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A STUDY OF FUEL ECONOMY AND EMISSION REDUCTION METHODS FOR MARINE AND LOCOMOTIVE DIESEL ENGINES

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PREFACE

This work was performed under the direction of the Transportation Systems Center of the Department of Transportation. Technical Contract Monitor for TSC was R. L. Mason. Technical assistance to the program was provided by R. A. Walter, Staff Member of TSC, and Lt. Cdr. J. R. Sherrard of the Coast Guard Office of Research and Development, and their assistance is gratefully acknowledged.

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- General Electric Diesel Engine Products Department, General Electric Company
- Missouri Pacific Railroad Company
- Southern Pacific Transportation Company

and the Commanding Officers and Engineering Staff Members of the Coast Guard cutters Chase, Decisive, and Hamilton.

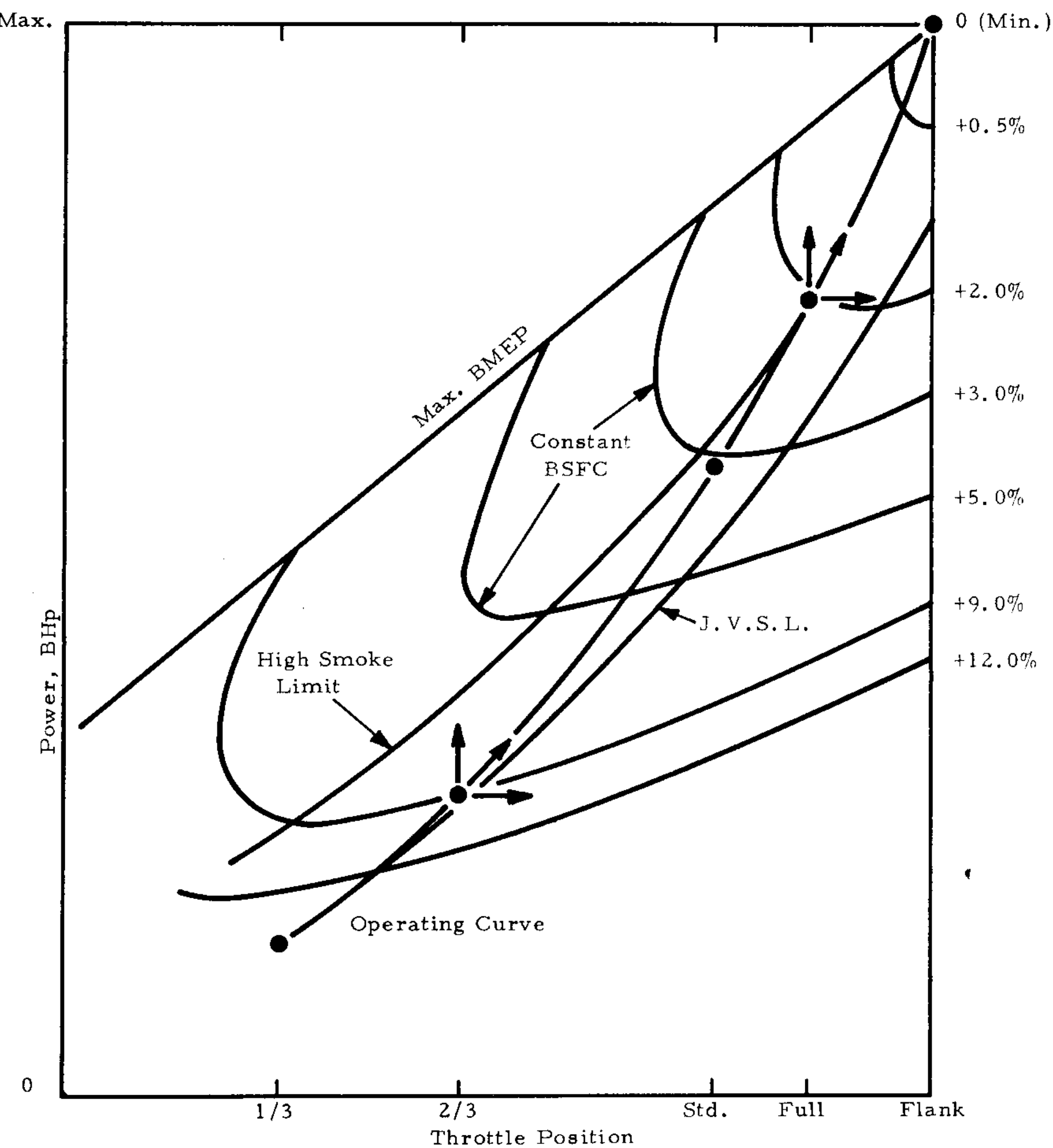

FIGURE 5.10 TYPICAL FUEL CONSUMPTION MAP--TWO-STROKE CYCLE TURBOCHARGED MARINE ENGINE

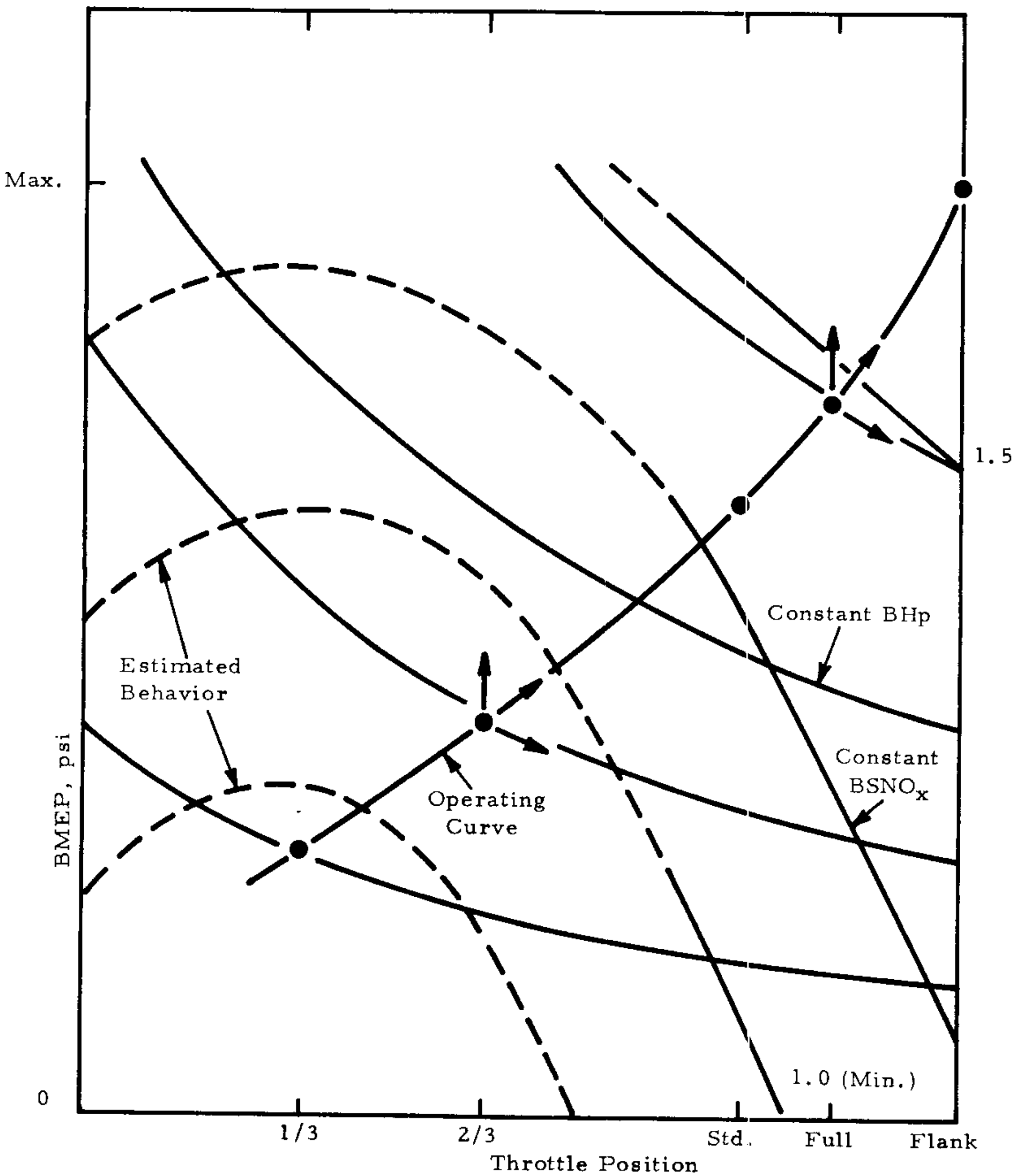

FIGURE 5.11 TYPICAL BRAKE SPECIFIC NO_x MAP--TWO-STROKE CYCLE TURBOCHARGED MARINE ENGINE

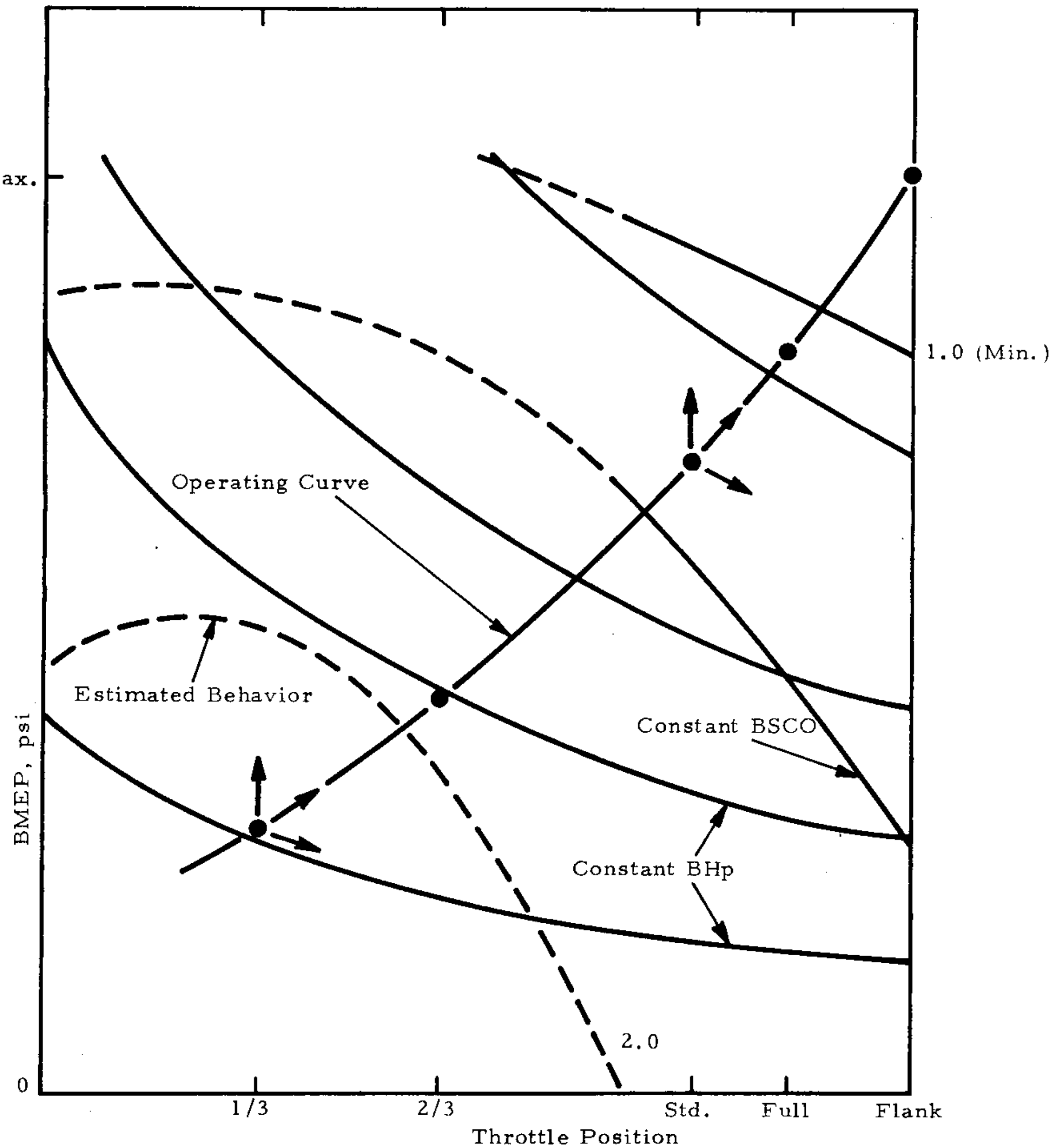

FIGURE 5.12 TYPICAL BRAKE SPECIFIC CO MAP--TWO-STROKE CYCLE TURBOCHARGED MARINE ENGINE

(It should be recalled that these engines operate at variable speed and load in this application and that the propeller pitch is constant.) The fuel map is shown in Figure 5.13.

It can be seen that the general shape of the map for the blown engine is very different from that of a turbocharged engine. Note that the area of minimum BSFC is in the upper left portion of the map (near the intersection of the maximum BMEP lines) and not at rated speed and load. This fact, together with the lack of propeller pitch control, makes it difficult to achieve reduced fuel consumption by operating mode changes. That is, the operating points cannot be moved upward along vertical constant engine speed lines, since propeller pitch cannot be adjusted to change the engine power requirement for a given speed. This lack of pitch control also prevents movement of the operating points in a horizontal manner, i.e., along constant power-variable engine speed lines. This latter circumstance is unfortunate, since the map reveals that such a change in operating modes could result in lower BSFC in the middle throttle positions.

The lack of flexibility in mode changes for engines in WAGB vessels caused us to halt this particular investigation; therefore, emissions maps were not developed for these engines.

5.3 Summary

A brief recapitulation of the main points of this analysis of changes in engine operating modes is in order.

First, it is obvious that the most direct way to improve fuel economy for line-haul locomotives is to reduce fuel consumption at rated speed and load (Notch 8) and at idle, since these two conditions account for about 60 to 70 percent of the operating time of these engines, and about 60 percent of their total fuel consumption. However, Notch 8 BSFC is a fixed quantity (insofar as mode changes are concerned) since it is not prudent to increase rated speed and BMEP above manufacturer-specified levels. Fuel consumption can be reduced at idle only by reducing engine speed, and this possibility is being investigated by EMD (see Section 3.2.7).

Therefore, one is restricted to considering mode changes for Notches 1 through 7, which are weighted about 30 to 40 percent of total operating time and account for about 40 percent of the fuel consumed by a line-haul locomotive. The derived fuel maps for both

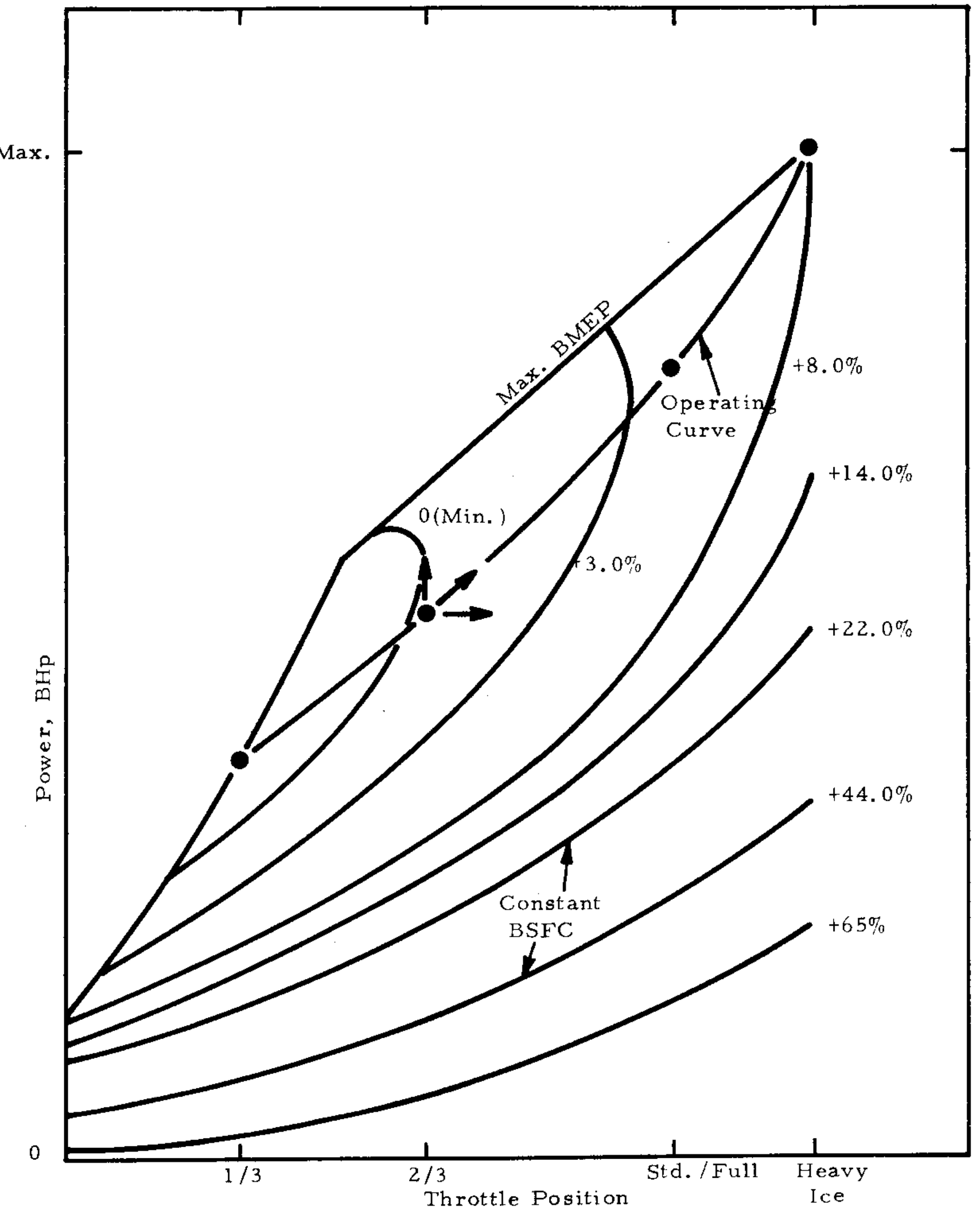

FIGURE 5.13 TYPICAL FUEL CONSUMPTION MAP--TWO-STROKE CYCLE BLOWN (NON-TURBOCHARGED) MARINE ENGINE

two-stroke and four-stroke locomotive engines indicate that the only way to achieve any significant reduction in BSFC for these notches is to maintain current engine speeds and to increase BHp. This action would be most effective in lower throttle positions, where line-haul locomotives spend only a small part of their operating time. In view of the small weight attached to these notches and the probable drastic change in low-speed flexibility that would result, it appears that any slight improvement in cycle composite BSFC would not offset the problems involved in this change. Increased smoke opacity would also likely result if this change was implemented.

Coast Guard engines, on the other hand, are apparently more likely to respond to changes in operating modes with a decrease in cycle composite BSFC. We say "apparently" because questions remain about the changes in propeller efficiency that could accompany the changes in engine speed. However, the indication that these engines spend little time at rated speed suggests that any improvement in part-load BSFC would have a favorable impact on cycle composite fuel consumption. Improvement in fuel consumption at idle is also potentially important for these engines.

6. EFFECT OF ENGINE WEAR ON PERFORMANCE AND EMISSIONS

An extensive search of the technical literature failed to find any results, either theoretical or experimental, that established a correlation between diesel engine wear and performance or emissions levels. Therefore, it was decided to develop a theoretical approach to relate changes in wear-sensitive performance parameters such as fuel/air ratio, compression/firing pressure and BMEP to corresponding changes in engine power, BSFC and brake specific emissions. The four-stroke cycle turbocharged engine was used as the subject of this theoretical study, but only a shortage of time prevented a similar investigation of the two-stroke engine.

One type of engine wear that has a direct effect on performance and emissions is that which results in leakage of compressed air charge and combustion gases from the cylinder, i.e., wear of piston rings, liner, and valves. Malfunctioning injectors perhaps have as much influence on performance and emissions (particularly unburned hydrocarbons and smoke) of the diesel engine; however, the existing cause-and-effect relationships are almost impossible to express in a quantitative manner, even with a broad theoretical approach as is employed here. On the other hand, cylinder leakage causes an increase in fuel/air ratio during the compression stroke and a decrease in cylinder pressure during the expansion stroke. Together, these changes can affect the concentration of emission constituents (especially NO_x and CO) and the effective power output from each affected cylinder. Since fuel consumption rate remains at its nominal value for each operating condition, BSFC can only increase; however, brake specific NO_x and CO could go up or down depending on the relative changes in concentration and BHp.

The method used to calculate the magnitude of these changes was based on the following simplifying assumptions: (1) leakage during the compression stroke resulted in an increase in fuel/air ratio that, in turn, produced a change in NO_x and CO concentration and BHp, (2) leakage during the expansion stroke resulted in loss of BHp, but had no effect on emission concentrations, and (3) the power losses in (1) and (2) were additive. Use of the first assumption allowed changes in concentration and power to be calculated according to the procedure outlined in Section 4. However, calculation of the power loss in (2) involved a somewhat lengthy procedure in which a quantitative relation was established between a certain percentage loss of the total mass in the cylinder and the resulting change in power. This procedure is presented in Appendix A in order to avoid interrupting the current discussion.

Before these methods could be applied to the problem, it was necessary to determine what constituted a reasonable amount of cylinder leakage for an engine in an advanced state of wear. The literature search produced estimates that a loss of approximately two or three percent of the total fuel/air mass represented highly excessive blowby into the engine crankcase. Data provided by other engine laboratories at Southwest Research Institute confirmed this figure in the case of a turbocharged, high-BMEP, four-stroke diesel truck engine. It was decided to use a mass leakage figure of at least twice this magnitude in order to present a worst-case condition that might include a similar amount of leakage from worn valves. Because of the nature of the analytical method that was employed and the underlying assumptions (see Appendix A), it developed that the mass loss associated with this worst-case condition was about 7.5 percent of the total mass in the cylinder during one cycle. This figure was comprised of a 2.5 percent loss during the compression stroke and a 5 percent loss during the expansion, or power, stroke.

The method outlined in the preceding paragraphs and in the appendix was used to calculate the percent change in BSFC, $BSNO_x$, and BSCO associated with this amount of cylinder leakage for low, medium, and rated engine speeds (corresponding to Notches 1, 4, and 8). The results are shown in Table 6.1, for the case where all cylinders were so affected and where only one cylinder was affected. Note first that the latter case can be dismissed at once since there is little or no effect on the total engine brake specific quantities. However, _if_ all cylinders had the same degree of wear (not an impossible circumstance for an engine that was long overdue for an overhaul) a significant increase in BSFC (caused by a decrease in BHp) is indicated, while $BSNO_x$ is relatively unchanged. Brake specific CO varies considerably either side of the baseline value due to the fact that CO concentration is highly dependent on fuel/air ratio (see Figure 4.3). The effect of more than one worn cylinder can be figured by a linear interpolation between the two sets of percentages given here.

Two points must be mentioned in regard to the results of this section. First, it is our opinion, based on interviews of various engine users, that many in-service engines would not be allowed to reach a state of wear represented by the cylinder leakage used in these calculations; rather, the engine would first undergo either regular preventive maintenance overhaul or an overhaul based on need as determined by observation of diagnostic parameters. The discussion of maintenance practices in the following section will expand on this statement. Second, the manner in which engine horsepower and emissions vary here with changes in certain parameters (such as fuel/air ratio) is determined solely by the assumptions that have been made in the absence of hard facts and data for these specific engines. Manufacturers of locomotive engines have begun studies to determine the behavior of performance and emissions as functions of operating time on a long term basis. However, even these investigations probably will not shed much light on the quantitative relationship between an engine's state of wear and these parameters.

TABLE 6.1 EFFECT OF CYLINDER LEAKAGE ON FUEL CONSUMPTION AND EMISSIONS

Throttle Position	% Change from Baseline Values		
	BSFC	$BSNO_x$	BSCO
All Cylinders Affected			
N1	+4	+2	-18
N4	+4	-3	+30
N8	+3	-3	+ 7
One Cylinder Affected			
N1	Nil	Nil	- 2
N4	Nil	Nil	+ 2
N8	Nil	Nil	Nil

7. EFFECT OF ENGINE MAINTENANCE ON PERFORMANCE AND EMISSIONS

An attempt was made in this program to establish a relation between the type of maintenance practiced by engine users and the resulting effects on in-service engine thermal efficiency and emissions. This study relied on a literature search, interviews of engine manufacturers and users, and a study of maintenance instructions issued by both types of companies. Not surprisingly, no relation was found that allowed direct correlation in a quantitative, statistical sense; however, effort expended in this area did produce qualitative information concerning the engine components that exert the most influence on performance and emissions and the nature of the maintenance they receive under various programs. These components are generally those that were mentioned in Section 3 as possible fuel consumption/emission reduction items. Whether of old or new design, however, these components and their maintenance are crucial to efficient operation of the engine.

The discussion presented here will first concentrate on general topics and considerations common to most of the maintenance programs that were studied, then take up specific items such as maintenance of the important components mentioned above. Methods and goals of both railroad and Coast Guard programs will be outlined, but it must be emphasized that interviews were conducted with only two railroads and three Coast Guard vessels; therefore, results presented here are based on a small sample of both types of maintenance organizations. This section ends with a discussion of a potential (rather than an actual) maintenance problem for some Coast Guard engines.

7.1 General Remarks

A study of maintenance instructions of manufacturers and users reveals that all programs employ procedures that are performed on a time-based schedule and observation of performance parameters on a more or less continuous basis. The rationale behind this type of approach is clear: the schedule of procedures has been laid out according to the experience gained by observing many engines in similar applications, while close monitoring of parameters such as exhaust temperatures, turbocharger pressure, and crankcase vacuum minimizes the possibility that a sudden deterioration in the engine will go undetected. Marine engines, which can, for all practical purposes, be considered stationary engines, usually are equipped with transducers needed to monitor these parameters, but locomotive engines are usually not so equipped.

The time-based schedule of maintenance items usually begins with relatively simple checks and inspections that are to be made on a short time scale such as daily or, in the case of locomotive engines, at the start of every trip out of the yard. Other, more detailed inspections and maintenance work are done weekly, monthly, biannually, annually, and so forth. Typical schedules for marine and locomotive engines are shown in Table 7.1. The information given in this table does not imply that the two types of engines are literally on the same maintenance schedule; Procedure No. 3, for example, is not the same for both engines. Both schedules do, however, have a factor in common: the schedules are cumulative in nature; that is, all work items included in Procedures 2, 3, and 4 are also performed when Procedure 5 is done, items included in No. 5, 6, and 7 are included in Procedure 8 and so forth.

TABLE 7.1 TYPICAL TIME-BASED ENGINE MAINTENANCE SCHEDULES

Schedule Time Scale	
Marine Engine	Locomotive Engine
Hourly	Trip
1 day	30 days
1 week	90 days
1 month	180 days
3 months	Annual
6 months	2 years
Annual	3 years
3 years	4 years
-	5 years
-	6 years

These procedures may be exactly as stated in the manufacturers manual, but it is more likely that the engine user will modify these recommendations to suit his particular situation. This modifying process may involve an improvement or degradation of any or all of the factory-recommended procedures, depending on constraints of time and/or money felt by the user. In general, the best maintenance programs are in an almost constant state of review in order to continuously update the cost-effectiveness ratio of the various procedures in the program and to determine which procedures are in need of modification.

An important part of the recommended maintenance programs for these engines is periodic analysis of the spectrochemical and physical properties of the lube oil. These two types of analysis are used to detect the presence and concentration of various wear metals, coolant, and fuel in the oil. Each test is a valuable tool for determining the condition of rings, liners, injectors, bearings and seals, parts that cannot be readily inspected without disassembling a considerable part of the engine. To obtain maximum benefit from an oil analysis program, it is necessary that samples be taken in the same (correct) manner each time; i.e., at the same engine location and operating time since the last sample was obtained.

The results of spectrochemical analysis are usually interpreted on a relative basis (that is, a percent change from the established baseline values) over a period of, say, several thousand hours of operating time. However, some maintenance programs initiate investigative action on the basis of a single test result that approaches predetermined condemning concentration limits of wear metals, fuel, and so forth. Such limits are difficult to set and require long experience and a broad data base in order to accurately reflect the state of engine wear. However, it is not an impossible task, and several railroads have successfully done so.

Interviews conducted in the course of the project revealed strong support for this analytical technique among engine railroads and manufacturers and less support among members of Coast Guard engineering staffs. (However, the Coast Guard Engineering Manual (CG-413) places strong emphasis on the use of spectrochemical analysis.) This lack of support from Coast Guard engineering personnel is apparently the result of a basic misunderstanding of the manner in which the results are to be interpreted. One staff member related how he had sent a sample of unused hydraulic fluid to be analyzed and that the test result indicated that his engine's rings were worn out. Therefore, he rejected the result as erroneous and misleading and began to doubt the efficiency of the technique. The mistake made in this case (if it is true) is that the test result was presented and interpreted in an absolute sense without the data base required to do so. (Also, the analytical laboratory showed a degree of incompetence and/or indifference by accepting and testing a sample of hydraulic fluid.) As mentioned previously, if results are plotted on a relative basis over a sufficient length of engine operating time, a realistic picture of the internal condition of the engine will emerge.

This discussion of spectrochemical oil analysis is presented here because it bears at least indirectly on the relation between maintenance and engine wear, performance, and emissions, and because it illustrates that the worthwhileness of correct maintenance is often not understood or believed by the man doing the actual work; hence, he must be convinced on this point by those persons responsible for formulating and implementing maintenance policy. There is no question that some diesel engines (even of the same make and model) perform better than others. However, there is also no question that every engine can be made to operate with minimum BSFC and smoke by proper maintenance.

It should also be pointed out that even though analysis of lube oil is (or can be) a valuable diagnostic tool, there is likely to be a lag of at least several days between the time the oil sample is taken and the result is available for interpretation. This problem is, of course, most severe for Coast Guard operation in which vessels are out of port for perhaps several weeks at a time. It is true that samples are taken from the engines even when a vessel is at sea and the analysis done later when the vessel returns to port. However, the delay in obtaining results could be significant. This discussion is not put forth as idle criticism of a situation that is beyond control of the Coast Guard, but rather to point out the need for other, virtually continuous diagnostic observations to supplement the oil analysis program.

7.2 Maintenance of Components

From the viewpoint of obtaining minimum levels of fuel consumption and emissions (especially smoke), it is most important to properly maintain those components that are directly involved in the combustion process; i.e., the introduction and burning of proper amounts of fuel and air in the cylinder. It is true that excessively worn rings, liners, bearings, and so forth will lead to a catastrophic situation which will prove to be the ultimate influence on performance and emissions (both will cease), but for the average in-service engine, the maintenance of injection pumps and/or injectors, turbocharger or blower, governor and air filters is more germain to the problem of achieving best overall performance.

The most important components are those that influence all cylinders in an engine rather than one. Therefore, an improperly adjusted governor or an inefficient turbocharger is much more critical to engine performance than is one malfunctioning injector or one liner with clogged air ports. Indeed, a program[31] conducted at Southwest

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[31] Investigation of Exhaust Gas Parameters for Automatic Diagnosis of Diesel Engine Fuel System Faults for Use with the Programmable Diagnostic Unit for Automatic Test Equipment for Internal Combustion Engine Powered Material (ATE/ICEPM), Technical Report 11932, U.S.A. TACOM, December 1974.

Research for the U.S. Army Tank Automotive Command (TACOM) dealt with the problem of diagnosing single injector malfunctions in a two-stroke six-cylinder engine by measurement of exhaust emissions, both gaseous and visible. The conclusion was that only the very worst malfunctions, such as an entire nozzle tip blown off (a possible but unlikely happening), could be consistently diagnosed by this method. Other, less severe malfunctions, such as two or four clogged tip orifices (out of a total of six) or a worn barrel and plunger assembly, could be spotted only because extensive tests had been conducted with the engine in perfect operating condition and the resulting baseline values were precisely known.

With these ideas in mind, we may proceed to look more closely at the maintenance of the more important components.

7.2.1 Governor

The effect on smoke density resulting from overfueling by the governor during engine acceleration has been mentioned in Section 3.2. While this type of governor action produces a highly visible smoke plume, there is actually little or no adverse effects on either fuel consumption or gaseous emission levels during this time. However, a governor that overfuels under steady-state conditions certainly causes increases in fuel consumption, smoke, and CO, as well as perhaps doing harm to the engine itself. One railroad maintenance officer interviewed in this program furnished a list of locomotives cited for state smoke limit violations (albeit, by the use of the highly subjective and unreliable Ringlemann method), and the corrective repairs made in each instance. It is judged significant that, while from one to all injectors were usually replaced, all but one case involved replacement of the governor.

This fact, together with the previously-cited diagnostic experience with single injector malfunctions in a two-stroke engine, indicates that these steady-state smoke problems were the result of overfueling by the governor. This particular company now calibrates governors annually and reports excellent results from this procedure. Most engine manufacturers recommend longer periods than this for governor adjustment; however, as with all maintenance procedures, the recommendation must be tempered by the judgment of the engine user. Any significant increase in steady-state smoke density should be viewed as the possible result of a governor problem and diagnostic action undertaken.

7.2.2 Turbocharger

The performance of this item also influences the fuel/air ratio to all cylinders. Turbochargers are subject to a gradual degradation in performance due to the buildup of a mixture of oil and dirt or dust on the impeller. They can, of course, fail in a sudden and catastrophic manner, but this is an obvious problem with an obvious solution. A gradual decrease in boost pressure over a period of time can be readily determined by frequent monitoring of this parameter. However, as mentioned before, locomotive engines are usually not equipped with pressure transducers. Instead, railroad maintenance practice involves either replacement of the turbo on a strict time or mileage basis or permitting the turbocharger to run until failure occurs. Marine engines, being generally equipped with boost pressure transducers and gauges, are in a position to avoid either premature replacement of the turbocharger or the risk of sudden failure. Typical recommended period of performance before overhaul or changeout is about three or four years. Inspection and cleaning of the turbocharger occurs periodically during its lifetime. An English railroad reports(32) the practice of fitting special spray nozzles to each turbocharger in order to inject a mixture of water and paraffin into the impeller while the engine is running. This action is taken every 600 to 800 hours of operating time and is said to produce a reduction in both smoke density and exhaust temperature. To our knowledge, no engine user in the United States uses this practice.

7.2.3 Intake Air Filter

Engine manufacturers and users alike are aware that air filter performance is highly dependent on the operating environment of the particular engine. Schedules for inspection and cleaning or replacement (if needed) range from 30 to 90 days. At least one user prefers filters that are slightly less efficient, but become restricted in a slow, progressive manner, to filters that are highly efficient and tend to become suddenly restricted. All engines apparently are equipped with transducers (or, at least, fittings where they can be connected) to indicate when the pressure drop across the filter becomes excessive. It seems likely that filters on locomotive engines undergo more severe service than those on ship power plants since the latter operate in a relatively dust-free environment.

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[32] Petrook, W., and W. A. Stewart, Critical Factors in the Application of Diesel Engines to Rail Traction, Proceedings of the Institution of Mechanical Engineers (England), 1969-70.

7.2.4 Fuel Injectors

As mentioned previously, a small number of injector malfunctions are not likely to result in a measurable change in engine performance or emissions, except in the case of certain extreme malfunctions that allow a large, uncontrolled flow of fuel into the cylinders. However, every injector is subject to a slow degradation in performance with operating time due to less severe problems such as carbon buildup on the spray tip, decreasing injection (crack) pressure, and leakage of fuel past a faulty needle valve. If the performance of several (or all) injectors is permitted to degrade significantly, then engine fuel consumption and emission characteristics will be affected accordingly.

In the railroad maintenance programs that were investigated, injector maintenance and replacement was approached in various ways. One program calls for replacement with either new or rebuilt units on a strict time schedule (every two years) or whenever the engine power assemblies are removed. Another program calls for annual injector change for one type of engine and a change whenever removal of power assemblies is necessary for another type of engine. Coast Guard maintenance apparently involves following the engine manufacturers' time-based maintenance schedules except when an obvious injector malfunction occurs. Of course, engines aboard these vessels are never "out of sight" of the engineering crew, so close scrutiny of various performance parameters is possible at all times. One vessel uses exhaust temperature of individual cylinders to determine when injector replacement is needed. This approach, which avoids unnecessary maintenance, is valuable provided the relation between changes in injector performance and exhaust temperature is accurately known.

There are other aspects of injector maintenance that deserve mention. Injection timing should be maintained at the recommended setting for minimum BSFC and smoke opacity, and all manufacturer and user programs call for timing checks and adjustments between injector replacements. Balancing of maximum fuel output per cycle at full rack position is also recommended in order to obtain a "matched" set of injectors, i.e., one that will equalize firing pressure among all cylinders. This equalization permits the engine to develop maximum BHp with minimum BSFC and mechanical and thermal stress. One railroad goes a step further by calibrating fuel delivery of EMD unit injectors at idle rack position as well as at full load. The objective of this procedure is to obtain a two-point check of an injector's fuel delivery curve and, therefore, to determine that fuel output is correct at all rack positions between these two extremes. Injectors that do not meet this two-point check are scrapped.

7.2.5 Internal Engine Components

It has been stated in Section 3, that the design of air inlet ports in cylinder liners of two-stroke cycle engines has a direct influence on combustion efficiency; similarly, if these ports become clogged with deposits the quantity and swirl direction of the incoming air charge can be altered to the extent that combustion efficiency is significantly reduced. The problem is aggravated by the prolonged periods of idling that many of these engines are subjected to (see Section 2). All maintenance programs for the two-stroke engines studied in this project provided for periodic inspection and, if necessary, cleaning of these ports. This is a relatively simple procedure that can be done without disassembling the engine.

In Section 6, it was stated that piston ring and liner wear were not significant factors in the fuel consumption and emissions levels of most of these large diesel engines because the users did not allow wear to reach that point. Most users follow manufacturers' recommendation on wear limits, though some users actually use more stringent limits in order to minimize the probability of ring breakage due to increased clearances. Indeed, in all maintenance programs the prime consideration is to avoid excessive clearances that can lead to catastrophic mechanical failure. No instances were found in which an engine that was "worn out" (by the user's criteria) was suffering from excessive fuel consumption or smoke production, provided that items such as governor, turbocharger, and injectors were still functioning at nominal performance levels.

It is a simple matter to check ring side clearance through the air ports in the cylinder liners of two-stroke engines. However, a clearance check is possible for a four-stroke engine only when the power assemblies are removed. One railroad first relies on results of the spectrochemical oil analysis to indicate when a four-stroke engine might be experiencing a high rate of ring and/or liner wear, then uses an optical device to inspect interiors of the power assemblies for visual confirmation of excessive liner wear. This somewhat indirect approach is said to work well, and again illustrates the value and reliability of a well-run lube oil analysis program.

7.3 Scheduled Versus As-needed Maintenance

The discussion up to this point has shown that maintenance is performed according to a schedule, on an as-needed basis, or by a combination of these methods. In general, engine manufacturers emphasize time-based maintenance in order to insure that their product receives adequate care in circumstances beyond their control. For their part, engine users usually adopt a time-based maintenance program which may or may not be the same as recommended by the manufacturer.

As-needed maintenance played a small role in the programs that were investigated during this project; in fact, it is our impression that most engine components either are maintained according to a conservative schedule or run until failure. Admittedly, only one or two instances were found where a component was operated until failure occurred, but it is also true that the programs included in this study are dedicated to the concept of strictly scheduled maintenance and are financially able to implement this policy. It seems reasonable to expect that financially-pressed railroads would either extend the time periods of their schedules or allow more components to operate until they failed.

Periodic replacement of components is not (so far as could be determined) based upon any objective principle dealing with either fuel economy or exhaust emissions or, for that matter, with overall economy of operation. The practice may be effective in keeping engines available for service (assuming that maintenance periods are reasonable), but it does not appear to be the most cost-effective method of maintenance.

Replacement of components that have failed is based upon obvious sensible factors -- e.g., excessive smoke or vibration, difficulty in starting, excessive dimensional changes of rings, and so forth. Such replacements are obviously overdue from the viewpoint of fuel economy and emissions since prior degradation has probably compromised these parameters. Engine downtime for repairs will occur at unpredictable intervals and, therefore, make smooth system operation more difficult.

The dilemma facing every maintenance organization is clear; an engine may be kept functioning at an acceptable level of performance and emissions by a conservative maintenance schedule that perhaps involves unnecessary replacement of components, or these performance and emissions parameters may be allowed to steadily degrade until a failure forces maintenance to be done. Ideally, a program can be established that permits each component to operate for its maximum effective life and no more; replacement takes place before fuel economy or emissions suffer and certainly before failure occurs. The method by which this ideal situation may be approached is discussed below.

First, it is necessary to determine the degree of component degradation that is allowable in terms of fuel economy, emissions, and overall economy of operation. The latter value includes cost of the replacement part and its installation, time required for installation (unit downtime), and "trade-in" value of the renewed component. Second, it is required to determine the means to sense or measure, on an operating engine, this allowable degree of degradation of the major components. This step implies development of equipment and test procedures. Last, use this equipment and procedures to provide continuous monitoring of component health and, hence, to guide maintenance actions.

Development of a program such as this would require a substantial effort, but the current state of electronics and engine diagnostics makes it appear possible, and the benefits derived from its implementation would be correspondingly great, for the Coast Guard as well as railroads. Maintaining an engine in peak operating condition is certainly the easiest way to improve fuel consumption and emissions levels, since no modification (radical or otherwise) need be done to the engine or to the manner in which it operates. Furthermore, costs arising from unnecessary maintenance could be drastically reduced and maximum availability of the locomotive or vessel obtained. This approach to engine maintenance is, in our opinion, the most cost-effective.

7.4 Post-Overhaul Checks

The Association of American Railroads has undertaken an evaluation[33] of light-obscuration smoke opacity measurement instruments, commonly referred to as smokemeters. Several makes have performed well enough on in-chassis locomotive engines to be recommended for use by member companies of the AAR. Accordingly, several railroads are using these instruments to determine smoke opacity levels of engines that have either been overhauled or had a major component, such as a turbocharger, replaced. The engine is connected to a load bank for a stationary power check, and smoke opacity is measured simultaneously. An AAR-approved smoke test procedure is available for use at this time. More will be said about this procedure in the next section of the report.

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[33] SP-A. A. R. Program to Develop Certification Procedures with Respect to Visible Emissions from New and Out-Shopped Locomotives, Final Report, Southern Pacific Transportation Company, August 1973.

7.5 Potential Wear Problems for Two-Stroke Cycle Coast Guard Engines

While gathering information on engine wear and maintenance practices for this project, we were made aware of a special concern on the part of Coast Guard engineering personnel about the possible effect of a proposed procedure change on ring wear rates in two-stroke engines. This procedural change involves the mixing of used engine lube oil with diesel fuel as a method of disposal.

Higher fuel costs and the difficulty involved in disposing of used lube oil (especially in large quantities) have prompted users of diesel engines to investigate the feasibility of mixing this oil with fuel. Manufacturers of most small (automotive-size) diesel engines have responded with recommendations concerning this practice. Some manufacturers of four-stroke cycle engines have approved this method of oil disposal and have prescribed procedures which render the practice compatible with their product. These procedures usually include methods of filtering and mixing the used oil and the maximum recommended percentage of the mixture.

Manufacturers of two-stroke cycle engines, however, have generally warned against use of the fuel/oil mixture in their engines. The reason for this is that the two-stroke engine is particularly susceptible to the formation of combustion chamber deposits, and it is thought that the burning of even small quantities of used lube oil will promote the formation of these deposits. (The deposit-forming agents in the used lube oil are the organometallics of the oil additive package and the unfiltered metallic wear particles. The mechanism by which these agents form deposits will not be discussed here.)

Such deposits can cause stuck piston rings and exhaust valves and clogged air ports in the cylinder liner. These problems lead, in turn, to poor engine performance, increased fuel consumption, and higher rates of ring and/or liner wear. In particular, stuck rings and hard deposits around the air ports, over which the rings must pass, are known to cause excessive rates of wear and eventual ring scuffing. Since two-stroke engines constitute a large part of the Coast Guard engine population, it would seem prudent to approach this situation with caution and to investigate, to the extent possible, the consequences of this method of oil disposal.

8. MEASUREMENT PROCEDURES FOR ENGINE THERMAL EFFICIENCY AND EXHAUST EMISSIONS

Evaluation of candidate changes in engine operation or equipment adjustment will require accurate measurement of the component factors needed to calculate thermal efficiency and emissions levels. As in all such evaluations involving two or more changes in a given independent test variable, a series of "back-to-back" tests for different configurations is most straightforward and yields the most information per unit of test time. However, there are many considerations in what may be termed the tactical area of these evaluations: which quantities are to be measured, what instruments are to be used, what engine operating conditions are to be run, and so forth.

Southwest Research Institute is fortunate to have experience in most of the matters mentioned above. This experience, plus that of other organizations that has been published in the technical literature, forms the basis of the recommended test approach contained in this section. Several topics covered here pertain to tests of both locomotive and Coast Guard engines, while others require a slightly different approach for each type of engine. These differences will be delineated as they occur in the discussion.

8.1 General Form of Test Results

It is proposed that the results generated by these tests be put in the same forms that have been used throughout this report, that is, brake specific fuel consumption in lb_m per horsepower-hour, brake specific gaseous emissions in grams per horsepower-hour, and exhaust smoke density in percent opacity. Use of these units will also permit direct comparison to most of the data already in existence, not only for the subject engines, but also for smaller truck diesel engines. Such a comparison is useful to keep the results of any evaluation in perspective and to lend confidence to the results by expanding the data base.

8.2 Exhaust Emissions Measurements

8.2.1 Gaseous Emissions

These emission constituents include NO_x, ($NO + NO_2$), HC, CO, CO_2, and O_2. Instruments will be those typically used in diesel emissions measurement: chemiluminescent analyzer for NO_x, heated flame ionization detector (F.I.D.) for HC, nondispersive infrared (N.D.I.R.) analyzers for CO and CO_2, and polargraphic analyzer for O_2. These instruments are incorporated into sampling systems that are built according to specifications given in either SAE Recommended Practices J215 (for HC) and J177 (CO, CO_2) or the Federal (E.P.A.) test procedure for diesel engine certification(34) of exhaust emission levels.

This type of emission instrumentation has been used by SwRI in many tests of locomotive and large stationary engines with complete success. The same type of instrumentation was used by Scott Research Laboratories in a study(35) of emissions for various Coast Guard vessels. This latter application required that the instrumentation be assembled into several modules that could be carried into the oftentimes close quarters of these vessels. It is anticipated that any Phase II on-board tests will require a similar approach. The Scott study also showed that placement of sample probes in the exhaust stacks and the use of longer-than-normal sample lines to the instruments present no difficulty. SwRI typically uses long sample lines in locomotive tests with no adverse effect on the accuracy of emission measurements.

8.2.2 Smoke

Smoke opacity can be determined by several available light-obscuration smokemeters; however, SwRI has had considerable success in measuring opacity by means of an enlarged version of the basic PHS (Public Health Service) instrument that is used in most E.P.A. smoke certification tests. The larger smokemeter is usually constructed around a 40 in. dia. ring (versus the usual 10. in. dia.) for use on locomotive exhaust stacks. Several engine manufacturers and railroads use the enlarged smokemeter for smoke determinations of both test bed and in-chassis engines.

Exhaust stack configuration varies among the various classes of Coast Guard cutters. From information concerning these configurations it appears that the PHS smokemeter can, with suitable mounting modifications, be used on the vertical stacks of high-endurance cutters and icebreakers. However, exhaust discharge of

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[34] Federal Register, Vol. 37, No. 221, Wednesday, November 15, 1972, pp. 24310-24312.

[35] A Study of Stack Emissions from Coast Guard Cutters, Department of Transportation.

medium-endurance cutters is through horizontal ducts that exit at the stern of the ship, just above the waterline. Since it is not practical to mount the smokemeter in such an awkward position and location, two alternate methods of smoke measurement have been examined. The first involves use of an L-shaped pipe or duct that is inserted into the main exhaust duct at the stern discharge opening and carries a part of the total exhaust flow to deck level where the smokemeter can be mounted in the conventional position. The second method utilizes a so-called inline smokemeter that can be mounted in the main exhaust duct at some point before the stern discharge port. Such smokemeters are commercially available, but they do not utilize the same optical system as the PHS instrument. While both of these methods depart from the more conventional measurement technique in one sense or another, it is thought that either one will yield the results needed for a relative comparison of smoke opacity from the same engine-vessel combination.

A final consideration involving smoke opacity is the effect of plume diameter on measured values. This effect need not be considered when only a relative comparison of smoke opacity is made for the same sampling configuration. However, if it is desired to compare opacities from vessels with different stack sizes or to relate, say, locomotive engine smoke opacity to that of a diesel-powered truck, then it is recommended that the Beere-Lambert Law be used to correct the data for this effect.

8.3 Fuel Consumption Rate

Determination of engine efficiency (BSFC) and brake specific emission values requires accurate measurement of fuel consumption rate. Weigh scales are typically used in tests of in-chassis locomotive engines, and this method has proved to be fairly accurate. The tests conducted aboard Coast Guard cutters by Scott Labs employed positive-displacement type flow meters, similar to common water consumption meters. These units are not influenced by vessel motion, as weigh scales would be, but they are incapable of distinguishing between fuel and air entrained in the fuel; hence, erroneous readings may result. This situation is aggravated by the fact that fuel flowing to the engine often contains air and excess fuel returning from the engine often contains air or combustion gases.

Use of a linear mass flowmeter gives highly accurate and continuous readings of net flow rate. These instruments are used by SwRI and many other laboratories for test bed work, and there is little doubt that on-site locomotive engine tests would benefit from their use. However, they have never been used on-board a moving ship, and it is known that they are sensitive to motion. Recent conversations with a manufacturer of such a flowmeter indicates that several options are available to minimize the effect of motion, and that a cooperative effort might be established between the manufacturer and SwRI to modify the device further and make it truly suitable for shipboard measurement.

Several other types of flowmeters have been briefly investigated for possible use in this difficult application, but none are without their drawbacks and none possess the ability of the linear mass flowmeter to accurately measure mass fuel flow rates over the range (perhaps 50 to 1) from rated speed and load to idle. Accurate fuel consumption data are imperative if reliable engine efficiency figures are to be obtained. It thus appears that development of a suitable method (including hardware) will have to occur in Phase II before meaningful BSFC data can be obtained from Coast Guard engines.

8.4 Engine Power or Torque

Measurement of locomotive engine power represents no fundamental technical problem provided, of course, one has access to a suitable test facility at a railroad maintenance operation. Diesel-electric systems in icebreakers cannot be loaded artificially while the vessel is at rest, but power output can be readily measured when the ship is underway. However, on-board measurement of engine power (or torque) for gear-reduction drive medium-endurance and high-endurance cutters constitutes a significant obstacle to obtaining accurate brake specific data that would be meaningful in terms of project objectives.

The two Class 378 vessels visited during the project were equipped with drive shaft-mounted torque transducers for each of the main diesel engines, and it is understood that all 378's are so equipped. These transducers were described as having never worked properly and have apparently fallen into disuse. They are of the "contacting" type; i.e., strain gauges are applied to the drive shaft and the resulting electrical signals are brought out through slip rings to the readout device. The engineering staffs of these vessels believed that the transducers were inoperative due to an oil film on the slip rings that prevented good electrical contact. This is a common problem with this type of transducer. It could not be determined through these interviews if the transducers had initially been calibrated to produce absolute or relative torque readings.

The Class 210B cutter we visited was not equipped with torque transducers, and it is our understanding that none of these vessels are. However, the amount (about four feet) of exposed, accessible drive shaft is sufficient to allow most types of transducers to be installed.

There are two problems associated with obtaining adequate torque output data for the engines in these two classes of vessel. Both difficulties are linked to the definition of what constitutes "adequate" data. It is our opinion that such data should be absolute (i.e., in actual lb_f - ft) rather than relative (i.e., proportional to actual torque) in nature, and that the accuracy of these absolute data must be equal to or better than one (1.0) percent across the entire range of values. The reasons behind our selection of these criteria are as follows.

First, concerning the desirability of absolute torque data, it is evident that relative values can be used to obtain a comparison of test results; that is, relative brake specific fuel consumption and emission values can be computed and compared in terms of a percentage change from some set of baseline values. However, without actual torque data there is no way to determine if the engine is operating at a realistic level of performance in its baseline (normal) configuration, and it is pointless to evaluate various candidate fuel consumption/emission reduction techniques using a "sick" engine. A general indication of the state of health of an engine can, of course, be obtained by analysis of important parameters such as exhaust temperature, turbocharger boost pressure, rack position and so forth, but this analysis cannot give a precise picture of engine performance. (For example, some engine manufacturers consider as "normal" variations of 250 F in exhaust temperature and ten percent in rack position for individual cylinders under rated load.) Only accurate absolute torque data for the entire operating range of the engine can allow a realistic assessment to be made.

Second, an accuracy of one percent in these torque readings is thought necessary in order to observe the relatively small changes in brake specific quantities that will probably occur when the candidate techniques are implemented. It is considered unlikely that any large changes (greater than, say, five percent) will occur in the baseline values of an engine that is in good operating condition. Therefore, torque (and, for that matter, fuel consumption) data that are accurate within even two or three percent will not be adequate to accomplish the project objectives. Note that even if the relative torque data mentioned above were used, the requirement on accuracy of both torque and fuel consumption would not be obviated.

A survey of commercially available torque transducers has turned up two or three models of the "noncontacting" type. These transducers, which also utilize small strain gauges mounted on the drive shaft, rely on solid state electronic circuitry to transmit the signal from the rotating member to a suitable receiver mounted in proximity to the shaft. The manufacturer's literature contains impressive performance specifications, including accuracy of ± 1.0 percent. However, the instruments all produce relative torque readings in their present configuration, although it may be possible to calibrate them to give absolute data. Calibration would involve applying a known torsional load to the shaft and observing the transducer reading. Application of the load would require "locking" the shaft to prevent rotation and making certain the shaft was subjected to only a pure torsional load. Calibration with several such loads that span the torque range of the engine would also verify the accuracy and linearity of the transducer.

8.5 Exhaust Mass Flow

The mass flow rate (in lb_m per minute) of engine exhaust gas must be known in order to calculate brake specific values of gaseous emissions. In tests of small diesel engines it is common practice to measure mass flow rates of intake air and fuel and then add these two quantities to obtain the exhaust mass flow. However, the large size of the subject engines renders conventional air flow measurement impractical. This problem is circumvented by use of a carbon balance technique which involves the fuel mass flow rate and concentrations of emission constituents that contain carbon. The basic procedure is explained in Reference 36, and a virtual treatise on the subject is available in Reference 37.

The carbon balance method places a premium on accurate measurement of fuel mass flow and CO_2 concentration. The method is also sensitive, to some extent, to the value of the fuel hydrogen-to-carbon ratio. However, it is reported(38) that only small differences result when an assumed value of 2.0, rather than the actual value

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[36] Exhaust Emissions from Uncontrolled Internal Combustion Engines, Part I, Southwest Research Institute, E. P. A. Report APTO-1490.

[37] Stivender, Donald L., Development of a Fuel-Based Mass Emission Measurement Procedure, SAE Paper 710604, June 1971.

[38] Hoffman, Four Cycle Diesel Emissions.

is used in the calculations. The exact H/C ratio can be analytically determined for the fuel batch used in each test if extreme accuracy is desired.

8.6 Supplementary Engine Data

Engine operating temperature and pressure data are usually acquired as a matter of course in most tests. Quantities such as exhaust temperature for individual cylinders, air box or blower pressure and temperature, exhaust backpressure, air filter restriction, and fuel transfer pump pressure are helpful in determining the overall condition of the engine and can be used to determine repeatability of operation in a series of tests. As mentioned previously, marine engines are usually instrumented for all or most of these parameters, but locomotive engines are not. However, any of the subject engines can be so outfitted if the user permits it.

8.7 Engine Test Conditions

Each engine test conducted in Phase II will consist of both steady-state and transient operating modes that are typical of the engine's normal operation. Power (or torque), fuel consumption, gaseous emission concentrations and smoke opacity will be measured at steady-state conditions, and brake specific fuel consumption, NO_x, HC and CO computed into final form. Smoke opacity only will be measured under transient operation.

Little is known of the transient operation of Coast Guard power plants, but it is assumed that most transients occur between operating conditions that lie fairly close together. The exact nature of the speed-power changes can be determined by direct observation of throttle operation and questioning of engine room personnel. These changes will be incorporated into the test procedure for these engines. Transient operation of locomotive engines appears to be adequately defined by a smoke test procedure developed by EMD, evaluated by Southern Pacific Transportation Company, and adopted by AAR(39). This procedure also includes steady-state operation at idle speed and the eight throttle notches. However, smoke opacity data for these conditions can be obtained during the emission measurements. The entire AAR test cycle appears in Table 8.1.

[39] SP-A.A.R. Certification Procedures.

TABLE 8.1 AAR LOCOMOTIVE SMOKE TEST PROCEDURE CONDITIONS

Mode Number	Throttle Position	Condition
1	Idle	Steady-state
2	N1	"
3	N2	"
4	N3	"
5	N4	"
6	N5	"
7	N6	"
8	N7	"
9	N8	"
10	N8 to Idle	Throttle sweep
11	Idle to N4	"
12	N4 to N8	"
13	N8 to N6	"
14	N6 to N8	"
15	N8 to Idle	"
16	Idle to N8	"
17	Idle	Steady-state

9. EFFECT OF AIR PROPERTIES ON PERFORMANCE AND EMISSIONS

This section presents a theoretical discussion of the relations between ambient or ingested air properties (pressure, temperature and absolute humidity) and engine performance and emissions of smoke and NO_x. We then review the methods used by various engine manufacturers to apply correction factors for these effects. The current state of correction factors used in E.P.A. diesel emissions tests is also summarized.

9.1 Effect on Engine Performance

The pressure, temperature and absolute humidity of ambient air varies with location, altitude, and season. It is generally acknowledged that humidity has a negligible effect on engine performance; therefore, the following discussion centers around the effect produced by changes in the other two parameters.

Changes in pressure and temperature of the air charge produce corresponding changes in the charge density and in volumetric efficiency of the engine. Experience has shown that for volumetric efficiency, v, and engine indicated mean effective pressure (imep), one has

$$v \propto \sqrt{T} \tag{9-1}$$

and

$$\text{imep} \propto \rho\, v \tag{9-2}$$

where ρ = air density. From the perfect gas law,

$$\text{imep} \propto \frac{P}{\sqrt{T}} \tag{9-3}$$

For a naturally aspirated engine, P and T are referred to ambient air and relation (9-3) can be easily applied to correct for the effects of these two parameters on engine performance. The correction factor usually involves the ratio of the imep at standard conditions to that at the test conditions:

$$C = \left(\frac{B_o}{B_t}\right)\left(\frac{T_t}{T_o}\right)^{1/2} = \frac{(\text{imep})_o}{(\text{imep})_t} \tag{9-4}$$

where the meaning of the subscripts is obvious.

For a turbocharged engine, however, the relation (9-3) applies to air conditions in the inlet manifold downstream from the turbocharger. Therefore, in order to find the effect of a change in ambient pressure and temperature on manifold pressure and temperature, it is necessary to know in some detail the characteristics of the turbocharger and any air charge cooling equipment (intercooler and/or aftercooler). Determination of these characteristics and their incorporation into a correction factor is a complicated process that involves analysis of many interdependent variables.

These complications have caused engine manufacturers to develop empirical correction factors for their own engines or to use no correction at all. For instance, one turbocharged engine is rated at a barometric pressure corresponding to 1000 ft altitude and 60 F air temperature, and the manufacturer supplies a nomograph to correct site horsepower to these standard conditions. Another engine is rated at 1500 ft and 90 F maximum air temperature, conditions adopted by the Diesel Engine Manufacturers Association (DEMA). However, it is interesting to note that the DEMA handbook(40) states that, for turbocharged and aftercooled engines, the correction factor becomes so complex that its use is impractical.

The Federal emissions test procedure does not employ a performance correction factor. However, the tests are conducted within a fairly narrow range of air temperature (68 to 86 F) and pressure (28.5 to 31.0 in. Hg). The procedure states that tests may be conducted at higher temperature and lower pressure than this, but that no allowance can be made for increased emission levels at these conditions.

In summary, it may be possible to develop a theoretically-based correction factor, but it will inevitably include a large number of terms and will probably not be applicable to a broad variety of engine. Hence, the present methods for correction used by engine manufacturers are considered to be satisfactory, especially in view of the deficiencies and difficulties of the alternative.

9.2 Effect on Smoke and NO_x

Values of pressure, temperature and absolute humidity determine not just air density, but also the thermodynamic properties of the

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40 Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines, 6th Edition, 1972, Diesel Engine Manufacturers Association.

ingested air. If it is assumed that fuel/air ratio is maintained constant with a change in air density, then the primary effect of such a change is to modify the fuel injection spray pattern and the heat transfer rates from the combustion chamber.

That is, reduced air density in the cylinder produces a more cohesive spray with less breakup and dispersion of fuel droplets. The core of this modified spray jet, being richer than before, burns later in the cycle and less completely, with a consequent increase in exhaust smoke. NO_x will probably be reduced under these conditions. Changes in heat transfer influence the gas temperature in the combustion chamber and both smoke and NO_x are sensitive to this parameter. However, the direction in which the emission constituents change is probably not predictable on theoretical considerations alone.

Moisture in the ingested air determines the specific heat of the air; an increase in absolute humidity increases the specific heat which, in turn, lowers combustion temperatures and the amount of NO_x formed. An increase in absolute humidity also, of course, increases air density with the results outlined in the preceding paragraph.

Therefore, changes in atmospheric conditions produce injection and combustion-related effects in the engine. The degree of these effects is largely dependent upon engine design features; e.g., fuel sprays from different types of injectors are affected differently by changes in air density. As in the case of performance correction factors, the relationships involved are very complex and usual practice is to correct NO_x for temperature and humidity only. Smoke is not corrected for any change in air properties since the effects involved vary greatly from engine to engine.

As far as is known, the only NO_x correction factor in use is that specified for use in E.P.A. diesel emissions tests. This factor has been derived empirically, but is of doubtful validity for large diesel engines or even for smaller engines other than those used in its derivation. The factor corrects observed NO_x concentrations to a standard condition of 85 F and 75 grains of water per lb_m of air.

10. SUMMARY AND CONCLUSIONS

This project was conducted as the first phase of a program to define and evaluate various candidate techniques to reduce fuel consumption and exhaust emissions of NO_x and visible smoke for in-service diesel locomotive and Coast Guard cutter engines. Phase I investigated several factors that, to varying degrees, influence engine efficiency and emissions characteristics; these factors included operating modes and duty cycles, improved component parts and operating methods, and the effects of wear and maintenance. This report also outlined possible test procedures to use in laboratory and field evaluations of candidate techniques in the second phase of the program.

Principal points and conclusions derived from investigation of these topics are summarized below:

(1) Engines employed in the applications of interest generally operate in distinct modes characterized by closely defined speed and load conditions. Selection of these operating modes is determined by the performance and flexibility desired of the locomotive or vessel.

(2) The duty cycle of line-haul (intercity) locomotives has been defined in several studies, with reasonable agreement in results. Engines in these locomotives spend about 40 percent of their time at idle and about 20 percent at rated speed and load (Throttle Position 8).

(3) Duty cycles of Coast Guard cutters have not been defined in a quantitative sense; however, interviews of Coast Guard personnel indicate that these engines spend a substantial fraction of their operating time at idle and little time at rated speed and load.

(4) It is essential that adequate quantitative duty cycle information be obtained for Coast Guard engines in order to evaluate overall cycle composite effectiveness of candidate fuel consumption and emission reduction methods.

(5) Improved component parts, suitable for retrofit of in-service engines, are available from several engine manufacturers, especially those that make locomotive engines. These components include fuel injectors, governors, turbochargers, pistons and cylinder liners

(for two-stroke cycle engines). Retrofit of all of these components to an engine not equipped with them will reduce fuel consumption, smoke opacity, CO, HC, and perhaps, NO_x. Their use is judged to be a cost-effective way to improve fuel consumption and emission levels provided that retrofit is accomplished on an as-needed basis during regularly-scheduled maintenance and overhaul.

(6) Effectiveness of many of these improved engine components has been documented in the technical literature and confirmed in our conversations with the engine users. Therefore, further testing is thought to be unnecessary in Phase II of this program.

(7) Retarded injection timing is a simple and apparently cost-effective method to reduce NO_x. However, its use entails penalties in fuel consumption and smoke opacity and may impose durability problems on the engine due to an increase in thermal stress. Some engine manufacturers are conducting laboratory evaluations to determine the exact nature and extent of these drawbacks.

(8) It has been demonstrated, in the case of one particular type of locomotive engine, that changes in operating modes and throttle operation can result in much lower transient and steady-state smoke opacity and slightly lower CO and NO_x. Cycle composite fuel consumption is said to be unchanged.

(9) Extended periods of engine idling result in several undesirable side effects; however, engine users can demonstrate several advantages to the practice which, in their opinion, offsets these deleterious features. No cost-effectiveness figures were available that could objectively demonstrate which point of view is correct. Several alternatives to the practice appear feasible in theory.

(10) Existing partial data on brake specific fuel consumption, NO_x, and CO was expanded into complete engine maps of these quantities. This was accomplished by the use of analytical techniques supported by numerous assumptions. The results are trendwise correct and proved to be adequate for project objectives.

(11) These fuel consumption and emissions maps, along with existing smoke opacity data, were used to predict consequences arising out of possible changes in engine operating modes (speed-load conditions) for the applications of interest. No changes in the manufacturer-specified rated speed and load conditions were considered.

(12) BSFC, $BSNO_x$, and BSCO associated with two-stroke and four-stroke cycle locomotive engines can be reduced by moving current operating points or modes in such a way that engine speed and BMEP are increased. However, this action would destroy the operating flexibility of the locomotive and impose high mechanical and thermal stresses on the engine.

(13) Operating modes where BSFC could be improved most (percentagewise) are those at low speed and power output. However, locomotive duty cycles show that these modes represent only a small part (8 to 12 percent) of the total operating time of line-haul units. Therefore, cycle composite BSFC would show only a very slight improvement. Low-speed operation of the locomotive would be radically altered.

(14) Changes in the speed-load values of operating points of Coast Guard engines must be accompanied by changes in propeller pitch and, therefore, in propeller efficiency. Insufficient data was obtained in Phase I to permit an analysis of the feasibility and practicality of these important ramifications.

(15) BSFC and brake specific emissions of Coast Guard engines could benefit from selected changes in operating modes, i. e., those changes allowed after consideration of propeller pitch and efficiency and of the effect on vessel operating flexibility. Since Coast Guard engines, unlike locomotive power plants, apparently spend little time operating at rated speed and load, any improvement in partial load fuel consumption could be of significant importance on a cycle composite basis.

(16) A theoretical approach, based on an ideal thermodynamical cycle, was developed and used to investigate the effect of cylinder leakage (due to wear of piston rings, liner and valves) on engine performance and emissions. Assuming an extreme amount of leakage from all

cylinders indicated that BSFC and $BSNO_x$ would increase by a moderate amount (2-4 percent) and BSCO would either increase or decrease, sometimes by a substantial amount (20-30 percent).

(17) A study of engine maintenance programs indicated that problems of high fuel consumption and smoke opacity are often caused by degraded performance of components that influence combustion in all cylinders simultaneously, i.e., governors, turbochargers, and, perhaps, air filters. Poor injector performance in several cylinders could produce the same results.

(18) Wear limits of rings, liners, and other internal components are conservatively set by engine manufacturers. Therefore, wear of these components probably has a negligible effect on performance and emissions as long as these limits are not exceeded by the engine user.

(19) Maintenance programs for the subject engines emphasize replacement of critical components according to a conservative, periodic schedule. This replacement is apparently not based on objective criteria regarding either fuel economy or emissions but, rather, is done on a "play safe" basis in order to avoid engine failures and consequent engine downtime or unavailability. Many such replacements are judged to be premature and, hence, unnecessary.

(20) In some instances, engine components are allowed to fail before replacement is undertaken. This policy undoubtedly compromises the performance and emissions levels of the engine since substantial degradation of the component(s) in question likely occurs prior to failure. This practice is probably more prevalent among railroads that are in financial difficulty.

(21) An ideal maintenance program, in which the performance of these critical components is monitored on a virtually continuous basis, is feasible in both railroad and Coast Guard operation. This program would increase the time between component replacements and use fuel economy and emissions as criteria for necessary maintenance.

(22) A tentative plan by the Coast Guard to dispose of used lube oil by mixing it with the on-board supply of diesel fuel could result in excessive wear rates of rings and liners in two-stroke cycle engines.

(23) On-site testing of locomotive engines in Phase II represents no fundamental problems and will utilize instrumentation and techniques employed by Southwest Research Institute (among others) for several years. Emissions measurements on board Coast Guard cutters have been shown to be feasible and no difficulty is anticipated in this regard.

(24) However, measurement of Coast Guard engine fuel consumption rate and absolute (or actual) engine torque to the accuracy called for by the program objectives represents a significant problem. Use of suitably modified, commercially-available flowmeters and torque transducers is possible.

(25) Power output of these turbocharged and aftercooled engines is only slightly affected by changes in ambient air properties (temperature, pressure, and absolute humidity). Such correction factors that exist are generally empirically derived for each particular engine, and these factors are thought to be adequate.

11. RECOMMENDATIONS FOR PHASE II

The following recommendations are based on the conclusions contained in the previous section. Their numerical sequence reflects only the order in which the topics were presented in the text and does not indicate rank of importance. Indeed, several of them are interdependent and constitute a package that is thought to be of fundamental importance to overall program objectives in Phase II. Specific work items are the following:

(1) Obtain adequate duty cycle information for three classes of Coast Guard vessels: high-endurance (378) and medium-endurance (210B) cutters and icebreakers. Acquisition of this information will continue throughout Phase II; however, partial results available after a few months will allow fairly accurate estimates of the real value of changes in fuel consumption and emissions characteristics of these engines.

(2) Establish a quantitative cost-to-benefit relation for the practice of extended engine idling, including an analysis of cost factors such as engine fuel consumption, use of lube oil/coolant heaters, heater consumption of fuel and/or electricity, and additional maintenance due to higher engine wear rates. Locomotive and Coast Guard situations will be analyzed separately.

(3) Analyze propeller pitch/efficiency data for medium-endurance and high-endurance cutters in order to determine the overall effect of engine mode changes on fuel consumption and vessel performance. Efficiency data will be obtained directly from the Coast Guard or derived from pitch specifications, if necessary.

(4) Conduct field tests of representative Coast Guard and locomotive engines in order to determine the degree of degradation which certain components (e.g., turbocharger, governor) must undergo before measurable (± 1.0 percent) changes in power and fuel consumption occur. Determine simultaneously any associated changes in smoke opacity and gaseous emissions.

(5) Determine and develop means to continuously monitor the performance of these components on an operating engine. Also, develop appropriate test rationales and procedures for each engine application. Install

prototype models of the test equipment on selected Coast Guard and locomotive engines.

(6) Conduct a laboratory investigation to determine if burning a mixture of diesel fuel and used lube oil results in a higher rate of piston ring wear for a two-stroke cycle diesel engine. Wear similitude theory will then be used to project any increase in wear rate from the laboratory engine to other types of Coast Guard two-stroke engines.

(7) Develop instrumentation to accurately (± 1.0 percent) measure engine fuel consumption rate and actual drive shaft torque of engines in medium-endurance and high-endurance cutters. Modification of existing instruments or design and fabrication of new equipment will be required. Methods and costs will be investigated to make this equipment a permanent part of the engine room installation for the purpose of furnishing performance data to engineering personnel for use in engine diagnostics and maintenance.

These recommendations convey several principal ideas that reflect our conception of the nature and direction of Phase II work.

First, emphasis has been placed on evaluating candidate changes involving operation of Coast Guard engines. The reason behind this is that (a) these engines appear more susceptible to improvement through changes in operating modes than do locomotive power plants, (b) these engines generally do not have the range of improved component parts that are available for locomotive engines, and (c) adequate data and documentation is available to demonstrate the effectiveness of most of the fuel consumption and emission reduction techniques currently available for locomotive engines.

Second, performance and emissions measurements of in-service Coast Guard engines should be predicated on the development of adequate fuel consumption and torque measurement devices so that the potential of candidate changes can be accurately evaluated. The determination of cutter duty cycles also helps insure that a realistic assessment will be made of the data that is obtained in these evaluations.

Third, the emphasis on development of an idealized maintenance program, based upon improved engine diagnostic procedures, reflects our conclusion that current maintenance is often not cost-effective since it does not utilize fuel economy, emissions levels, and overall economy of operation as its criteria. It appears that this area holds the greatest potential for improving fuel economy and minimizing emissions for the broadest class of engines and applications. This conclusion is enforced by the apparent inability of other options (e.g., changes in operating modes) to improve fuel consumption of in-service engines without producing undesirable side effects.

APPENDIX A
THEORETICAL BASIS OF CYLINDER LEAKAGE CALCULATIONS

The general method and discussion are taken from Taylor and Taylor, The Internal Combustion Engine, International Textbook Company, 1961. A simplified but useful method of engine cycle analysis is to use the air standard cycle, in which a so-called "perfect gas", air, is the medium and the limiting case of an actual engine cycle may be obtained. It should be noted that this limiting case represents the theoretical or ideal situation for the actual diesel engine cycle. Hence, theoretical values of important performance parameters such as IMEP are not of the same magnitude as those associated with the actual cycle. The idealized air standard cycle can be represented by a pressure vs. specific volume diagram (Figure A-1) for a constant volume combustion process.

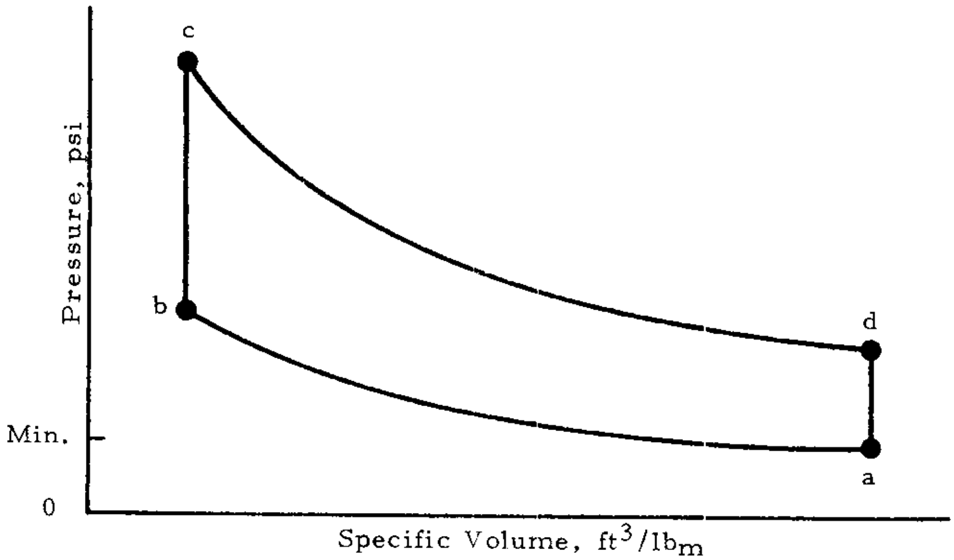

FIGURE A-1. IDEAL CONSTANT VOLUME COMBUSTION DIAGRAM

The cylinder is filled with air at the pressure and specific volume designated by Point a. The air charge is then compressed reversibly and adiabatically to Point b, where heat is added under constant specific volume condition. The pressure increases to Point c, where reversible and adiabatic expansion occurs to Point d. The heat input necessary to obtain the pressure increase between Points b and c is given by the equation

$$Q(b \rightarrow c) = FQM\left(\frac{r-1}{r}\right) \qquad \text{(A-1)}$$

where F = fuel-air ratio, Q = heat of combustion per unit mass of fuel, M = mass of air in the cylinder, and r = compression ratio. The work per cycle per lb_m air is represented by the area bounded by the curve abcda and is determined by the initial pressure and temperature at Point a, the value of Q, and the compression ratio of the engine. Such an area was computed for the subject engine for the idealized case of no leakage from the cylinder.

For the case where cylinder leakage is present, it is assumed that leakage on the compression stroke occurs reversibly and adiabatically to Point b' in Figure A-2, and that leakage on the expansion stroke occurs at the beginning of the stroke, i.e., from Point c to c'. Expansion continues to a higher value of specific volume (Points e and f) than before because of the decreased mass of air in the cylinder. The work performed during this cycle is represented by the area bounded by the curve fb'c'ef minus the area b''b'c'c'' and the area under the compression part of the curve from b' to b. The first of these two (shaded) areas is the work lost due to leakage during expansion, while the second is the work not performed in compressing the part of the air charge that is lost during compression.

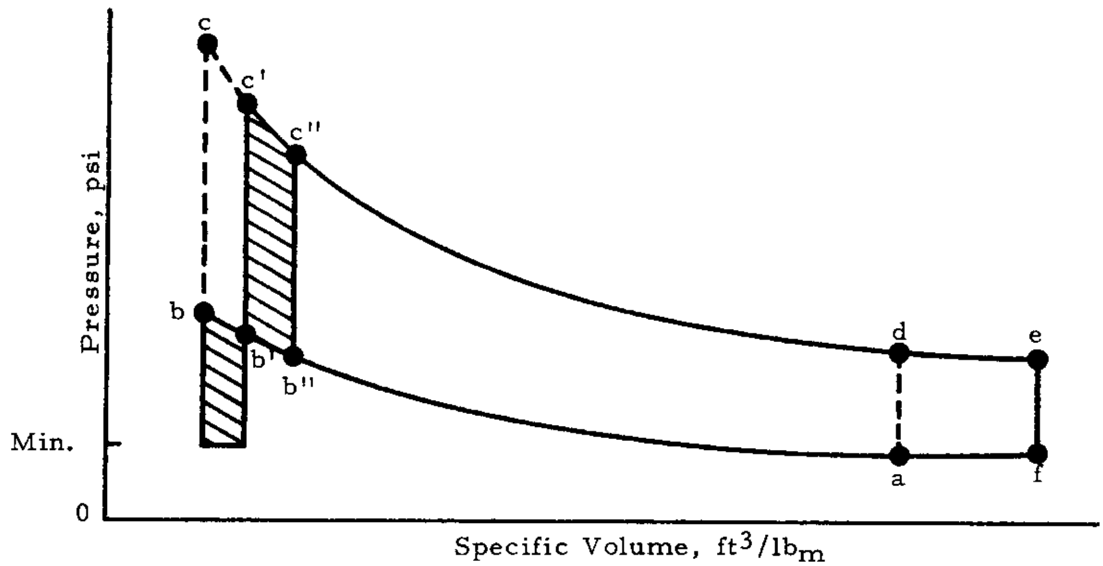

FIGURE A-2. CONSTANT VOLUME COMBUSTION WITH LEAKAGE

The ratio of the net area obtained for the leakage case, to the area for the ideal case (no leakage present) is the percent reduction in work per cycle per lb_m of air for the two cycles. This change is then assumed to represent the percent reduction in actual BHp as well; i.e., the baseline BHp is reduced by this amount, and new brake specific quantities are calculated with the lower value of BHp. This approach is necessary since IMEP values obtained from use of the air standard cycle are not of the correct absolute magnitude needed to calculate the new BHp values directly by means of the formula BHP = IHP - FHP.

It is necessary to establish the relation between the mass of air lost during the compression stroke and the mass of exhaust gas lost during the expansion stroke. The masses will differ because of the different pressures and temperatures that are present in the cylinder during these two strokes. We make the simplifying assumption that the points where leakage occurs are represented by a single area and that choked flow occurs through this area during both compression and expansion strokes. Then the maximum rate of mass loss under the conditions is given by

$$\dot{M} = (.53)\, C_d \left(\frac{PA}{\sqrt{T}}\right) \qquad \text{(A-2)}$$

where C_d = discharge coefficient, P = average pressure during compression or expansion stroke, A = leakage area, and T = average temperature during compression or expansion stroke.

Letting the subscript c represent the conditions present during compression and subscript e the conditions present during expansion, the ratio of the corresponding mass losses is

$$\frac{\dot{M}_c}{\dot{M}_e} = \left(\frac{C_d PA}{\sqrt{T}}\right)_c \left(\frac{\sqrt{T}}{C_d PA}\right)_e \qquad \text{(A-3)}$$

Since the area and its shape are assumed constant in time, eq. (B-3) is simplified to

$$\frac{\dot{M}_c}{\dot{M}_e} = \left(\frac{P_c}{P_e}\right)\left(\frac{T_e}{T_c}\right)^{1/2} \qquad \text{(A-4)}$$

Typical (average) values of these quantities for the cycle under consideration are P_c = 500 psi, P_e = 2000 psi, T_c = 1200R, and T_e = 4800R. Inserting these values in eq. (B-4) gives

$$\dot{M}_c = \frac{1}{4}\sqrt{4}\ \dot{M}_e = \frac{1}{2}\ \dot{M}_e \qquad \text{(A-5)}$$

Therefore, about twice as much mass per unit of time is lost during expansion as is lost during compression. Since the assumption is made that compression leakage occurs within a very short time interval t_c and that expansion leakage takes place in a similar time interval t_e, we set these two intervals equal to each other and multiply through eq. (A-5) to obtain

$$M_c = \frac{1}{2} M_e \qquad \text{(A-6)}$$

where M_c and M_e is the total mass lost from the cylinder during compression and expansion, respectively.

APPENDIX B
REPORT OF INVENTIONS

A diligent review of the work performed under this contract has revealed no new innovation, discovery, improvement, or invention.

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8 Walter, USCG Pollution Abatement Program, p. 2.

9 Ephraim, Max, Jr., Status Report on Locomotives as Sources of Air Pollution, p. 10, SAE Paper 720604, 1972.

10 Hoffman, J. G., Jr., K. J. Springer, and T. A. Tennyson, Four Cycle Diesel Electric Locomotive Exhaust Emissions: A Field Study, p. 3, ASME Paper 75-DGP-10, 1975.

11 Ephraim, Status Report on Locomotives, p. 10.

12 Hoffman, Four-Cycle Diesel Emissions, p. 9.

13 Private Communication to C. D. Wood, Southwest Research Institute, from H. A. Williams, Electromotive Division, January 24, 1975.

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21 Hoffman, Four-Cycle Diesel Emissions, p. 15.

22 Hare, Locomotive Exhaust Emissions, pp. 11-12.

23 Hoffman, Four-Cycle Diesel Emissions, p. 12.

24 Exhaust Emissions from Uncontrolled Internal Combustion Engines, Part I, pp. B2-B9 and C2-C11, Southwest Research Institute, E.P.A. Report APTD-1490.

25 Hoffman, Four-Cycle Diesel Emissions, pp. 12-16.

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30 Exhaust Emissions from Uncontrolled Internal Combustion Engines, Part I, pp. B2-B9 and C2-C11, Southwest Research Institute, E.P.A. Report APTD-1490.

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LIST OF ABBREVIATIONS AND SYMBOLS

EMD	Electro-Motive Division of General Motors Corporation.
G.E.	General Electric Company's Diesel Engine Products Department.
rpm	Revolutions per minute.
BHp	Brake horsepower.
BSFC	Brake specific fuel consumption in units of lb_m per brake horsepower-hour.
$BSNO_x$	Brake specifie oxides of nitrogen (NO_x) in grams per brake horsepower-hour.
BSHC	Brake specific unburned hydrocarbons (HC).
BSCO	Brake specific carbon monoxide (CO).
FMEP	Friction mean effective pressure.
BMEP	Brake mean effective pressure.
IMEP	Indicated mean effective pressure.
$\dot{F}$	Engine fuel consumption rate in lb_m per hour or minute.
Q	Heating value of the fuel in BTU's per lb_m.
n_t	Engine thermal efficiency, a dimensionless number.
n_m	Engine mechanical efficiency, a dimensionless number.
$\frac{\dot{F}}{A}$	Engine combustion (or trapped) fuel-to-air ratio.
ρ	Density of air in lb_m per ft^3.
v	Engine volumetric efficiency, a dimensionless number.
P	Air pressure in lb_f per in^2.
ppm	Parts per million.

List of Abbreviations and Symbols (Cont'd)

$\dot{M}_x$	Engine exhaust gas flow rate in lb_m per minute.
$\dot{M}_A$	Engine intake air flow rate in lb_m per minute.
PHS	Public Health Service.
E.P.A.	U.S. Environmental Protection Agency.

GLOSSARY OF TERMS

Diesel-electric drive system -- a system in which a diesel engine turns a generator to obtain electricity for motive power.

Governor -- an attachment to an engine that provides automatic control or limitation of engine speed and/or power output.

Load control -- a system consisting of the engine throttle and governor, often employing information feedback of engine power demand, that determines the engine speed and power output at a given throttle setting.

Rated engine speed and power -- the maximum speed and power output of an engine, as designated by the manufacturer.

Turbocharger -- a turbine compressor driven by exhaust gases for supplying air to the engine at pressures above atmospheric.

Propellor pitch -- the distance advanced by a propellor in one revolution.

Engine map -- a set of brake specific fuel consumption or gaseous emissions data for various engine speed - power points, sufficient in number to permit lines to be drawn connecting all equal values of fuel consumption, etc. (lines of constant BSFC, $BSNO_x$ and so forth).

Friction mean effective pressure (FMEP) -- the average pressure exerted on the piston during the power stroke necessary to overcome friction of the various engine components at a given engine speed.

Brake mean effective pressure (BMEP) -- the average pressure exerted on the piston during the power stroke necessary to develop the flywheel or brake power output of the engine at a given speed.

Indicated mean effective pressure (IMEP) -- the sum of the friction and brake mean effective pressures.

Engine brake thermal efficiency (n_t) -- the ratio of the heat equivalent flywheel or brake power output to the total heat input of the fuel.

Engine brake mechanical efficiency (n_m) -- the ratio of the brake horsepower to the indicated horsepower.

Engine volumetric efficiency (v) -- the ratio of the volume of inducted air charge to the swept engine volume (displacement).

Roots blower -- a mechanically driven positive-displacement rotary blower used on two-stroke cycle diesel engines to supply air to the engine cylinders.

Smokemeter -- an instrument to measure the opacity of an exhaust smoke plume.

1. INTRODUCTION

1.1 Background

Southwest Research Institute began work on Contract DOT-TSC-920 for Transportation Systems Center of the U.S. Department of Transportation in November 1974. The project was initiated to investigate methods to reduce fuel consumption and improve emissions of large in-service diesel engines used in locomotives and several classes of Coast Guard vessels. This initiative, in turn, was the result of recent emphasis on controlling emissions from most types of power plants and the even more recent events that caused diesel fuel first to be in short supply and then to be available, but at a sharply increased price.

The number of engines employed in these applications is approximately 27,400; of this number, some 27,285 are in locomotives(1) and slightly over 100 are in Coast Guard vessels(2). A recent DOT report(3) places the average yearly fuel usage by the entire CG cutter fleet at 4.6×10^7 gallons, of which about 70 percent or 3.2×10^7 gallons are consumed by vessels equipped with the diesel engines of interest. However, it should be noted that this figure includes fuel consumed by gas turbines and boilers, as well as main and auxiliary diesel power plants, aboard some of these vessels. The exact proportion of usage between main diesel engines and these other power sources was not derived in the report referenced above, but it seems reasonable to assume (and it is only an assumption) that at least two-thirds of this amount, or somewhat more than 10^7 gallons per year, are consumed by the large main diesel engines.

In contrast, fuel consumption by locomotive engines totaled about 4.1×10^9 gallons in 1973(4). Most of this fuel was consumed by line-haul locomotives engaged in intercity movement of freight, as opposed to units engaged in yard switching service. It has been

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[1] Statistics of Railroads of Class I in the United States, Years 1963 to 1973, Association of American Railroads, Economics and Finance Department, August 1974.

[2] Janes Fighting Ships, 1972-1973.

[3] Walter, R. A., et al, USCG Pollution Abatement Program; A Preliminary Study of Vessel and Boat Exhaust Emissions, Report No. DOT-TSC-USCG-72-3, Department of Transportation - U.S. Coast Guard, November 1971.

[4] Statistics of Railroads of Class I, 1963-1973.

calculated[5] that a line-haul locomotive consumes almost 410,000 gallons of fuel per year. Such freight movement is highly efficient; figures derived from data in Reference 1 give a figure of about 210 ton-miles per gallon of fuel consumed.

Consideration of the fuel consumption figures mentioned above shows that the engines of interest consume well over one billion (10^9) gallons of diesel fuel per year. Therefore, a one percent reduction in fuel consumption in all engine operating modes involves the conservation of ten million (10^7) gallons per year.

In regard to emission production by these engines and their contribution to the total yearly amount of air pollution in the United States, reference is made to a study[6] conducted for the Environmental Protection Agency by Southwest Research Institute. This study concluded that, on a mass basis for 1970, locomotive engine emissions were responsible for approximately 0.8 percent of hydrocarbons from mobile sources, 0.2 percent of carbon monoxide, and 5.2 percent of oxides of nitrogen. On an all-source basis (i.e., mobile and stationary) these percentages are reduced to approximately 0.5, 0.1, and 2.7, respectively. On a mass basis, the quantity of particulate matter emitted by these engines (in the form of visible smoke) is a small fraction of the mass of gaseous emissions[7]; however, its high degree of visibility has made it a matter of public discussion and thus caused locomotive engine manufacturers and the railroads to concentrate their efforts towards its reduction.

An emissions impact study[8] dealing solely with Coast Guard engines indicated that the Coast Guard is an insignificant contributor

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[5]Private Communication to Mr. C. D. Wood, Southwest Research Institute, from Mr. H. A. Williams, Product Development, Electro-Motive Division of General Motors Corp., January 24, 1975.

[6]Hare, C. T., K. J. Springer, and J. A. Huls, Locomotive Exhaust Emissions and Their Impact, ASME Paper 74-DGP-3, 1974.

[7]Bascom, R. C., L. C. Broering, and D. E. Wulfhorst, Design Factors that Affect Diesel Emissions, 1971 SAE Lecture Series, 1971.

[8]Walter, USCG Pollution Abatement Program.

to nationwide gaseous emissions. However, in-port visibility of exhaust smoke is a cause of public and Coast Guard concern, and smoke reduction techniques are therefore pertinent to this engine application.

1.2 Objectives

Principal objectives of Phase I of this study were as follows:

a. Identify existing methods; including operating procedures and equipment modifications and adjustments, to reduce exhaust emissions and improve thermal efficiency of diesel engines with individual cylinder displacements greater than 150 in^3.

b. Define duty cycles of these engines in their principal applications in order to establish representative test modes.

c. Determine thermal efficiency and emission measurement procedures which are applicable to these engines and produce emissions data that are compatible with data obtained by established Environmental Protection Agency procedures for diesel engines.

d. Evaluate the effects of engine wear and maintenance on fuel consumption and emission levels.

e. Determine how properties of ingested air influence fuel consumption and emissions.

The investigation of these topics was to yield candidate fuel consumption and emission reduction techniques that would be evaluated in a series of laboratory and field tests in Phase II of the program.

1.3 Approach

Work performed in Phase I was divided into three areas: data collection, data analysis, and preparation of recommendations for Phase II.

Data collection involved personal interviews with manufacturers and users of the engines of interest, as well as an extensive search of published technical literature. Five manufacturers were interviewed: Alco Engines Division of White Industrial Power, Inc., Electro-Motive Division of General Motors Corp. (EMD), Enterprise

Engine and Compressor Division of Delaval Turbine, Inc., Fairbanks Morse Engine Division of Colt Industries, Inc., and General Electric Diesel Engine Products Department of the General Electric Company (G.E.). Users of these engines were represented by railroads and the U.S. Coast Guard, and interviews were conducted with Missouri Pacific Railroad Company, Southern Pacific Transportation Company, and engineering staffs of WHEC 378 Chase, WHEC 378 Hamilton, and WMEC 210 Decisive.

Data analysis ranged from merely summarizing the available information, to lengthy and involved application of analytical techniques to derive complete engine maps for fuel consumption and emissions from partial data. These and other analytical methods were used to estimate the effect of cylinder wear on performance and emissions, an area where data for these engines were especially lacking.

Preparation of recommendations involved estimating the relative importance of each possible work item for Phase II. The list of recommendations at the end of the report includes those items which, on the basis of this initial investigation, appear to be promising candidates for evaluation or are tasks which must be accomplished before adequate evaluation can take place.

1.4 Comments

A copy of the first draft of this report was sent to each of the five engine manufacturers and two railroads interviewed during the project. A copy was also sent to the Association of American Railroads (AAR). Each organization was requested to review the draft report and comment on its contents.

Several changes were made in the draft report as a result of the comments generated by this review. The decision to change any part of the text was based on a desire to clarify certain points and to make the contents as accurate as possible.

2. OPERATING MODES AND DUTY CYCLES

This section presents a discussion of typical modes of operation of the engines of interest and the results of a study of existing time-based duty cycles for these engines in their principal applications. As such, the section serves as necessary preparation for the analysis of fuel conservative operating procedures that follows in Section 5.0.

2.1 Operating Modes

2.1.1 Diesel-electric Locomotive Application

The large medium-speed engines used in diesel-electric locomotives operate in a manner that is quite different from engines in highway trucks and farm and construction equipment. In these latter applications, engines are usually equipped with so-called full-range governors that permit engine operation throughout the recommended speed range (from approximately peak torque to rated speed) at any load between zero and maximum. Hence, these engines can operate at an infinite number of speed-load settings within their design limits.

Locomotive engines, on the other hand, operate in a discreet number of throttle positions or "notches", each of which corresponds to a unique engine speed and power output. There are eight power notches and an idle notch. Many line-haul locomotives are equipped with a dynamic brake mode in which the engine operates at a predetermined "notched" speed (specified by the engine manufacturer), but at a light load, sufficient to drive cooling fans and excite the electrical fields of the locomotive wheel traction motors. The dynamic braking system is activated by a selector switch rather than by the throttle.

All EMD and most G.E. locomotives utilize eight engine speeds to obtain the eight power outputs, i.e., a different speed for each power. However, late model G.E. units use just two engine speeds to produce the same power outputs that were formerly obtained with the eight-speed throttle schedule. Hence, eight power notches are still employed. The two speeds of the new throttle schedule are joined with the same low idle speed to yield a three-speed schedule. (The reason behind this change will be discussed in the section dealing with emission reduction techniques.) Both G.E. schedules and that of EMD are presented in Table 2.1.

TABLE 2.1 TYPICAL LOCOMOTIVE OPERATING THROTTLE SCHEDULES

Throttle Position	EMD[9] Nominal Engine Speed, rpm	EMD[9] % Nominal Rated BHp	G.E.[10] Nominal Engine Speed, rpm 8-speed	G.E.[10] Nominal Engine Speed, rpm 3-speed	G.E.[10] % Nominal Rated BHp
Idle	315	1.0*	450	450	1.0*
1	315	5.0	450	790	4.0
2	395	12.0	535	790	10.0
3	480	23.0	620	790	20.0
4	560	35.0	705	1050	30.0
5	645	51.0	790	1050	48.0
6	730	66.0	880	1050	65.0
7	815	86.0	965	1050	82.0
8	900	100.0	1050	1050	100.0
Dynamic Brake	645	3.0*	1050	1050	6.0*

*Auxiliary Load Only

It should be noted that the engine horsepower developed for each throttle notch is, for all practical purposes, a constant that is independent of the load demand on the engine. That is, the load control is designed to deliver a (different) preset quantity of fuel for each position of the throttle schedule. If the load demand on the engine exceeds the fixed available horsepower for that throttle notch, as in the case where a train begins to climb a grade, the load control reduces the demand to match the available power, and the train slows down. To increase train speed requires use of a higher notch where more horsepower is available. This load control system allows locomotives in a multiple-unit arrangement to operate in a like manner and also insures that the engine(s) remain loaded to an efficient level regardless of train speed[11].

[9]Ephraim, Max, Jr., Status Report on Locomotives as Sources of Air Pollution, SAE Paper 720604, 1972.

[10]Hoffman, J. G., Jr., K. J. Springer, and T. A. Tennyson, Four Cycle Diesel Electric Locomotive Exhaust Emissions: A Field Study, ASME Paper 75-DGP-10, 1975.

[11]Ephraim, Status Report on Locomotives.

Another point worth noting is that locomotive diesel engines operate at manufacturer-specified maximum Brake Mean Effective Pressure (BMEP) only at rated speed (Notch 8). Therefore, in all throttle positions except Notch 8 the engine operates at a fixed, partial load condition. While most locomotive engines thus operate at full load at rated speed, some engines that operate in situations where air density is much below normal (as in mountain and/or tunnel operation) are equipped with a load control feature that reduces the amount of fuel delivered to the engine in a given notch position. This fuel-limiting feature is designed to limit exhaust temperature and, hence, to protect exhaust valves and the turbocharger.

2.1.2 Reduction Gear-Diesel Drive Coast Guard Application

Some of the engines of interest are utilized as main propulsion power plants in Class 210 B medium-endurance cutters (designated WMEC) and Class 378 high-endurance cutters (WHEC). Engines in these vessels also operate only in distinct modes that are determined by preselected throttle settings. Modes are defined by engine speed and propeller pitch. This latter quantity is programmed into the throttle control so that a given engine speed is always associated with the same propeller pitch.

Engines in this application are equipped with governors that maintain the preselected speed for a particular throttle position so long as the load required does not exceed the maximum engine capacity at that speed. If load demand exceeds this maximum figure, then propeller pitch is reduced until the load demand equals the power available at that engine speed. In other words, variable fuel rate (or power) at constant engine speed is possible with this type of load control system. However, the load does not vary from zero to maximum; rather, it remains in a fairly narrow band about some nominal power output that is required to achieve the desired propeller speed under average vessel operating conditions (wave height, wind velocity, draft, and so forth). Changes in these conditions will increase or decrease the required engine power output relative to this nominal value, and the engine will respond accordingly.

Typical throttle schedules for engines in these two cutter classes are shown in Table 2.2. It should be noted that engines in WHEC vessels actually have continuous throttle controls and, therefore, can be operated at conditions that lie between those shown in the Table, though this is not done in normal practice.

TABLE 2.2 TYPICAL COAST GUARD CUTTER OPERATING THROTTLE SCHEDULES

WMEC		
Throttle Position	Engine Speed, rpm	Prop. Pitch, % of Max.
Idle	300	-
1	415	58
2	498	79
3	580	100
4	664	100
5	747	100
6	830	100
7	913	100
8	1000	96

WHEC		
Throttle Position	Engine Speed, rpm	Prop. Pitch, % of Max.
Idle	300	-
1/3	453	35
2/3	574	100
Std.	755	100
Full	815	100
Flank	900	100

WAGB		
Throttle Position	Engine Speed, rpm	Propeller Pitch
Idle	350	-
1/3	420	Constant
2/3	550	Constant
Std. -Full	720	Constant
Heavy Ice	810	Constant

2.1.3 Diesel-Electric Coast Guard Application

Diesel-electric propulsion systems that utilize large medium-speed engines are found in icebreakers (designated WAGB). Engine operating modes are either typical of engine-driven generator sets (i.e., almost constant speed and variable load) or utilize the variable speed and load throttle schedule shown at the bottom of Table 2.2. All icebreakers are equipped with fixed pitch propellers; therefore, operating modes are defined in terms of horsepower output only (in the case of constant engine speed operation) or in terms of engine speed and horsepower. With either type of operation the desired propeller shaft speed (or vessel speed) determines the horsepower required of the engine.

2.2 Duty Cycles

The duty cycle of an engine is the fraction of time (relative to some total time) that the engine spends in each operating mode under consideration. The "total time" is usually taken as the total operating time of the engine. Each operating mode included in the duty cycle should be one that is typically encountered in the engine's normal service. It should be noted that duty cycles are concerned only with steady-state engine operation and not with any transient aspects of actual engine usage.

2.2.1 Locomotives

Since locomotive engines typically operate only in fixed throttle positions or notches, their duty cycles are particularly simple to study and define. Data in the form of cumulative elapsed time for each notch is obtained by a recording device; these data are then analyzed to generate a set of weighting factors, one factor per notch. These factors are used to calculate so-called cycle composite values of gaseous emissions and fuel consumption. Such composite values usually take the form of brake specific data. Each set of weighting factors can also be used to determine the most significant speed-load conditions for engines in locomotive application and, therefore, can assist the engine builders in design and development work.

Several duty cycle studies have been conducted by cooperative efforts between the engine manufacturers and various railroads or by railroads alone. The results of these studies are presented in Table 2.3, along with a compromise schedule proposed several years ago by the Association of American Railroads (AAR). All but one of these cycles are for line-haul or road operation; the exception is the cycle for switch engine service compiled by the Atchison, Topeka and Santa Fe Railway (ATSF).

Several points concerning these weighting schedules should be made. First, the G.E. schedules shown in Table 2.3 constitute the minimum and maximum values obtained during their studies and two average schedules that were presumably derived from analysis of the "Min" and "Max" values. The average duty cycles are quite similar; however, recent tests[12] of in-service locomotives sponsored by G.E. have utilized the "First Average" schedule. Second, the two EMD cycles represent an upper (or "High") and an average (or "Medium") amount of Notch 8 utilization. EMD uses the "High" schedule for establishing engine reliability performance, while the "Medium" schedule is used to determine cycle composite fuel consumption and emissions levels of line-haul locomotives[13]. Third, the ATSF "High" and "Medium" schedules also represent an upper and average level of Notch 8 usage. The switch duty cycle is, of course, heavily weighted towards idle and the low-power notches; in fact, there is essentially no time allotted to operation in Notches 7 and 8. Finally, as mentioned previously, the AAR duty cycle is apparently a suggested compromise between the "High" EMD schedule and one of the G.E. schedules and is not the result of an actual study.

As far as is known, these ten weighting schedules represent the current state of knowledge concerning domestic locomotive operation, although at least one manufacturer continues to obtain duty cycle data. There is, in general, good agreement between several of the schedules, e.g., the "First Average" G.E., the "Medium" EMD, and both ATSF line-haul cycles. The greatest difference among them is the percentage of time allotted to idle. These differences may be due to the interpretation of "total" engine operating time, i.e., whether it is the actual total time the engine is running or this total time <u>less</u> the time that the unit is unavailable for revenue operation, such as time spent awaiting routine maintenance or service (fueling, loading with sand, and so forth). More likely, however, the difference in idle time is the result of different types of revenue service and the varying work patterns of the locomotives involved in the duty cycle studies. In any case, it is evident that idle time is significant even for line-haul service, since the lowest idle weighting factor is 40 percent.

A given set of weighting factors can be used to analyze the relation between fuel consumption of any given mode and the

12 Hoffman, <u>Four-Cycle Diesel Emissions.</u>

13 Private Communication to C. D. Wood from H. A. Williams.

TABLE 2.3 LOCOMOTIVE DUTY CYCLES

Throttle Position	Weighting Factor Schedules, % of Operating Time									
	G.E. (14)				E.M.D. (15)		A.S.T.F. (16)			A.A.R. (17)
	Min.	Max.	1st Avg.	2nd Avg.	High	Medium	High	Medium	Switch	
Idle	59.0	40.0	54.0	53.0	41.0	46.0	46.0	59.0	77.0	43.0
1	6.5	2.5	5.0	5.1	3.0	4.0	5.0	5.0	10.0	3.0
2	6.5	2.5	2.5	3.9	3.0	4.0	3.0	4.0	5.0	3.0
3	6.5	2.5	2.0	3.4	3.0	4.0	3.0	3.0	4.0	3.0
4	6.5	2.1	5.0	3.3	3.0	4.0	3.0	2.0	2.0	3.0
5	2.9	1.7	2.0	2.8	3.0	4.0	2.0	2.0	1.0	3.0
6	2.9	1.7	2.0	3.4	3.0	4.0	3.0	2.0	1.0	3.0
7	2.5	1.8	2.5	2.6	3.0	4.0	2.0	1.0	nil	3.0
8	5.2	38.0	21.0	17.0	30.0	17.0	24.0	20.0	nil	28.0
Dynamic Brake	1.5	7.0	4.0	5.5	8.0	9.0	9.0	2.0	*	8.0

*Switch engine not equipped with dynamic braking.

14 Hoffman, Four Cycle Diesel Emissions.

15 Private Communication to C.D. Wood from H.A. Williams.

16 Bryant, A.H., and T.A. Tennyson, Exhaust Emissions of Selected Railroad Diesel Locomotives, ASME Paper 74-WA/RT-1, November 1974.

17 Exhaust Emissions from Uncontrolled Vehicles and Related Equipment Using Internal Combustion Engines, Part I - Locomotive Diesel Engines and Marine Counterparts, Southwest Research Institute, Environmental Protection Agency Report APTD-1490, October 1972.

total fuel consumed in all operating modes. For example, using modal fuel consumption rates (either mass or volume) for a typical 16-cylinder turbocharged locomotive engine -- and assuming that the engine spends 40 percent of its operating time at idle, 20 percent in Notch 8, and the remaining 40 percent equally distributed in Notches 1 through 7 -- it is easily calculated that (on a mass or volume basis) some 4 percent of the total fuel consumption occurs at idle, about 56 percent occurs in Notch 8, and the remaining 40 percent of the fuel is consumed in Notches 1-7.

A similar calculation, using modal fuel consumption rates for a switcher locomotive engine and the weighting factors of the ATSF switch duty cycle, reveals that about 45 percent of total fuel consumption occurs at idle and the remaining 55 percent is more or less evenly distributed in Notches 1-6 (no weight is attached to Notches 7 and 8).

The effect of the various locomotive weighting schedules on cycle composite brake specific calculations is worth noting. Table 2.4 contains brake specific emission and fuel consumption values calculated with several of the line-haul schedules under discussion. The "raw" data (emission concentrations, engine horsepower and fuel consumption figures) used in these calculations are similar to test data obtained during a locomotive test conducted by Southwest Research Institute.

TABLE 2.4 EFFECT OF LOCOMOTIVE WEIGHTING FACTOR SCHEDULE ON CYCLE COMPOSITE VALUES

Quantity	G.E. 1st Avg.	G.E. 2nd Avg.	EMD High	EMD Medium	ASTF Medium
$BSNO_x$ (g/bhp-hr)	8.5	8.5	8.2	8.4	8.6
BSHC (g/bhp-hr)	0.7	0.7	0.6	0.7	0.7
BSCO (g/bhp-hr)	4.1	4.4	3.8	4.5	4.0
BSFC (lbm/bhp-hr)	0.39	0.40	0.38	0.39	0.39

	Max. Value	Min. Value	% $\frac{\text{max.}}{\text{min.}}$
$BSNO_x$ (g/bhp-hr)	8.6	8.2	5
BSHC (g/bhp-hr)	0.7	0.6	17
BSCO (g/bhp-hr)	4.5	3.8	18
BSFC (lbm/bhp-hr)	0.40	0.38	5

The most important thing to notice about the values in Table 2.4 is that the variation for a particular quantity (NO_x, CO, and so forth) is generally slight; in fact, this amount of variation could easily be encountered in a series of consecutive tests with the same locomotive. Therefore, for this particular locomotive test, the brake specific cycle composite values are relatively insensitive to differences in weighting factor values. However, from this statement it should not be concluded that any weighting schedule will suffice, since at least the general trend of the duty cycle should be accurately defined in order to permit a realistic interpretation of test results. In these terms, it appears that such accurate definition has been accomplished for domestic line-haul locomotive operation.

2.2.2 Coast Guard Vessels

No definitive study of duty cycles for Coast Guard vessels has been undertaken by either engine manufacturers or the Coast Guard. However, from interviews of engineering personnel on medium- and high-endurance cutters it has been determined that engines in these vessels may spend a considerable amount of time at idle and almost no time at maximum (rated) speed and load. The available information is, of course, inadequate to allow a more precise analysis, or one that would extend to WMEC or WHEC vessels in general. Even less is known about duty cycles of diesel-electric-drive icebreakers since no interviews were conducted with personnel from these vessels.

It is essential that duty cycles for the Coast Guard vessels of interest be defined in at least a general quantitative sense so that effects produced by changes in engine equipment adjustment can be adequately evaluated. For instance, any such change that produced a favorable effect on emission levels or fuel consumption in operating modes where little time was spent, and a detrimental effect in modes that were heavily weighted, should obviously be avoided. The highly qualitative remarks of the previous paragraph concerning operational trends of engines in Coast Guard cutters are not adequate for this purpose.

Acquisition of duty cycle data for the vessels of interest could be obtained by two methods. First, medium- and high-endurance cutters, whose engines operate in distinct throttle positions only, could be equipped with recording devices similar to those used on locomotives. Diesel-electric propulsion systems on board icebreakers would require devices to record generator output, propeller shaft speed or throttle position, whichever is most convenient. Second, engine room personnel could be asked to keep a record of all bell order changes and the time that these changes occur. (Such records are already kept on high-endurance cutters, but not on medium-endurance vessels. It is thought that such records are probably not kept on icebreakers because of the rapid change in operating modes that takes place during ice breaking operation.)

Regardless of the method used to obtain duty cycle data, several vessels of each class should be included in the study in order to broaden the data base, and data should be taken for a period of time that is long enough to minimize the effect of short-term fluctuations in vessel operating characteristics.

3. FUEL CONSUMPTION AND EXHAUST EMISSION REDUCTION TECHNIQUES

Manufacturers of large medium-speed diesel engines have expended much effort in the investigation of methods to improve efficiency (fuel consumption) and emissions characteristics of their products. Since the basic designs of these engines change very slowly with time, most investigations are in the area of component modification and improvement, rather than in drastic redesign of the entire engine. The improved components are almost always suitable for retrofit to existing, older engines in field service. Indeed, the improved component is often the only one manufactured and supplied to engine users. Therefore, as existing engines are rebuilt they tend to be upgraded by use of the newer parts.

This section of the report will summarize these component areas and the current status of the improvements, with an eye towards possible retrofit and modification of existing engines. However, the section begins with a brief discussion of basic causes of diesel engine emissions in order to provide the reader with an understanding of the influence of various reduction techniques on the emission constituents.

3.1 Diesel Engine Emissions

The primary emissions produced by diesel engines are smoke, NO_x, HC, and CO. Smoke is ranked first in spite of the fact that it is generally recognized as being a nuisance and not a potential health hazard. Oxides of nitrogen are the only gaseous diesel emission whose level approaches that produced by gasoline engines; NO_x reduction is therefore given considerable attention by government and industry groups. Production of HC and CO by diesel engines is generally low (compared to that of gasoline engines) and has been reduced to even lower levels by several techniques that will be discussed later in this section.

First, however, consider the causation factors for these diesel emissions, beginning with exhaust smoke. Diesel smoke is conventionally categorized as white (or blue) smoke, or black smoke. The former is considered to be primarily unburned fuel, or fuel only partially reacted and still substantially in the liquid state. The latter consists primarily of carbon particulates in the solid form.

White smoke is associated with the absence of combustion in a substantial part of the fuel charge and generally indicates a malfunctioning engine: grossly incorrect injection timing, poorly atomized fuel spray due to injection system malfunction, loss of compression, overcooled chamber walls, and so forth. It is also observed during starting of the engine or during prolonged idling. Hence, white smoke is the result of either an engine malfunction or a particular operating mode, and is not due to engine design. Therefore, white smoke is best avoided by proper engine maintenance and/or by avoiding operating modes characterized by low speed, low temperature operation.

Black smoke indicates a partial combustion process in which the fuel is stripped of its hydrogen, leaving carbon. This process occurs in regions where inadequate oxygen is available for a complete reaction, and its presence indicates poor mixing of the fuel and the available air, or lack of air. In these terms, then, black smoke is definitely a function of engine design, particularly in the areas that influence fuel injection, combustion (chamber or piston shape), and air motion or swirl.

However, it should be noted that all diesel engines inevitably produce excessive black smoke if the fueling rate is high enough (due to the decline of air availability), and the increase in smoke with fuel/air ratio is normally at a rate greater than a linear increase. In this context, black smoke can become excessive because of a malfunction or a change in operating mode that increases fuel/air ratio.

NO_x formation is a function of combustion temperature and oxygen availability (two factors that are in turn determined by fuel/air ratio) and of the time that conditions favorable to NO_x formation exist in the cylinder. In a given engine at constant speed, NO_x increases almost linearly with fuel/air ratio up to a point where sufficient fuel is being injected to consume most of the available oxygen (i.e., a fuel-rich condition exists in the cylinder). Beyond this point, NO_x concentration may level off or even decline. It should be noted that NO_x formation is intimately related to factors that, in part, determine the combustion efficiency of any given diesel engine; that is, maximum formation of NO_x usually occurs when the engine has attained maximum efficiency and minimum brake specific fuel consumption.

Concentration of unburned hydrocarbons in diesel exhaust is determined more by design and performance of the injection system, and by combustion chamber geometry, than by operating variables such as engine speed and load. High HC emission results when fuel droplets which are too large to be vaporized and burned (almost) completely, are introduced into the cylinder. Hence, fuel spray geometry and the amount of fuel remaining in the injector nozzle after injection is completed are most important in controlling HC formation.

CO concentration is a function of the same factors that influence black smoke. Therefore, it is generally true that smoke and CO change together, in the same direction. The exception to this similarity in behavior involves retarded injection timing, which increases smoke opacity, but has little or no effect on CO due to the fact that the fuel/air ratio remains the same when timing is changed.

3.2 Improved Engine Components and Operation

3.2.1 Fuel Injectors

Fuel injectors constitute the heart of the engine, and their design can have a tremendous effect on both efficiency and the formation of exhaust emissions. In this regard, the most critical part of the injector is the spray tip, which determines the number and location of the spray plumes in the combustion chamber and, in conjunction with some other injection parameters, influences the shape of the plumes and the degree of fuel atomization. Therefore, it is not surprising that tip design has received so much attention from engine manufacturers. The greatest change in tip design has been in the reduction of sac volume, i.e., the volume contained between the needle valve or plunger and the spray tip. It was found[18] several years ago that the uncontrolled quantity of fuel in this space was a major contributor to unburned hydrocarbon emission and black smoke from an otherwise well-developed diesel engine. The effect of the new tip design on other emission constituents has generally been reported to be a decrease in carbon monoxide (CO) and oxides of nitrogen (NO_x), although in some cases NO_x has increased slightly due, probably, to an increase in combustion efficiency and the resulting increase in gas temperature. Fuel consumption, of course, benefits from any such increase in efficiency.

These so-called low sac injectors are available for most of the subject engines (though some manufacturers are still engaged in development work) and are a relatively simple and inexpensive retrofit

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[18] Ford, H.S., D.F. Merrion, and R.J. Haines, Reducing Hydrocarbons and Odor in Diesel Exhaust by Fuel Injector Design, SAE Paper 700734, 1970.

item if retrofit is accomplished by replacement of worn old-style injectors during normal maintenance. This type of replacement policy is stated in the U.S. Coast Guard Naval Engineering Manual (CG-413) and is followed by most railroads.

3.2.2 Governors

The governors employed on this class of engines are precision devices to control engine speed and/or fuel delivery rate under steady-state operation, and ideally, fuel delivery under transient conditions. The most common governor inexactitude manifests itself as overfueling during acceleration of an engine that uses a free-running (i.e., exhaust gas-driven) turbocharger, thus causing the familiar smoke puffs during the time it takes the turbocharger to increase its speed (and, hence, air flow to the engine) and to restore the proper fuel/air ratio. It should be noted that the two-stroke turbocharged locomotive engine utilizes a turbocharger that is gear driven at low and midrange engine speeds where exhaust gas energy is relatively low and turbocharger lag time is highest.

Manufacturers of four-stroke turbocharged engines use various methods to control fuel delivery during transient operation. The most common method is to use a device, often termed a rack delay, that "senses" either turbo speed or boost pressure and maintains a restricted fuel rack position until the pertinent parameter assumes its nominal steady-state value. A sometimes undesirable side effect of a rack delay is to increase the response time of the engine and to introduce a degree of sluggishness into the operation of the locomotive or vessel.

Older governors that are scheduled for changeout can be replaced with new units that have the rack limiting feature. Or, governors without this feature can often be modified during overhaul with parts that limit rack travel during transient operation. One engine manufacturer reports(19) that, in field tests of this governor modification, acceleration smoke puffs were reduced from an approximate Ringlemann 4 rating lasting for 15 seconds to a Ringlemann 2 rating of two seconds duration.

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[19] Bellis, Max W., What's Your Problem?, Railway Locomotives and Cars, May 1973.

3.2.3 Turbochargers

Some well-developed diesel engines are equipped with turbochargers that are matched to the particular engine and its application. This matching process consists of selecting a turbocharger that provides sufficient steady-state air flow throughout the operating range of the engine and possesses good transient response characteristics in order to minimize acceleration smoke puffs and response time. The need for a strongly-damped rack delay is obviated as turbo response time is decreased, since acceleration smoke can still be adequately controlled.

Therefore, engine manufacturers -- whether they make their turbochargers or obtain them from a supplier -- have placed considerable emphasis on obtaining the best possible match between engine and turbocharger. It is possible that older engines could benefit from use of newer matching concepts. One marine engine company has developed a retrofit kit consisting of an exhaust manifold and turbocharger arrangement that is said to virtually eliminate acceleration smoke. However, such kits are very expensive (about $20,000 per engine for parts alone), and retrofit would be economically feasible only in the course of normal engine overhaul or replacement of a failed turbocharger. Besides lower acceleration smoke, retrofit of an improved turbocharger could also result in lower steady-state smoke, CO, and fuel consumption. Oxides of nitrogen might well be higher if combustion efficiency is markedly increased.

3.2.4 Pistons and Cylinder Liners

Most diesel engines have the combustion chamber in the top of the piston; hence, piston design is, together with injector design, all-important in determining combustion efficiency. Accordingly, several engine manufacturers have evolved piston designs that reportedly reduce exhaust smoke by achieving better mixing of air and fuel spray, and any reduction in smoke density usually means that CO emission is also reduced. Levels of NO_x emission have likely not been reduced by the new piston designs.

Cylinder liner design of two-stroke cycle engines also exerts a strong influence on engine performance and emissions since the liner air ports determine the amount and swirl direction of the scavenging/charging air flow admitted into the cylinder. Manufacturers of these engines offer liners of improved design that can be retrofitted to older engines. These new liners are teamed with the improved pistons to obtain maximum benefit from both.

The remarks made above concerning effects of improved piston design on fuel consumption and emissions are generally applicable here, too.

It should be noted that older EMD two-stroke locomotive engines with a displacement of 567 in^3 per cylinder can be rebuilt with new piston/liner assemblies featuring the above design improvements and a displacement of 645 in^3 per cylinder. New fuel injector tips of low sac design are also installed at this time. This modernization of older engines results in the advantages of reduced fuel consumption and emissions that are inherent in use of the new components. However, cost of this conversion is high, with parts costs alone for a 16-cylinder engine amounting to $8,000 to $18,000, excluding necessary modifications to the electrical system if the rated power output of the engine is increased over the former 567 rating.

3.2.5 Injection Timing

No method of emission control for diesel engines has received more attention than that of injection timing. Current large engine practice is to select the timing that results in minimum fuel consumption at rated engine speed, without regard for exhaust emission levels. Exhaust smoke is then controlled at this timing by the improvements in injectors, turbocharger, and combustion system that have been previously discussed. There is no doubt that this approach has produced a generation of diesel engines that has excellent fuel consumption and visible smoke characteristics. Emissions of HC and CO have likewise benefited, while NO_x remains the most persistent (and resistant) diesel emission constituent.

Retarded timing appears, at first glance, to be a cost-effective method for reducing NO_x. The modification is relatively simple to perform and requires that no other changes be made to other engine components as long as the amount of retard is moderate, say, 4° or less relative to the standard setting. In one laboratory study(20) conducted with a two-cylinder revision of a non-turbocharged two-stroke locomotive engine, and in the case of stationary (trackside) tests of a four-stroke locomotive engine(21), retarded timing of this magnitude resulted in a 10 to 25 percent reduction in brake specific NO_x for the line-haul locomotive duty cycle.

[20] Storment, J.O., K.J. Springer, and K.M. Hergenrother, NO_x Studies with EMD 2-567 Diesel Engine, ASME Paper 74-DGP-14, April 1974.

[21] Hoffman, Four Cycle Diesel Emissions.

However, the other effects of retarded timing have been to increase fuel consumption, smoke opacity, and, in some cases, CO. HC has generally not been affected. The reported increase in fuel consumption has been between one and four percent, and there is at present no known way to avoid this penalty. Increases in smoke and CO are moderate if the engine in question is equipped with low sac injectors and other improvements; otherwise, retarded timing can cause a high smoke situation.

A further concern of engine manufacturers and users is that retarded timing may adversely affect engine durability in two ways. First, if the quality of diesel fuel is lowered, the rate of deposit formation in the cylinder could increase substantially and lead to much higher wear rates of piston rings and liners. (Smoke opacity would also increase under the combination of retarded timing and poor fuel quality, but this is not a durability problem.) It is feared that such a reduction in fuel quality is almost a certainty in the event that crude oil supplies are again restricted. Second, it is thought that the higher exhaust temperature that results from retarded timing will impose extra thermal stress on critical engine components. The effects of these added stresses would most likely be long term in nature and would be aggravated by severe conditions encountered in field operation.

Some engine manufacturers, therefore, plan long-range field and/or laboratory evaluations to determine the penalty incurred in fuel consumption and the magnitude of mechanical degradation caused by retarded timing. Both principal manufacturers of locomotive engines are now, or soon will be, engaged in such evaluations. The results should do much to determine the true cost/benefit ratio of this emission control technique.

3.2.6 Other Engine Modifications

At least one railroad equips some of their EMD line-haul units with a special exhaust manifold with four outlets or stacks rather than the standard two stack arrangement. This modification can only be performed on locomotives that have non-turbocharged (Roots blown) engines and that are not equipped with dynamic braking hardware. This latter factor involves the space available above the engine for the extra stacks. Further, all turbocharged engines have only one exhaust outlet, and the subject modification does not apply to them.

For those engines that can be so modified, the change is said to result in lower exhaust backpressure and to permit better scavenging and charging of cylinders, expecially at high altitude. Maximum power at high altitude is reportedly increased by 200 BHp (or 10 percent), with better fuel consumption figures. Exhaust gas temperature is also reduced and the life of valves and cylinder heads is extended. While this modification may be cost-effective for locomotives that operate in mountainous terrain, it has not been determined if the change is equally worthwhile for units that operate only at low altitude. Also, no cost figure for the modification was available. Apparently, older locomotives can be retrofitted with the required hardware and new locomotives can be ordered with it. The hardware must, however, be compatible with the exhaust spark arresters that some railroads are required by law to use.

One engine manufacturer whose engines are used mainly in marine and stationary applications offers a special "kit" option for engines that are required to idle for long periods of time. The kit consists of special camshaft and injectors, together with an engine coolant heater to maintain the water jackets at normal operating temperature. The manufacturer maintains that engines so equipped are not subject to many of the harmful effects usually associated with prolonged idle operation (see Section 5.1) and that performance throughout the speed-load range is not compromised. The apparent reason why all engines sold by this company are not equipped with these items as standard equipment is that engine cost is increased by the kit and many engine users do not engage in extended idle as a normal practice. The exact cost of this option was not available, but was described as "high". And it is usually more expensive to retrofit an older engine than to order a new engine so equipped.

3.2.7 Changes in Engine Operation

Several changes in locomotive engine operation have been implemented or proposed to reduce smoke or fuel consumption. The most drastic change to date involves the adoption by G.E. of an eight-notch throttle schedule based on two engine speeds rather than the former eight speeds (Section 2.1.1). The new throttle schedule came about when G.E. engineers found that their engine produced higher steady-state smoke at low speed-low power notches than at high speed-high power conditions. The solution was to keep the same horsepower outputs for the eight notches, but to operate the engine at higher speed: Notches 1 through 3 are at the former Notch 5 speed, and the remaining notches are at Notch 8 (rated) speed. The change to just two engine speeds for the power notches also reduced acceleration smoke, CO and NO_x (slightly)[22]. Flexibility of the locomotive and overall fuel consumption are said to be unaffected.

A recommendation has been made by G. E. concerning locomotive throttle usage during transient operation of engines with governors that have not been modified to reduce acceleration smoke. It is proposed that sudden throttle sweeps covering several notch positions be avoided, but rather that the throttle be delayed in each notch for two or three seconds. Peak smoke opacity during acceleration has been reduced by up to one-half by this operating change[23]. This operating change could be applied to any engine equipped with a gas-driven turbocharger.

The large amount of idle time accumulated by locomotive engines has been mentioned previously. EMD is currently evaluating a lower idle speed for their engines that reportedly reduces idle fuel consumption by about 25 percent. On a cycle composite basis, the reduction would be about 10 percent for a switch engine and less than one percent for a line-haul engine. The evaluation of this change will include a lengthy field test to determine all pertinent ramifications.

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[22]Hare, Locomotive Exhaust Emissions.

[23]Hoffman, Four Cycle Diesel Emissions.

4. ANALYTICAL DERIVATION OF FUEL CONSUMPTION AND EXHAUST EMISSION DATA

The modes of operation of locomotive and Coast Guard prime movers have been discussed in Section 2. An important objective of the project was to investigate possible fuel conservative operating procedures that involved changes in these operating modes. To thoroughly analyze the effect of these changes required not only complete fuel consumption maps, but also maps of exhaust smoke opacity and oxides of nitrogen (NO_x), the two primary diesel exhaust emissions; i. e., these maps could be used to determine the effects of any contemplated change in operating modes upon both fuel consumption and emissions.

However, it was soon evident that such maps would, in general, not be available to the project. Indeed, an intensive literature search and interviews with the engine manufacturers produced only one fuel map and no emissions maps at all. Some fuel consumption and emission data were, however, obtained from published technical reports(24, 25, 26, 27) and from the manufacturers. This lack of fuel consumption and emissions data in "map" format made it necessary to develop methods to expand the available data into this form.

This section of the report outlines the type and extent of the fuel consumption and emission data that were obtained and explains how these data were extended into complete maps. The section thus does not directly pertain to the presentation of a stipulated project work requirement. However, the work described here constituted a major project effort. Further, it is essential that all persons who read and/or make use of this report and its conclusions be made aware of the assumptions involved in the generation and use of these engine maps.

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[24] Exhaust Emissions from Uncontrolled Internal Combustion Engines, Part I, Southwest Research Institute, E. P. A., Report APTD-1490.

[25] Hoffman, Four Cycle Diesel Emissions.

[26] Bryant, Emissions of Selected Locomotives.

[27] A Study of Stack Emissions from Coast Guard Cutters, Department of Transportation, U.S. Coast Guard Office of Research and Development, Report No. CG-D-13-73, September 1973.

It was decided to consider only turbocharged models of the engines of interest. This decision was based on the fact that most of the data on hand were for this type of engine and that turbocharged engines are most numerous in the applications under consideration. Indeed, some manufacturers do not even make a naturally aspirated version of their engine. It was also decided that fuel consumption and emissions maps would be generated for three basic engine designs: four-stroke cycle, conventional two-stroke cycle, and opposed-piston two-stroke cycle. Thus, any differences in fuel maps due to the influence of fundamental engine design would be noted, but it would not be necessary to consider the effect on the fuel map of relatively minor design differences.

4.1 Fuel Consumption Maps

It was desirable to have a map of brake specific fuel consumption (BSFC) that extended from low to maximum load over the normal speed range of the engine. The BSFC data on hand were, however, for the usual engine operating modes, such as locomotive throttle notch positions or variable speed-variable torque points along a propeller curve. Figure 4.1 shows the typical available data (top) and the desired extended data (bottom).

A complete BSFC map for a particular turbocharged engine can be generated if values of certain parameters (e.g., thermal efficiency, turbo boost pressure, and so forth) are known for the operating range of that engine. As mentioned previously, BHp and BSFC for one power output at each of several speeds was usually known. However, boost pressure and intake manifold or air box temperature were usually available only for maximum power at rated speed. No actual data were available on engine friction mean effective pressure (FMEP) or thermal efficiency. Therefore, several important assumptions were necessary, as was the use of some data that are valid for diesel engines in general. These assumptions and generalized data were then used to establish relationships between FMEP and piston speed, thermal efficiency and fuel/air ratio, and between boost pressure and engine speed. These relationships will be discussed separately in the following paragraphs.

The friction mean effective pressure (FMEP) is a function of various power losses associated with an engine and is expressed as FMEP = IMEP - BMEP, where IMEP (indicated mean effective pressure) includes the work of compression and expansion strokes

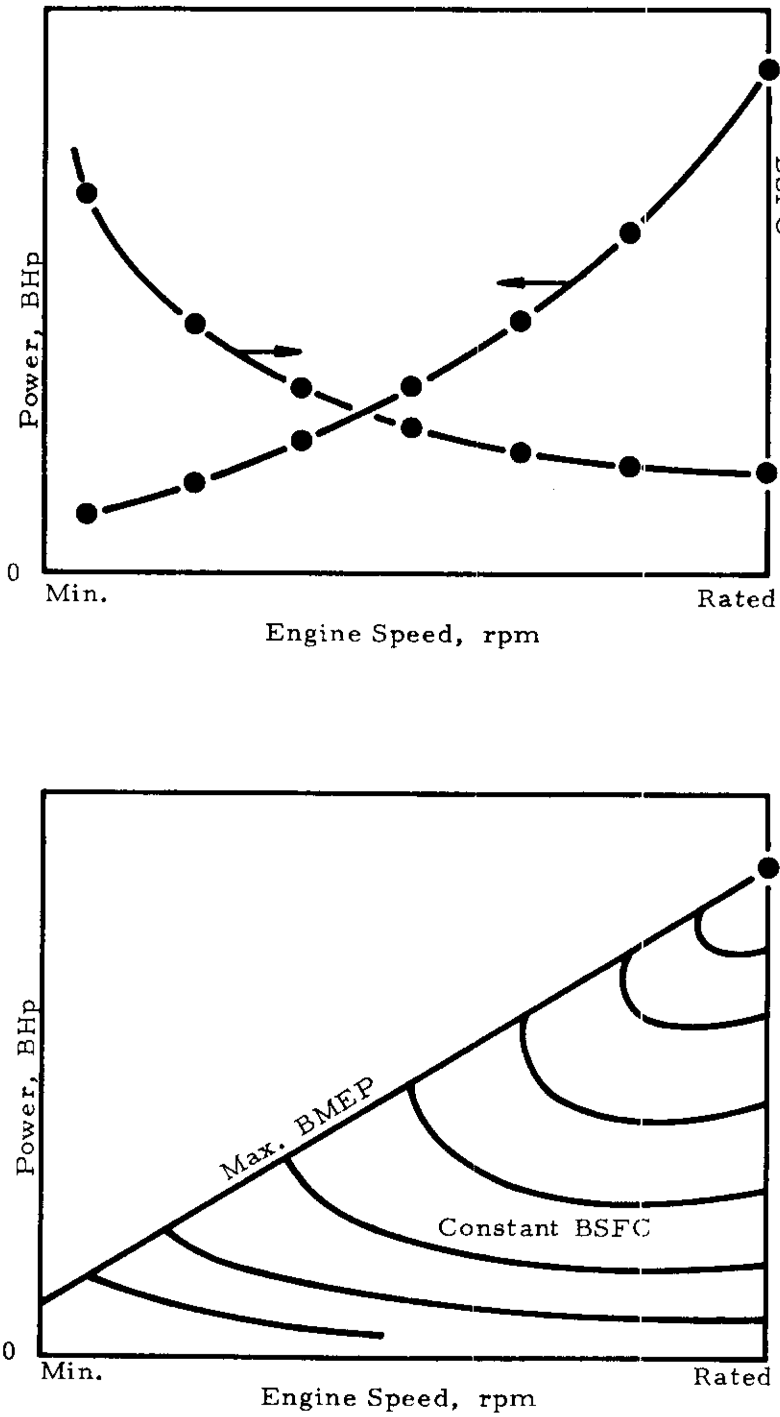

FIGURE 4.1 TYPICAL POWER AND BSFC DATA FOR TURBOCHARGED ENGINE--AVAILABLE DATA (TOP) AND EXTENDED DATA (BOTTOM)

only and BMEP is brake mean effective pressure. If power to drive auxiliary components is neglected, the major contribution to FMEP of a two-stroke cycle engine is mechanical friction. Similarly, for a four-stroke cycle engine the major contribution comes from mechanical friction and pumping losses. The approximate magnitude of the FMEP for each type of engine may be obtained from several engineering texts; the reference used here was the well-known book by Taylor and Taylor[28]. This reference also furnished the thermal efficiency for two-stroke and four-stroke cycle open-chamber engines.

Turbocharger boost pressure and (sometimes) manifold or air box temperature was known for the maximum power-rated speed operating point of these engines. It was assumed that the turbocharger output pressure varied directly with fuel rate and with the square of the engine speed. The curves that result from these assumptions are shown in Figure 4.2. Further, since these turbocharged engines

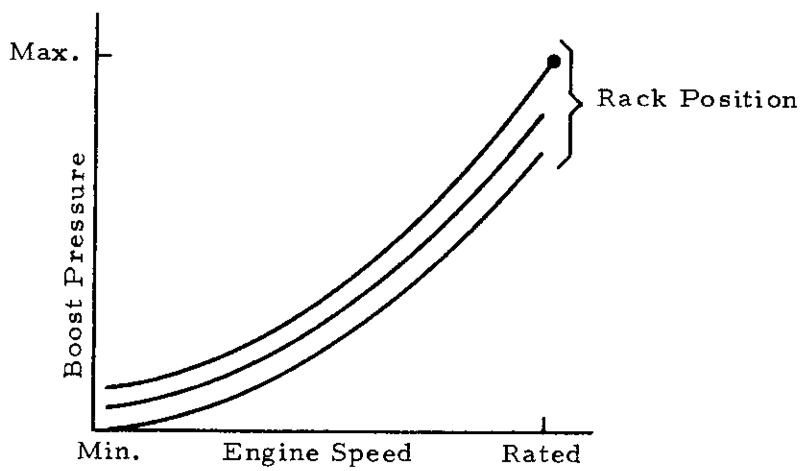

FIGURE 4.2 ASSUMED TURBOCHARGER BOOST PRESSURE CURVES

[28]Taylor, C. Fayette, and Edward S. Taylor, The Internal Combustion Engine, International Textbook Company, Scranton, 1961.

are equipped with aftercoolers, it was assumed that the aforementioned air temperature remained constant at the various engine speed-rack conditions.

It is believed that assumptions described in the preceding paragraph are the most critical of all and, unfortunately, the ones that are most likely to deviate considerably from the "real life" situation. Their importance rests on the nature of the analytical method used to derive the BSFC maps; that is, the method utilizes turbocharger discharge air density to obtain fuel/air ratios which, in turn, are used to obtain engine thermal efficiencies at the different speed-rack conditions. Any deviation from what actually occurs for each engine is due to the fact that pressure and temperature were usually known for only one operating condition and that the three types of engines being studied use very different turbocharging techniques. For instance, all the four-stroke engines use turbochargers that are driven solely by exhaust gas energy, the conventional two-stroke engine uses a turbocharger that is mechanically driven at low engine speed and exhaust gas driven at higher speed, and the opposed-piston two-stroke engine uses a mechanically-driven blower at low and midrange engine speed and a conventional turbocharger at high speed. Without detailed turbocharger performance data or, at least, boost pressures at several engine speeds and loads, assumptions were necessary in this important area. There is little doubt that these assumptions represent a greater compromise than do those concerning FMEP or thermal efficiency.

With these ideas in mind, we can now develop the method used to generate the BSFC maps. First, we have[29]

$$\text{BMEP} \propto \dot{F} Q n_t \, n_m \tag{4-1}$$

where $\dot{F}$ = combustion fuel rate, Q = fuel heating value, n_t = thermal efficiency, and n_m = mechanical efficiency. Suppose that values of BHp, BSFC, boost pressure and temperature are known for maximum power at rated speed. Let this operating point be designated by the subscript 1 and a second operating point by the subscript 2. Then,

$$\frac{\text{BMEP}_2}{\text{BMEP}_1} = (\dot{F}_2 \, Q_2 \, n_{t_2} \, n_{m_2}) \Big/ (\dot{F}_1 \, Q_1 \, n_{t_1} \, n_{m_1}). \tag{4-2}$$

[29]Ibid.

By using the definition of mechanical efficiency,

$$n_m = \frac{BMEP}{BMEP + FMEP}, \qquad (4\text{-}3)$$

and noting that $Q_1 = Q_2$, eq. (4-2) becomes

$$BMEP_2 = \left(\frac{\dot{F}_2}{\dot{F}_1}\right)(BMEP_1 + FMEP_1)\left(\frac{n_{t2}}{n_{t1}}\right) - FMEP_2 \qquad (4\text{-}4)$$

The FMEP values can be obtained immediately from the aforementioned reference; however, the following preliminary steps are required before the value of thermal efficiency can be found.

First, we note the relationship

$$\frac{\left(\frac{\dot{F}}{A}\right)_2}{\left(\frac{\dot{F}}{A}\right)_1} = \left(\frac{\dot{F}_2}{\dot{F}_1}\right)\left(\frac{A_1}{A_2}\right) = \left(\frac{\dot{F}_2}{\dot{F}_1}\right)\frac{(\rho_1 \; D_1 \; N_1 \; v_1)}{(\rho_2 \; D_2 \; N_2 \; v_2)} \qquad (4\text{-}5)$$

where ρ = air box or manifold air density, D = engine displacement, N = engine speed, and v = volumetric efficiency. Assuming that the volumetric efficiencies change very little with changes in ρ and N, their ratio is set equal to one and we have

$$\left(\frac{\dot{F}}{A}\right)_2 = \left(\frac{\dot{F}}{A}\right)_1\left(\frac{\dot{F}_2}{\dot{F}_1}\right)\left(\frac{\rho_1 \; N_1}{\rho_2 \; N_2}\right) \qquad (4\text{-}6)$$

Finally, use of the assumptions regarding the behavior of air pressure P and temperature with changes in engine speed and fueling rate gives

$$P_2 = P_1\left(\frac{N_2}{N_1}\right)^2\left(\frac{\dot{F}_2}{\dot{F}_1}\right) + P_o \qquad (4\text{-}7)$$

where P_o is atmospheric pressure. It is convenient to express this relation in terms of air density:

$$\rho_2 = \rho_1 \left(\frac{N_2}{N_1}\right)^2 \left(\frac{\dot{F}_2}{\dot{F}_1}\right) + \rho_o \qquad (4\text{-}8)$$

Substituting eq. (4-8) into eq. (4-6) allows the fuel/air ratio at the second operating point to be expressed in terms of known parameters and the constant ρ_o. It is now possible to use the reference data to find the thermal efficiency at this point and to then solve eq. (4-4) for $BMEP_2$.

A computer program was developed to calculate the BMEP and associated BHp and BSFC values. Reference data for FMEP and thermal efficiency was included in the program. Several points were calculated at various engine speeds ranging from minimum operating speed to rated, and the BSFC map was constructed from these points. The accuracy of the map was checked by comparing calculated values with actual data at those points where the latter were available. Good numerical agreement was noted for points in the upper right-hand part of the map, while less agreement was found for points at low speed. This discrepancy at low speed is probably due to the fact that all BSFC values are derived from the performance parameters for maximum power at rated speed and that the assumptions made are less valid at low speeds. However, the overall trend and relative values of the constant BSFC lines were judged to be generally correct. The manner in which these maps are used in the next section is based on their trendwise, rather than numerical, validity.

4.2 Emission Maps

Maps of the brake specific NO_x ($BSNO_x$) and CO (BSCO) were derived from partial data obtained from the same sources that provided fuel consumption data. The development of a map for unburned hydrocarbons was not attempted since HC emissions are not primarily a function of performance parameters such as speed, BMEP and fuel/air ratio, but are more dependent on combustion chamber geometry and injection system design features. Similarly, a complete map for smoke opacity was not developed; rather, data were analyzed in a qualitative manner as a function of fuel/air ratio only. The results were plotted on the BSFC maps to illustrate the general behavior (i. e., increase or decrease) of smoke opacity that could be expected to result when engine speed-load conditions were changed.

The procedure for developing gaseous emissions maps began with an analysis of available brake specific values. It was determined that data from the various sources differed considerably in a numerical sense, but demonstrated the same general trends when viewed as a function of parameters such as engine speed and load. Therefore, data consisting of brake specific NO_x and CO values and associated concentrations and BHp values were selected from one[30] of the available sources.

These data were used to calculate mass exhaust flow rate at each test condition by means of the equation

$$\dot{M}_x = \frac{\text{BS }(NO_x \text{ or CO}) \times \text{BHp}}{\text{Concentration }(NO_x \text{ or CO}) \times C} \qquad (4\text{-}9)$$

where the concentration is in parts per million (ppm) and C is a constant. The assumption was then made that the $\dot{M}_x$ for any condition is equal to the air mass flow rate $\dot{M}_A$ into the engine. However, the air mass flow rate of interest is the combustion (or trapped) air flow; therefore, in the case of two-stroke cycle engines it was necessary to allow for the part of this air that is used to scavenge the cylinders. It was assumed that these engines used a nominal 30 percent of the total air flow for this purpose, and all values of $\dot{M}_x$ were reduced by this amount. The combustion fuel/air ratio could then be calculated for each operating point.

The next step in the procedure was to determine the changes in emission concentration and BMEP that occurred when engine speed was held constant ($N_1 = N_2$) and fuel rate was allowed to vary above and below the baseline fuel rate ($\dot{F}_2 > \dot{F}_1$ or $\dot{F}_2 < \dot{F}_1$). The same assumptions and equations employed in the derivation of BSFC maps were used to calculate BMEP, BHp, and $\dot{M}_x$ values for the new operating conditions.

The method used to determine concentrations of NO_x and CO was based on the assumption that concentration of each pollutant was, for constant engine speed, solely a function of combustion fuel/air ratio. A search of diesel emission technical literature produced several graphs that depicted this functional relationship, and it was noted that the curves often had a similar shape, though with different absolute values. Hence, two such representative curves, one for NO_x and the other for CO, were selected and

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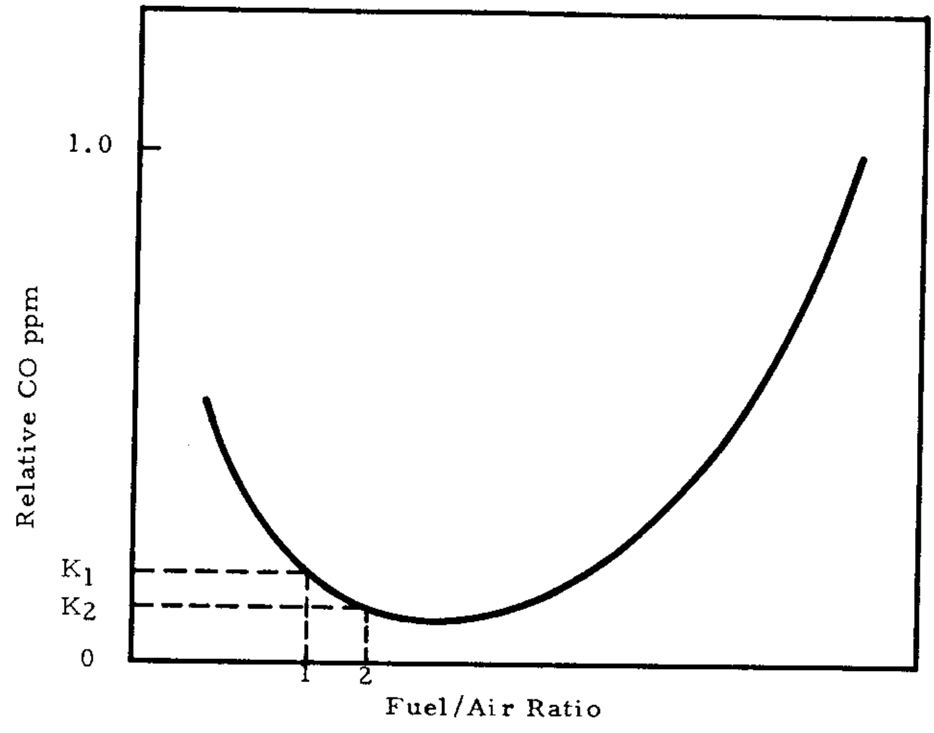
30 <u>Exhaust Emissions from Uncontrolled Internal Combustion Engines, Part I,</u> Southwest Research Institute, E.P.A., Report APTD-1490.

redrawn with relative concentrations plotted on the ordinate (Figure 4.3). The combustion fuel/air ratio at the original operating condition (designated, as before, as Point 1) was used to find a relative concentration, K_1. The fuel/air ratio for a new condition (Point 2) was used to find a new relative concentration, K_2. These relative values, together with the actual known concentration for the first condition, were used to calculate the concentration at the second point:

$$(\text{ppm})_2 = \frac{(\text{ppm})_1}{K_1} \times K_2 \qquad (4\text{-}10)$$

Finally, the derived values for BMEP, BHp, $\dot{M}_x$ and emission concentration were used to calculate brake specific NO_x and CO at the non-standard operating condition. Accuracy of the resulting emission maps was checked, where possible, by existing partial data and, as in the case of the BSFC maps, good numerical agreement was found at all points except those at low engine speed, and overall prediction of trends was judged to be acceptable for project objectives.



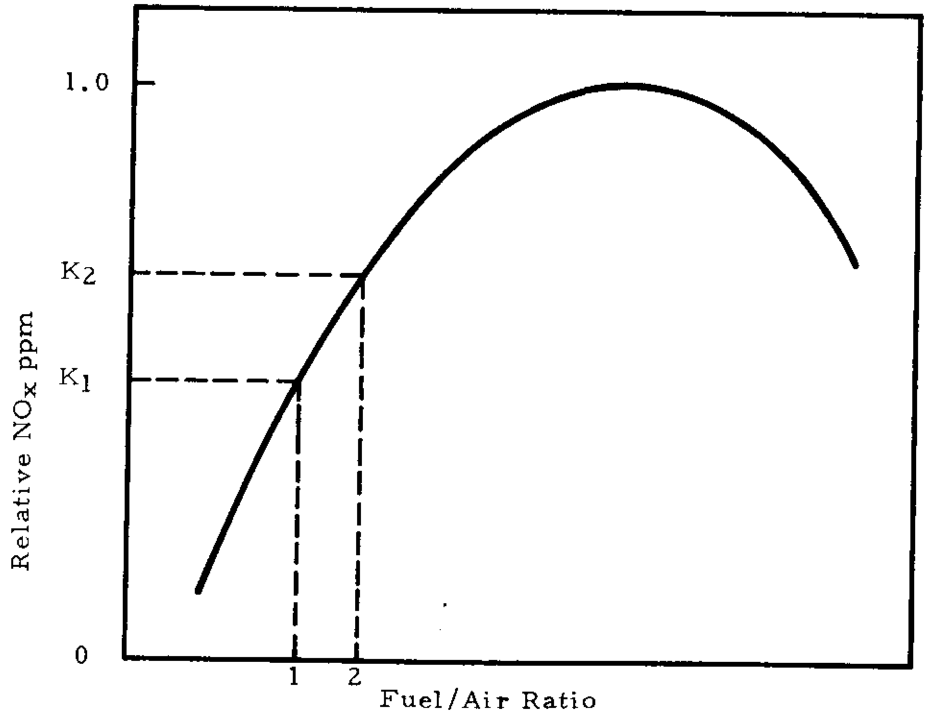


FIGURE 4.3 RELATIVE CONCENTRATIONS OF CO AND NO_x AS FUNCTIONS OF FUEL/AIR RATIO

5. ANALYSIS OF FUEL CONSERVATIVE OPERATING PROCEDURES

This section will discuss and evaluate candidate changes in operating characteristics of locomotive and Coast Guard prime movers and the trendwise effects of these changes on fuel consumption and exhaust emission levels. Two general methods of reducing fuel consumption will be considered, a reduction in nonessential operating time and a modification of operating modes in order to achieve higher brake thermal efficiency of in-service engines. The analysis of engine operating modes and duty cycles, together with the fuel consumption and emissions maps discussed in the previous section, will be used in this evaluation.

5.1 Reduction in Nonessential Engine Operation

In this context, nonessential engine operation is defined as non-productive (i.e., non-revenue-producing) locomotive operation or operation of a Coast Guard prime mover in a manner that does not contribute to accomplishment of the vessel's assigned duty. It is assumed that locomotive engineers and captains of Coast Guard cutters do not engage in such engine operation in power-producing modes; that is, when the locomotive or cutter is underway, it is for a purpose that is consistent with its principal function. Furthermore, it is assumed that these engines are not operated at a power level and fuel consumption rate above that required in a particular instance.

Given this definition of nonessential engine operation, and granting the reasonableness of these assumptions, it is apparent that attention must be focused on the amount of time these engines spend at no-load idle speed. The various duty cycles of Section 2 show that even line-haul locomotive engines idle at least 40 percent of their operating time, and interviews of Coast Guard engineering personnel indicate that main diesel engines sometimes idle for hours or days at a time.

Prolonged idle operation has several undesirable ramifications other than consumption of fuel (typically, three to six gallons per hour) and production of exhaust emissions. Many engine manufacturers assert that wear rates of piston rings and cylinder liners are highest at idle due to the fact that unburned fuel and condensation wash the lubricating oil film from liner walls. This action also leads to dilution and contamination of oil in the sump. Low combustion/exhaust temperatures permit lube oil to accumulate in exhaust manifolds where it becomes a fire hazard if the engine is quickly brought to a high load condition. Finally, carbon buildup on injector tips can plug the tip holes and, hence, degrade injector performance and combustion efficiency.

It would thus appear that, while some idle time is inevitable, extensive idling is a practice that could not possibly benefit the engine user and, therefore, should be avoided. However, it cannot be concluded what part, if any, of idle operation is unnecessary and/or undesirable without investigating the reasons behind it.

The principal reason for allowing large diesel engines to idle for extended periods of time is to avoid the problems of coolant leakage and condensation into the crankcase and cylinders upon engine shutdown. While modern engines benefit from better water seal design, leaks are still a problem for older engines and those that have run for several thousand hours without overhaul. Crankcase contamination involving treated cooling water or condensation is detrimental to lube oil properties and can result in high wear rates of critical engine components. However, leakage of coolant that contains ethylene glycol antifreeze is a more serious situation, since the antifreeze reacts with the lube oil to produce a heavy sludge that has almost no lubricating properties. Catastrophic engine failure is almost a certainty if operation continues under such conditions. Engine users have, therefore, generally avoided the use of antifreeze, with the consequence that prolonged idling has become even more necessary for engines that operate in ambient temperatures below freezing.

Restarting an engine that has been stopped for more than a few hours and has cooled to ambient temperature is usually a time-consuming procedure. Most manufacturers recommend that the engine be slowly "barred over" for one or two revolutions with compression relief valves open in order to insure that any water that has leaked and/or condensed in the cylinders during shutdown is expelled before starting is attempted. (A sufficient quantity of water in a cylinder can cause damage to piston and connecting rod during engine cranking.) Also, it may be necessary to pump oil from an external supply through the engine to insure that all critical parts are lubricated prior to starting. In any case, an engine that has cooled to 30-40 F would probably start only with some difficulty, and might not start at all below this temperature unless the coolant and lube oil were heated by an external source.

The Coast Guard vessels visited utilize coolant heaters that maintain water jacket temperature at 120 to 140 F. However, these engines are still required to idle for long periods, supposedly in order to be ready to furnish motive power on short notice. This is a somewhat perplexing situation, for manufacturers of generator sets that operate in temperatures down to -40 F claim that engines of this same type, equipped with heaters that keep lube oil and coolant at 120 F, can start in less than ten seconds and assume full load operation in less than thirty seconds. Certainly, the engine room of a vessel offers an environment that is at least as conducive to fast engine start-up, and full load operation in thirty seconds would seem to be sufficiently rapid for most situations. Commanding officers of these vessels undoubtedly have their own opinion on this matter, however.

Locomotive engines generally do not use heaters for either oil or coolant, though both types are commercially available. The principal reason that such heaters are not popular is that their use dictates the locomotive must be parked only at specific yard locations where the heater/circulating pump and/or its power source is located. This inflexible arrangement is undesirable to most railroads from the point of view of efficient yard operation. Also, cost of heaters, electricity, and related equipment would be high, and only accurate cost analysis by individual roads could determine whether or not the cost-benefit ratio would be favorable.

In summary, all parties concerned, engine manufacturers as well as engine users, generally regard extended idle operation as, at best, a necessary evil. However, they see no reasonable alternative to the practice at the present time.

5.2 Improvement of Engine Efficiency

The measure of engine efficiency is here taken to be engine brake thermal efficiency, n_t, which is proportional to the inverse of the product of fuel heating value and brake specific fuel consumption (BSFC):

$$n_t \propto \frac{1}{Q \times BSFC} \tag{5-1}$$

If fuel heating value is regarded as constant, then brake thermal efficiency is a function of BSFC alone; i.e., lower BSFC implies higher thermal efficiency. Therefore, this discussion will center on changes in the baseline BSFC that occur when current operating modes are modified. This approach is straightforward and has the advantage of using a performance parameter (BSFC) that is a realistic measure of engine efficiency: units (lb_m) of fuel consumed per unit of useful work (brake or flywheel horsepower per hour) delivered.

Before proceeding with this analysis, it must be emphasized that changes in engine operating modes cannot, in the end, be separated from changes in the operation of the total unit (locomotive or vessel). That is, since most of these prime movers operate in well-defined modes, any "fuel conservative operating procedure" must involve changes in these modes and, hence, in the operating characteristics of the locomotive or vessel. The discussion that follows will attempt to assess, in a qualitative sense, the overall impact of the various engine operating changes on the performance of the unit in question.

5.2.1 Locomotive Engines

The derived maps of BSFC, $BSNO_x$, and BSCO for locomotive power plants are presented in this section. Several points concerning the data representation in these maps should be mentioned.

First, curves for constant BSFC are labeled in a purely relative manner along the right-hand side. The minimum BSFC value is set equal to zero (0), and higher values are expressed as a percentage increase over the minimum. Hence, a curve labeled by the number +4.0 represents a BSFC that is 4.0 percent higher than the minimum value. The presentation of relative values only was adopted in order to avoid the direct comparison of absolute BSFC values for the various engines. Such a comparison of the derived fuel consumption figures might unfairly depict the performance of one make of engine as compared to the others and would serve no useful project purpose. The use of relative values, on the other hand, allows an analysis of general trends that is adequate for this study of engine operating modes and related BSFC.

The line that cuts across the curves of constant BSFC is the typical baseline operating curve for the engine. The points on this operating curve correspond to the eight notches of the locomotive throttle schedule. (Note that only an eight-speed schedule is used for the four-stroke engine.) The two lines that bracket and/or merge into this operating curve represent so-called smoke opacity limits;

the lower line corresponds to the just-visible smoke limit (J. V. S. L.), defined here as being equivalent to about 3 percent PHS opacity, and the upper line represents an opacity that is approximately seven to ten times this lower value. It is important to note that opacity does not change linearly with distance between these two curves; i. e., the point equidistant from both curves does not represent an opacity that is halfway between the two extreme values. However, the curves do allow the directional change in smoke opacity to be determined whenever an operating point is moved from its indicated baseline position.

Next, the curves for constant brake specific emissions have also been labeled in a relative manner, with the least such value set equal to one (1.0) and higher values expressed as a multiple of this minimum. The emissions maps have BMEP, rather than BHp, as the ordinate; therefore, lines of constant brake horsepower have been added to these figures to assist in going from fuel map to emissions maps. Also shown is the baseline operating curve for the engine. The points on this latter curve again correspond to the present eight throttle notches.

With these preliminary considerations in mind, the analysis of possible changes in current operating modes can begin. This analysis will first consider those changes that indicate (according to the derived fuel maps) an improvement in BSFC. The contemplated mode changes will be of three types: (1) current operating points, defined by speed-power (or BMEP) values, are moved upward along the operating curve; i. e., both speed and power (or BMEP) are increased; (2) operating points are shifted upward along vertical constant engine speed lines so that power (or BMEP) is increased; and (3) operating points are moved to the right so that engine speed is increased, but power remains unchanged. It is important to note that in no case is the Notch 8 speed and power changed, as these values are fixed by engine design limits. Each fuel consumption and emissions map shows one or two operating points drawn with small arrows showing the direction of these three changes. The effect on smoke opacity, $BSNO_x$, and BSCO will be noted.

Looking first at the fuel map (Figure 5.1) for the two-stroke cycle engine, it can be seen that the current operating curve generally intersects each BSFC line at its "turning point" or bend. This

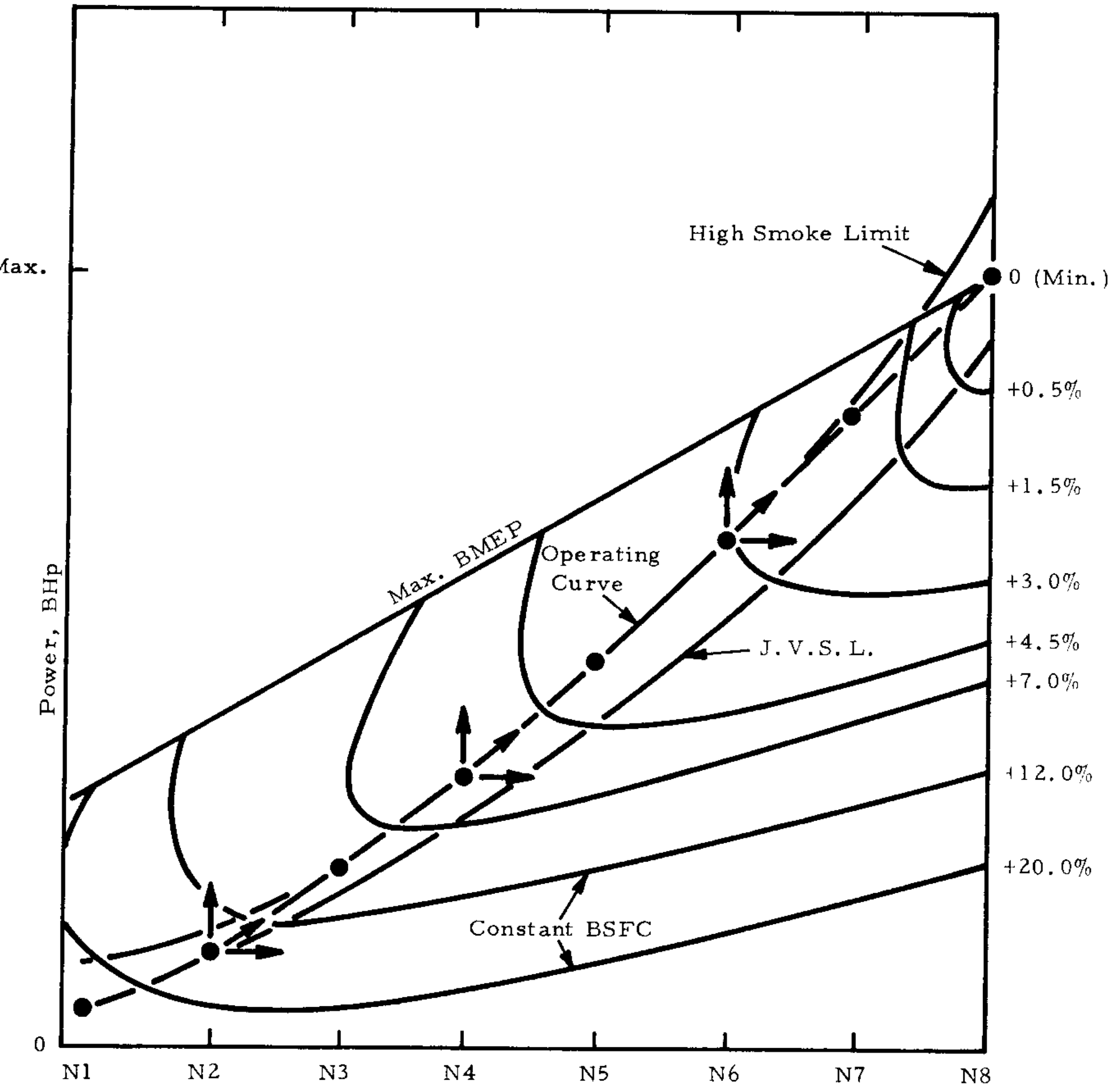


FIGURE 5.1 TYPICAL FUEL CONSUMPTION MAP--TWO-STROKE CYCLE TURBOCHARGED LOCOMOTIVE ENGINE

situation indicates that, in order to obtain a more favorable BSFC, the present points (throttle notches) must be moved upward along the operating curve. Of course, all of the points cannot be grouped into the low-BSFC part of the map, since to do so would destroy the operating flexibility of the locomotive and force the engine to operate at consistently high BMEP levels and fuel consumption rates. Such operation would actually waste fuel and cause high wear rates throughout the engine. The maps of $BSNO_x$ and BSCO (Figures 5.2 and 5.3, respectively) do, however, indicate that these quantities would be reduced by shifting the operating points into the high-BMEP region.

No significant overall improvement in BSFC can be obtained by holding speed constant and allowing BHp to increase, since the operating points will generally remain on the same BSFC curve. However, it appears that some improvement would result if the first two or three notches were modified in this way. Unfortunately, the fuel map is for a turbocharged line-haul engine that spends only a small amount of operating time in these lower notches. Note also that increased power at constant engine speed would likely result in higher smoke opacity. The effect on $BSNO_x$ and BSCO can be determined by allowing points on the operating curve to move upward along constant speed lines, thus increasing BMEP at each point except Notch 8. It can be seen that $BSNO_x$ would be decreased at most points by this practice, while BSCO would probably decrease in the low-power modes and increase elsewhere.

The remaining alternative is to allow speed to increase while power output is held constant. Again, the modification would result in little or no improvement in BSFC except, possibly, for throttle positions whose operating points lie in the upper right-hand corner of the maps (Notch 8 excepted). Here, a horizontal movement would place these points in a region of lower BSFC without an increase in smoke opacity. The decrease in BSFC for these notches would be on the order of one or two percent; hence, cycle composite BSFC for a line-haul locomotive would decrease by a negligible amount (based on nominal ten percent total weight attached to Notches 5-7). Smoke opacity and engine wear rates would probably not be adversely affected since BMEP would be lower for the same power output at the higher speed. By means of the constant BHp lines drawn on the emission maps, it is found that $BSNO_x$ tends to increase and BSCO tends to decrease. Both effects are most pronounced at the lower throttle positions.

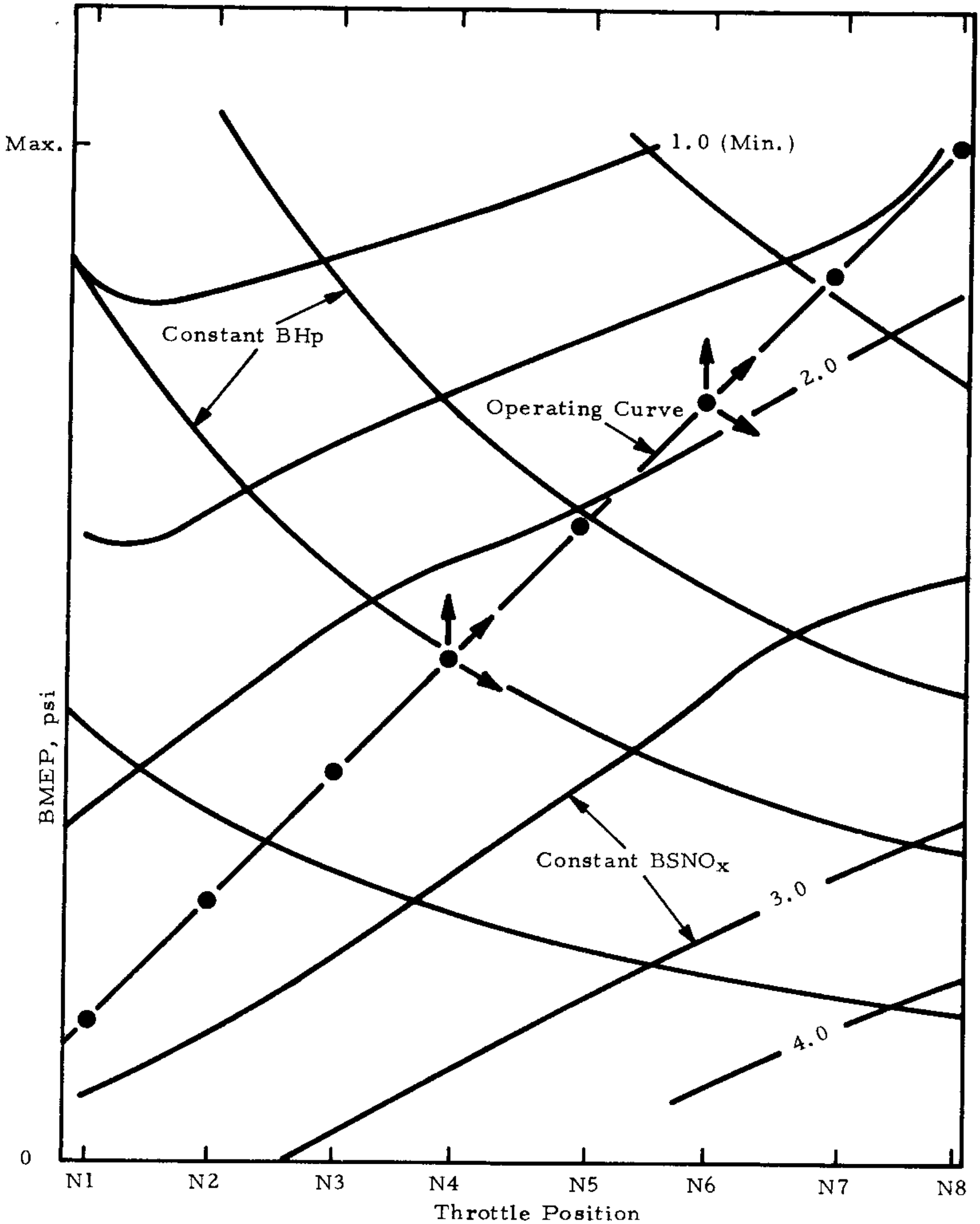


FIGURE 5.2 TYPICAL BRAKE SPECIFIC NO_x MAP--TWO-STROKE CYCLE TURBOCHARGED LOCOMOTIVE ENGINE

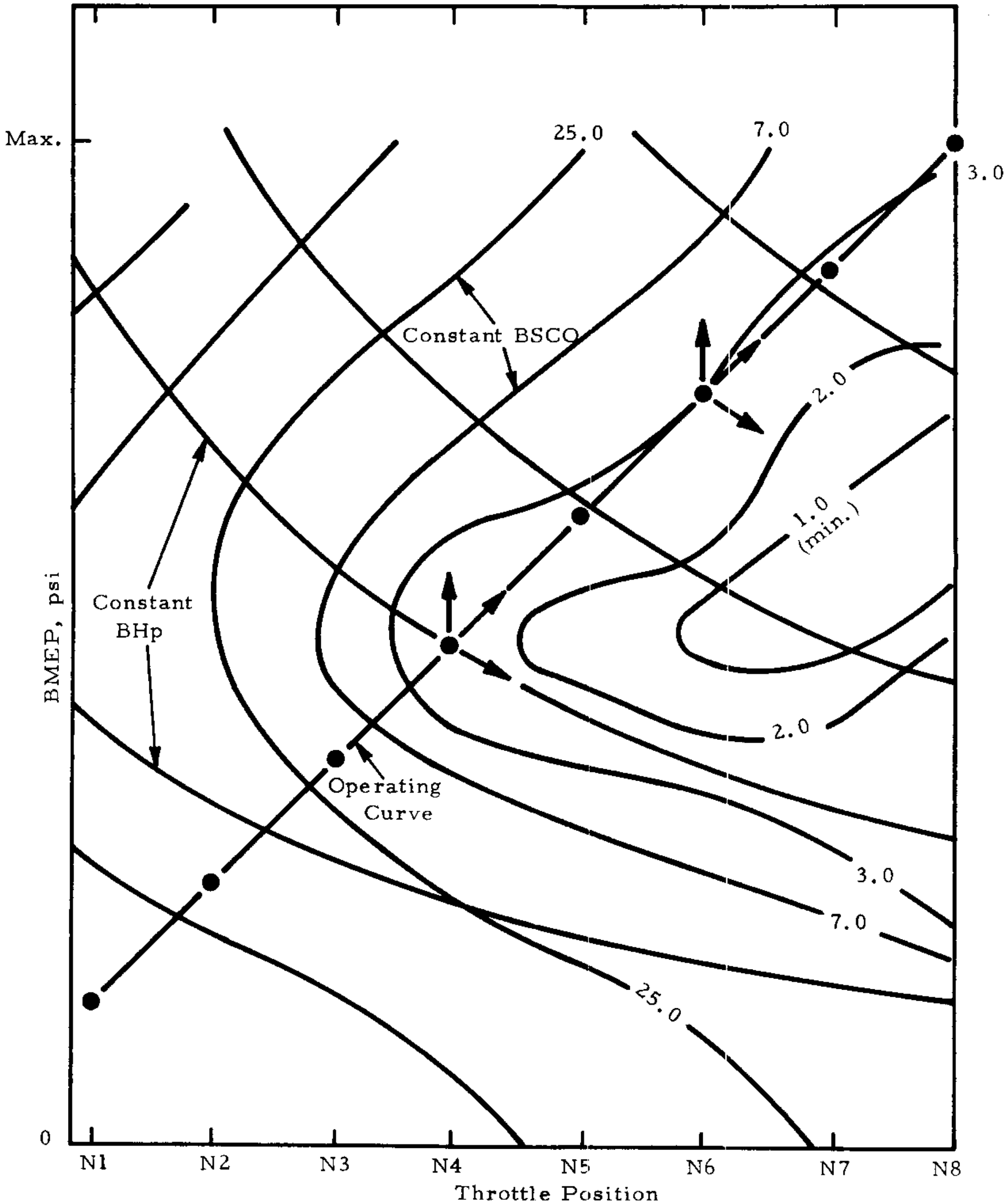


FIGURE 5.3 TYPICAL BRAKE SPECIFIC CO MAP--TWO-STROKE CYCLE TURBOCHARGED LOCOMOTIVE ENGINE

The discussion contained in the preceding paragraph pertains, of course, to the usual eight speed-eight notch throttle schedule. However, an analysis of the effects produced by a change to a two speed-eight notch schedule (per the modified G.E. schedule) is similar in nature. Assume that the first four notches are all at the usual Notch 5 engine speed and the last four notches are at the usual Notch 8 speed. Horsepower is to remain constant; that is, the points on the current operating curve are moved horizontally to the new vertical speed line. It can be seen that this change would cause BSFC to increase for the first five notches and to decrease or remain the same for the last three notches. A line-haul locomotive engine would probably show little improvement in cycle composite BSFC with this throttle schedule since, although the total weight for the last three notches is somewhat greater than for the other notches, the Notch 8 BSFC remains unchanged. The effects on $BSNO_x$ and BSCO would be similar to those discussed in the previous paragraph. However, by having the last four notches at rated speed, the operating points lie in the part of the BSCO map where values are very low.

The fuel map (Figure 5.4) for the four-stroke engine indicates that BSFC for the low and middle notch positions might be improved by operating at the same engine speeds and higher power outputs. The improvement realized could be quite large (percentage-wise) in the low notches and much smaller in the middle notches. Once again, however, the fuel map is that of a line-haul locomotive engine that spends a small percent of its operating time in these notch positions. Note also that smoke opacity is very sensitive to any increase in fuel/air ratio at these points. Operating points for the upper notches (Notch 8 excepted, of course) generally lie on the bend of the BSFC curve; hence, fuel consumption can be improved only by moving these points upward along the operating curve. The maps of $BSNO_x$ and BSCO (Figures 5.5 and 5.6, respectively) show that the changes in operating modes would tend to slightly reduce $BSNO_x$ and to perhaps significantly increase BSCO.

It is interesting to look at the results of changing from the eight-speed throttle schedule to one of just two speeds, all power outputs remaining the same. As mentioned, such a modification has been recently made by G.E. The new schedule has the first three notches at the former Notch 5 speed and the remaining notches at rated speed. The derived fuel map indicates that BSFC is higher for the first four notches and is unchanged or decreased slightly for the other notch positions. The overall cycle composite BSFC would probably be unchanged or slightly higher with this throttle schedule,

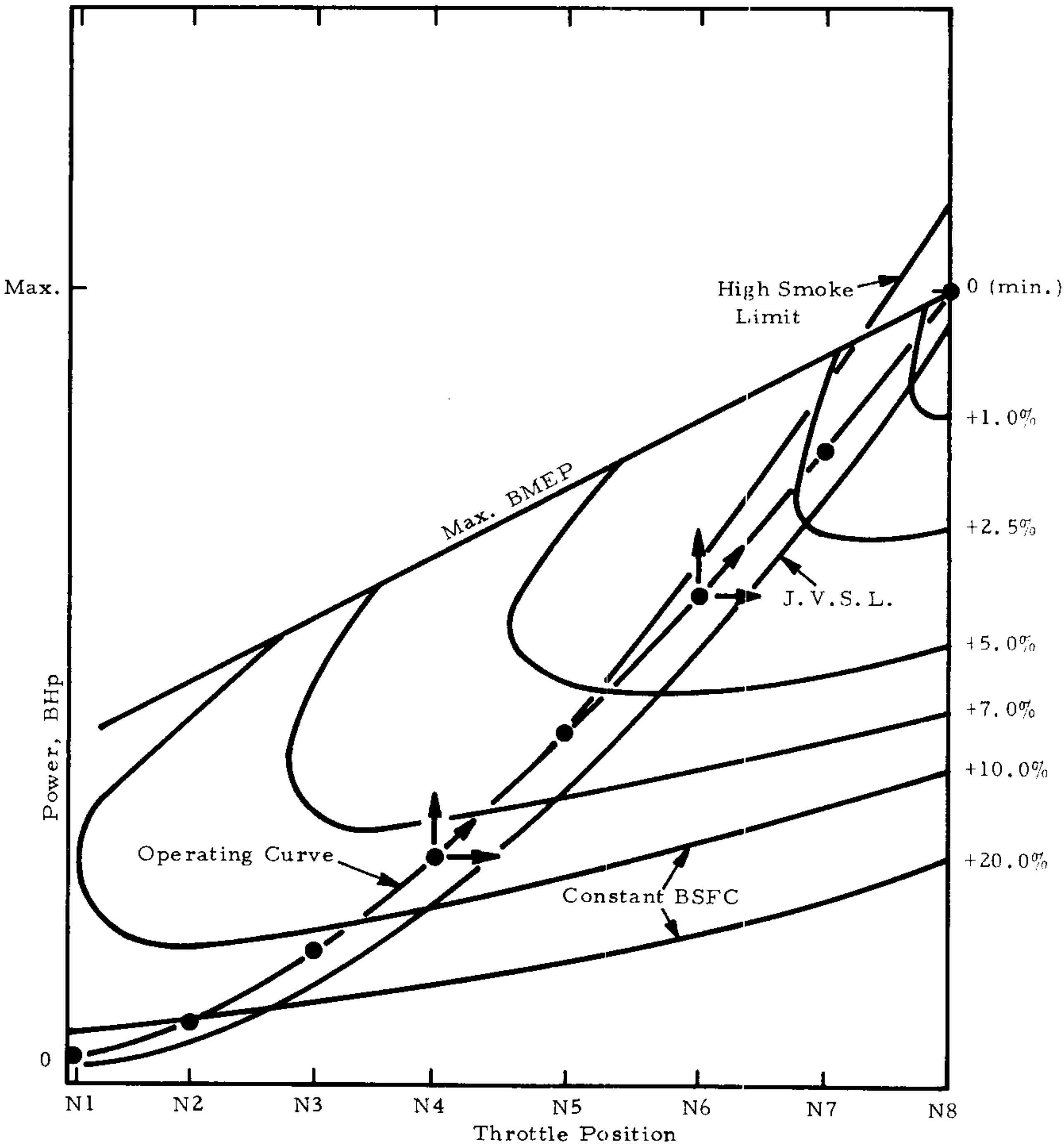


FIGURE 5.4 TYPICAL FUEL CONSUMPTION MAP--FOUR-STROKE CYCLE TURBOCHARGED LOCOMOTIVE ENGINE

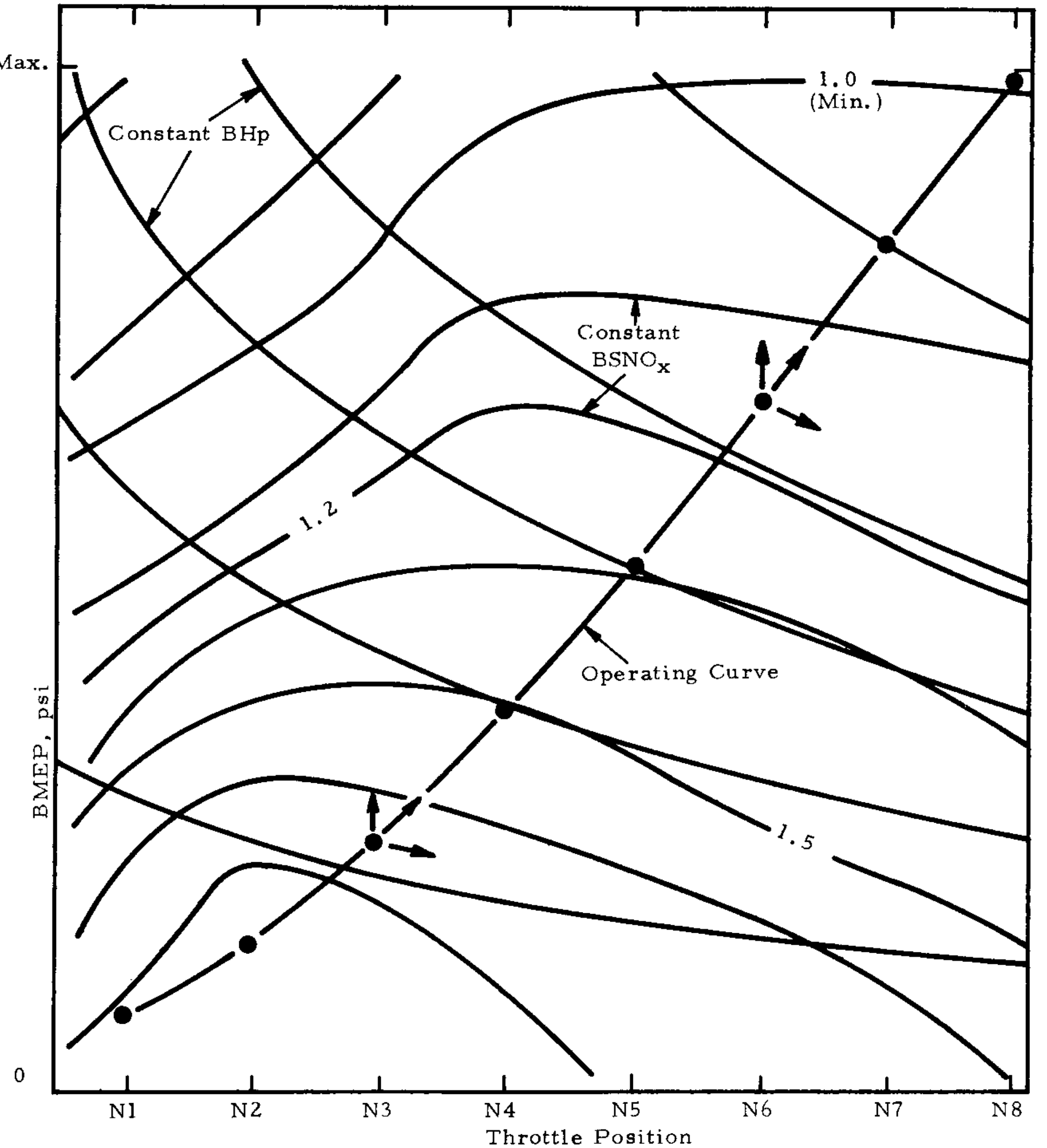


FIGURE 5.5 TYPICAL BRAKE SPECIFIC NO_x MAP--FOUR-STROKE CYCLE TURBOCHARGED LOCOMOTIVE ENGINE

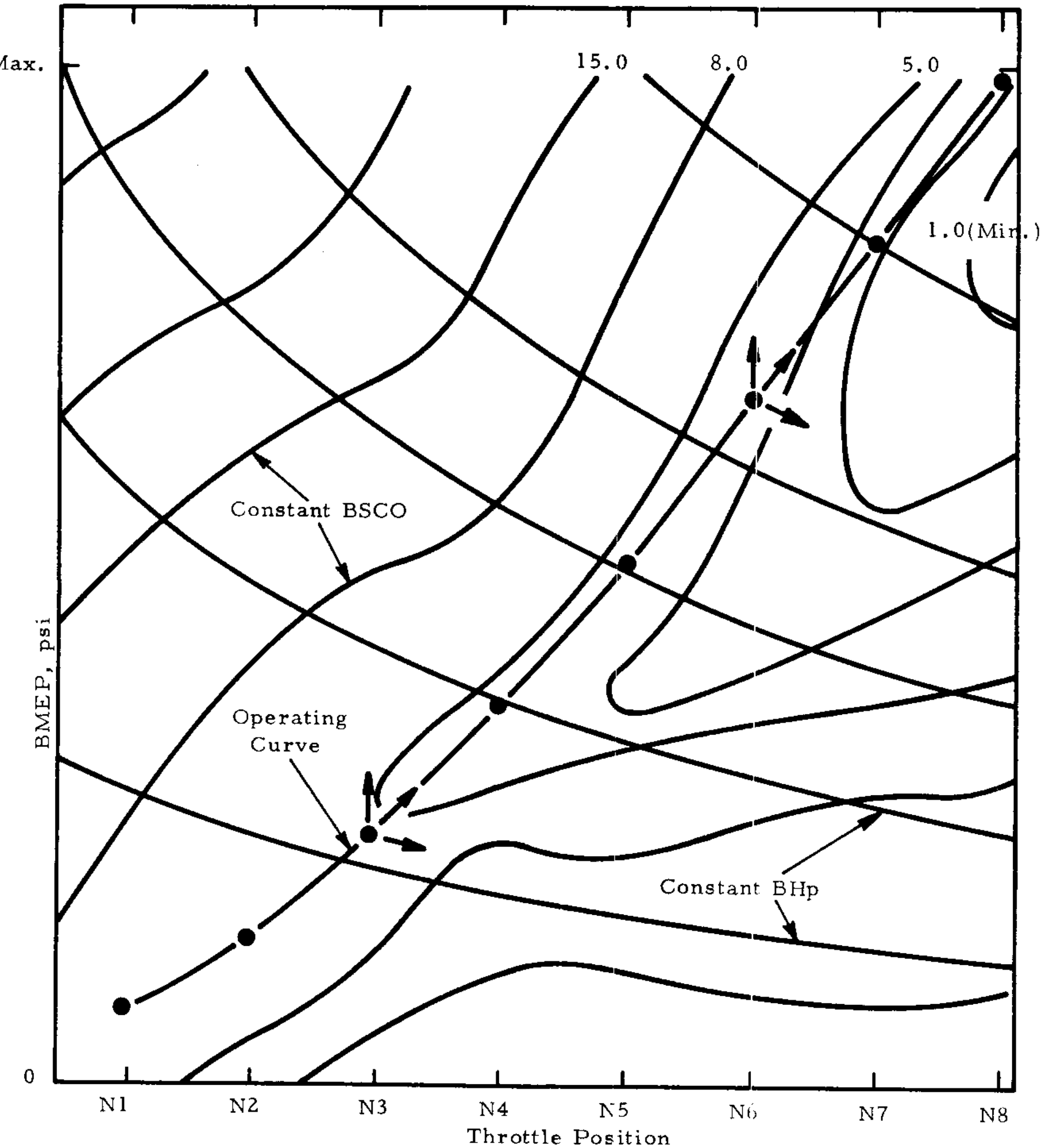


FIGURE 5.6 TYPICAL BRAKE SPECIFIC CO MAP--FOUR-STROKE CYCLE TURBOCHARGED LOCOMOTIVE ENGINE

depending on the value of the weighting factors used for the various notches. Exhaust smoke opacity, according to the smoke limit curves on the fuel map, is somewhat lower for steady-state engine operation. Engine performance and flexibility are presumed to be essentially unchanged since the power delivered to the generator is the same for both schedules. The engine operates at a higher average speed with the modified schedule, but BMEP is less, so wear rates are probably not adversely affected. Finally, $BSNO_x$ is reduced by a small amount, while BSCO is reduced in the upper throttle positions only. The preceding analysis generally agrees with the quantitative results presented in Reference 10; however, the latter are in the form of cycle composite, rather than modal, values so that direct comparison is not possible.

5.2.2 Coast Guard Engines

Maps of BSFC and gaseous emissions for four-stroke cycle turbocharged locomotive engines have been used for four-stroke marine engines that operate on either an eight speed-eight notch throttle schedule (direct drive vessels) or at constant speed and variable load (diesel-electric vessels). The use of locomotive engine maps is not intended to imply that certain important differences do not exist between specific makes of these four-stroke engines. However, in other important aspects--such as size, speed range, BSFC values, and use of turbocharging--there are sufficient similarities to warrant use of the relative data presented in the discussion of locomotive engines. Further, no actual brake specific emissions data were available for the four-stroke marine engines of interest.

Maps for the two-stroke opposed-piston marine engine are not the same as those for the two-stroke locomotive engine. In fact, emission maps for the former are not complete because of a lack of data for the entire speed range of this engine, and it was necessary to estimate behavior of the $BSNO_x$ and BSCO curves for most of the speed range. The strictly estimated parts of the curves are clearly marked in the figures. It should also be noted that even the available fuel consumption and emissions data were not for the same engine configuration (turbocharger, injection timing, and so forth) as used in Coast Guard WHEC vessels. However, it is thought that the relative trends shown in the maps are generally accurate, with the possible exception of the estimated data mentioned above.

As mentioned in Section 2, Coast Guard medium-and high-endurance cutters utilize the direct-drive (more precisely, fixed-ratio reduction-drive) concept and variable-pitch propellers. This variable pitch factor complicates the analysis of operating mode changes and, it is thought, limits the number of ways in which these modes can -- or, at least, should -- be modified.

For example, each throttle position corresponds to essentially constant values of engine speed and prop pitch. Also, at a given throttle position the engine is confronted with an essentially constant-load situation, provided wave height and wind and current force and direction are relatively constant. Therefore, it is not possible to maintain the same engine speed at that throttle position and to vary the load on the engine <u>without</u> varying the prop pitch. Similarly, it is not feasible to maintain the same engine load and to change engine speed without a change in pitch. While it is probably possible to program new propeller pitch settings into the vessel's throttle control, it is very likely that the present pitch settings are those that result in maximum propeller efficiency for each prop speed. (This circumstance is termed "very likely" because, although it is not known for certain to be so, there is no apparent marine engineering reason why it should not be so.) If this is true, any change in pitch will result in a loss of propeller efficiency. Hence, it would be a futile exercise to make an operating mode change that resulted in higher engine efficiency and lower propeller efficiency.

It is beyond the scope of Phase I of this project to obtain and analyze propeller efficiency data for these Coast Guard cutters. Therefore, the analysis that follows will admit of prop pitch changes that allow the operating modes to vary in ways that may not be possible or desirable in practice. In other words, consideration will be given only to the effect of the mode changes on engine efficiency (BSFC) and emissions. Questions of feasibility and/or practicality of these changes must be determined later.

The maps of fuel consumption and emissions for the four-stroke cycle engines are presented as Figures 5.7 through 5.9. The operating curves for the eight speed schedule of medium-endurance cutters have been drawn on these maps; note that marine engines do not operate at BMEP levels as high as those of locomotive engines and that rated speed is slightly lower. Smoke limit curves have been omitted from the BSFC map because smoke opacity data (obtained by PHS smokemeter) were not available.

Since the operating curves for marine and locomotive engines are of similar shape, analysis of these modified maps yields

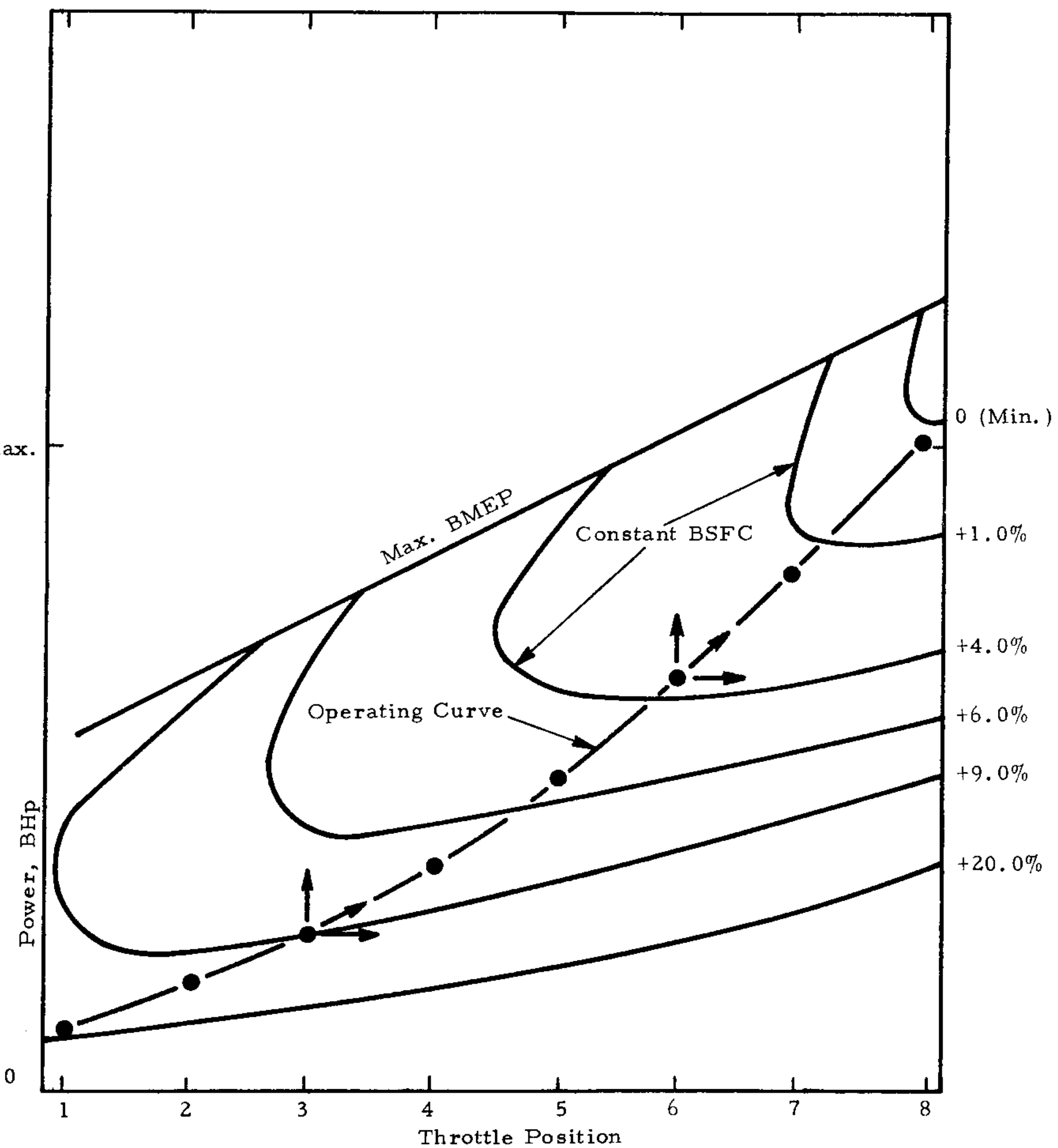


FIGURE 5.7 TYPICAL FUEL CONSUMPTION MAP--FOUR-STROKE CYCLE TURBOCHARGED MARINE ENGINE

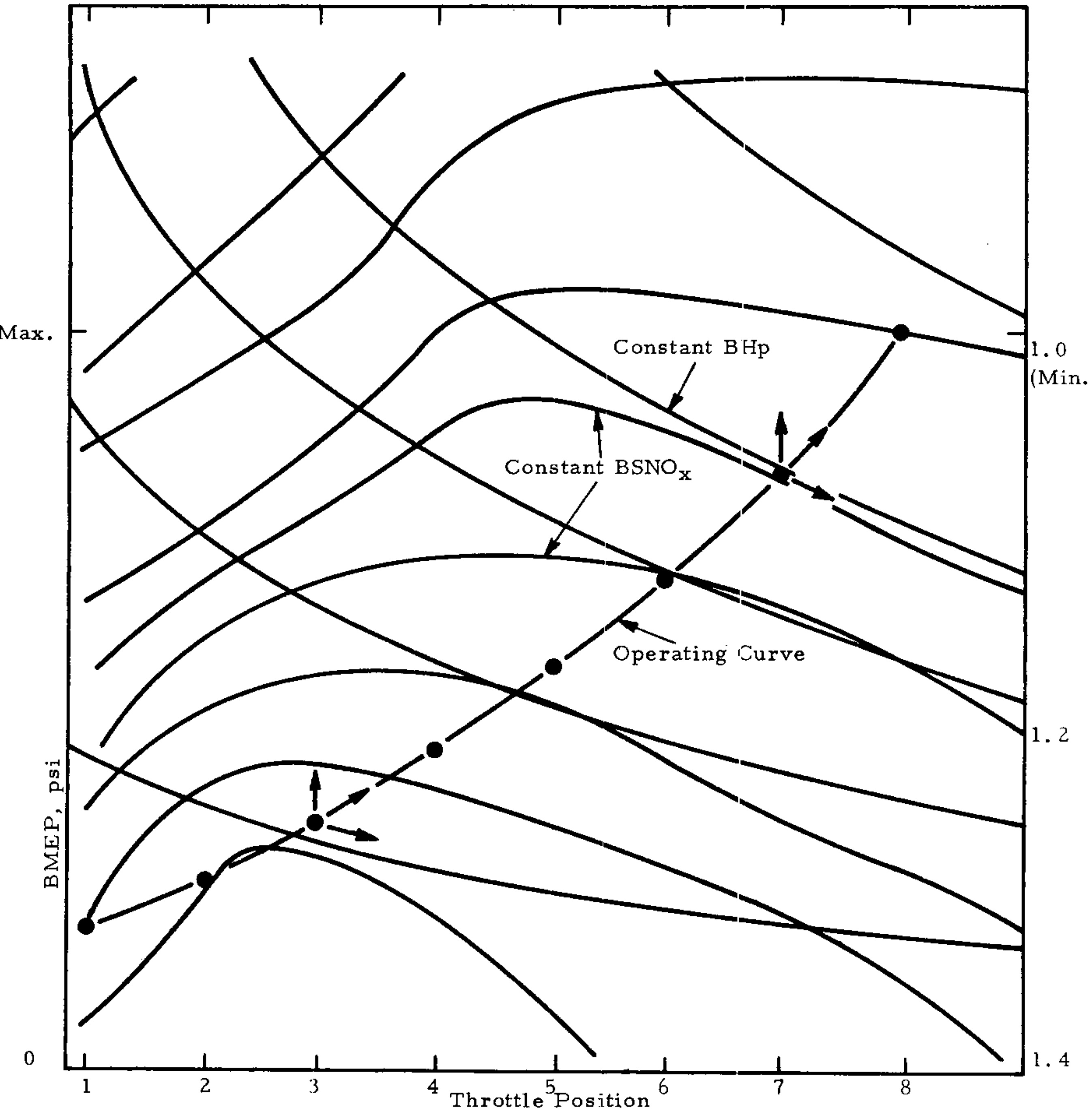


FIGURE 5.8 TYPICAL BRAKE SPECIFIC NO_x MAP--FOUR-STROKE CYCLE TURBOCHARGED MARINE ENGINE

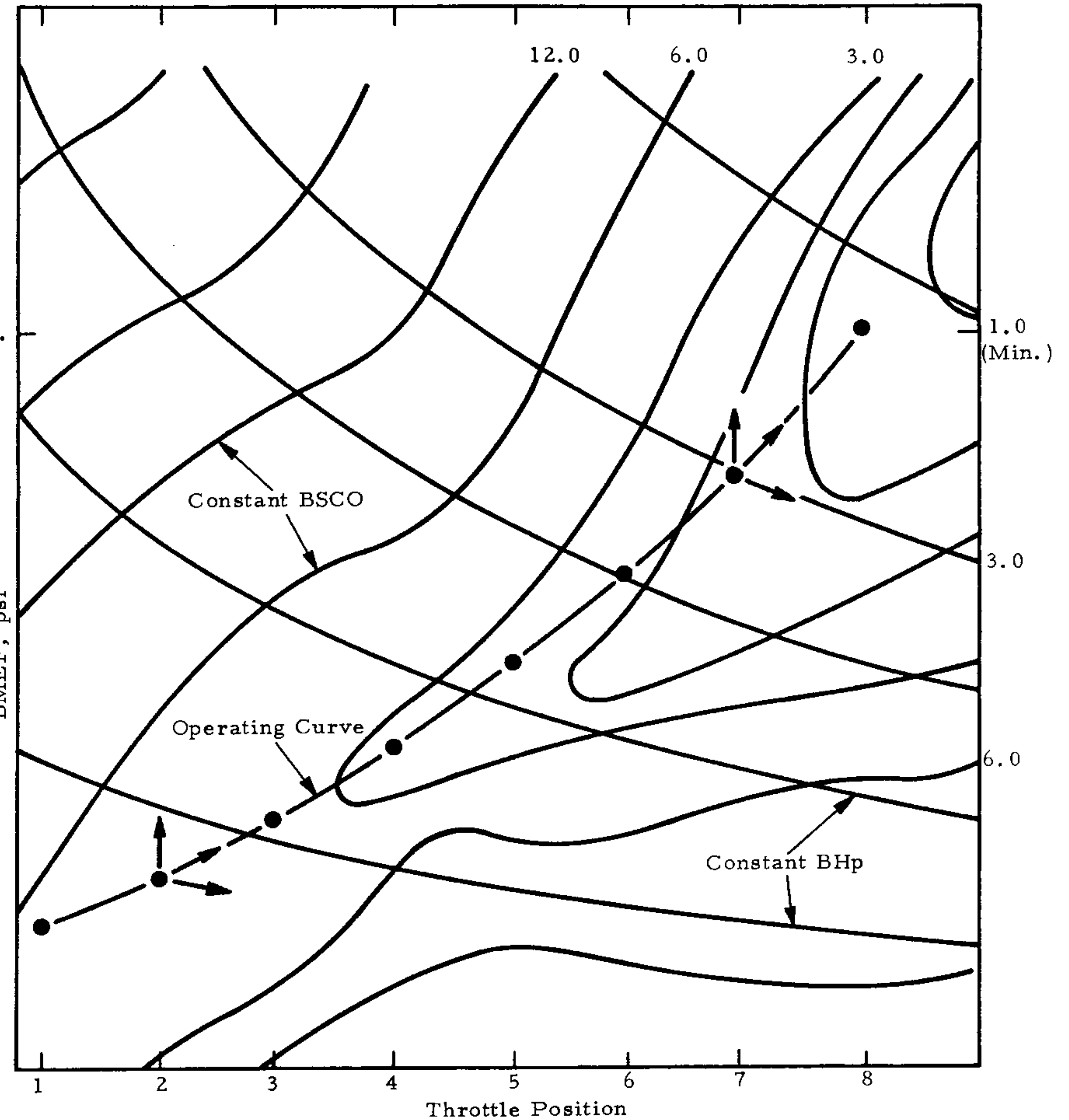


FIGURE 5.9 TYPICAL BRAKE SPECIFIC CO MAP--FOUR-STROKE CYCLE TURBOCHARGED MARINE ENGINE

similar results. That is, BSFC in the low and middle speed range could be improved by either operating the engine at higher power output at the same speeds or moving the operating points upward along the curve, while improvement at higher speed could be obtained only by the latter method. No significant improvement would occur if power was held constant at each point and speeds were increased; indeed, considerable degradation in BSFC might result instead. Brake specific NO_x would be lowered slightly by any of the methods that improve fuel consumption, but BSCO would show overall reduction only if operating points were grouped in the high speed and BMEP region of the map. Without accurate duty cycle information for these engine, it is not possible to evaluate various changes in operating modes in terms of the trade-off that almost always marks attempts to simultaneously lower fuel consumption <u>and</u> emissions.

In the case of the constant speed four-stroke engines that are used in diesel-electric drive systems of some icebreakers, it is sufficient to select an engine speed at the high end of the rpm range and then allow load to vary along a vertical constant speed line. It is seen that fuel consumption and gaseous emissions are at minimum values when BHp or BMEP are at rated levels. Of course, it is not required to operate these engines at maximum power at all times, but multiple engine installations could benefit from operating fewer engines at high loads rather than many engines at low loads. Changing to another (lower) operating speed is not desirable since BSFC increases as speed decreases, even under maximum load conditions. Any such change would probably require modification of the electrical system as well.

Figures 5.10 through 5.12 illustrate derived data for the two-stroke cycle turbocharged engine employed in high-endurance cutters. Placement of points on the operating curve indicates that BSFC would be improved by the methods cited earlier, i.e., by increasing power output at lower engine speeds or moving all points (except the one representing Flank Speed) upward on the operating curve. Either method would likely result in slightly lower $BSNO_x$ and slightly higher BSCO. No improvement in any of these quantities would result if power output was held constant and engine speed was increased for each operating point.

Some BSFC data were available in map form for an engine similar to the two-stroke opposed-piston Roots blown (non-turbocharged) model used in most icebreaker diesel-electric drive systems.