equipment used by the contractor.

Engine monitoring systems, which are typically permanently installed and that do not record cylinder pressure and vibration signatures, are not included in this discussion as they are not true engine analyzers. This review covers hand held, portable engine analysis equipment specifically designed to record and analyze engine signatures as well as PC-based engine analysis systems. Table 5-1 summarize the features and capabilities of these devices. The equipment vendors provided the data in Table 5-1. The table provides a summary of the recorded parameters and engine analysis methods available for each make and model. Table 5-1 is by no means a complete list of all available equipment. This technology is changing rapidly; readers are advised to check for updated information.

Table 5-1 Engine Analysis Equipment Summary

Vendor	Model	Description	Measured Inputs	Output Values	Built-in Analysis Capabilities	Unique Features	Comments	Vendor Contact Info.
Liberty Technologies/ Beta	Recip-trap (RT-9240)	Hand-held engine analyzer	Cylinder pressure, crank angle, vibration ultrasonic, temperatures	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration both frequency time domain ultrasonic	Historical data trending and analysis with PC software	Data can be exported to PC for analysis using Beta Windows 95/NT analysis software. Also capable of monitoring compressors and rotating equipment.	Battery powered. Two hot-swappable batteries, 5 hrs./pair. 486-50 processor. 10 MB Data storage.	Liberty Technologies/Beta 1-800-950-BETA
	Diesel Trap 1 channel + channel	Hand-held engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic	Historical data trending and analysis with PC software	Data can be exported to PC for analysis using Beta DOS-based analysis software	Battery powered	
	Beta trap	Hand-held balancing analyzer	Cylinder pressure	P_{max}, P_{max} statistics	None	Data can be downloaded to PC	Peak firing pressure balancing only. Battery powered.	
NKF/CARMA	CARMA 8D	Portable PC-based engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic	Historical data trending and analysis	Includes analysis software	--	NKF Engineering 703-358-8600
PMC/Beta	Beta 250	Portable oscilloscope-based engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic, other instrument channels	None	Digital data storage. Data can be exported to PC.	--	PMC/Beta 713-821-4175
	Beta 360	Portable oscilloscope-based engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P-θ, P-V, vibration, ultrasonic, other instrument channels. P_{max} and θ_{Pmax} can be read from screen.	None	None	No data storage capability	
CSI/Windrock	Model 6100	Engine monitoring system	Up to 250 inputs, including cylinder pressure, crank angle, engine system pressures and temperatures, vibration ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic. Others depend on inputs and can include air/fuel ratio, lube oil condition, emissions.	Historical data trending and analysis, comparisons versus baselines	Online monitoring via PC workstation	Permanently mounted engine monitoring system	Windrock Inc. 423/539-5944
	Model 6300	Hand-held engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic	Historical data trending and analysis	Digital data storage. Data can be exported to PC.	Battery powered	

Table 5-1
Engine Analysis Equipment Summary (continued)

Vendor	Model	Description	Measured Inputs	Output Values	Built-in Analysis Capabilities	Unique Features	Comments	Vendor Contact Info.
Liberty Technologies/ Beta	Recip-trap (RT-9240)	Hand-held engine analyzer	Cylinder pressure, crank angle, vibration ultrasonic, temperatures	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration both frequency time domain ultrasonic	Historical data trending and analysis with PC software	Data can be exported to PC for analysis using Beta Windows 95/NT analysis software. Also capable of monitoring compressors and rotating equipment.	Battery powered. Two hot-swappable batteries, 5 hrs./pair. 486-50 processor. 10 MB Data storage.	Liberty Technologies/Beta 1-800-950-BETA
	Diesel Trap 1 channel + channel	Hand-held engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic	Historical data trending and analysis with PC software	Data can be exported to PC for analysis using Beta DOS-based analysis software	Battery powered	
	Beta trap	Hand-held balancing analyzer	Cylinder pressure	P_{max}, P_{max} statistics	None	Data can be downloaded to PC	Peak firing pressure balancing only. Battery powered.	
NKF/CARMA	CARMA 8D	Portable PC-based engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic	Historical data trending and analysis	Includes analysis software	--	NKF Engineering 703-358-8600
PMC/Beta	Beta 250	Portable oscilloscope-based engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic, other instrument channels	None	Digital data storage. Data can be exported to PC.	--	PMC/Beta 713-821-4175
	Beta 360	Portable oscilloscope-based engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P-θ, P-V, vibration, ultrasonic, other instrument channels. P_{max} and θ_{Pmax} can be read from screen.	None	None	No data storage capability	
CSI/Windrock	Model 6100	Engine monitoring system	Up to 250 inputs, including cylinder pressure, crank angle, engine system pressures and temperatures, vibration ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic. Others depend on inputs and can include air/fuel ratio, lube oil condition, emissions.	Historical data trending and analysis, comparisons versus baselines	Online monitoring via PC workstation	Permanently mounted engine monitoring system	Windrock Inc. 423/539-5944
	Model 6300	Hand-held engine analyzer	Cylinder pressure, crank angle, vibration, ultrasonic	P_{max}, θ_{Pmax}, dP, IMEP, IHP, P-θ, P-V, vibration, ultrasonic	Historical data trending and analysis	Digital data storage. Data can be exported to PC.	Battery powered	

5.2 Cylinder Pressure Transducers

5.2.1 Pressure Transducer Characteristics

Most engine analyzers are provided with pressure transducers tailored for the specific analyzer. However, some analyzers do provide for the use of a user-selected pressure transducer and amplifier. A comprehensive discussion of all of the issues related to cylinder pressure transducer selection is beyond the scope of this guide. Transducers are available in many different types, each with unique advantages and disadvantages. References 10–22 provide guidance on pressure transducer characteristics and selection. If use of user-selected pressure transducers and amplifiers is desired, the following characteristics should be considered:

- Linearity—A linear transducer delivers an output signal strictly proportional to input pressure. Linearity is the deviation from strict proportionality.
- Sensitivity—The sensitivity of a transducer determines how large a change occurs in the output signal for a given change in the input pressure.
- Damping—Damping describes how quickly the output signal of a transducer settles.
- Dynamic response—The transducer must be capable of measuring rapid pressure changes associated with combustion.
- Hysteresis—Transducer hysteresis is the maximum difference in output signal for the same measured input pressure when that pressure is approached from above and below.
- Mounting characteristics—The transducer must be able to be mounted within the space available and using methods that will not result in additional errors.
- Cooling requirements—Many transducers require external cooling to prevent damage or drift.

Each of these characteristics can be discussed with the transducer or engine analysis equipment manufacturers.

5.2.2 Transducer Preparation

Cylinder pressure transducers are exposed to rapid (for example, 300 times per minute in a 600 RPM 4-stroke engine) thermal transients. Many transducers are adversely effected by these temperature transients. References 14, 18, 22, and 23 discuss the actions necessary to minimize the effects of the thermal cycles. Specifically, some vendors recommend that the face of the transducer be coated with a thin layer of RTV for flush mounted transducers. Most EDGs in nuclear service have relatively long indicator passages, which reduce the magnitude of the thermal cycles. The manufacturer of the analyzer or transducer should be consulted before the cylinder pressure transducers are prepared. Also, it should be ensured that any vendor-recommended cooling is always available during data collection.

5.2.3 Transducer Calibration

Calibration of the transducer and associated amplifiers contributes directly to the accuracy of the engine balancing and analysis data. Calibration instructions should be provided with the engine analyzer. However, experience shows there are three areas regarding calibration that merit special note.

1. Calibration should include all components of the engine analyzer including the amplifier, cables, read-out devices, and mechanical mounting hardware. This "end-to-end" calibration is the only way to account for all system variances. The system should be recalibrated following change out of any component.
2. Some pressure transducers react to dynamic pressure only; static pressures decay rapidly. This is particularly true of piezoelectric or "crystal" transducers. Special dynamic pressure calibration devices are available to calibrate these transducers accurately. However, static calibration devices, such as dead weight testers, can be used with success as long as the operator is aware of the dynamic effects and the transducer vendor supports static calibrations.
3. Traditional cylinder pressure transducers are somewhat fragile. They should be checked periodically during use and any time the transducer is dropped, overheated, or the transducer mounting is changed.

Transducer calibration requirements vary by vendor and historical experience with specific devices. However, the calibration should always be spot checked following balancing to ensure that the transducer has not drifted. Cylinder pressure transducers typically cannot accurately measure low or absolute pressures. Therefore, many engine analyzers support direct or indirect input of a reference pressure as the baseline for the pressure transducer every revolution. Usually, the cylinder pressure at BDC (before the compression stroke for four-stroke engines) is assumed to be equal to the pressure of the inlet manifold [24]. This correction should always be used when it is supported by the engine analyzer. Failure to input a reference pressure will allow the pressure reading to drift from cycle to cycle.

5.2.4 Transducer Mounting

Most pressure transducers are mounted in an adapter that facilitates the connection to the cylinder test connection. Forces exerted on the transducer housing can pre-load the transducer and cause an offset in the output. For this reason, it is recommended that the transducer be mounted in the adapter prior to calibration. Further, the transducer calibration should be checked each time it is mounted in the adapter.

5.3 Vibration and Ultrasonic Accelerometers

5.3.1 Accelerometer Characteristics

Accelerometers transform mechanical vibration and ultrasonic signals into electrical signals proportional to the acceleration. Most accelerometers use piezoelectric materials as the sensing element. Accelerometers should be used to measure signals well below their natural frequencies. Usually, the operating frequency range is limited to approximately ten percent of the natural frequency [25]. The frequency response of an accelerometer is the range of frequencies that can be measured without significant error in the output signal due to resonance of the accelerometer. The high end of the frequency response for an accelerometer is typically defined as the frequency at which the increase in vibration amplitude due to resonance of the accelerometer is 5%. Note that the usable frequency range of an accelerometer is not limited by its frequency response. However, the error in vibration amplitude must be considered when measuring signals with the frequency components greater than the frequency response of the accelerometer [26]. The frequency response of an accelerometer is usually provided by the accelerometer manufacturer.

5.3.2 Accelerometer Calibration

Accelerometers are calibrated to determine their sensitivity. Accelerometers should be checked prior to every use from “front-to-back,” including cables and the amplifier. Hand-held calibration devices that generate a calibrated “vibration” (frequency and magnitude) are available from a number of vendors.

5.3.3 Accelerometer Mounting

The method of mounting an accelerometer effects its frequency response. The objective is to use a mounting technique that provides a flat response within the frequency range of interest. Several different methods of mounting accelerometers, along with their effect on frequency response, are described below in order of preference [27]:

- Stud mounting—The accelerometer is mounted to the measurement surface using a threaded stud. Of all the mounting methods, stud mounting provides the best frequency response. However, convenient stud mounting locations are not present on many diesel engines.
- Adhesive mounting—The accelerometer is directly cemented to the measurement surface or a pad mounting is cemented to the surface. A pad mounting includes a stud for attaching the accelerometer. The frequency response of an accelerometer mounted with adhesive is almost equivalent to a stud mounting. This is the recommended method for vibration analysis of engines where stud mounting locations are not available.
- Magnetic mounting—A magnet is attached to the base of the accelerometer. The magnet is then attached to the measurement surface. The frequency response of an accelerometer with a magnetic mount is acceptable for mounting accelerometers in temporary locations.
- Hand-held probe—These probes are typically cylindrical and end in a point which is held against the measurement surface. Although hand-held probes are very easy to use, they are not recommended because their frequency response is usually much worse than other mounting methods.

Accelerometers with a magnetic mount have been found to produce acceptable vibration measurements if adhesive-mounted studs cannot be used. The vibration data shown in Figures 5-1 and 5-2 were measured using a Kistler Model 8622 accelerometer with a magnetic mount. The manufacturer's listed frequency response for the accelerometer is 1.00 Hz to 8.00 kHz. This frequency response is reported for a stud mounted accelerometer. The manufacturer indicated that the frequency response of the accelerometer would decrease when using a magnetic mount. However, if the accelerometer is mounted on a relatively flat, smooth surface such as a cylinder head stud, and a sufficiently powerful mounting magnet is used, the frequency response of the accelerometer might not degrade significantly due to the magnetic mount.

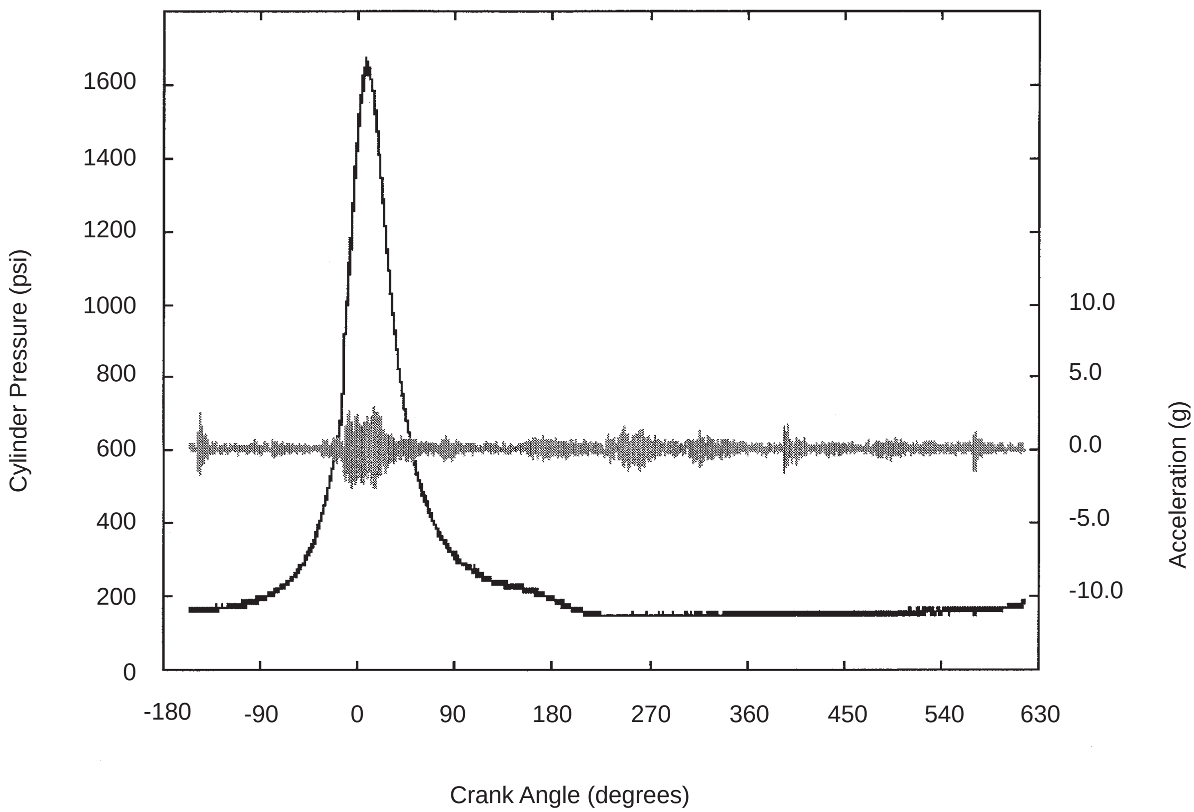

Figure 5-1
Typical Engine Pressure and Vibration Data Measured on a Cylinder Head Stud

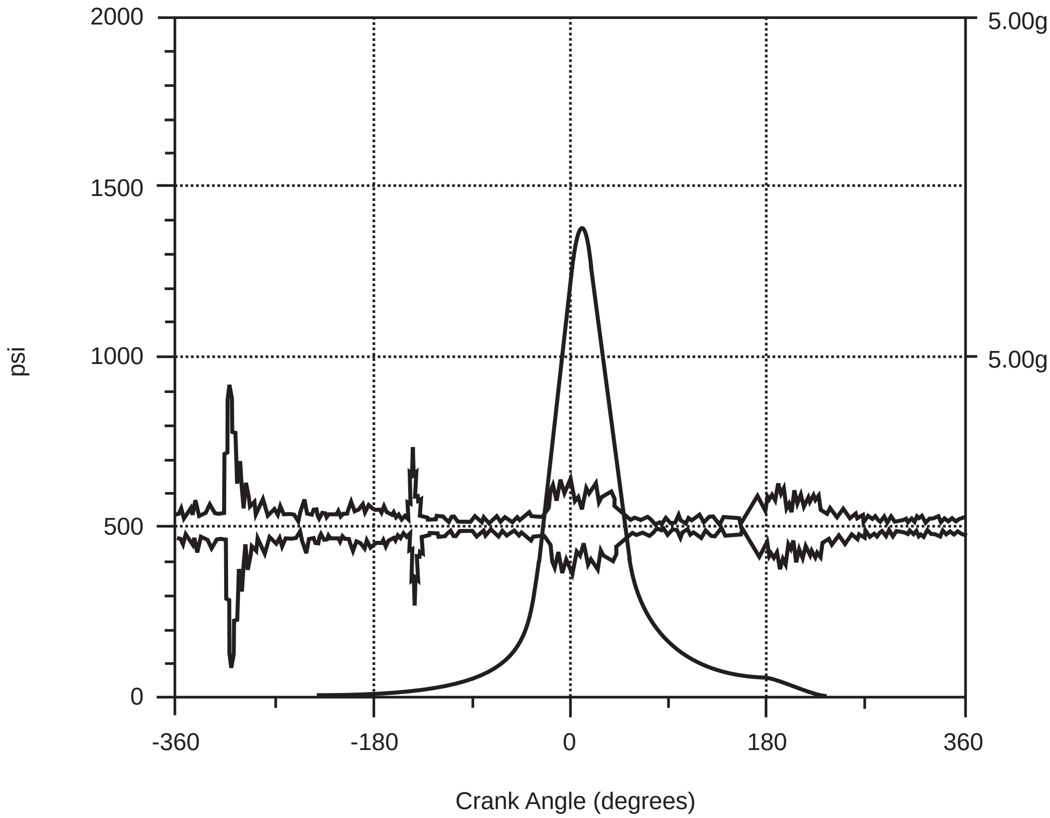

Figure 5-2
Typical Engine Pressure and Vibration Data Measured on the Cylinder Liner Flange

Vibration measurements are typically taken on the engine structure (that is, cylinder liner flange or cylinder head stud). The location of the accelerometer has a significant effect on the measured vibration signal. In general, mounting the accelerometer closer to the source of vibration will produce a stronger vibration signal. Placing the accelerometer at one location can give a strong signal for exhaust valve closure while another location can give a strong signal for piston impacts. It is desirable, however, to use a location where all the various vibration signals can be measured conveniently.

When first taking vibration readings on an engine, various accelerometer locations should be used to identify the best location for measuring a clean signal. As an example, Figure 5-2 is a pressure and vibration trace for an engine measured from the cylinder liner flange. If the vibration trace of Figure 5-2 is compared to the vibration trace of Figure 5-1, it can be seen that the vibration signature is much stronger when measured from the cylinder head stud and, therefore, the cylinder head stud would be the preferred location.

Successive measurements on a cylinder should always be taken at the same location (with the same accelerometer and mounting) such that comparisons between the data are consistent. In addition, since it is useful to compare data from different cylinders, measurements should be taken at the same location on each cylinder. For most diesel engines, it has been found that the cylinder head studs provide a good location for identifying mechanical impacts in the engine. Note that the same cylinder head stud must be used for every cylinder, since different head studs produce different vibration signatures. For the Fairbanks Morse Opposed Piston engine, it is recommended that the accelerometers be mounted on the cylinder liner next to the fuel injector.

5.4 Filtering and Signal Conditioning

Engine analyzers with built-in vibration measurement capabilities often have internal filtering and signal processing. Figure 5-1 shows an unfiltered vibration signal from a Cooper-Bessemer Model KSV engine. It is important when using an engine analyzer with vibration measurement capabilities, that the user understand the filtering and signal processing that is performed by the analyzer, if any. For example, the BETA RECIP-TRAP engine analyzer has vibration measurement and analysis capabilities. Figure 5-2 is an example of a vibration signature of a Cooper-Bessemer Model KSV engine taken with the BETA RECIP-TRAP. The vibration signature looks similar to the signature shown in Figure 5-1. However, the RECIP-TRAP displays a filtered and processed vibration signal, rather than the raw vibration signal. The same filtering and signal processing must be used for vibration data to be compared from one session to another.

The measured vibration signal contains vibration from other sources besides the desired source wave form. Signal processing is required to remove the unwanted components of the vibration signal and recover, as best possible, the source vibration signal. Various types of signal processing can be used. Two examples are listed below:

- Band-pass filtering—Filtering is used to remove unwanted frequency components of the vibration signal. The high-pass filter removes frequencies associated with the engine speed and the low-pass filter removes frequencies above the mounted frequency response of the accelerometer. The low-pass filter should be determined based on the frequency response of the accelerometer and the sampling frequency.
- Time-domain (synchronous) averaging—Data points one period (engine cycle) apart are averaged. This type of averaging removes components of the signal that are not linked to the engine period (that is, random noise). This method of filtering is typically not used by engine analyzers.

Signal processing can be performed as the measurements are being recorded or after measurements are taken (post-processing). If the analyzer supports post-processing, it is recommended that filtering be accomplished after the data is acquired and recorded. The use this method allows the user to refer back to raw data if questions arise. All frequencies above half the sampling frequency range of the accelerometer should be removed from the signal. For example, if the sampling frequency is 50 kHz then a low-pass filter should be used to remove all frequencies above 25 kHz.

Until a sufficient base of data is built up for the engines at a particular station, it is recommended that vibration be recorded for individual engine cycles (no averaging) if your analyzer supports post-data acquisition averaging. Data can be taken for multiple cycles on the same cylinder and time-domain averaging can be performed as a part of post-processing.

5.5 Sampling Rate

The sampling rate used to acquire vibration data should be determined based on the frequency components associated with vibration signal. The sampling rate should be a minimum of twice the maximum frequency components of the vibration signal to ensure that signal aliasing is avoided.

5.6 Sources of Error

The usefulness of engine analyzer data for analysis and balancing is highly dependent on the accuracy of the data. The potential for error exists at every step of the engine analysis process; from setup of analysis equipment to review of analysis data. Many of the calculated engine performance parameters, such as IMEP and IHP, are very sensitive to small data acquisition errors. While EDG engine analysis is not intended to be a research project, it is prudent to keep data acquisition errors as low as practical. Potential sources of error are discussed in this section and include:

- Improper selection, calibration, preparation, and mounting of cylinder pressure transducers
- Effects of the cylinder pressure transducer indicator passage
- Phasing errors between the cylinder pressure measurement and the crankshaft position
- Load drift during data collection
- Analyzer setup

The analyzer measures cylinder pressure and crank position and uses these two variables to calculate other engine performance parameters of interest. In order for the analyzer to make these calculations correctly, the correct engine geometry information such as cylinder bore and piston stroke must be entered.

5.6.1 Selection, Calibration, Preparation, and Mounting of Cylinder Pressure Transducers

Selection, calibration, preparation, and mounting of cylinder pressure transducers are discussed in Section 5.2.

5.6.2 Indicator Passage Effects

On most large diesel engines, internal cylinder pressure is recorded by an indicator passage machined into the cylinder head (or cylinder liner for FMOP engines). These passages range in length from several inches to a few feet. An indicator valve, or Kiene valve, is mounted to the end of the indicator passage by a cast iron boss. Pressure transducers capable of recording pressures inside the cylinder typically cannot be mounted to the Kiene valve without the use of an extension. The resulting configuration (a long, narrow passage) interferes with accurate dynamic cylinder pressure measurements. For this reason, engine research has usually been performed with the cylinder pressure transducer mounted flush with the cylinder head flame face. However, this is not practical for field measurements of cylinder pressure. The effects of the indicator passage, indicator valve, and extensions are well documented. References 28 through 36 discuss the effects:

- Delays of the cylinder pressure signal, inducing a significant phase shift between the actual peak pressure angle and the measured peak pressure angle
- Damping of the cylinder pressure signal, reducing the magnitude of the signal
- Resonance effects due to indicator passage geometry that could cause a standing pressure wave. Any resonance could result in pressure oscillations and indicated peak pressures that are much higher than what actually occur in the cylinder.

These effects are present to varying degrees in all indicator passages. The phase shift between crank angle and cylinder pressure will typically be the most significant effect. The most accurate method of ensuring proper phasing between the TDC reference mark and engine cylinder pressure is by using the thermodynamic loss angle (TLA) method during a compression motoring test on the reference cylinder. This method will also accommodate for the effect of the indicator passage.

Many analyzers allow the operator to input the known pressure delay or phase shift manually into the analyzer, and the analyzer will then make all the necessary corrections. Otherwise, the crank angle timing must be adjusted to offset the time delay or phase shift. This can be accomplished by mechanically shifting the crank angle measurement by the known delay angle.

5.6.3 Crank Angle Phasing and Correction

Many sources [24,25,28,30–35] address the sensitivity of correct phasing of crank angle and cylinder pressure. Clearly, errors in crank angle measurement directly offset "timing" measurements on the P-θ diagram, such as θ_{Pmax} and the start of combustion (SOC); while other crank angle insensitive parameters such as P_{max} are unaffected. Calculated parameters, in particular IMEP and IHP, are significantly sensitive to phasing errors between cylinder pressure and crank angle. Reference 24 reports sensitivities as high as 5% per degree error for full-load IMEP and 40% per degree for motoring IMEP.

The literature recommends that the crank angle (and TDC) be accurately measured to within 0.1° of crank angle for activities that involve studying engine combustion. However, this value is likely to be unrealistic for non-laboratory settings. Instead, it is recommended that the total error on TDC and crank angle be limited to approximately 0.5°

using reasonable methods, and that measurements of the total error be made to ensure the total error remains in the acceptable range.

There is more than one method for accurately determining the true TDC for each cylinder and ensuring the cylinder pressure is phased correctly with the crank angle. The timing marks on the engine flywheel are the traditional method for determining TDC. However, flywheel marks can be mismarked or painted over. Therefore, an alternative method for determining TDC should be used at least once to verify the TDC readings. The most accurate method for determining TDC is the thermodynamic method. Also, it is typically the most time effective method. The mechanical method was historically used prior to the development of electronic engine analysis methods. However, the mechanical method is very time consuming. Further, the results of the mechanical method must be corrected for the effects of the indicator passage, which is frequently not a straightforward effect, whereas the thermodynamic method is self-correcting.

Thermodynamic method—The location of peak pressure in an engine cylinder with no combustion must fall slightly before TDC. This phenomenon, which has been referred to as thermodynamic loss angle (TLA), [29] can be explained by considering the three factors acting to change the cylinder pressure in a motoring cylinder:

- Volume change caused by cylinder motion
- Heat transfer
- Gas leakage

During the majority of the motoring engine cycle, piston motion is the dominant effect. But as the piston approaches TDC, its velocity slows. At the instant the piston is at TDC, the piston is motionless. During the approach to TDC, the pressure and temperature of the cylinder gases are reaching their highest points. Consequently, leakage and heat transfer, both of which serve to reduce cylinder pressure, assert their greatest influence. The rise in pressure due to compression is overwhelmed by leakage and heat transfer as the piston approaches TDC, causing the pressure to begin decreasing before the piston reaches its peak location. Only during isentropic (that is, quasi-static with no heat transfer or leakage) compression would peak pressure occur precisely at TDC. All engines are subject to at least some heat transfer and leakage loss and therefore have peak pressure located slightly before TDC. Noise or indicator passage errors could cause measured pressure histories to appear to contradict this fact. Independent observations have given some insight to the proper location of peak monitoring pressure for ensemble averaged pressure histories. Thermodynamic lag angle has been reported to be inversely proportional to engine speed [29] and to be on the order of one degree [39]. However, these results cannot be considered universal because the leakage and heat transfer will vary from engine-to-engine (and from cylinder to cylinder on a given engine).

If leakage and heat-transfer rates were known exactly, a first law of thermodynamics model could predict the exact location of the peak compression pressure. Unfortunately, these quantities are difficult to predict accurately enough to use such a model to establish the location of TDC. The first law of thermodynamics can be used to write an expression for the polytropic constant that depends on engine speed, engine geometry, trapped mass, pressure, heat transfer, and leakage [38].

$$n = \gamma + \frac{1}{w}\left[(\gamma - 1)\frac{\dot{Q}_l}{PV} + \gamma\frac{\dot{m}_l}{m}\right]\frac{V}{\frac{dV}{d\theta}} \qquad \textit{Eq. 5-1}$$

where

n	=	polytropic constant
γ	=	specific heat ration
w	=	engine frequency
$\dot{Q}_l$	=	rate of heat loss
P	=	cylinder pressure
V	=	cylinder volume
$\dot{m}_l$	=	rate of blowby mass loss
m	=	mass of cylinder contents
$\frac{dV}{d\theta}$	=	rate of change of cylinder volume

This equation shows that near TDC, the polytropic constant changes rapidly as the rate of change of cylinder volume approaches zero. Because the rate of change of cylinder volume changes sign, the polytropic constant goes from negative infinity to positive infinity as the piston moves through TDC. Expanding Equation 5-1 by using a Taylor series and simplifying the resulting expression by neglecting the higher order terms, an expression for the polytropic constant as a function of crank angle can be obtained:

$$n(\theta) = K\left(1 - \frac{\theta_P}{\theta}\right) \qquad \textit{Eq. 5-2}$$

where

n	=	polytropic constant
K	=	constant
θ_P	=	angle of peak pressure
θ	=	crank angle

The values of K and θ_P are dependent on the constants of the first order terms in the Taylor series, which approximate the heat-transfer and leakage terms. These constants would be more difficult to evaluate explicitly than the leakage and heat transfer terms that they approximate. However, the measured pressure data can be used to calculate the polytropic constant at two particular crank angles, θ_1 and θ_2, using the following equation.

$$n = \frac{\ln(P_1/P_2)}{\ln(V_2/V_1)}$$

θ_1 and θ_2 should be less than 10° to ensure that the small angle assumption used in the derivation of Equation 5-2 is valid. These values can then be substituted into Equation 5-2 to solve for K and θ_p. Equation 5-2 can be compared to the measured polytropic constant at each crank angle near TDC. This process can be iterated with incremental changes to the TDC phasing of the measured pressure until the measured polytropic exponent versus crank angle is closest to the relationship predicted by Equation 5-2. This method is accurate to within approximately 0.3° of crank angle [38].

Mechanical Method of Checking / Setting TDC—Most diesel engines are provided with timing marks inscribed on the engine flywheel or barring gear. These timing marks provide a reference for TDC for each cylinder. Typical factory-provided timing marks are accurate to within ± 1°. However, this timing mark setting accuracy can and should be improved to support accurate engine balancing. For engines with single-acting pistons a dial indicator with a long extension is required to determine TDC accurately using the mechanical method. In addition, a bushing or guide to center the dial indicator accurately in the fuel injection port of the cylinder head should be fabricated.

An initial estimate of TDC should be made by using the existing timing marks and aligning the pointer with the TDC mark on the flywheel when the piston is at this position. This initial setting is only approximate because the piston vertical position in the cylinder changes very slowly with crank angle near TDC. This makes the exact TDC difficult to establish. A more accurate determination of TDC is made by utilizing the more rapid change in piston position with crank angle that occurs at approximately 40° on either side of TDC. With the dial indicator tip contacting the top of the piston crown, the engine is rotated about 60° from the approximate TDC in either direction. It is then rotated back toward TDC and stopped at 40° from TDC (as indicated by flywheel markings). A dial indicator reading is taken. The engine is now rotated 60° past TDC in the other direction and then rotated back toward TDC until the dial indicator reading is the same as the one taken at 40° BTDC on the other side. The crank angle at that position is read. By taking both readings as the piston is moved toward TDC, the effects of changes in mechanical clearances are minimized. TDC is the crank angle at the midpoint between equal dial indicator readings. The engine is rotated until this midpoint is aligned with the pointer, setting the piston at minimum cylinder volume. The TDC mark is then prescribed with the pointer. The above procedure might have to be repeated to refine the readings. Additionally, it could be done for each cylinder.

The precision of the mechanical method for checking TDC is dependent on the precision of crank angle readings taken from the flywheel. The degree markings machined on the flywheel are assumed to be spaced accurately. Any errors in readings are then due to the width of the pointer, the width of the flywheel degree markings, and parallax. This mechanical method can be used to determine TDC to within 0.1°–0.25°.

TDC for each cylinder can be determined in this fashion and marked on the flywheel for each EDG at a station. The baseline of the engine timing marks need only be performed once.

Once TDC is established using this technique, fuel injection, and inlet and exhaust valve timing can be checked by the refined flywheel timing marks. Further, the flywheel timing marks can now be used to set up the crank angle probe or signal pick-up connected to the engine analyzer. The method of accurately calibrating the crank angle transducer is dependent upon the specific analyzer involved. The analyzer vendor's instructions should be followed in this area.

Note that crank angle, as measured at the flywheel, will vary due to torsional vibrations or alternating twist in the crankshaft. These effects are small and do not effect the results of engine balancing and analysis.

5.6.4 Load Drift

Often, engine load will drift during the data acquisition process, resulting in variations in cylinder pressure. To preclude this, it is recommended that the governor be locked to preclude fuel rack movement, if possible. The governor should only be locked if the implications of this action are understood—that is, it might make the EDG inoperable—and if it needs to be locked due to load drift or changes. See Section 5.7 for additional information.

5.6.5 Configuring the Analyzer

The cylinder pressure transducer calibration should be checked both before and after engine data is recorded. In addition, checks to ensure TDC and crank angle position are properly aligned with the crank angle transducer are appropriate each time the analyzer is used. Errors in crank angle and TDC of only a few degrees will skew the calculations of IHP and IMEP, as well as the measurements of θ_{Pmax}. As a result, this data will not be comparable to the baseline engine data to support diagnostics, if necessary.

The cylinder pressure transducer must be calibrated in accordance with utility calibration programs and based on operating experience. It is a good idea to check the operation of the transducer prior to starting and warming the engine to ensure that the analyzer is functional. A dead-weight tester provides the most accurate static calibration of the cylinder pressure transducer. Some utilities have used high pressure gas in conjunction with a calibrated pressure gauge. Both of these calibration methods are static and must be used with caution when using piezoelectric pressure transducers. These transducers are best for measuring dynamic pressures such as cylinder pressure in a reciprocating engine. Some companies manufacture dynamic pressure transducer calibration machines, but they are expensive and their use is not necessary. The transducer should be calibrated over the entire range of expected pressure values.

Crank angle and TDC can be checked by barring the engine in the direction of normal rotation and ensuring that the TDC measurement coincides with the TDC mark on the flywheel for the cylinder of interest. Typically, TDC for other cylinders is entered as an offset to the TDC value of the cylinder for which TDC is measured directly.

There are a number of methods of generating crank angle signals. These include magnetic pickups operating off of crankshaft, camshaft, or flywheel gear teeth, an optical pickup using reflective tape on the crankshaft or flywheel, and optical encoders rigidly attached to the crankshaft. All engine analyzers are supplied with some type of crank angle signal generator. It is desirable to have both a once per revolution signal to reference the TDC position, usually referred to as the TDC pulse, and a more frequent signal, usually occurring every degree or every few degrees. It is possible to use a only a once per revolution signal with the analyzer interpolating all of the crank angles between each TDC position. However, if the crankshaft rotational speed is not constant during each revolution, this will introduce some error into the engine analysis parameters that are dependent on crank position.

Once the proper TDC location has been determined, the next step is to install the crank angle measurement device so that the crank angle measurement will be properly phased with the mechanical TDC. The indicator passage will delay the cylinder pressure signal. As a result, the actual peak firing pressure will occur before the measured peak firing pressure. If the mechanical method of phasing TDC is used, this delay must be accounted for in the analyzer setup. Some engine analyzers allow input of an indicator passage delay angle and correct the crank angle signal accordingly. If the analyzer does not allow for input of a delay angle, the crank angle signal generator will have to be manual offset to account for the delay. If the thermodynamic method of phasing TDC is used, the indicator passage delay is already accounted for because the indicated cylinder pressure is used to phase TDC rather than the actual cylinder pressure or piston position.

5.7 Preparing the Engine for Taking Data

A number of steps must be accomplished prior to performing engine analysis. These include: starting and loading the engine, allowing it to warm up to operating temperature, and setting the engine load.

5.7.1 Start and Load the Engine

The engine should be started and operated at rated load in accordance with utility and vendor procedures. The engine should then be operated at full load for at least one hour prior to taking data to ensure that the engine is at steady-state.

In many cases, the generator load is indicated in the engine bay as well as being repeated in other locations, such as the main control room. Often, differences in observed readings between various power meters for a given diesel generator exist. For this reason, it is desirable to select a single power meter for each engine that will be used for all future engine analysis run for that engine. Further, the power meter readings should be checked against the generator voltage, current, and power factor readings. In this way, the same load will be used for all tests, ensuring that test results are comparable to the baseline data for that engine.

Engine analysis data should be collected at or near the load at which the engine is expected to operate. Note that the rated load per cylinder in terms of brake mean effective pressure (BMEP) might vary between sites for the same model engine due to different rated loads or engine configurations. As a result, engine analysis data from different sites and engines might not be directly compatible.

5.7.2 Lock the Governor

Often the load on the diesel generator will vary somewhat due to changing plant and electrical grid conditions. Changes in overall engine load will produce changes in individual cylinder loads. To minimize load swings, the load should be set by “locking the governor” by the load limit knob on the governor actuator, if possible. Locking the governor is accomplished by raising the load of the diesel generator (by the governor speed setting) known to some value slightly greater than the final desired test value. The diesel generator load will control the speed; therefore, the diesel generator should be synchronized to the grid before locking the governor so that adequate speed control is maintained. This method essentially prevents the fuel racks from moving unless the diesel experiences a load rejection or over-speeds.

None of the safety features of the engine are disables by this technique. However, locking the load opens the formerly closed-loop speed control system. The diesel generator should not be loaded by plant hotel loads (unless synchronized to the grid) during engine analysis. The station operations department and appropriate engineering personnel should be consulted prior to locking the governor. If it is not appropriate to lock the governor at your site, special attention must be paid to load swings during the test. Finally, the load limit setting must be returned to its original value once testing is completed.

5.7.3 Set Engine Load

Establishing nearly the same load every time the engine analysis is performed is important from the standpoint of data trending. Many utilities have experienced relatively large discrepancies between local power meters and the power meters located in the main control room. It is recommended that operations consistently choose a single meter from the purpose of engine loading.

5.7.4 Record Initial Balance of Plant Data

Once the engine attains steady-state operating temperatures, the engine auxiliary system temperatures and pressures should be recorded. The purpose of this data is to document the operating conditions of the engine in case questions arise later. This data will assist in diagnosing engine problems if any are identified, and to compare against data taken at the end of the test to ensure operating conditions have not changed.

5.8 Measuring Cylinder Pressures

The actual measurement of cylinder pressure versus crank angle (and cylinder volume) is straightforward. The actions needed to record cylinder pressure will vary among engine analyzers. Documentation supplied with the engine analyzer should be consulted as to the actual procedure. The following provides some general guidelines for cylinder pressure measurement.

5.8.1 Indicating Valve Extension

If any adjustments have been made for the indicator passage, indicating valve, and valve extension, check the adjustment by performing a motoring test on one or more cylinders to ensure that the TLA is approximately 1° BTDC. In any case, the length and type of the valve extension used should be recorded.

5.8.2 Check Cylinder Pressure Transducer Operation

Cylinder pressure transducers are available in a large number of types. Many of these devices are relatively fragile and tend to drift with changes in temperature. Few can withstand more than one hundred hours of engine operation before replacement is required. Because of this, the cylinder pressure transducer should be checked before and after the pressure readings have been taken. Some utilities require that the cylinder pressure transducer be calibrated or checked more or less frequently, such as once per engine bank. Note that the pressure readings should be rerecorded if the transducer goes out of calibration during the readings. Ensure that any necessary cooling methods, such as cooling air directed at the fittings or transducer cooling water systems, are used during the calibration.

5.8.3 Reference Pressure

The inlet air manifold pressure at rated load should be recorded. This pressure should also be used as the reference pressure during data acquisition. The inlet air manifold pressure is assumed to be the same as the cylinder pressure with the piston at BDC (during the exhaust stroke for four-stroke engines). The reference pressure is used to correct dynamic pressure transducers, such as piezoelectric devices, that do not measure static pressures.

5.8.4 Record Cylinder Pressure

Specific instructions for recording cylinder pressure are available from each engine analyzer vendor. However, there are a few technical issues that should be highlighted.

The indicator passage should be cleared prior to recording cylinder pressure. This is accomplished by opening the indicating valve fully without the cylinder pressure transducer attached to the valve extension for several engine cycles or until all debris has been discharged. The purpose of this is to discharge any carbon or other products of combustion that might have become trapped in the indicator passage. The presence of foreign material will cause erroneous readings if it is not blown out. The indicator valve must be closed prior to attaching the cylinder pressure transducer.

Some cycle-to-cycle variation in combustion characteristics for a given cylinder is normal. Because of this, cylinder pressure should be recorded for approximately 200 cycles to obtain an average reading. If the analyzer is not capable of recording 200 cycles, record as many as possible. Some variation in cycle-to-cycle peak firing pressures are expected [33,35,36]. Averaging numerous cycles eliminates many of these effects as well as reducing the effect of signal noise. Many engine analyzers provide statistics on cycle-to-cycle variation in cylinder pressure. These statistics are important and should be recorded. Examples include standard deviation, combustion quality index, and combustion variation index. All are based on standard deviation calculations. Because there is not an industry-wide standard for combustion variation, standard deviation will be used throughout this manual.

The P-V and P-θ curves for each cylinder should be recorded at this time. (In most cases, cylinder vibration and ultrasonic signatures are also recorded concurrently.) If the engine analyzer is capable, both individual and average P-θ curves should be recorded. Given the choice of plotting one cycle or a composite, which represents the average of several cycles, the engine average (composite) is preferred. The composite reading provides the data on the average cylinder combustion characteristics.

Cylinder pressure data is taken for each cylinder during the balancing process. The order in which the cylinder pressure data is recorded is not important. However, if the balancing process is interrupted for any reason, such as to make adjustments to the fuel injection system and the engine is shut down, then the entire balancing process must be repeated for all cylinders. Adjusting the load carried by one cylinder changes the load carried by all of the cylinders.

A brief inspection of each curve should be performed at the time the data is taken. Abnormal cylinder pressure data could be caused by:

- A cylinder pressure transducer that has drifted or malfunctioned
- A loose connection between the cylinder pressure transducer and the indicating valve extension
- A blocked indicator passage
- A malfunction of the engine analyzer
- An actual combustion anomaly

The cylinder data should be retaken if any anomalies are present to ensure that equipment malfunctions are not the reason for the anomalies.

After the cylinder pressure data has been recorded for each cylinder and the operator is satisfied that the data is accurate, then the cylinder pressure transducer calibration should be checked to determine if any drift in the transducer has occurred. The cylinder pressure transducer should not drift more than 15 psi from the beginning of the balancing procedure to the end of the balancing procedure. This amount of drift represents about one percent of typical full load peak firing pressures. This value is only a rule of thumb, but it should be recognized that it still could represent as much as 10% of the allowable variation between all cylinders.

5.9 Recording the Data

As soon as the cylinder pressure data for all of the cylinders has been recorded and the data is determined to be satisfactory for analysis (cylinder pressure transducer calibration checks, etc.), then the remaining balance of plant data and cylinder exhaust gas temperatures should be recorded. The difference between the start of test and end of test balance of plant data should be calculated and checked against acceptable limits. The purpose of this data is to determine if any engine parameters have changed during the data collection process, including load, which could adversely skew the results of the test.

The data that is derived from the P-V and P-θ curves, such as P_{max} and θ_{max}, should be recorded. The derived θ_{max} should be corrected for the delay introduced by the indicator passage and the indicator valve extension, unless the correction was made in the engine analyzer during the data acquisition process.

6

ENGINE BALANCING AND TUNING

Ideally, engine balancing is the process of tuning an engine so that all of its cylinders produce the same amount of power for a given EDG load. An imbalanced engine will allow:

- One or more cylinders to hog the load
- Excessive vibration and crankshaft stresses
- Accelerated wear and degradation
- Increase the possibility that engine design limits will be exceeded

A balanced engine maintains:

- Equal load levels in each cylinder
- Minimizes engine vibration and wear
- Reduces stresses on internal engine components
- Reduces exhaust gas emissions

Much of the data recorded for engine balancing when engine load is constant for all data taken is also used for combustion analysis. This section covers engine balancing only. Section 7 of this guide addresses when to shift from engine balancing to engine troubleshooting and diagnostics.

6.1 Purposes of Engine Balancing

The two main purposes of engine balancing are:

1. Tune the engine to some nominal state specified by the engine vendor for which the vendor has considered such parameters as performance and fuel oil consumption
2. Balance the average power within each cylinder so as to minimize the engine vibration and dynamic stresses on the engine components

To accommodate both purposes, the recommended analysis guidelines for parameters used to balance the engine normally consist of two parts:

1. The *magnitude* of each parameter to ensure the nominal state of tune is achieved
2. A *span* or *range* about the engine average value to ensure the balance (or variation between cylinders) is satisfactory

In addition, some cycle-to-cycle variation in peak firing pressures and associated combustion parameters is normal and expected in diesel engines due to random variations in the combustion process [7,8]. The amount of this cycle-to-cycle variation might also be indicative of the condition and state-of-tune of the engine. Therefore, balancing criteria for some parameters should include limits on cycle-to-cycle variation in combustion parameters for a given cylinder.

6.2 What is a Balanced Engine?

In a perfectly balanced engine, each cylinder would produce identical pressure versus crank angle curves and, therefore, would produce the same amount of power. However, our inability to precisely control the amount of fuel injected and the timing of the injection for each cylinder prevents perfect balancing. An attainable goal, however, is to ensure that the pressure curves for all cylinders have the same characteristics and the power per cylinder is nearly equal.

The characteristics of the pressure curves for each cycle in each cylinder are determined by the combustion process in the cylinder. A deviation from the normal or expected pressure curve indicates a problem in the combustion process, in the mechanical components of the engine, or in the collection of the data. Identifying and diagnosing combustion anomalies falls within the scope of combustion analysis and are discussed in Section 7.

In many cases, combustion anomalies are small and indicative of normal engine degradation. Engine balance can sometimes be considered satisfactory with this type of degradation. When engine balance cannot be achieved, combustion analysis is used to identify, diagnose, and direct corrective actions.

6.2.1 Balancing Criteria

Indicated cylinder power can be calculated from the P-θ data recorded during engine combustion analysis. However, as discussed in Section 5, cylinder power calculations are very sensitive to small phasing errors between cylinder pressure and crank angle. Even a perfectly balanced engine can exhibit large cylinder-to-cylinder variations in calculated power if small phasing errors are present.

As a result, although equalization of cylinder power is the primary objective of balancing, itself should be a secondary measurement. Instead, measured data that is indicative of power should be used for balancing and combustion analysis to confirm that the characteristics of the cylinder pressure curves are within specifications. If the mechanical condition of two cylinders are approximately equal and the same amount of fuel is injected into each cylinder at the same time in the mechanical cycle, the resulting P-θ curves will have the same shape. As a result, if the timing and magnitude of the peak firing pressure of two cylinders can be equalized, the power produced by each cylinder will be approximately equal. The cylinder peak firing pressure (P_{max}) and the peak firing pressure angle (θ_{Pmax}) represent the coordinates of a single point on the P-θ curve. These are the two key variables that should be used to determine engine balance.

A range on the engine average values for both parameters should be imposed to ensure engine structural and performance limits are satisfied. Cylinder power measurements (if they are available) should be used as a secondary confirmation of cylinder balance. However, it should be recognized that the margin of error for cylinder power calculated from P-θ data is much larger than the margin of error for the P-θ data itself.

Different sources recommend the use of different variables for engine balancing. Cylinder peak firing pressure, fuel rack position, fuel injection pump timing, exhaust gas temperatures, cylinder peak firing pressure angle, indicated horsepower, and IMEP have all been mentioned as parameters that should be equalized from cylinder to cylinder in order to balance an engine. The benefits and drawbacks of the various balancing methods are discussed in Table 6-1.

Table 6-1
Advantages and Disadvantages of Various Balancing Methods

Balance Parameter	Advantages	Disadvantages
Cylinder peak firing pressure	Avoids problems associated with phasing crank angle with piston position. Easy, minimal equipment required.	Most likely will not result in a well-balanced engine if peak firing pressure angles or mechanical conditions vary from cylinder to cylinder.
Fuel rack position	Does not require special instrumentation and can be accomplished quickly.	Most likely will not result in a well-balanced engine due to variations in fuel injection pump timing. Also, this is an open-loop balancing because only an input to the combustion process is controlled.
Fuel injection pump timing	Does not require special instrumentation and can be accomplished quickly.	Most likely will not result in a well-balanced engine due to variations in the fuel rack. Also, this is an open-loop balancing because only an input to the combustion process is controlled.
Exhaust gas temperatures	Does not require special instrumentation.	Temperature is not a good indicator of cylinder power. Actually, it leads to poor power balance. Because the amount of fuel delivered to each cylinder is different, the amount of power produced in each cylinder will be different. In addition, exhaust manifold effects (standing waves, cylinder interactions, and cylinder location) can introduce significant exhaust temperature variations in some engines even though cylinder power balance might be relatively good.
Cylinder peak firing pressure angles	When used in combination with peak firing pressures, results in proper timing and value of peak of P-θ curve. If curves are similar in shape, it will result in good balance.	Requires special instrumentation, training, and time.
Indicated horsepower or indicated mean effective pressure	Equalizing indicated horsepower from all cylinders results in a balanced engine.	Difficult to equalize indicated horsepower across many cylinders. Small TDC phasing errors can lead to large errors in indicated horsepower.

Balancing just peak firing pressures will not necessarily ensure a well-balanced engine. The timing of the peak is just as important as the value of the peak. If the peak occurs too early, the compression stroke work transfer from the piston to the cylinder gases increases because cylinder pressure is higher during the compression stroke. This results in reduced engine efficiency. If the peak is delayed, it occurs later in the expansion stroke and is reduced in magnitude (due to the increased cylinder volume). This results in reduced expansion stroke work transfer from the cylinder gases to the piston, also reducing engine efficiency. The optimum peak firing pressure occurs when these two opposing trends just offset each other. Maximum engine efficiency occurs when the peak firing pressure angle is about 15° ATDC [5,38].

Perfect balance is unlikely to be attainable on any engine. Small differences in engine component dimensions and conditions will result in finite limits on engine balance.

6.3 Engine Analysis Guidelines

In order to evaluate the state-of-tune of an engine, engine analysis guidelines are needed. The parameters for which numerical analysis guidelines are appropriate were discussed in Section 4. Here, the methodology for developing the appropriate values for the guidelines is developed and discussed. Each of the parameters for which numerical guidelines should be developed are discussed, too.

6.3.1 Maximum Engine Average Peak Firing Pressure and Maximum Cylinder Peak Firing Pressure

Analysis guidelines for maximum engine average and maximum cylinder pressures are usually based on the structural limits of engine components such as cylinder heads, cylinder head studs, connecting rods, and connecting rod bearings. Most engine vendors have published operating limits for either maximum engine average peak firing pressure or maximum cylinder peak firing pressure. If available, these limits should be used as the analysis guidelines. A guideline for only one of these two parameters is sufficient, because if a limit for engine average peak firing pressure exists, the maximum cylinder firing pressure is controlled by the limit on the maximum range of peak firing pressures.

6.3.2 Maximum Cylinder-to-Cylinder Variation in Peak Firing Pressures

Published guidelines on the maximum cylinder-to-cylinder variation of peak firing pressures are rare. Where they exist, they are typically based on limiting engine vibration and crankshaft stresses. In addition to the normal, random cycle-to-cycle combustion variation, the tolerance should include allowances for:

- Instrument error
- Variations in engine load during data collection
- Variations in fuel rack settings between cylinders
- Differences in swept volumes between cylinder banks (for engines with articulated master-link connecting rods)

For operators of commercial engines, a total allowance (maximum to minimum) of 5% is typical [9]. For engines in nuclear standby service, a total allowance of 10% is considered acceptable.

6.3.3 Variation of Cylinder Peak Firing Pressure Over a Number of Cycles

Published guidelines on the variations of peak firing pressure in a single cylinder over a number of cycles are also rare. Because the mixing of fuel and air in the cylinder prior to and during combustion and combustion itself are random processes, combustion and parameters that are measured during engine analysis vary from cycle to cycle on a random basis [7]. Large variations from cycle to cycle are an indication of poor combustion performance (detonation or misfire) or engine component problems (inconsistent fuel injection equipment performance, blowby, and so on). Limits on cycle-to-cycle variation should be expressed in terms of the standard deviation of the cylinder peak firing pressure over a number of cycles. The cycle-to-cycle variability of peak firing pressure typically has a standard deviation on the order of 1%–2% over 50 cycles [8].

6.3.4 Engine Average Corrected Peak Firing Pressure Angle

Engine specific guidelines on the engine average corrected peak firing pressure angle are also rare. However, maximum IMEP and, therefore, maximum mechanical efficiency will occur with corrected peak firing pressure at about 15° ATDC [5,38]. In this case, piston motion during the rapid combustion phase will be small and will not greatly effect the cylinder pressure. Peak firing pressure angle is strongly dependent on fuel injection timing. Early injection results in long ignition delays due to the lower cylinder temperature and pressure when injection begins. The inward piston motion and rapid combustion combine to give high rates of pressure rise.

If injection timing is advanced far enough, the peak firing pressure can occur prior to TDC leading to reduced mechanical efficiency. If the delay is long enough, a large quantity of fuel can burn during the rapid combustion phase, leading to an excessive rate of pressure rise and high peak firing pressure with resulting high stresses and vibration. This is known as detonation or knock. Similarly, late injection timing also results in long ignition delay and high rates of pressure rise. However, in this case, piston motion is increasing the cylinder volume, leading to greatly reduced peak firing pressures and increased peak firing pressure angles.

Some of the important considerations when establishing guidelines for the engine average peak firing pressure angle include:

- The angle should be as close as practicable to the angle that gives the highest IMEP. This will result in the most efficient engine operation. However, with increasing emphasis being placed on engine emissions, the desired angle might need to be retarded to reduce undesirable emissions [40,41].
- The angle must be large enough to avoid the possibility of detonation, which could damage the engine.

The guideline should also allow sufficient margin for drift such that detonation will not occur. It should be noted that there will be one corrected peak firing pressure angle that will be best for mechanical efficiency and fuel economy and another that will be best for exhaust emissions. The actual angle chosen will depend on the situation at each nuclear power station.

6.3.5 Maximum Range of Peak Firing Pressure Angles Across All Cylinders

Guidelines on the range of peak firing pressure angles across all cylinders should be strict, with a range of no greater than 2°–4°. This is to ensure that proper power balance can be achieved.

6.3.6 Engine Average Fuel Injection Timing

Fuel injection pump timing is set to default values by engine manufacturers through the use of timing marks on the engine flywheel. The actual timing varies from model-to-model and from engine to engine in order to meet peak firing pressure limits and load requirements. It is often not published, but is mechanically set through the design of engine components. The pump timing is set with the engine shut down using, for example, timing windows (ALCO 251) or fuel injector timing gauges (EMD 645). However, this timing is static fuel injection timing.

The actual dynamic fuel injection timing, also known as start of injection (SOI), can vary from the static timing by as much as several degrees due to camshaft windup and variations in fuel injection nozzle pop pressure. Camshaft windup is the twist imparted to the camshaft because the camshaft is driven from one end, but has loads along its entire length. Camshaft windup results in retarded fuel injection timing at the end of the engine opposite the camshaft drive. As a result, equalizing the fuel injection timing for all cylinders will not necessarily result in the same start of injection for all cylinders. For this reason, it is recommended that θ_{Pmax} be used for the timing characterization of engine balance.

6.3.7 Maximum Range of Fuel Injection Timing Across All Cylinders

The design range of fuel injection timing across all cylinders is zero. However, due to tolerance in engine parts and wear, it is not possible to set the fuel injection timing exactly the same for all cylinders. In addition, the effects of camshaft windup must be accounted for. As a result, some differences between cylinders will exist. However, the differences should be kept small enough that they do not mask other problems. A range of ± 4° around the engine average fuel injection timing is considered appropriate.

6.3.8 Maximum Range of Fuel Rack Settings Across All Cylinders at Full Load

Some engine vendors have published limits for fuel rack balance. These limits are typically quite tight and are in the order of ± 1%–2% of full load fuel rack position. Because imbalances in the quantity of fuel injected into the engine lead directly to power imbalances, fuel rack balance should be maintained within the specified tolerances.

6.3.9 Allowable Range of Fuel Injection Nozzle Pop Pressures for New and Used Fuel Injection Nozzles

Most engine vendors have established limits for fuel injection nozzle pop pressures. Because the pop pressure will have an effect on fuel injection timing and duration, it is prudent to control nozzle pop pressures within the vendor recommended bands to prevent undesirable variations in fuel injection timing or fuel delivery. Adjustments to nozzle pop pressure are relatively easy to make when the nozzles are removed for testing.

6.3.10 Maximum Cylinder Exhaust Gas Temperature

Limits for maximum cylinder exhaust gas temperatures have been published by most engine vendors. These limits are typically based on the melting temperatures of engine components such as pistons, valves, exhaust manifolds, and turbochargers. If the engine vendor has published an exhaust temperature limit, it should be used as the engine analysis guideline. If a limit is not available, the analysis guidelines should be established at a value below the melting temperature of engine components. The guidelines should include an allowance for instrument error.

6.3.11 Maximum Range of Exhaust Gas Temperatures Across All Cylinders

Most engine vendors have published limits for the maximum range of exhaust gas temperatures across all cylinders. With most engines, if all of the cylinders are doing the same amount of work, the cylinder exhaust temperatures should be closely matched. As a result, prior to the general availability of engine analysis equipment, exhaust gas temperatures were used to check engine power balance. While this method is acceptable for some engines—for engines that do not exhibit a strong relationship between load and exhaust gas temperatures—it is less than adequate. An engine that has been power-balanced using engine analysis could have large cylinder to cylinder exhaust temperature variation due to standing waves in the exhaust manifold, cylinder interactions through the exhaust piping, or other effects.

Parameters for which numerical limits are not appropriate and that, instead, should be trended against past results for the same cylinder and against other cylinders to provide indications of engine condition:

- Combustion start angle
- Expansion reference pressure
- Terminal pressure
- Compression reference pressure
- Cylinder vibration signature
- Cylinder ultrasonic signature

6.4 Factors Affecting Balance

In addition to factors that operators and mechanics can adjust to affect engine balance, such as fuel rack position, fuel injection pump timing, and fuel injection nozzle pop pressure, there are a number of factors outside of their control that can affect engine balance or engine performance.

6.4.1 Mechanical Variations or Manufacturing Tolerances of Engine Parts

At least one station has experienced difficulty in balancing an engine due to differences in valve pushrod lengths from one cylinder to another. In addition, differences in fly-wheel timing mark locations, connecting rod lengths, cam lobe profiles, fuel injection pump calibrations can all affect engine analysis results.

6.4.2 Fuel Cetane Number

If engine analysis runs are conducted with fuels with different cetane numbers, the results will be affected. Higher cetane numbers will cause the fuel to ignite more quickly in the cylinder, resulting in shorter ignition delays and increasing the possibility of detonation. Because the fuel storage tanks at nuclear power stations are large and fuel turnover is slow, this will likely only be a factor when the entire fuel inventory is replaced. However, it is noted that most nuclear power stations only have requirements for minimum cetane number in their Technical Specifications.

6.4.3 Engine Load

As discussed in Section 5, changes in engine load during a test or from test to test will change engine analysis and balance results. In order to ensure adequate engine balancing and engine analysis data trending, engine analysis should be performed at the same engine load each time. One way to minimize load changes during engine analysis is to lock the governor.

6.4.4 Fuel Rack Giggle

Fuel rack giggle will affect the amount of fuel delivered to the cylinders.

6.4.5 Variations in Ambient Conditions

As discussed in Section 5, changes in ambient conditions from test to test will affect engine analysis results.

6.5 Adjustments to the Fuel Injection System

6.5.1 Fuel Injection Pump Timing

Both fuel injection timing and the quantity of fuel injected into the cylinder can be adjusted to tune the fuel injection system and balance the engine. Fuel injection timing determines the dynamic fuel injection and, therefore, determines both the peak firing pressure angle (assuming normal combustion occurs) and the peak firing pressure.

Advancing the fuel injection timing shifts P_{max} towards top dead center. The rapidly expanding products of combustion expand into a relatively smaller volume, increasing P_{max}. Retarding fuel injection has the opposite effect. On most engines, fuel injection timing can be adjusted for each cylinder individually.

Fuel injection pump timing is fixed by the mechanical configuration and adjustments to the camshaft and fuel injection system. However, SOI might deteriorate due to degradation in the fuel injection pump or nozzle tip holder. In particular, SOI is advanced with decreasing nozzle holder pop pressures and delayed by increasing pop pressures. Changes in SOI will have a direct impact on P_{max}.

θ_{Pmax} is strongly related to fuel injection timing and fuel rack position. Changes in θ_{Pmax} correlate directly to changes in the nominal fuel injection timing. Therefore, when adjustments to θ_{Pmax} are required, and the fuel racks are balanced (precluding further adjustments), they are affected by changing the fuel injection timing. However, θ_{Pmax} could degrade independently of fuel injection timing if SOI is affected by other faulty fuel injection system components.

θ_{Pmax} must not be too close to TDC or else excessive peak firing pressures and combustion instability will occur.

Note also that on some engines, adjustments to the individual fuel rack positions also adjusts the fuel injection timing at less than full load due to the shape of the helix on the fuel injection pump plunger. While fuel rack adjustments are normally made for fuel delivery purposes, the effect on timing should also be considered.

6.5.2 Fuel Delivery

The fuel rack position on each fuel injection pump controls the volume of fuel delivered to each cylinder. In addition, at low- and mid-level loads on some engines, the fuel rack position also effects the fuel injection timing. In any case, equal amounts of fuel must be injected into each cylinder to balance the power produced. As a result, relatively tight tolerances are placed on the cylinder-to-cylinder fuel rack variation.

6.5.3 Inlet Valve Timing Adjustments

For engines with inlet values, proper inlet valve timing directly impacts the effective cylinder compression ratio. Some engine operators have noticed improvements in engine balance after correcting inlet valve timing. It is strongly recommended that the inlet valve timing be checked and corrected (if necessary) for all cylinders prior to implementing a rigorous engine analysis program.

6.6 Balancing Methods

6.6.1 Operating Factors Affecting Balance

In addition to the variables listed in Section 6.4, several other factors affect engine balance. However, it is not possible to completely compensate for these effects. Engine imbalance can be attributed to:

- Inlet and exhaust manifold standing waves
- Cylinder interaction from the exhaust manifold
- Flow losses
- Mechanical variations in engine components
- Fuel pump calibration errors
- Valve clearances
- Gas leakage past piston rings and ports / valves

In addition, changes in atmospheric conditions (temperature, pressure, and humidity) during tests or between tests will result in variations in the recorded data. However, there is no fixed threshold at which the extent of imbalance prevents an engine from operating and producing power, although engine performance will degrade with increasing imbalance. As a result, this guide recommends three types of engine balancing:

1. Static balancing (set-up) of the engine—This method is used to establish the nominal settings of fuel injection pump timing, fuel rack settings, and fuel injector nozzle pop. This method should be used when the engine becomes so far out of adjustment that the operator determines that it is easier to start over, or when a complete engine overhaul is accomplished and the original balanced engine fuel injection timing cannot be replicated. Calibrated fuel injection pumps are recommended when setting up an engine using this method. Static balancing is mechanical only. No combustion analysis is conducted during static balancing.

   Because the static setup does not account for such factors as minor piston ring or port / valve leakage, camshaft windup, or other small mechanical differences between cylinders, static set-up of an engine might not result in a balanced engine when the engine is run. Static balancing is usually only required once for the entire engine, but can be used whenever questions arise regarding the settings on the engine.

2. Dynamic balancing of the key engine analysis parameters (P_{max} and θ_{Pmax})—This method recognizes the variations in cylinder mechanical components, such as leakage past ports / valves and rings, fuel injection pump calibration differences, camshaft windup, and other minor component variation or degradation. It is used to balance the engine quickly so that performance will be satisfactory for one nuclear power plant fuel cycle. This method could involve adjustments to the fuel rack, fuel

injection pump timing, or nozzle replacement. In essence, an imbalance of fuel is used to compensate for normal engine degradation and to obtain a balanced engine. This method might not be applicable to all engines.

3. Fine-tuning to optimize engine performance—Fine-tuning ensures that the engine is precisely balanced, all cylinders receive equal fuel, and all balancing performance parameters are within "like-new" specifications. Fine-tuning should be used during each fuel cycle, if possible. An engine optimized using this method should accommodate normal degradation of the fuel injection and other systems as well as compensate for camshaft windup. This balancing method is recommended following major engine overhauls, or after one nuclear power plant fuel cycle of operation without fine-tuning.

Using these three methods, the operator will have consistent methods to set up the engine following extensive maintenance in an initial state that minimizes further balancing, to quickly balance the engine when time is limited, and to optimize performance when the time is available.

Balancing or tuning the engine so that it just meets the established analysis guidelines leaves no margin for performance parameter drift over time. Therefore, it is recommended that the engine be initially balanced or tuned to a state better than the analysis guidelines to account for drift that will occur over time. Recommended balancing or tuning goals should be established for dynamic balancing and fine-tuning, which have tighter allowance bands than the analysis guidelines.

7

USING ENGINE ANALYSIS TO DETERMINE AND TROUBLESHOOT ENGINE PERFORMANCE AND CONDITION

7.1 Integrated Maintenance Program

The goals of an engine maintenance program are to maximize the availability and the reliability of the engine. As discussed in Section 2, an integrated maintenance program uses elements of preventive-, condition-, and performance-based maintenance programs to achieve these goals. A number of techniques for evaluating engine condition and performance have been developed and used by operators of diesel engines. Among these techniques are:

- Operating log review
- Periodic engine testing
- Non-intrusive visual inspections by borescopes, videoprobes, and thermography
- Visual inspections of disassembled parts
- Exhaust gas analysis
- Combustion analysis
- Vibration and ultrasonic signal analysis
- Lube oil, fuel oil, and cooling water chemical analysis

Most nuclear power stations already use several of these techniques in evaluating the condition and performance of nuclear standby EDGs. For example, all stations are required to perform periodic surveillance testing of EDGs, and most stations sample and analyze lubricating oil on a periodic basis. As discussed in Section 2, many of these techniques are most effective when used in combination with one or more of the others. Frequently, one technique will provide an indication of a problem, but will not provide enough information to isolate the problem to a single component. In these cases, other techniques will frequently permit positive identification of the source causing the problem.

7.1.1 Identifying Potential Problems

The first step to solving any problem is to identify that a problem exists and what it is. However, this is not always as straightforward as it might seem to be. In some cases, the problem could cause only subtle changes in performance. In others, prior maintenance actions might have been taken to correct the problem, but have actually masked it until a more serious condition becomes evident. For example, at one station a cracked cylinder head on an EDG allowed cooling water to leak into the combustion chamber, causing low firing pressures. Contract personnel performing maintenance work on the engine increased the fuel rack on that cylinder to increase the firing pressures to within the normal range. However, that cylinder remained low relative to the remaining cylinders. The cracked head was not found until lube oil samples showed increased amounts of chemicals used for jacket water system corrosion control.

Had the cause of the low firing pressures been investigated when it was first noted, a great deal of effort to find the source of the leak could have been avoided. Each engine problem will cause its own set of symptoms. It is up to the engine analyst to diagnose the problem based on the symptoms.

Typically, one of the techniques discussed in Section 7.1 will first provide indications of an engine fault. For the head crack example discussed above, combustion analysis would have shown low peak firing pressures for the cylinder, lube oil analysis would have showed contamination by jacket water chemicals, or engine logs would have showed an increased fuel rack position for the cylinder and decreasing water level in the jacket water system expansion tank. Which technique shows the first indication is often only a matter of timing.

7.1.2 Identifying and Characterizing the Failure

If a failure cannot be identified from the initial indication, the next step is to attempt to identify and characterize the degraded or failed component using one or more of the additional inspection techniques. In order to accomplish this, it is best to have a plan of attack worked out based on all the possible failures that could cause the symptom or symptoms discovered. Additional tests or inspections should be performed to confirm or eliminate suspected causes of the problem. The most likely causes of the problem should be investigated first. It is helpful to have a table or flow chart of typical or expected symptoms for the each suspected failure being investigated.

Most engine vendor technical manuals include some form of troubleshooting chart. However, they tend to be rather qualitative approaches to problems, and not based on actual measured data that might be available to the engine analyst. The table or flow chart generated to solve a specific problem should identify how the problem will manifest itself; that is, which inspection techniques will identify the problem and how the results of each inspection will be effected.

Table 7-1 depicts a table for combustion, vibration, and ultrasonic analyses. The left most column contains the observed potential symptoms and some of the potential problems that could cause it. The other columns contain other engine analysis parameters that could be affected by each of the problems. For example, if the peak firing pressure for a cylinder is low, potential causes include:

- Improper fuel rack setting

- Retarded fuel injection timing
- Late inlet valve closure
- Valve or port leakage
- Piston ring blowby
- High fuel injection nozzle pop pressure
- Bent connecting rod
- Decreased ignition delay (caused by fuel property changes for example)
- Water leakage into the cylinder

Table 7-1
Example of Simplified Engine Troubleshooting Chart

	Combustion Analysis												Vibration Analysis		Ultrasonic Analysis		Other Analysis
Symptom/Cause	Peak Firing Pressure	Peak Firing Pressure Angle	Start of Combustion	Expansion Reference Pressure	Terminal Pressure	Compression Reference Pressure	Start of Injection	IMEP	Compression Pressure	Maximum Pressure Rise Rate	Fuel Rack Position	Nozzle Pop Pressure	Time Domain	Frequency Domain	Time Domain	Frequency Domain	
LOW PEAK FIRING PRESSURE																	
Low Fuel Rack Setting	Low			Low				Low			Low						
Retarded Fuel Injection Timing	Low	Retarded	Retarded				Retarded	Low		Low					X		
Late Inlet Valve Timing	Low			Low		Low			Low				X				
Valve Leakage	Low			Low		Low			Low				X		X		
Piston Ring Blowby	Low			Low		Low		Low	Low	Low			X		X		
High Fuel Injection Nozzle Pop Pressure	Low						Retarded					High					
Bent Connecting Rod	Low			Low		Low		Low	Low								
Decreased Ignition Delay	Low	Advanced	Advanced							Low							Fuel Analysis
Water Leakage	Low																

For each of these potential problems, the effects on other engine analysis parameters are identified in Table 7-1. For example, if blowby is the cause, the expansion and compression reference pressures and peak compression pressure would be expect to be low and indications of blowby should be seen on the vibration and ultrasonic signal traces for the cylinder. In addition, the maximum pressure rise rate and IMEP for the effected cylinder might also be lower than the other cylinders. If retarded fuel injection is the cause, the peak firing pressure angle and start of combustion will be delayed, the indicated mean effective pressure and maximum pressure rise rate will be reduced, and the late nozzle opening event would be visible in the ultrasonic signal trace for the cylinder.

It is highly recommended that each station or EDG owners group develop troubleshooting tables or flow charts for their particular EDGs, or, at the very least, keep records of failures that are found.

Detection of engine problems using engine combustion, vibration, and ultrasonic analyses is a skill that takes time to develop. For the first several engine analysis runs, it is best to place the emphasis on identifying major anomalies such as:

- Detonating or misfiring cylinders
- Cylinder power imbalance
- Cylinder liner scoring or scuffing
- Blowby
- Leaking valves or ports

In some cases, engine faults will not be detected prior to the failure of a component. In these cases, it is important to review engine analysis data from the prior test runs to determine if any signs of the impending failure were present but not detected.

Finally, the use of engine analysis as part of an integrated maintenance includes several key elements:

1. An understanding of the difference between symptoms and causes of failures is required. It is beneficial to construct a flow chart or table of typical or historical symptoms for the specific equipment that you are operating. This effort can be rather large and can benefit tremendously from consideration of the experience of other similar equipment or identical equipment installed at other locations. A cooperative effort is this area by EDG owners group would be beneficial.
2. A database or record of prior failures and their symptoms is required to provide a guide during the diagnosis of symptoms.
3. The operator must understand the results of any tests, trending, inspections, analysis, etc. before they are accomplished. This is perhaps the most important point. This point cannot be over-emphasized. Performing numerous tests and inspections, recording and trending data, etc. without some understanding of the meaning of the data and the appropriate actions to take in the event of any deviations is of little

value. Understanding the tests, inspections, and analysis is important to accomplishing such procedures successfully. In addition, without this insight, the operator might inadvertently question good results and schedule unnecessary maintenance, or could miss indications of impending failure.

8

CASE HISTORIES

This section includes sample engine analysis data for many of the diesel engines in nuclear standby service, in addition to case histories of typical problems and failures. The case histories include:

- Power cylinder imbalance
- Misfire
- Piston slap
- Piston ring blowby or leaking valves/ports
- Unequal valve lash setting
- Excessive exhaust valve lash
- Faulty fuel injection
- Cylinder liner scoring or scuffing
- Port bridge problem
- Improved valve seating
- Bent connecting
- Burned exhaust valve

This section also includes plots of typical cylinder pressure and vibration traces for the following engines:

- ALCO model 251
- Cooper-Bessemer model KSV
- EMD model 645
- Cooper-Enterprise model DSRV
- Fairbanks Morse Opposed Piston model 38TD 8-1/8
- Colt-Pielstick model PC2

8.1 Power Cylinder Imbalance

The plot (see Figure 8-1) shows the P-θ curves for all cylinders of a 20-cylinder engine superimposed. In this engine, the spread in peak firing pressures is 10%, which is regarded as the maximum acceptable spread.

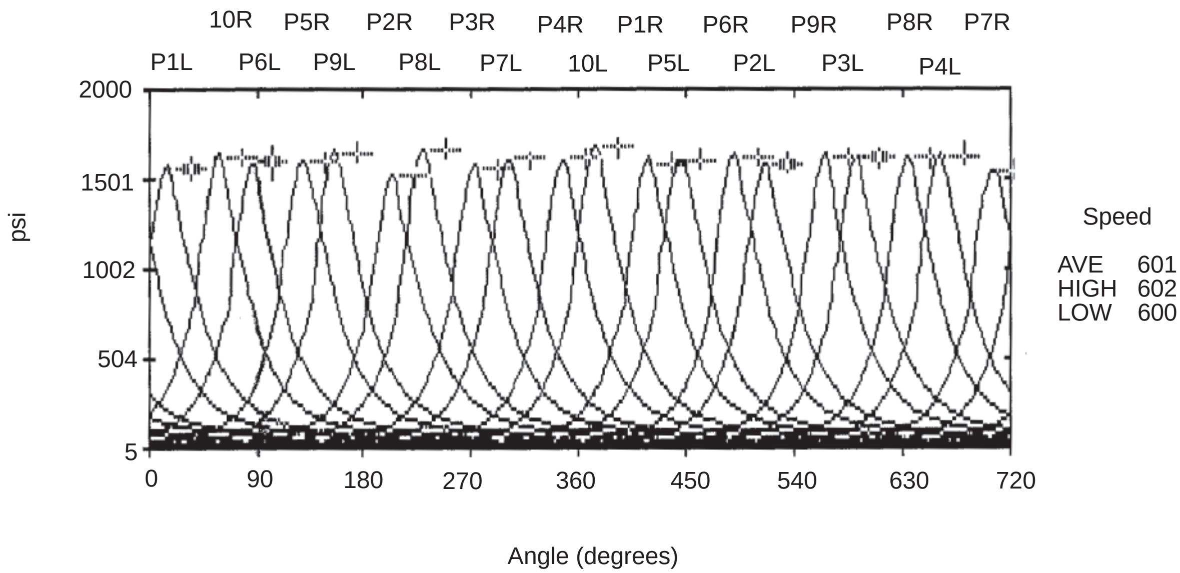

Figure 8-1
Power Cylinder Imbalance

8.2 Misfire

A misfiring cylinder would be indicated as shown above on the engine analyzers manufactured by Liberty Technologies / Beta. The vertical portion of the icon next to the peak of each pressure curve represents the minimum and maximum peak firing pressures which occurred in each cylinder over a number of cycles (see Figure 8-2). The lower end of the vertical bar for cylinder 5R, which extends to the compression pressure, indicates that the cylinder had at least one complete misfire during the cycles analyzed.

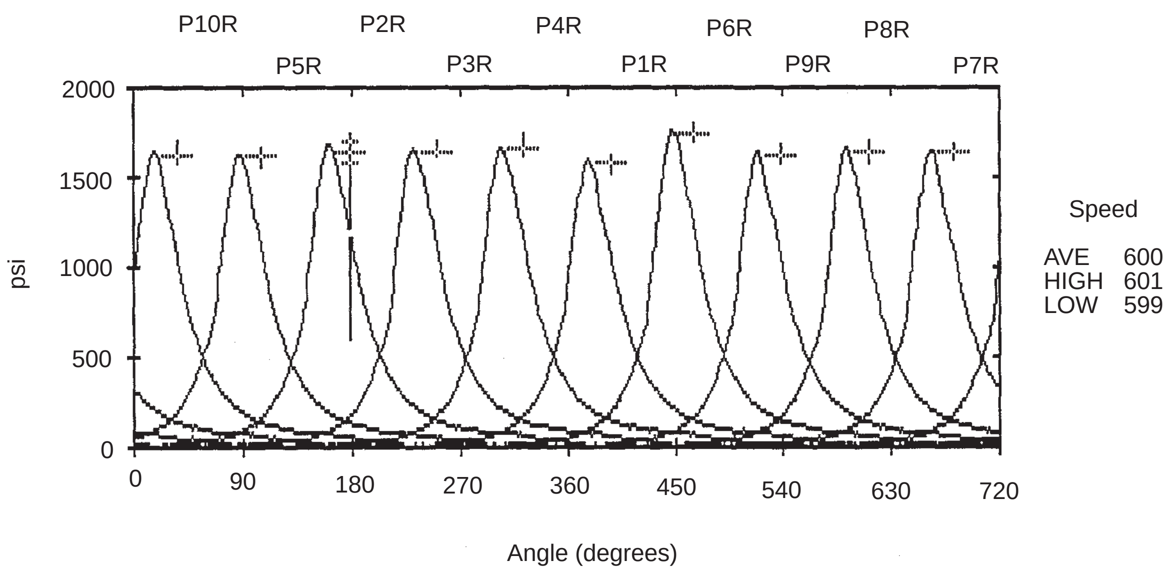

Figure 8-2
Misfire

8.3 Piston Slap

Figure 8-3 shows the vibration traces for several cylinders of the same engine. The traces are arranged so that they are all in phase in this plot. Piston slap shows up as a sharp mechanical impact occurring at about the angle of peak firing pressure as shown for cylinder 2L (second trace from the top on the left side).

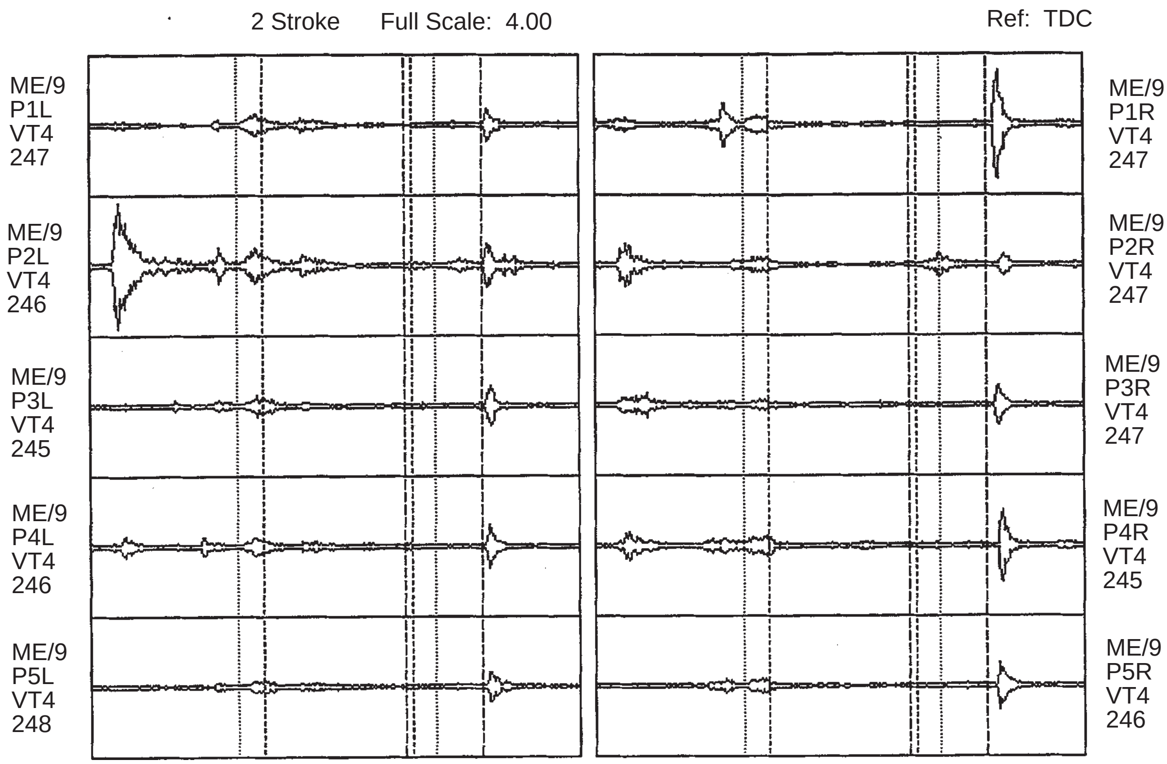

Figure 8-3
Piston Slap

8.4 Blowby/Valve Leak

Leakage past rings or valves will be detected as a gas flow signature during the compression process before the beginning of fuel injection. In Figure 8-4, such a pattern can be seen beginning prior to the -36° position. Of course, the leakage continues during the injection and combustion processes, but the signal is usually obscured by other energy. If significant leakage is occurring, the compression reference pressure for the cylinder will be low.

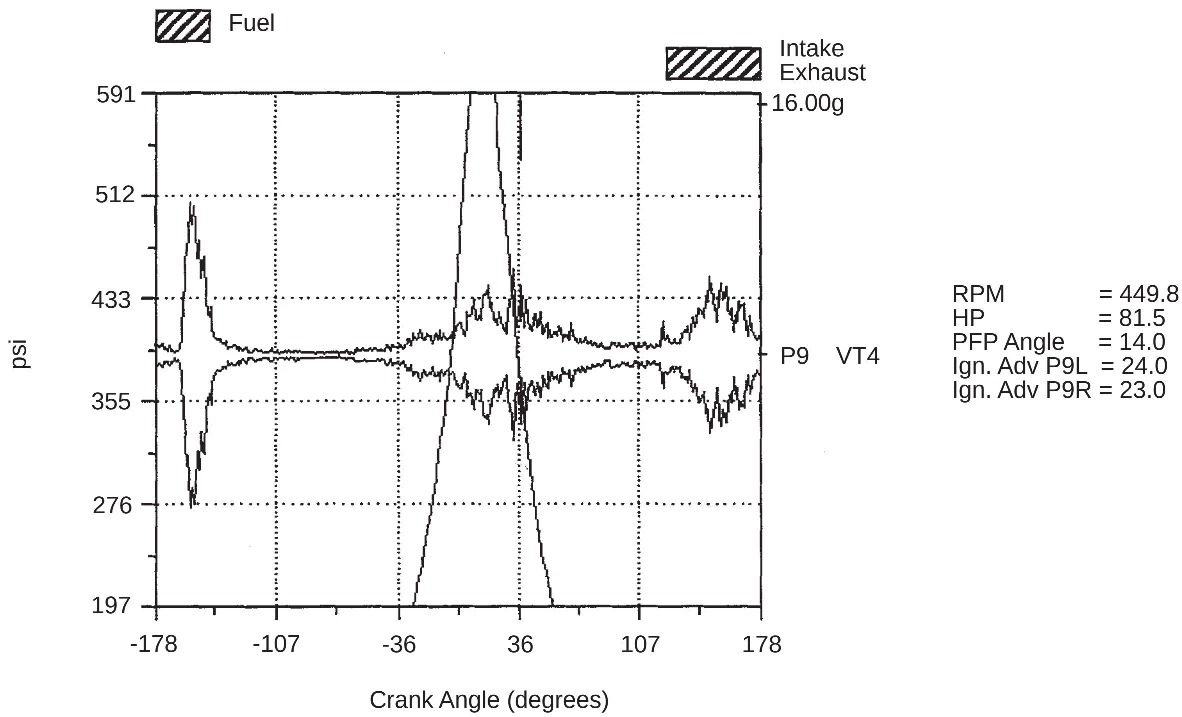

Figure 8-4
Piston Ring Blowby / Valve Leak

8.5 Unequal Lash Settings

In engines with two intake and exhaust valves, a pattern similar to that shown in Figure 8-5 will often be seen, where the two intake or exhaust valves (exhaust valves are shown in the middle of the figure) are closing at slightly different times. This is due to unequal lash settings or unequal wear in components. Some difference is permissible, but that shown is probably unacceptable.

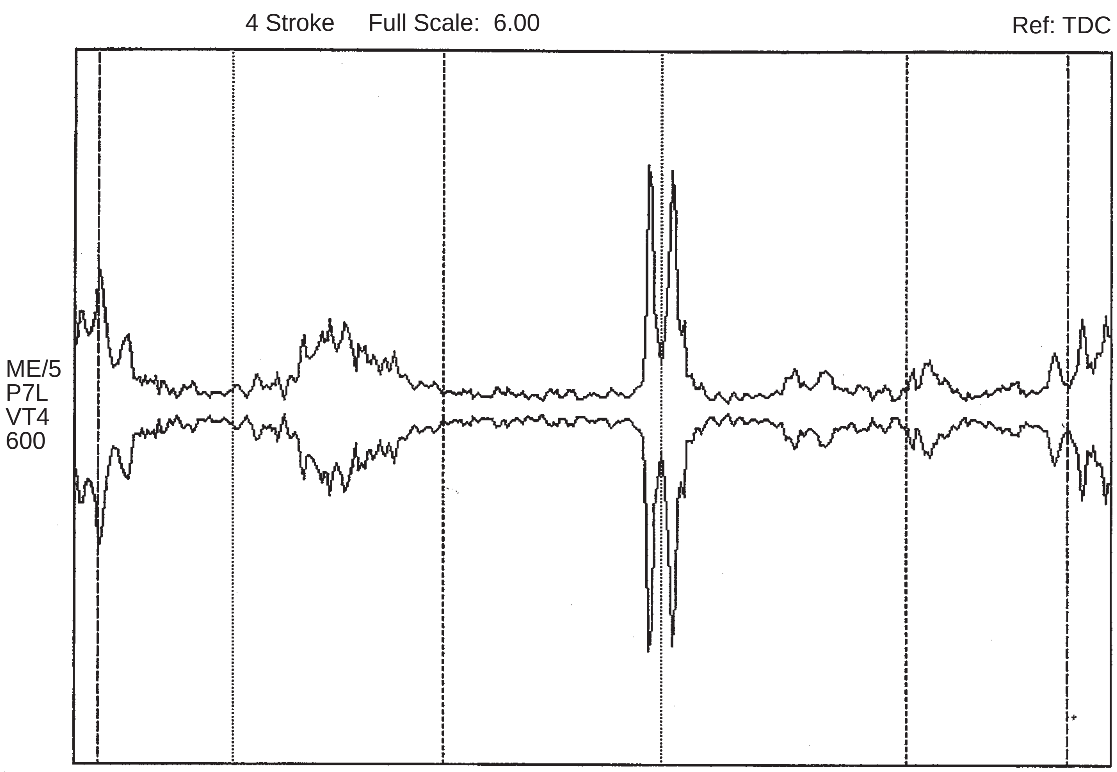

Figure 8-5
Unequal Valve Lash Settings

8.6 Excessive Exhaust Valve Lash

Excessive valve lash causes valve dynamics to deviate from design. The most observable effect is that the impact amplitude at closure starts to increase, and the closure becomes earlier (see cylinder 10L in Figure 8-6). Eventually, an impact would also be seen at the valve opening and the valve opening will occur late.

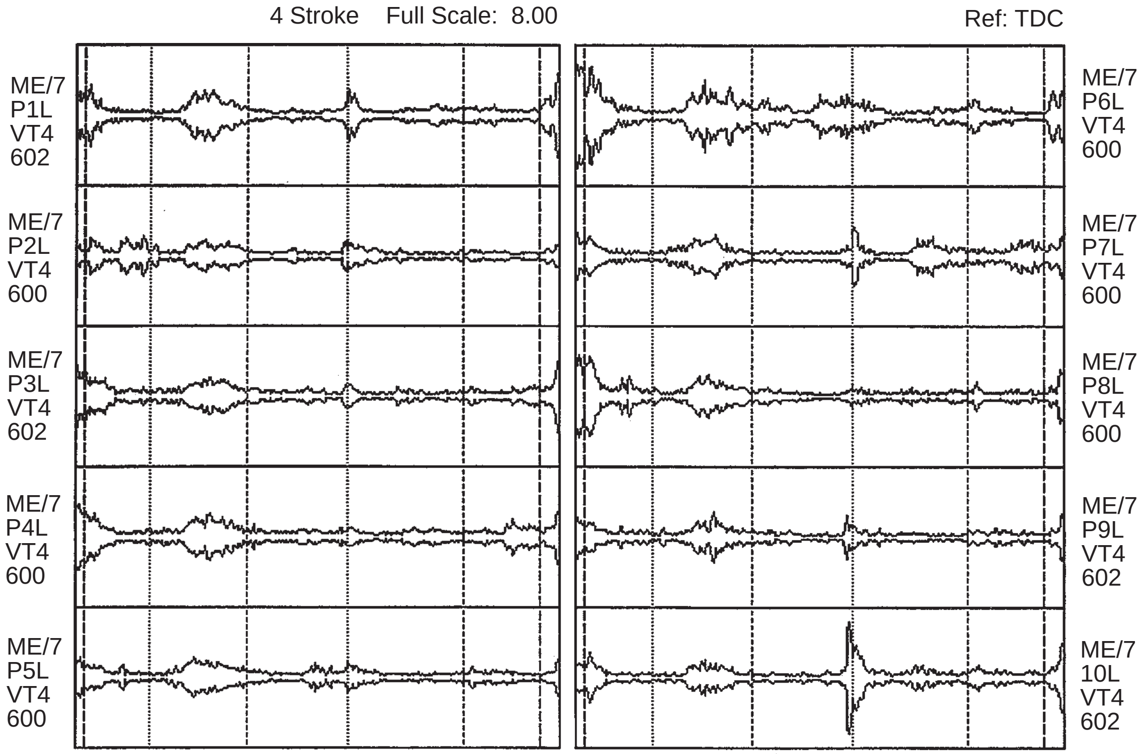

Figure 8-6
Excessive Exhaust Valve Lash

Possible causes include:

- A collapsed lifter
- A bent pushrod
- A worn rocker arm bushing
- A valve lash misadjustment

8.7 Faulty Fuel Injection

Table 8-1 shows the combustion analysis results for a six cylinder engine. Note that P6 indicated horsepower is low compared to other cylinders, even though its peak firing pressure is close to engine average. This would happen if the fuel injection duration was short.

Table 8-1
Combustion Analysis Results for a Six Cylinder Engine

Cylinder	Speed (rpm)	IHP	Press at 20° BTDC (psi)	Comb. Start deg. ATDC	Max. Rise Rate psi/deg.	Peak Firing Pressures					Peak Firing Pressure Angle ATDC	Diff. psi at 180° ATDC
						Ave (psi)	Dev (psi)	Max (psi)	Min (psi)	Delta (psi)		
P1	396	629	408 LO	0 LO	42 HI	1276	4	1295	1249	18	16	24
P2	394	616	425	2 HI	33	1202	3	1234	1178	-57	18	23
P3	397	632	419	0 LO	39	1257	4	1280	1236	-2	16	23
P4	398	629	419	0 LO	33 LO	1231	3	1253	1211	-28	16	25
P5	398	666	444 HI	0 LO	39	1332	4	1351	1301	74	16	26 HI
P6	401	597	422	0 LO	35	1253	4	1277	1229	-6	16	22 LO
Eng:	397	3769	423	0	37	1259	4			30	16	24
Sprd:	2%	11%	9%	2 deg.	23%	10%	27%				2 deg.	17%

The vibration patterns reveal the start and end of fuel injection. The closeup of the firing pressure and vibration curves for a typical cylinder is shown in Figure 8-7. The start and end of injection can be seen clearly.

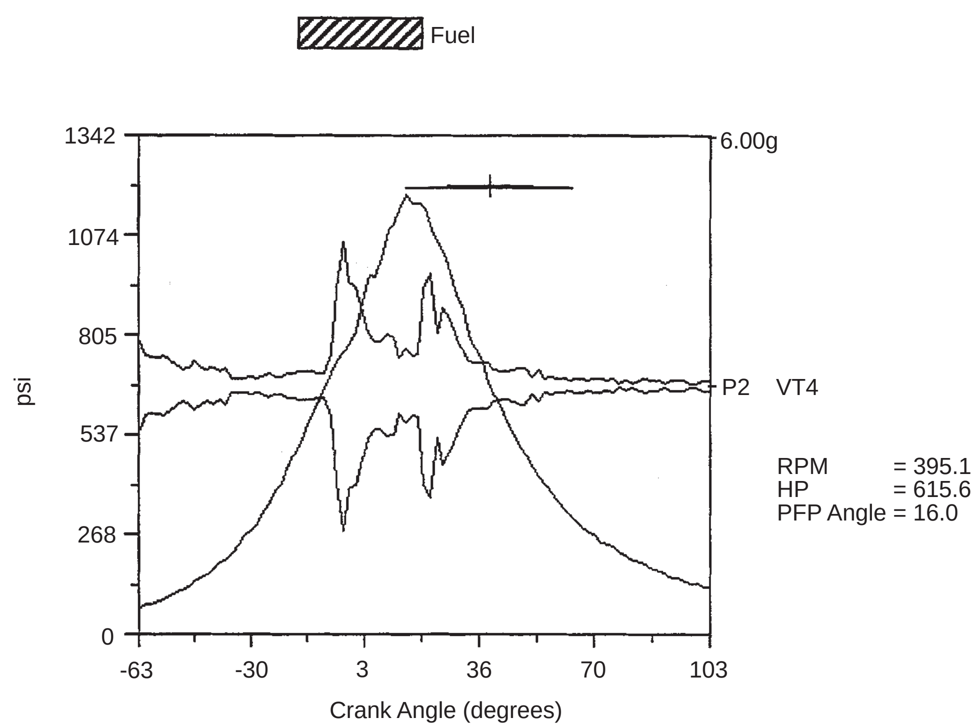

Figure 8-7
Normal Fuel Injection Vibration Pattern

In cylinder P6, the fuel cutoff is less obvious and seems earlier than in other cylinders.

Cylinder	Injection Begins (BTDC)	Injection Ends (ATDC)	Duration (deg)
P1	8.0	20.1	28.1
P2	7.5	21.5	29.0
P3	6.3	20.9	27.2
P4	6.6	20.6	27.2
P5	6.3	22.0	28.3
P6	6.7	18.0	24.7

The table above compares the fuel injection duration for each cylinder in the last column. Note that the duration for cylinder P6 is three to four degrees shorter than for the other cylinders. The suspected cause is a leaking fuel pump. Figure 8-8 depicts a faulty fuel injection vibration pattern.

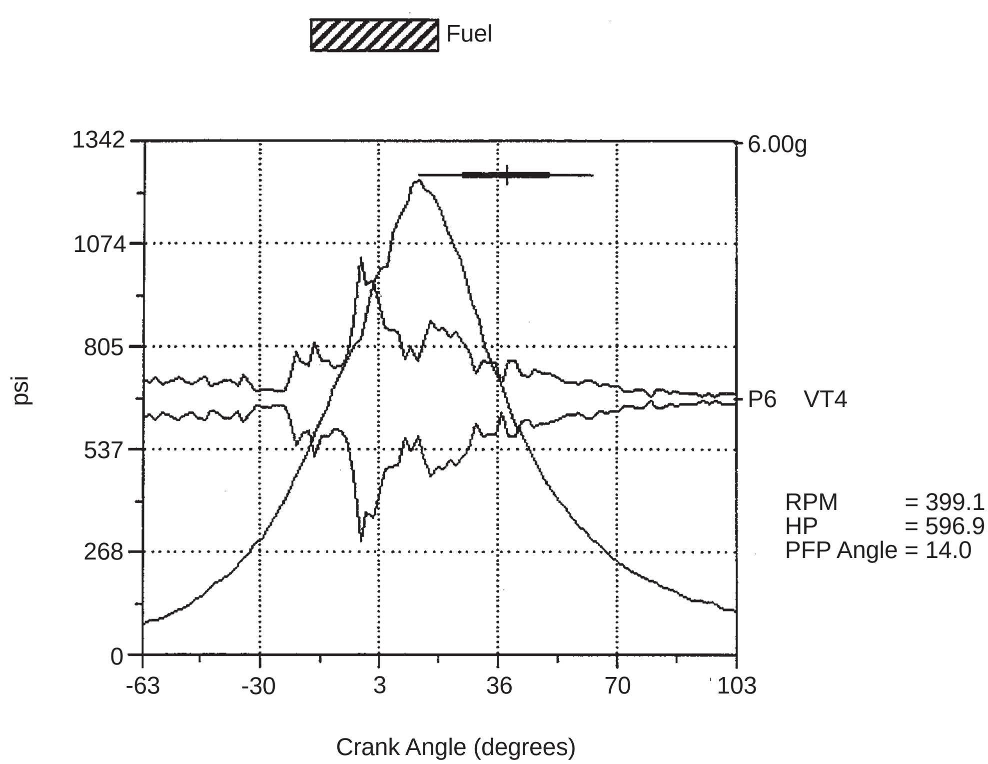

Figure 8-8
Faulty Fuel Injection Vibration Pattern

8.8 Scored Liner

Comparing all the vibration patterns, the most significant anomalies appear on P2R and P3R (see Figure 8-9). Because these two cylinders are adjacent to each other, it is important to check for cross talk. The anomalous components seen on P2R actually are from P3R.

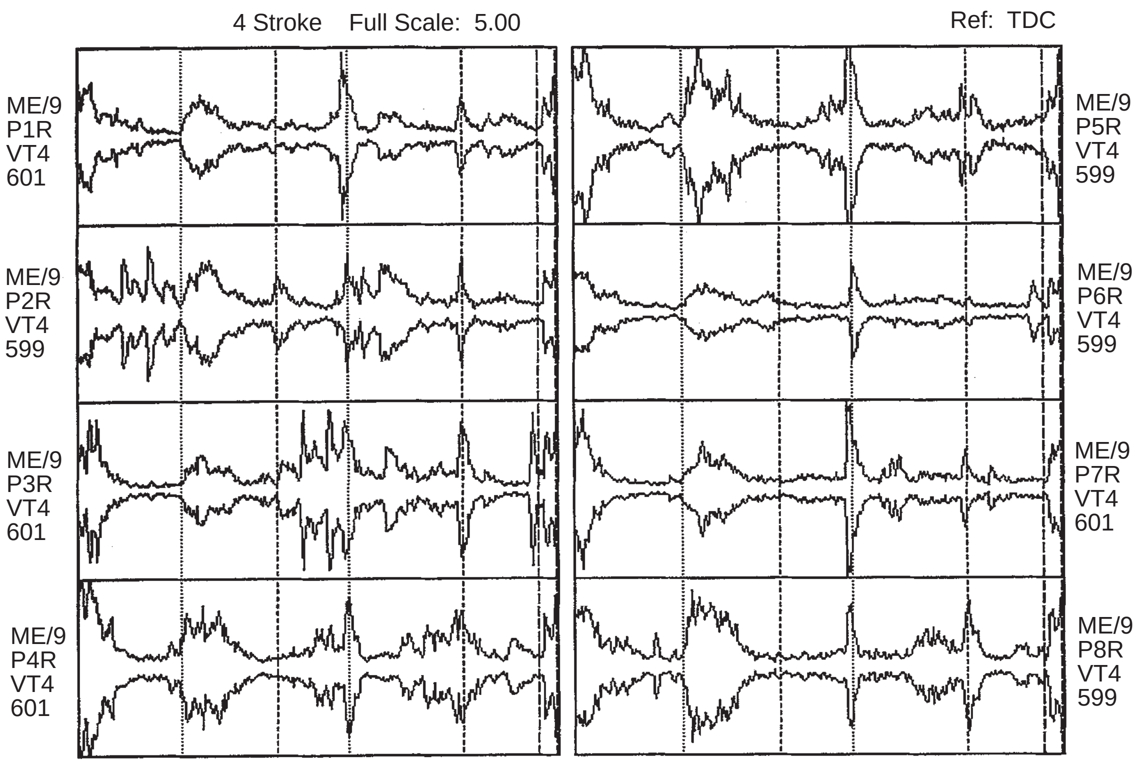

Figure 8-9
Scored Cylinder Liner

In P3R, the mechanical impact events occur at several points in the 720° cycle and are found symmetrically placed with respect to TDC. This is consistent with liner deterioration, such that piston rings cause repeated impacts.

8.9 Port Bridge Problem

The vibration patterns in Figure 8-10 are from an EMD 645 engine, with intake ports and exhaust valves. Note how the P2L intake port closure event shows up as a large mechanical impact. Normally the piston and rings should travel upward across the ports with little if any vibration energy.

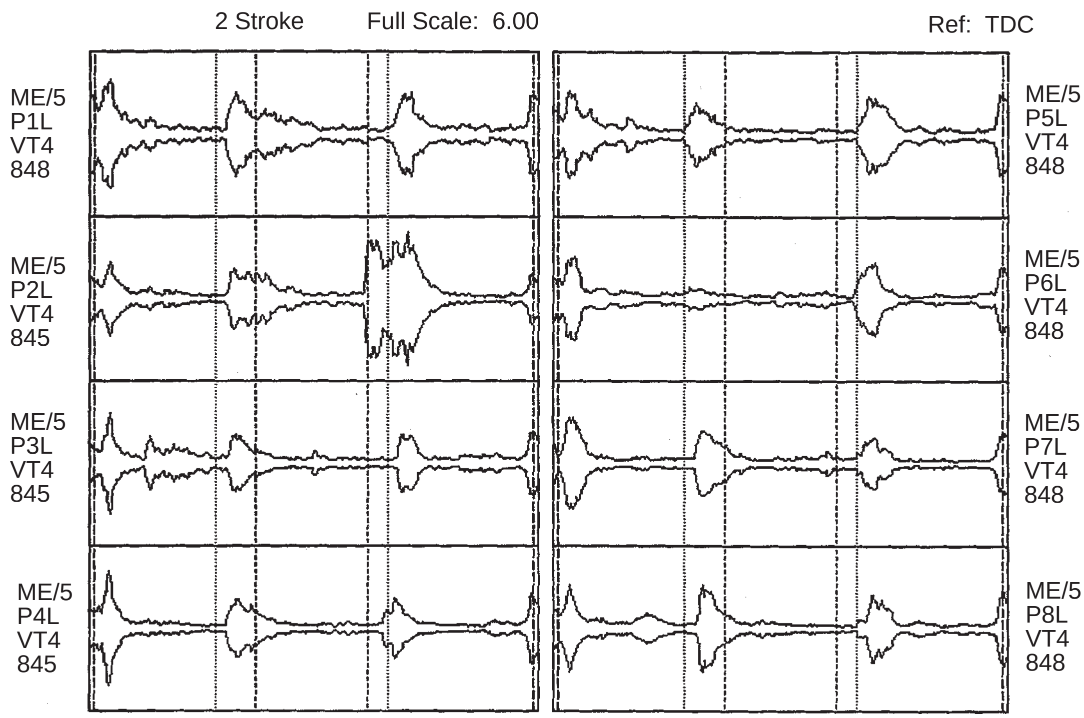

Figure 8-10
EMD 645 Vibration Patterns

The implication is that there is some type of blockage or barrier. One possibility is that there might be carbon buildup in the ports. Another possibility is that the port bridges have worn in such a way that there is a ridge impeding the ring travel. Figure 8-11 shows the P-θ and vibration plots for cylinder 2L.

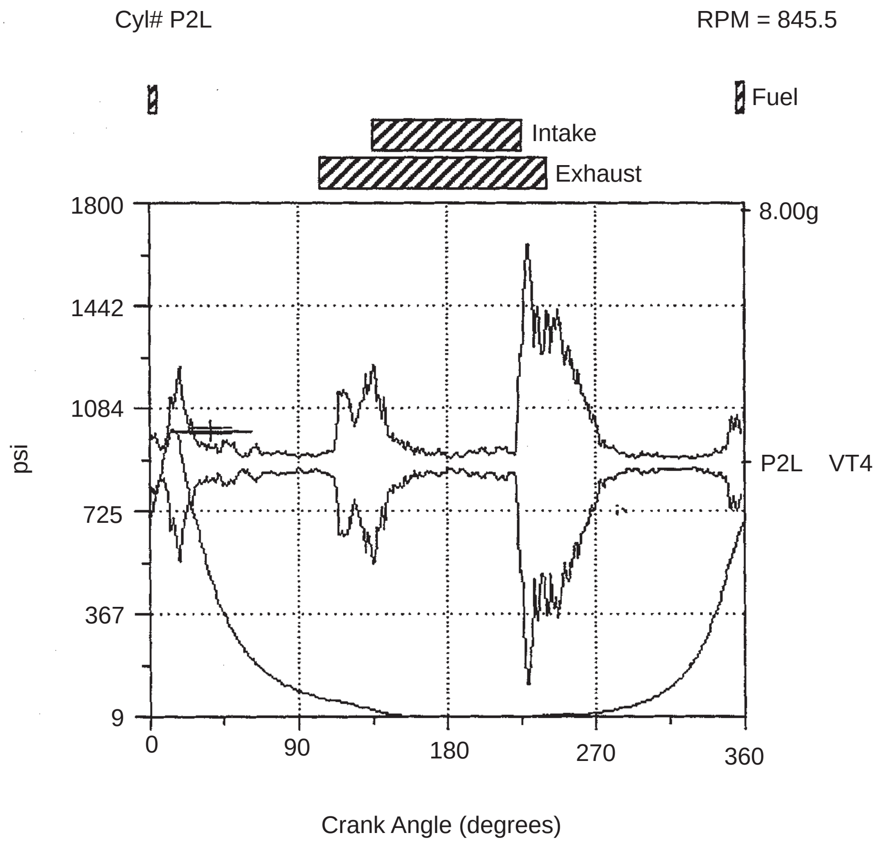

Figure 8-11
Pressure and Vibration Plots for Cylinder with Port Bridge Problems

8.10 Improper Valve Seating

The cylinder P9L at intake valve closure has a pattern typical of that occurring when a valve fails to seat properly, which can be seen in Figure 8-12. There is a leak past the valve for some degrees until the pressure inside the cylinder forces the valve to seat. This condition can exist because of a worn guide, a bent valve, or a damaged seat.

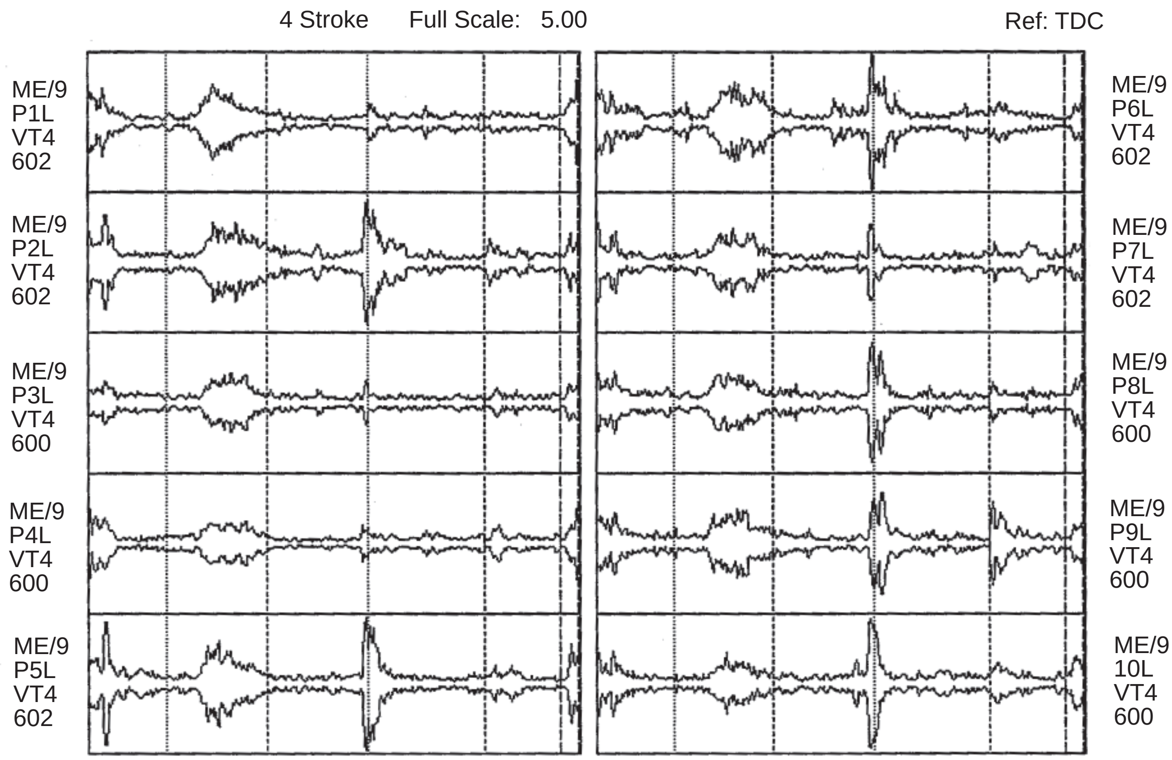

Figure 8-12
Vibration Pattern with Improper Valve Seating

8.11 Bent Connecting Rod

The plot of the P-θ curves for this engine shows good balance between cylinders (see Figure 8-13) except for P6L (see Figure 8-14), which is low by a significant amount. Another important observation is that all cylinders are firing consistently, including P6L, as shown by the tight vertical dimensions of the statistical icons.

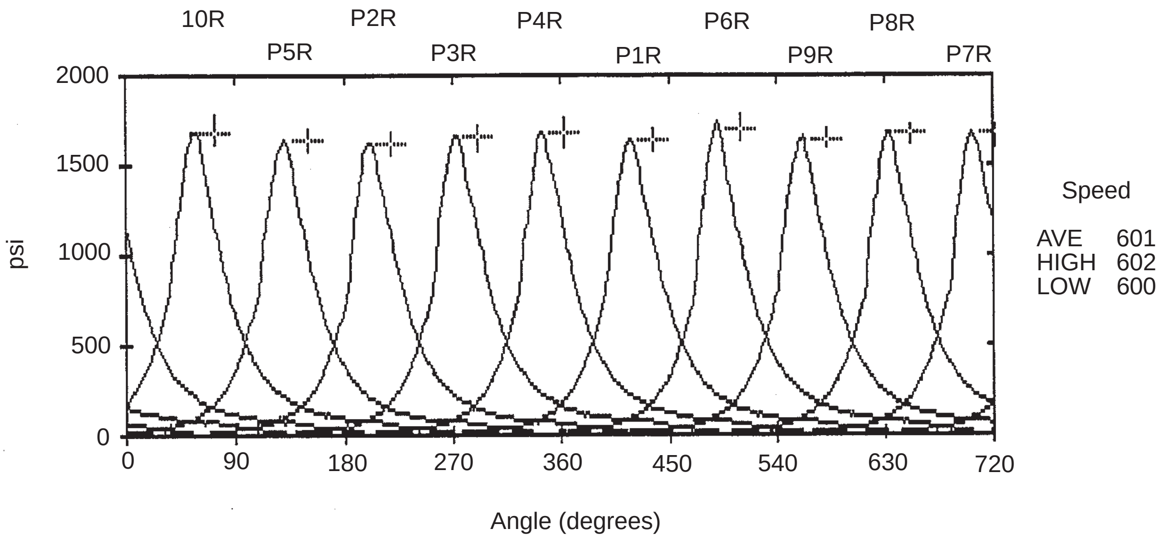

Figure 8-13
Pressure Curves for Right Bank Cylinders

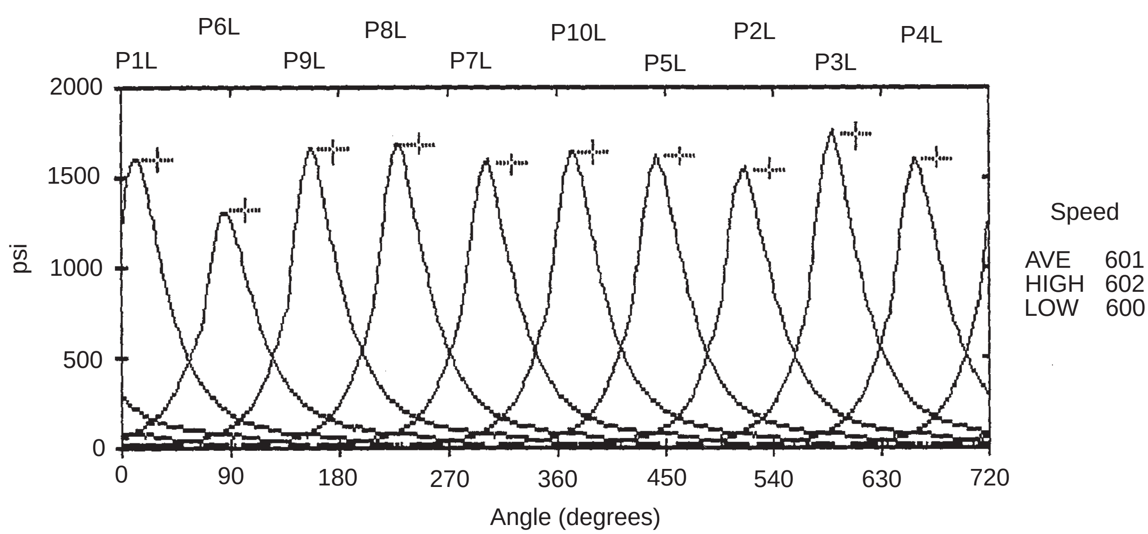

Figure 8-14
Pressure Curves for Left Bank Cylinders

Table 8-2 shows that P6L firing pressure is 317 psi below engine average. Another important piece of information is that the P6L compression reference pressure is 465 psi, compared to about 560 psi in other cylinders. The immediate cause of the low firing pressure is clearly the low compression pressure.

Low compression pressure means that the minimum combustion chamber volume in this cylinder is too large or that a compression leak is occurring. The vibration patterns show no evidence of leak. The increased combustion chamber volume could result from an incorrect piston installed, a worn wrist pin, or a bent connecting rod. Inspection revealed the latter.

Table 8-2
Combustion Report

DIESEL GENERATOR

Unit Name: Model: KSV-20T

Location: Unit Mfr: COOPER-BESSEMER Serial No:

Stroke: 4 Marker Correction Angle: .0 degrees Engine runs clockwise and is articulated

Cyl	Speed	MEP	IHP	Comp Ref Press	Comb Start	Max Rise Rate	Peak Firing Pressures AVE	DEV	MAX	MIN	DELTA	PFP Angle	Term Ref Press	Exhaust Ref Press
	rpm	psi	ihp	@ 703 deg	deg ATDC	psi/deg	psi	psi	psi	psi	psi	ATDC	psi	psi
1L	602	245	455	597 HI		70	1609	10	1663	1549	-29	10	125	54
2L	600	238	441	591		72	1547	9	1600	1502	-91	10	122	54
3L	600	270	500	562		64	1741	11	1804	1676	103	10	131	56
4L	600	253	468	582		64	1611	8	1668	1562	-27	10	125	55
5L	600	256	473	581		64	1626	8	1678	1588	-12	10	132	56
6L	602	249	462	465 LO	-6 HI	58 LO	1321	12	1392	1262	-317	14	134	62 HI
7L	600	254	470	546		60	1585	10	1638	1536	-53	12	128	56
8L	600	264	489	575		64	1695	9	1759	1643	57	10	127	53
9L	600	264	488	575		70	1660	10	1709	1598	22	12	134	58
10L	600	259	481	585		68	1650	10	1717	1581	12	10	132	55
1R	600	262	471	574		68	1647	9	1702	1594	9	12	129	55
2R	600	246	442	556		73	1633	10	1699	1567	-5	10	119	53
3R	602	258	463	586		75 LO	1675	11	1736	1597	37	10	127	57
4R	602	259	463	570		67	1693	12	1768	1617	55	10	129	55
5R	602	247	444	580		66	1642	10	1709	1595	4	12	118 LO	51
6R	602	283	508	536		73	1710	14	1789	1645	72	12	135 HI	58
7R	602	260	466	580		74	1680	10	1738	1609	42	10	122	53
8R	602	261	468	562		67	1689	9	1738	1637	51	10	126	58
9R	600	268	479	561		60	1649	9	1709	1603	11	12	129	57
10R	602	271	487	560	-34 LO	68	1698	14	1790	1632	60	12	127	50 LO
Eng:	601	258	9419	566	-20	67	1638	10			53	11	128	55
Sprd:	0%	17%	14%	23%	28 deg	25%	26%	59%				4 deg	13%	22%

NOTES:

1. The peak firing pressure (PFP) is the highest pressure in the cylinder during the cycle.
2. PFP statistics are based on up to 64 cycles. Ignition statistics are based on 20 cycles.
3. DEV means the average deviation of each cycle from the cylinder average. It is used to determine how consistently a cylinder fires.
4. DELTA is the difference between the cylinder and engine average PFP and indicates cylinder balance.
5. PFP Angle is the angle ATDC at which the PFP occurred.

8.12 Burned Exhaust Valve

Leakage past valves or rings shows up in vibration patterns as a rounded, gas flow-type pattern (see Figure 8-15). The amplitude will be related to the pressure in the cylinder. Injection and combustion occurring near TDC tend to obscure any such leakage pattern. Therefore, the best place to look is before the start of injection. In the case of Figure 8-15, the vibration energy can be seen starting to increase well before TDC as compression pressure increases. This was caused by an exhaust valve leak.

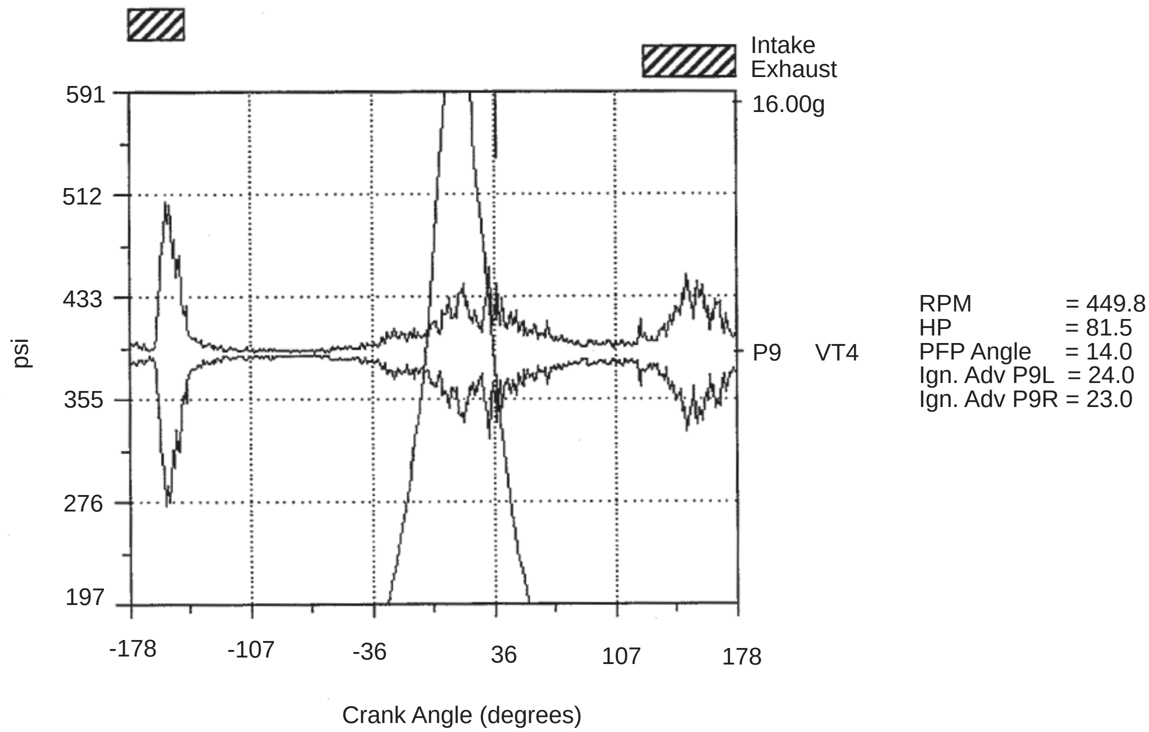

Figure 8-15
Burned Exhaust Valve

8.13 Typical Alco 251 Cylinder Pattern

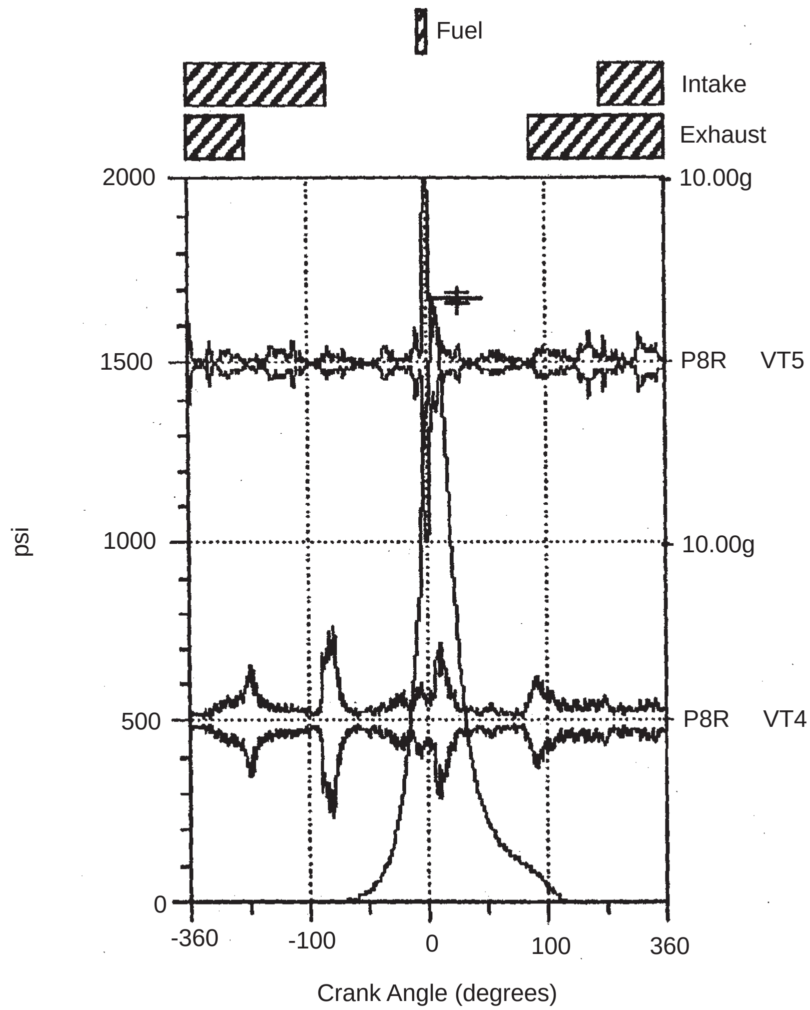

Figure 8-16
Typical ALCO 251 Cylinder Pattern

8.14 Typical Cooper-Bessemer KSV Cylinder Pattern

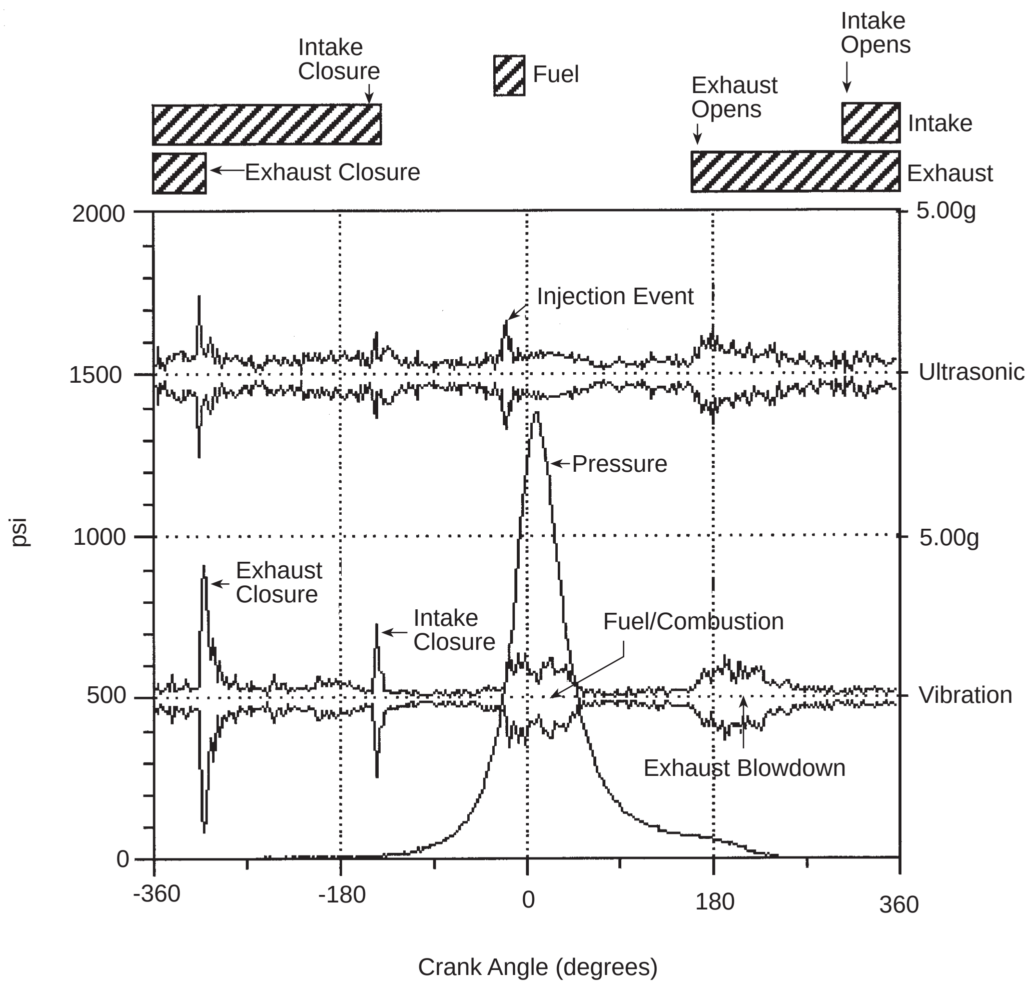

Figure 8-17
Typical Cooper-Bessemer KSV Cylinder Pattern

8.15 Typical EMD 645 Cylinder Pattern

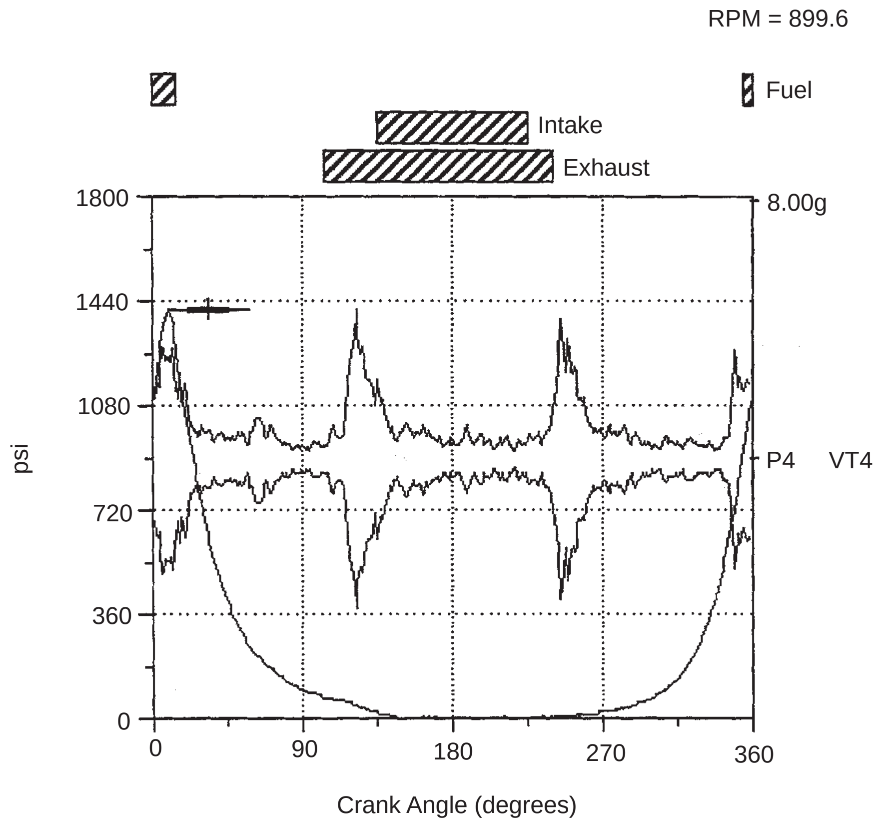

Figure 8-18
Typical EMD 645 Cylinder Pattern

8.16 Typical FMOP 38TD 8-1/8: 12 Cylinder Pattern

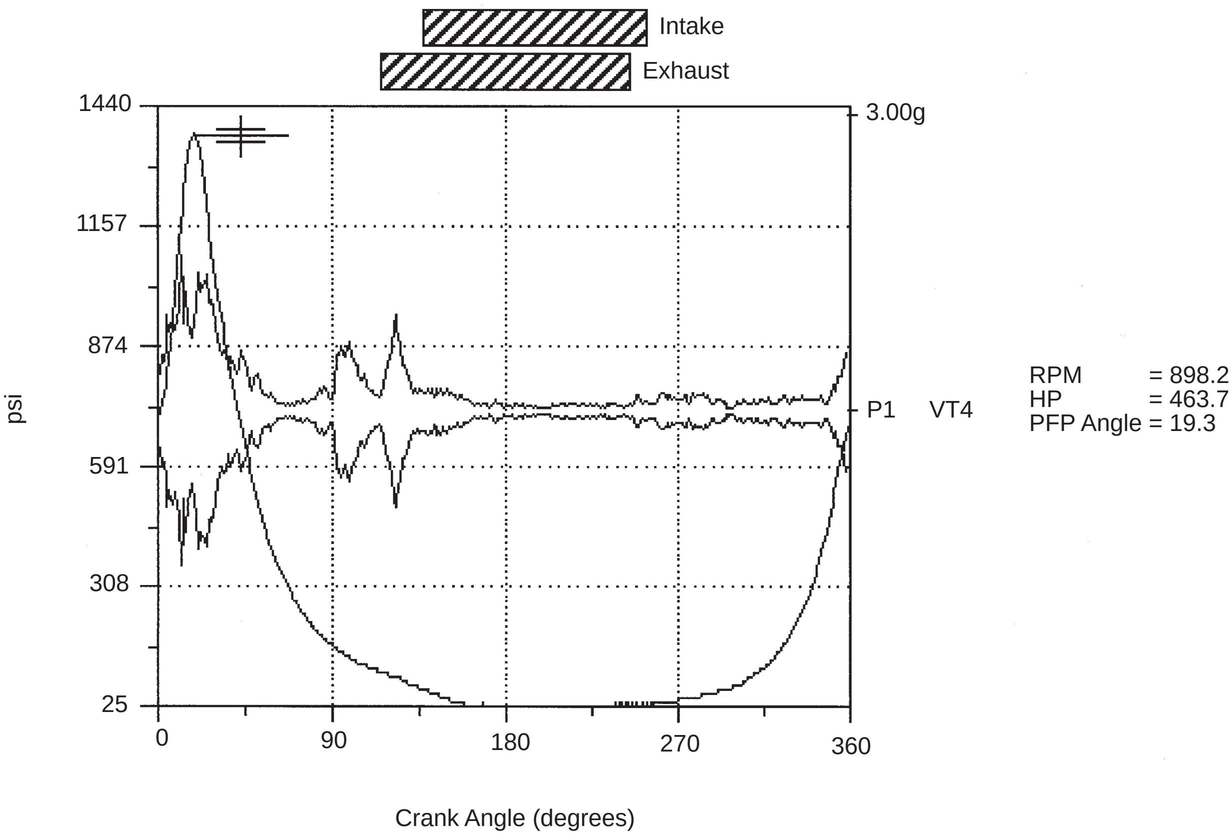

Figure 8-19
Typical FMOP 38TD 8-1/8 Cylinder Pattern

8.17 Typical Enterprise DSRV Cylinder

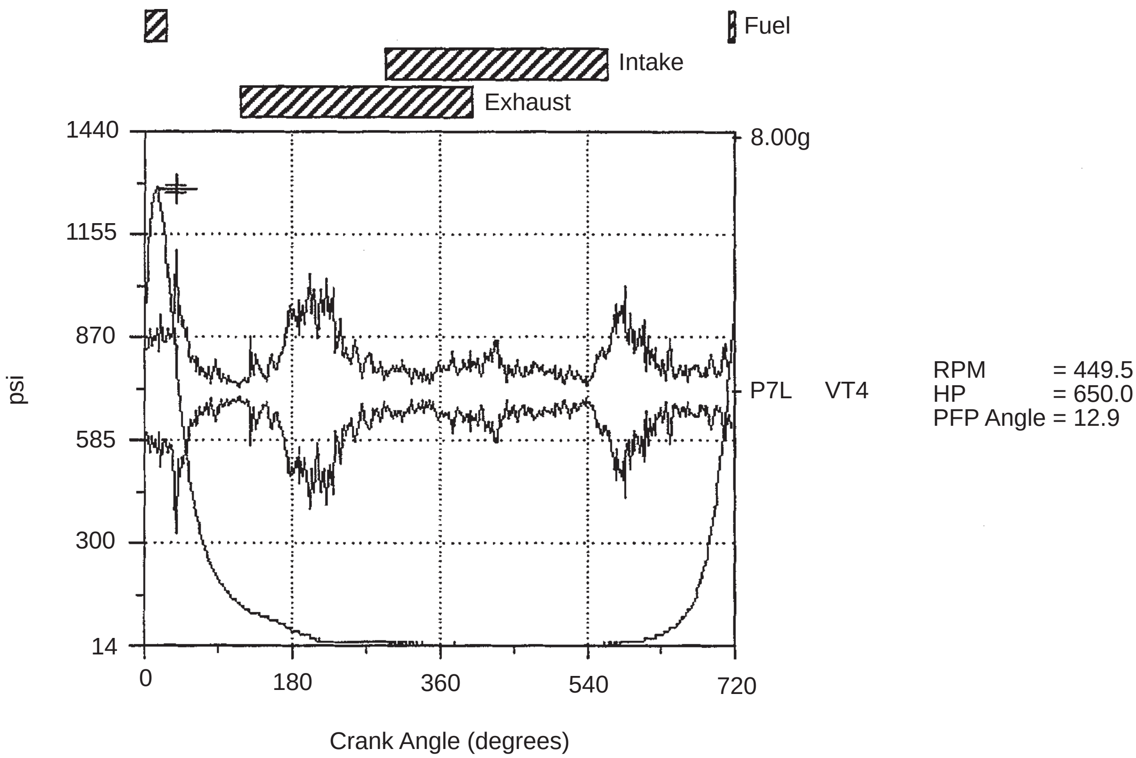

Figure 8-20
Typical Enterprise DSR 6-4 Cylinder Pattern

8.18 Typical Colt Pielsick PC2 Cylinder Pattern

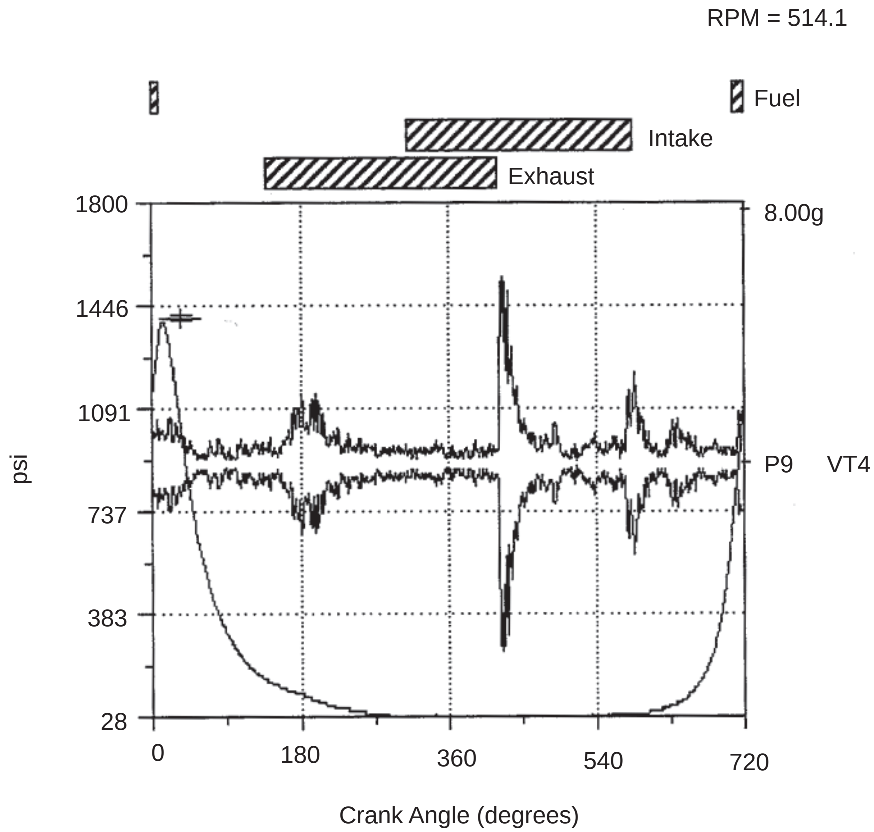

Figure 8-21
Typical Colt Pielstick PC2 Cylinder Pattern

9

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