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Future Fuels and Engines for Railroad Locomotives

Volume II: Technical Document

S. G. Liddle, Task Manager
B. B. Bonzo
G. P. Purohit
J. A. Stallkamp

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Prepared for
U.S. Department of Energy
Through an Agreement with
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ABSTRACT

A study was made of the potential for reducing the dependence of railroads on petroleum fuel, particularly Diesel No. 2. The study takes two approaches: (1) to determine how the use of Diesel No. 2 can be reduced through increased efficiency and conservation, and (2) to use fuels other than Diesel No. 2 both in Diesel and other types of engines. The study consists of two volumes; Volume I is a summary and Volume II is the technical document.

The study indicates that the possible reduction in fuel usage by increasing the efficiency of the present engine is limited; it is already highly energy efficient. The use of non-petroleum fuels, particularly the oil shale distillates, offers a greater potential. A coal-fired locomotive using any one of a number of engines appears to be the best alternative to the Diesel-electric locomotive with regard to life-cycle cost, fuel availability, and development risk. The adiabatic Diesel is the second-rated alternative with high thermal efficiency (up to 64%) as its greatest advantage. The risks associated with the development of the adiabatic Diesel, however, are higher than those for the coal-fired locomotive. The advantage of the third alternative, the fuel cell, is that it produces electricity directly from the fuel. At present, the only feasible fuel for a fuel cell locomotive is methanol.

Synthetic hydrocarbon fuels, probably derived from oil shale, will be needed if present Diesel-electric locomotives are used beyond 1995. Because synthetic hydrocarbon fuels are particularly suited to medium-speed Diesel engines, the first commercial application of these fuels may be by the railroad industry.

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The various subtasks in the project are represented by the sections in this volume. The individuals responsible for the various subtasks are:

Section I - Locomotive and Train Characteristics
S. G. Liddle

Section II - Diesel Engine and Electric Transmission Controls
J. A. Stallkamp

Section III - Locomotive Modeling
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Section IV - Locomotive Modifications
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Section V - New Diesel Engines
S. G. Liddle

Section VI - Alternative Fuels for Diesel Engines
S. G. Liddle

Section VII - Rankine Cycle (Steam) Engines
S. G. Liddle

Section VIII - Gas Turbine Engines
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Section IX - Stirling Engines for Railroad Locomotives
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Section X - Fuel Cells for Locomotives
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Section XI - Other Engines and Transmissions
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Section XII - Regenerative Energy Storage Systems
S. G. Liddle

Section XIII - Locomotive Operations
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Section XIV - Cost Analysis
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CONTENTS

SECTION I
LOCOMOTIVE AND TRAIN CHARACTERISTICS

A. INTRODUCTION

A considerable volume of data about the train, the locomotive, and its components was required to conduct an in-depth analysis of the locomotive system. Gathering this information was one of the major tasks of the project. This section presents and discusses typical data for trains and locomotives.

For convenience, the data is divided into five categories. The first category is the general characteristics of representative locomotives made by the four North American manufacturers; General Motors, General Electric, Bombardier Inc., and Morrison-Knudsen. The second category is the resistance characteristics of the "consist" (the cars making up the train). This category includes aerodynamic characteristics, curve resistance, weights, and rail-flange losses among others. In other words, all of the forces that act on the drawbar. The third category is the resistance and tractive effort characteristics of the locomotive, its aerodynamic drag, weight, wheel size, etc. The fourth and fifth categories are related to the internal characteristics of the locomotive, that is, the components themselves. Those two groups are the engine with its mechanical accessories and the electrical equipment (i.e. alternator, motors, etc). No effort has been made to compile an all inclusive set of data but rather to limit it to those items which are necessary for the modeling effort and those necessary to help understand the system as a whole.

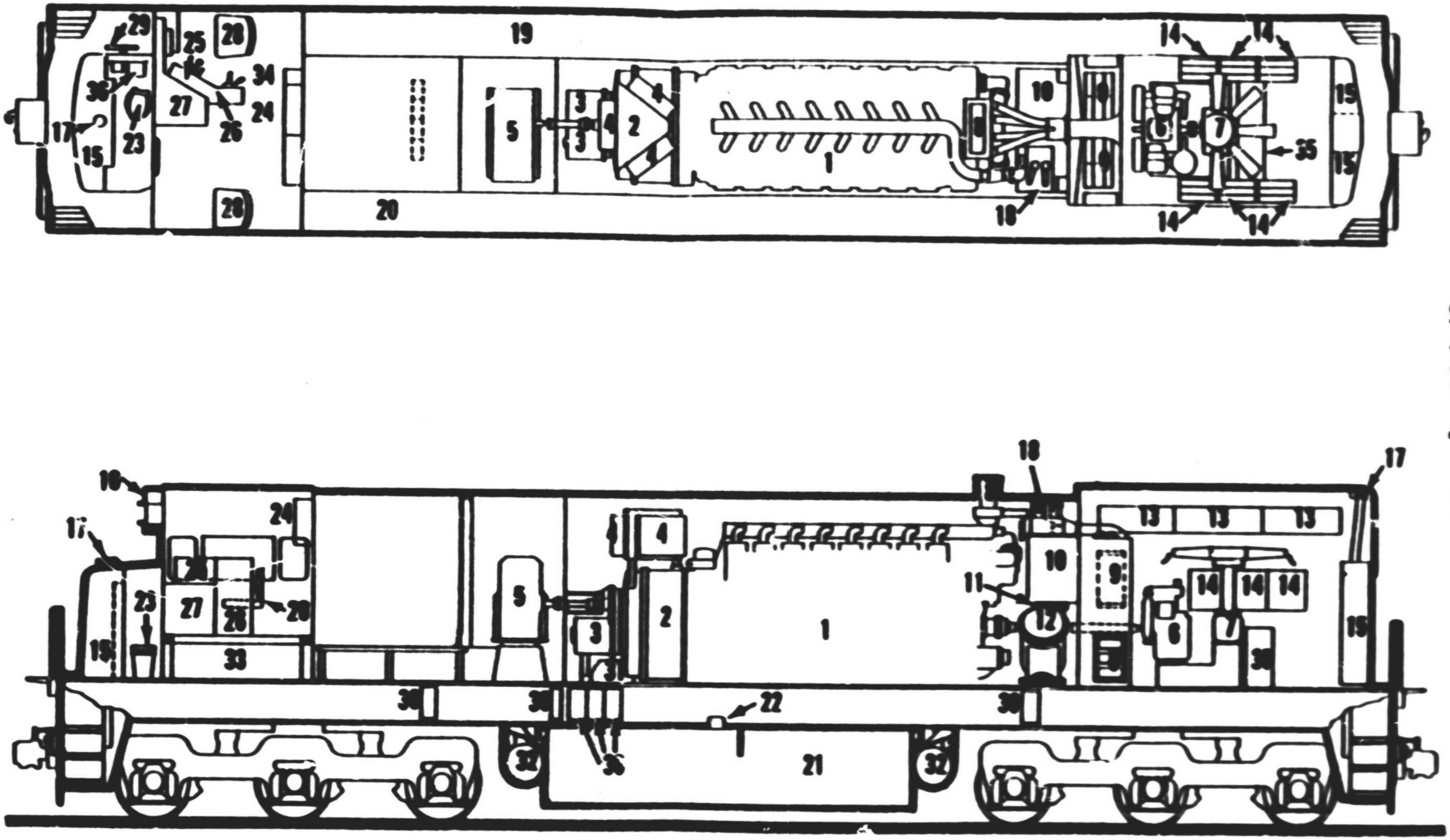
B. THE PRESENT DIESEL-ELECTRIC LOCOMOTIVE

There are a number of different locomotives being built by the North American manufacturers. In all of them, a Diesel engine is used to drive an alternator whose output is rectified to drive dc traction motors mounted on each axle. Though there are similarities between the locomotives, there are also some marked differences in the design details and in the locations of the components in the locomotives.

Figures 1-1 through 1-3 show some of the equipment, both inside and outside modern locomotives. The width, height, and weight per axle of different locomotives are similar because of the constraints of the railroads themselves. The length can vary, but for a 3000 hp locomotive, it is usually 60 to 70 ft. Locomotives also differ in the number and arrangement of the axles, and in the way the traction motors are connected to the axles. The different types of axles and trucks are shown in Figure 1-4 together with the standard terminology used to describe the axle arrangements. Table 1-1 lists some of the characteristics of modern locomotives with values for four typical units in the 3000 hp class.

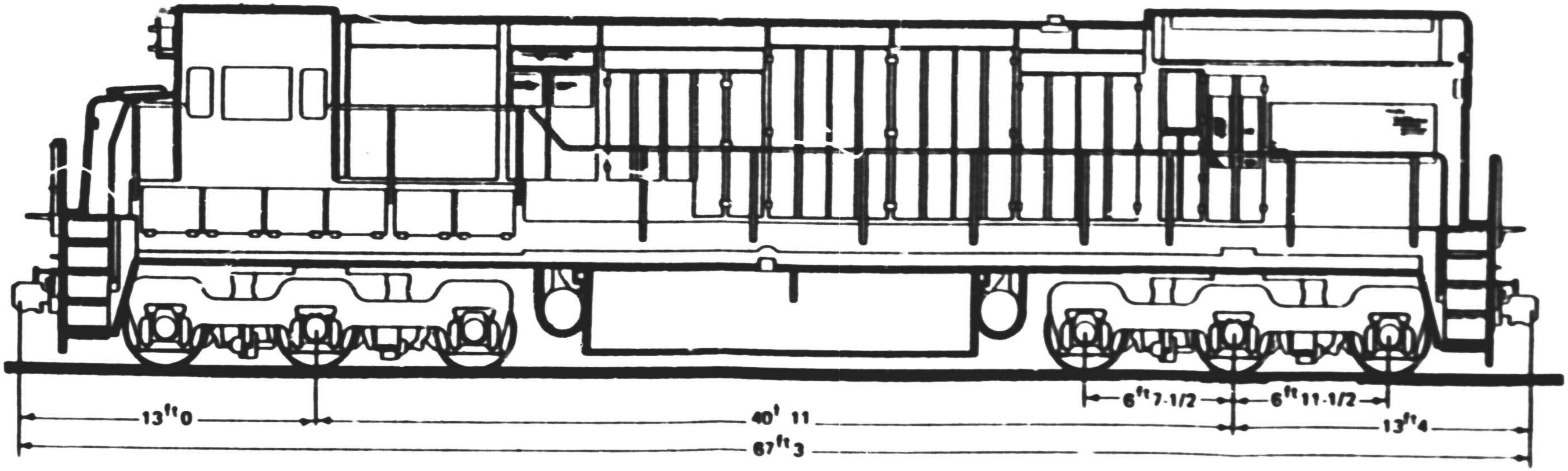
C. TRAIN RESISTANCE CHARACTERISTICS

As stated earlier, these characteristics are related to the drawbar forces. They include bearing friction, flange resistance, aerodynamic drag, grades, curve resistance, and acceleration forces.

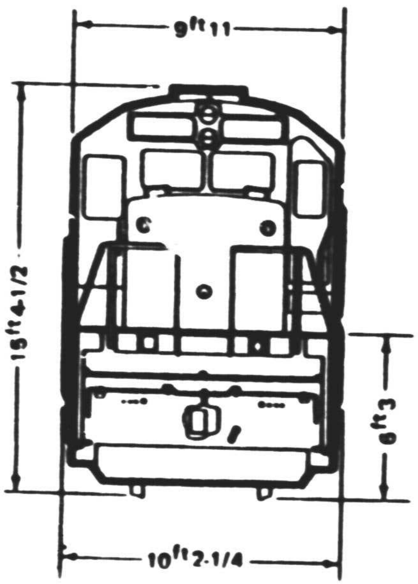


Note: Dimensions and
Specifications
on Following Page

Figure 1-1. General Electric U30C Locomotive
(From Ref. 1-1)

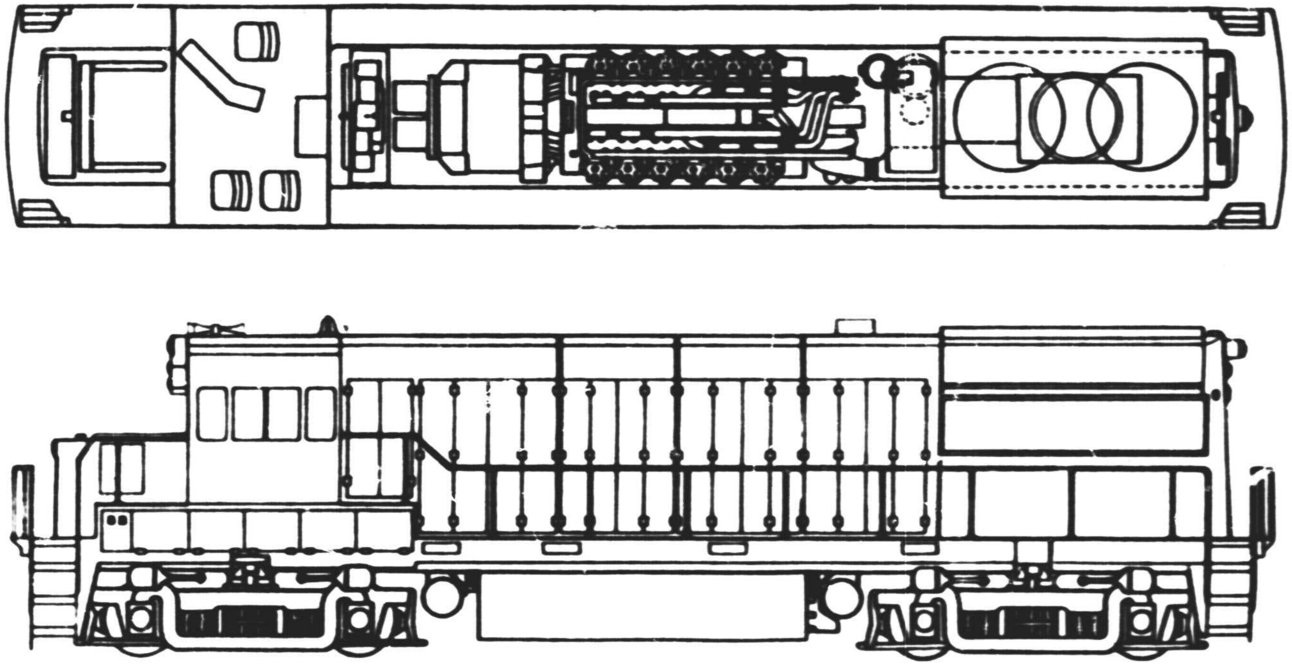


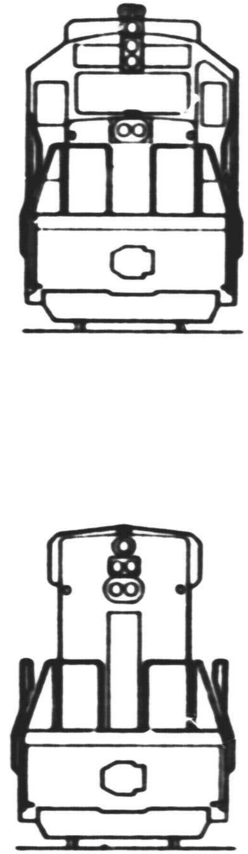
Dimensions and Specification



1. Engine - G.E. Model 7FDL16
2. Alternator
3. Auxiliary Generator
4. Rectifiers
5. Equipment Blower
6. Air Compressor
7. Gear Unit & Radiator Fan
8. Engine Exhaust Stack
9. Engine Air Filters
10. Engine Water Tank
11. Lube Oil Cooler
12. Lube Oil Filter
13. Radiator
14. Braking Resistors (Optional)
15. Sand Box
16. Number & Light Box
17. Sand Filler
18. Fluid Amplifier
19. Batteries
20. Control Compartment
21. Fuel Tank
22. Fuel Filler
23. Toilet (Optional)
24. Engine Control Panel
25. Control Stand
26. Air Brake Valve
27. Cab Heater
28. Sliding Seats
29. Hand Brake
30. Equipment Air Filters
31. Air Duct
32. Air Reservoir
33. Air Brake Equipment
34. Battery Switch
35. Ext. Range Brk. Equip. (Optional)
36. Head Lite Resistors

Figure 1-1. General Electric U30C Locomotive (cont'd)





General information.

Model designation	TE 70-4S
Locomotive ty[illegible]	Bo-Bo (0-4-4-0)
Horsepower	2800 HP
Maximum weight on rails	279,000 lbs.
Diesel engine	Sulzer 12ASV 25/30
Main generator	5GT598C4
Auxiliary generator	5GY27
Exciter	5GY50
Traction motors	5GE752E3
Cooling package	Behr-hydrostatic driven fans
Equipment blower	Hydrostatic driven
Air compressor	3CWDL
Air brake equipment	26L
Dynamic braking	Sta[illegible]dard range, flat
Batteries	420 amp.-hour (8)
Cab environment equipment	Electric heat/air conditioning
Gear ratio	74:18
Maximum speed	70 MPH
Tractive effort (cont.)	60,400 lbs.
Minimum cont[illegible]eed	10 MPH

Dimensions

Length (pulling face of couplers)	60'-2"
Height (over horn)	15'-10⅝" Bell 16'-5"
Width (outside armrests)	10'7½"
Center of bolsters	36'-2"
Axle spacing	9'-4"

Figure 1-2. Morrison-Knudsen TE70-4S Locomotive (From Ref. 1-2)

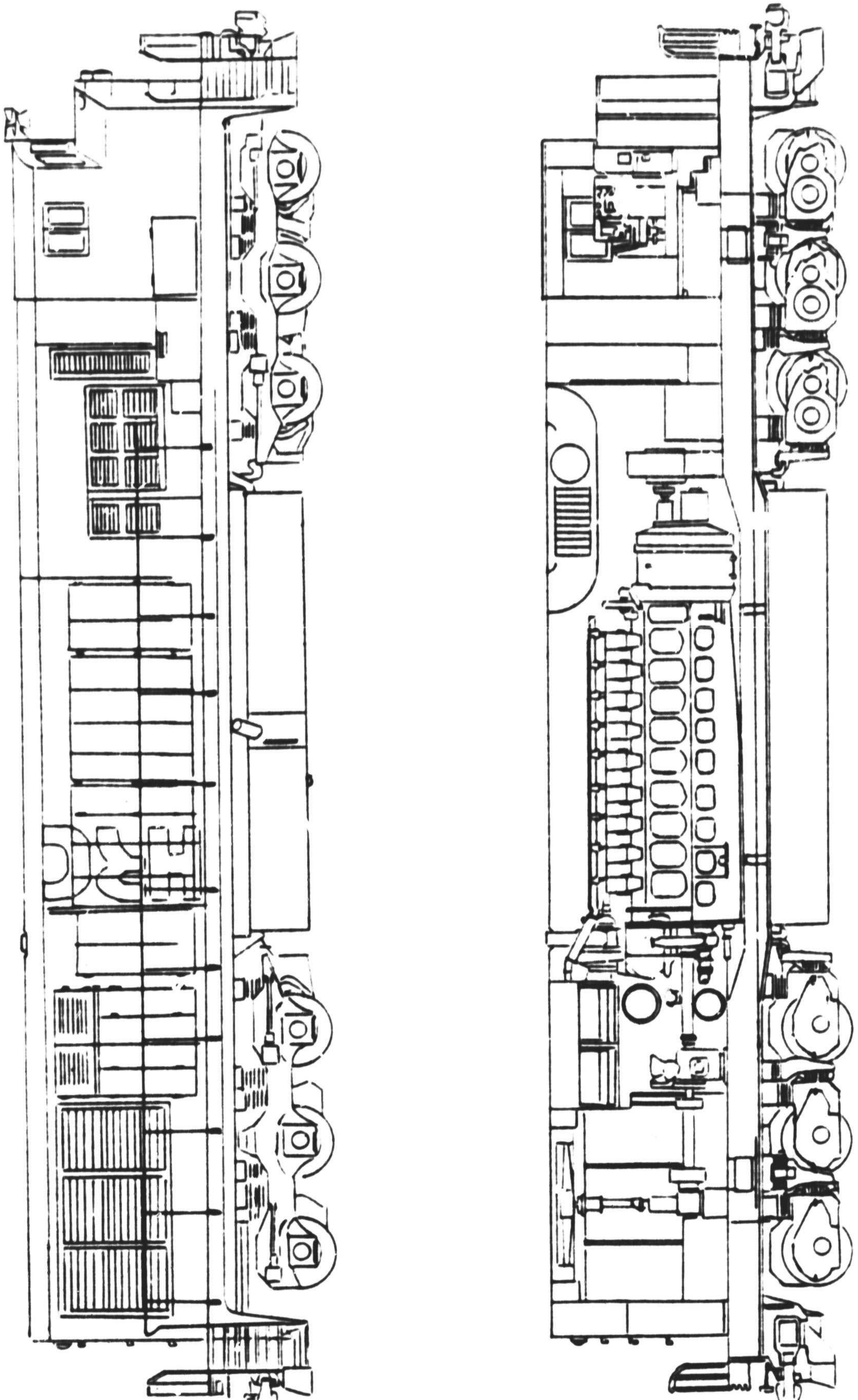


Figure 1-3. Bombardier HR-618 Locomotive
(From Ref. 1-3)

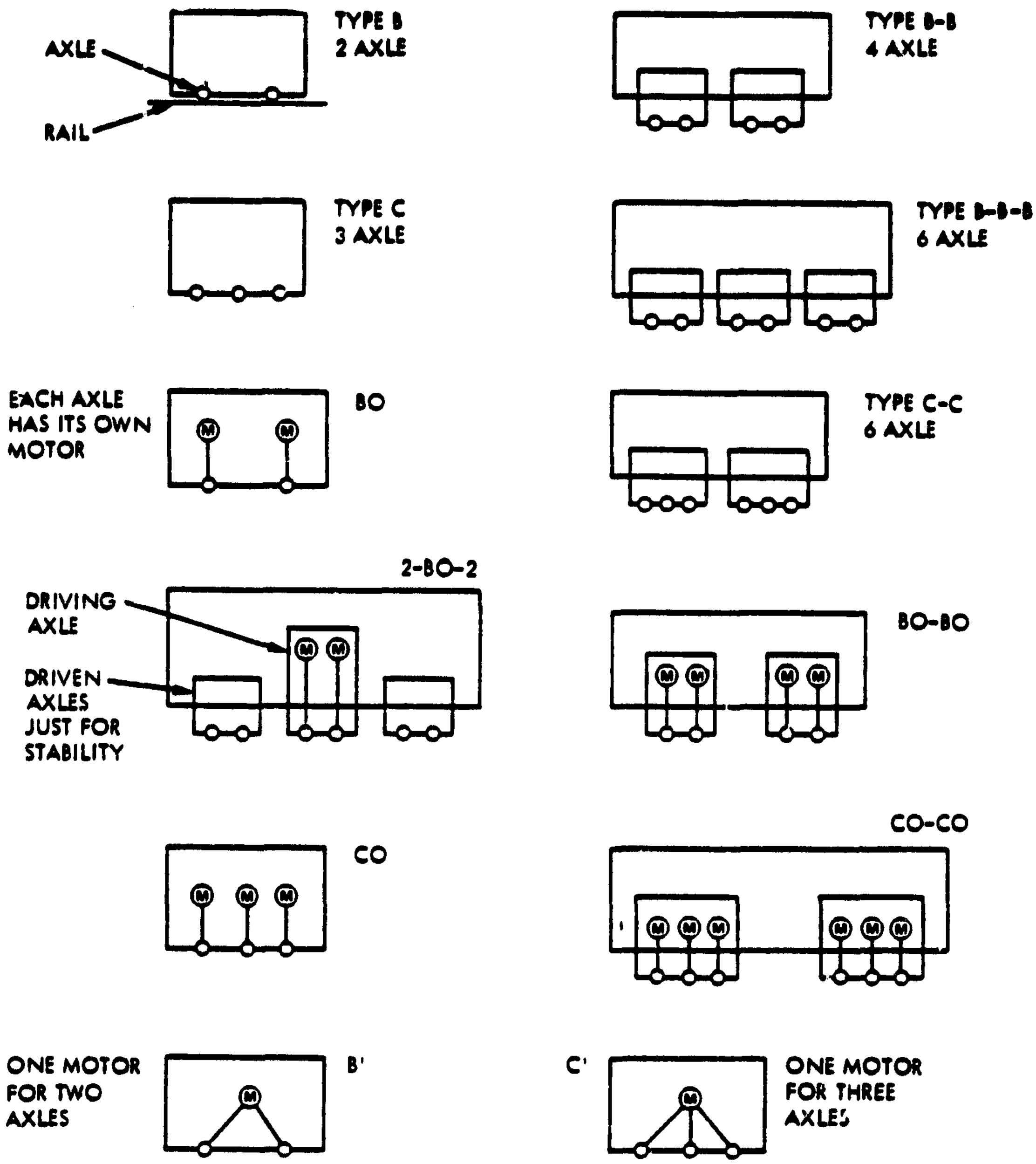


Figure 1-4. Locomotive Terminology

Table 1-1. Baseline Diesel-Electric Locomotives

Parameter	M-K TE70-4S Locomotive[a]	Bombardier HR-616 Locomotive[b]
Engine Power (hp)	2,800	3,200
Locomotive Length (ft)	60	69
Height (ft)	16	15
Width (ft)	10	10
Wheel Arrangement	B_o-B_o	C_o-C_o
Diameter (in.)	40	40
Maximum Speed (mph)	70	70
Gear Ratio for Maximum Speed	74/18	65/18
Total Loaded Weight (lb)	279,000	420,000
Total Loaded Weight Per Axle (lb)	70,000	70,000
Fuel Tank Capacity (gal)	N.A.[c]	N.A.
Engine Lube Oil Capacity (gal)	N.A.	N.A.
Cooling Water (gal)	N.A.	N.A.

Note: [a]Ref. 1-2
[b]Ref. 1-3
[c]N.A. - Not Available

Parameter	EMD Locomotive[d]	GE Locomotive[e]
Engine Power (hp)	3,000	3,000
Locomotive Length (ft)	59	67
Height (ft)	15	15
Width (ft)	10	10
Wheel Arrangement	B_o-B_o	C_o-C_o
Diameter (in.)	40	40
Maximum Speed (mph)	70	70
Gear Ratio for Maximum Speed	62/15	74/18
Total Loaded Weight (lb)	256,000	366,000
Total Loaded Weight Per Axle (lb)	64,000	61,000
Fuel Tank Capacity (gal)	3,600	4,000
Engine Lube Oil Capacity (gal)	243	380
Cooling Water (gal)	254	365

Note: [d]Ref. 1-5
[e]Ref. 1-4

The resistance of the journal bearings of freight cars (Ref. 1-6) is:

$$R_j = 1.3 + 29/W \text{ lb/ton}$$

where W is the car weight in ton/axle and N is the number of axles. The flange-rail resistance (Ref. 1-6) is only speed dependent and is:

$$R_f = 0.045 \text{ V lb/ton}$$

where V is the speed in miles per hour. The aerodynamic drag or wind resistance (Ref. 1-6) is given as:

$$R_w = 0.0005 \text{ AV}^2/\text{WN lb/ton}$$

where A is the frontal area of the car in ft^2. The sum of these three resistances is called the train resistance. The effect of grade is accounted for by a grade resistance (Ref. 1-7) which is defined by the following equation:

$$R_g = 20 \text{ P}_g \text{ lb/ton}$$

where P_g is the percent grade. The percent grade is defined as the number of feet of rise or fall per 100 ft of track. As a freight car goes around a curve, the flanges on the wheels and the rails interact to increase the resistance. This effect is known as curve resistance and its magnitude varies with the degree of curvature. The degree of curvature is defined as the number of degrees of angle subtended by a 100 ft length of track. The degree of curvature and the radius of curvature are related by the equation:

$$r_c d_c = 5730$$

where r_c is the radius of curvature in feet and d_c is the curvature in degrees. The curve resistance (Ref. 1-7) is:

$$R_c = 0.8 \text{ d}_c \text{ lb/ton}$$

The force per ton required to accelerate the train is given by the equation:

$$R_a = 91.17a \text{ lb/ton}$$

where a is the acceleration in mi/hr/s. The rotational acceleration of the wheels and axles of the freight car are small compared to the car itself and are ignored. The total resistance of the train is the sum of the six factors just discussed. These resistances, however, are in pounds of drawbar force per ton of train weight. It is necessary, therefore, to specify the train weight. Table 1-2 lists the empty weights and the capacities of a number of types of freight cars. The list is neither complete nor up-to-date but provides some idea of the weights involved.

Table 1-2. Freight Car Weights

Description	Weight Empty (lb)	Capacity (lb)	Length (ft)
Box, all steel, T&G lining	53,500	100,000	52
Box, all steel, plywood lined	43,300	100,000	42
Stock, steel frame	54,700	80,000	42
Refrigerator	56,600	79,000	42
Refrigerator	62,900	100,000	42
Refrigerator	95,000	40,000	53
Hopper, all steel	43,900	100,000	32
Hopper, all steel	48,400	140,000	42
Ore, all steel	54,600	190,000	42
Gondola, all steel	42,300	100,000	42
Gondola, all steel	59,000	140,000	47
Tank	45,700	10,000 gal	38
Flat	50,400	100,000	54
Flat	52,500	140,000	50
Auto transport	99,800	12 Std	86
Auto transport	99,800	15 Compact	86
Trailer flat (piggyback)	56,000	130,000	90

D. LOCOMOTIVE RESISTANCE AND TRACTIVE EFFORT CHARACTERISTICS

The locomotive, like the consist, has certain resistance forces acting on it. The bearing, grade, and curve resistance equations are the same for the locomotive as for the cars in the train. Some of the other equations are different from train resistance. The flange resistance is:

$$R_f = 0.03\ V\ \text{lb/ton}$$

where V is the speed in mi/hr. The aerodynamic or wind resistance (Ref. 1-7) is now given as:

$$R_w = 0.0024\ AV^2/WN\ \text{lb/ton}$$

where V and W are as defined previously. N is the number of axles and A is the frontal area in ft^2. Typical values of A are 105 ft^2 for a 50 ton locomotive, 110 ft^2 for a 70 ton locomotive and 120 ft^2 for a 100 ton locomotive. The coefficient is .0024 for the leading locomotive and the coefficient for trailing locomotives is .00034 per unit.

The acceleration resistance is modified by the addition of a term to compensate for the rotational inertia of the wheels, gears, and motors.

The equation for the sum of the translational and the rotational acceleration resistance (Ref. 1-7) is:

$$R_a = 100\ a\ \text{lb/ton}$$

where a is the acceleration in mi/hr/s.

The six resistances for the locomotive and the consist when multiplied by their respective weights and then summed, gives the total drag force of the train. This force is opposed by the tractive effort of the locomotive. The tractive effort is the horizontal force produced by the locomotive's powertrain on the rail. The tractive effort depends on the weight on the driving wheels, the rating of the engine, the characteristics of the electrical transmission, and the usable adhesion. Adhesion is the coefficient of traction between the wheel and the rail and is usually expressed as a percentage. By definition, it is the ratio of the horizontal force at the interface to the vertical force with the result multiplied by one hundred. Its value depends on the rail condition, weather, condition of the wheel tread, and the wheelslip equipment. The usable adhesion, in addition, depends on the grade and curvature of the track, on the drawbar strength, and on the length of the grade relative to the train length. Adhesion can be thought of as a force path linking the wheel and the rail. If the force produced at the wheel is too large, the path is broken and wheel slip occurs. Because adhesion, like the coefficient of traction involves both friction and mechanical engagement, it can be varied over a wide range by changes to either effect. A wet track lowers friction and adhesion dramatically. The usual method for overcoming slipping is to increase the mechanical engagement by spreading sand on the rail. Typically the starting adhesion is about 24% to 25% while running adhesion is more commonly 20% or less. The latest wheel slip control systems achieve running adhesions of about 25%. On dry rails with the use of sand, an adhesion of 50% can be achieved.

The tractive effort controls the acceleration of the train and tonnage hauled over grades. If the tractive effort is less than the total drag force, then the train will either decelerate if it is moving or be unable to move if it is at rest. If the tractive force matches the total drag, the train runs at a steady velocity known as the balancing speed.

The effect of tractive effort on tonnage is shown in Figure 1-5 (Ref. 1-8) which shows the number of trailing tons that can be hauled by a 65,000 lb per axle locomotive over a grade. Using this curve, the number of axles required to haul a given size train up a specific grade can be determined. For example, if the ruling grade is one percent, each axle can pull 500 tons. If the train size is 6,000 trailing tons, a total of 12 axles is required to handle this train. These twelve axles could be on two C_o-C_o locomotives with six axles each or three B_o-B_o locomotives of four axles each. Figure 1-6 through 1-10 show typical tractive effort curves (Ref. 1-4 and 1-5) for EMD, GE, M-K, and Bombardier locomotives.

In addition to tractive effort there is locomotive power. There are, however, several different power parameters of interest, such as engine power, alternator or generator power, traction motor power, and rail power. The rail power is the product of the speed and the tractive effort. The

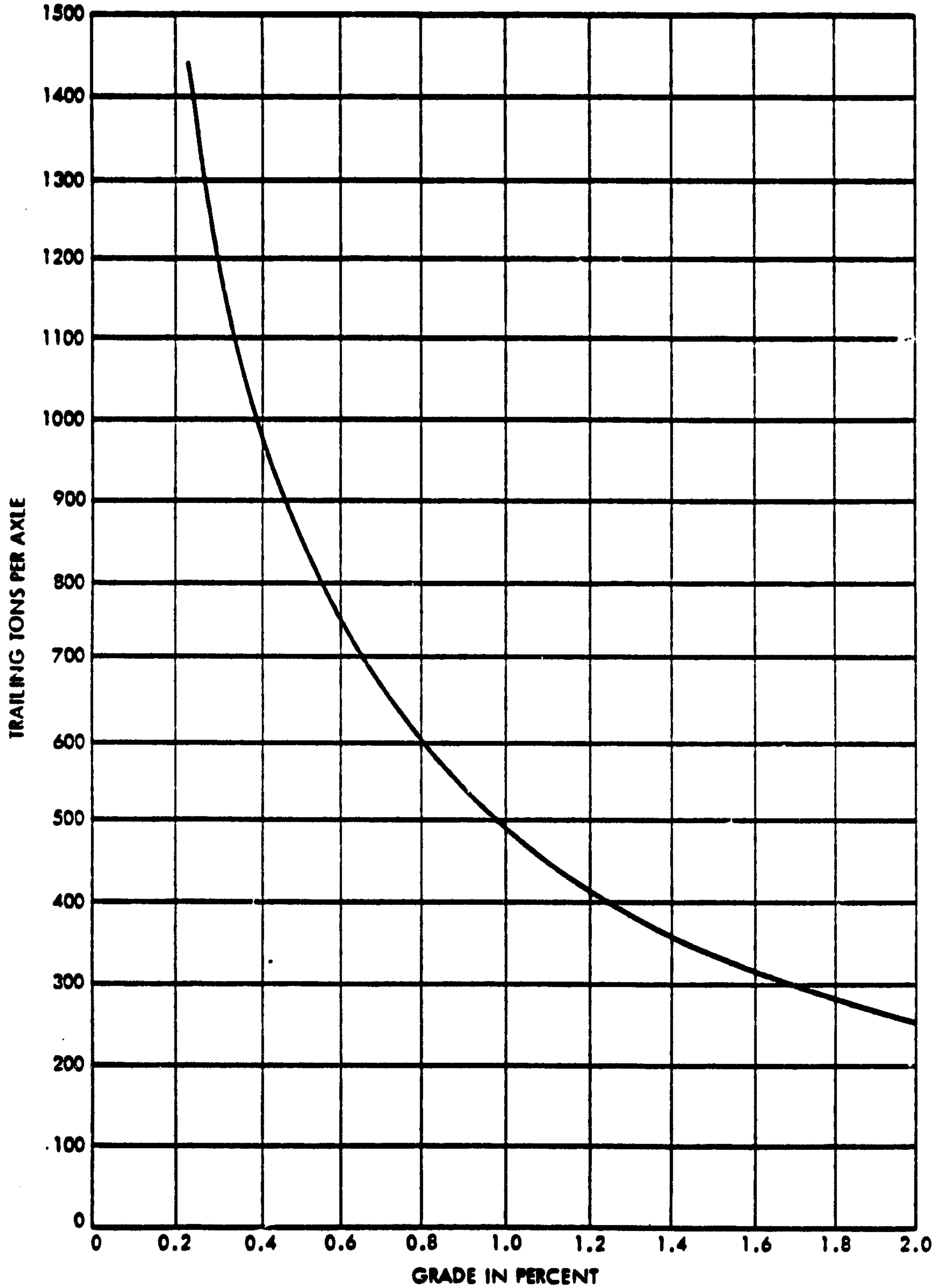


Figure 1-5. Grade Performance of 65,000 lb per Axle Locomotive

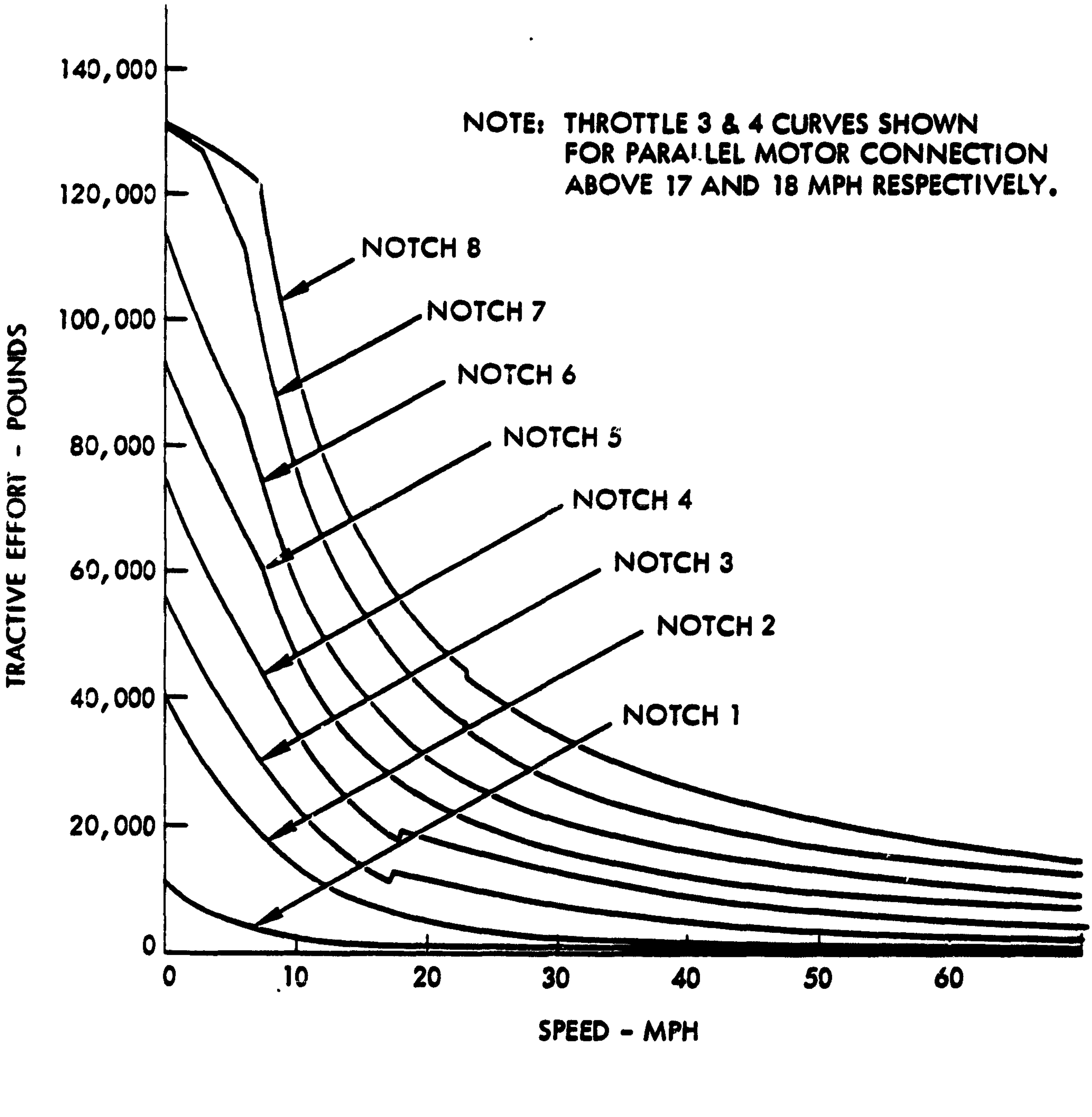


Figure 1-6. Tractive Performance of EMD SD40-2 Locomotive (From Ref. 1-5)

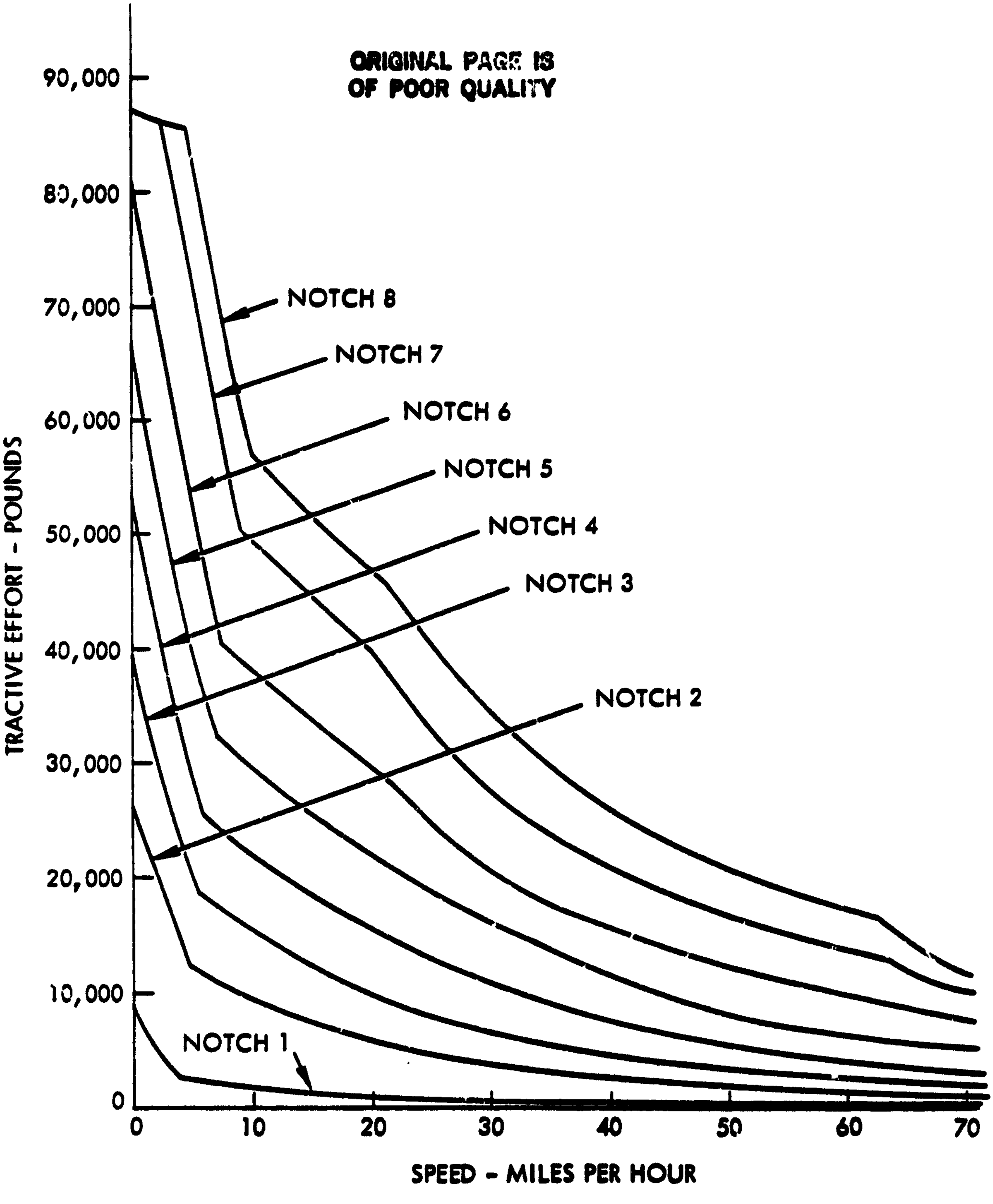


Figure 1-7. Tractive Effort vs. Speed for GP40-2 Locomotive (From Ref. 1-5)

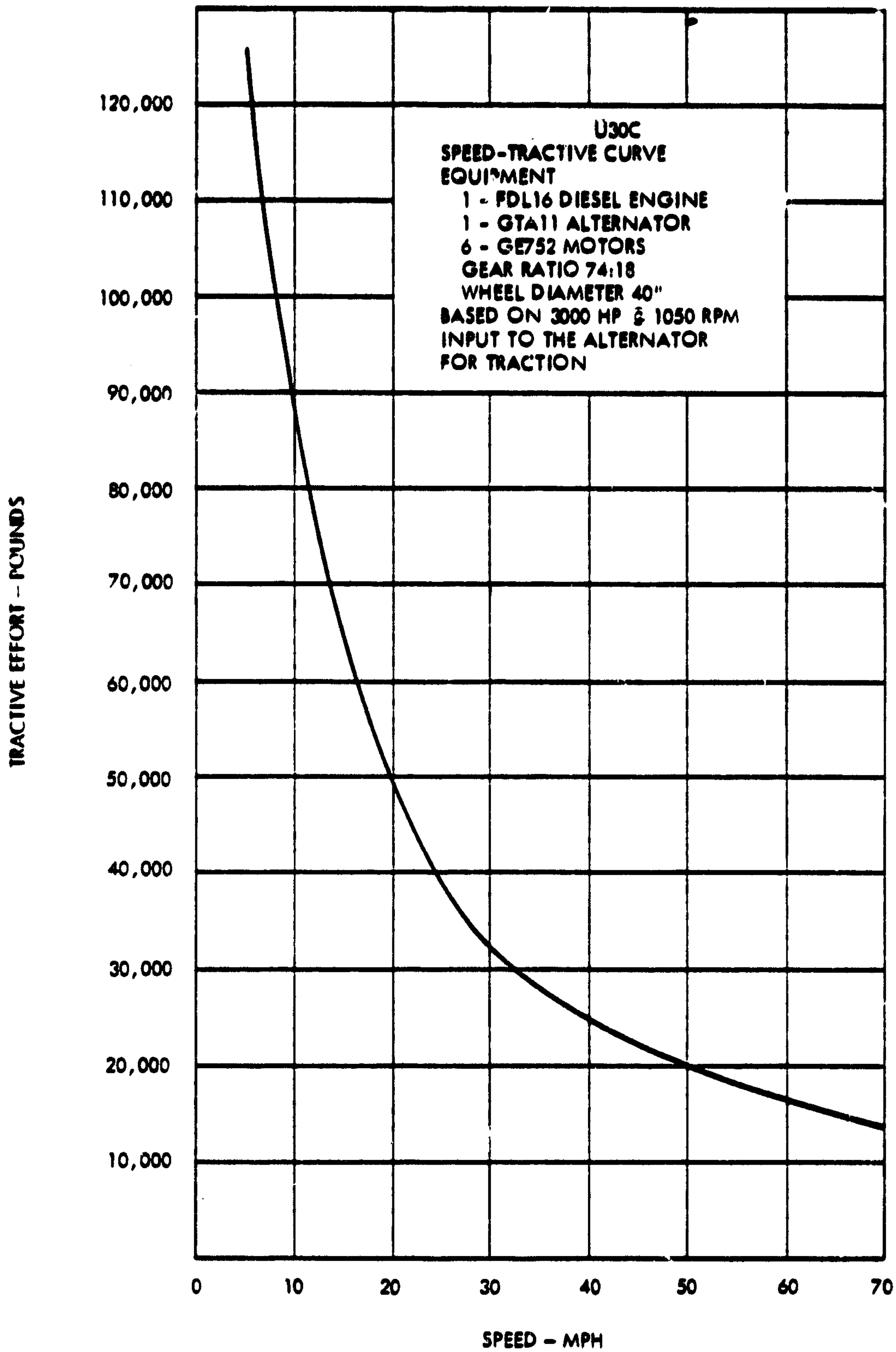


Figure 1-8. U30C Locomotive Tractive Effort
(From Ref. 1-1)

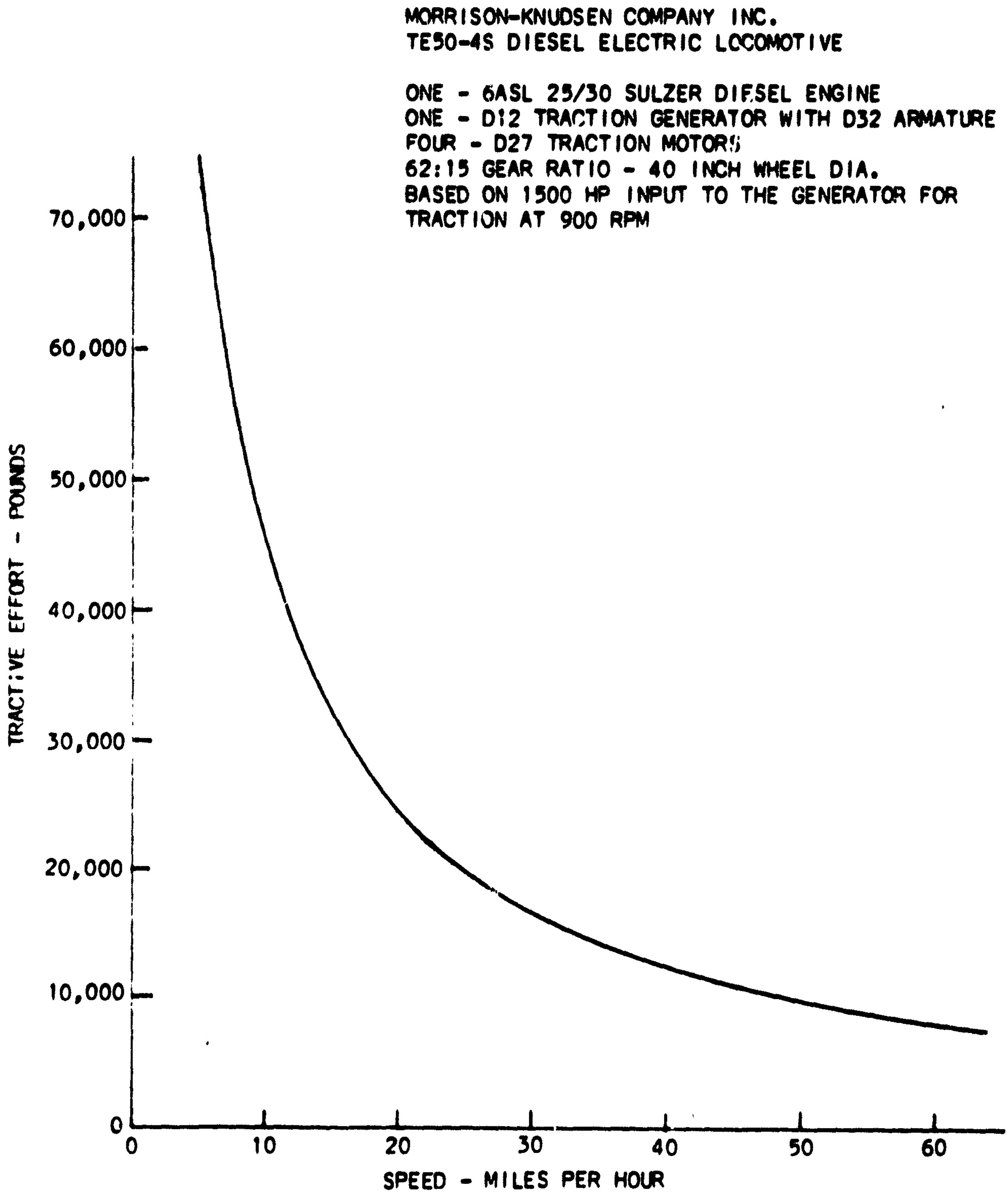


Figure 1-9. TE50-4S Locomotive Tractive Effort
(From Ref. 1-2)

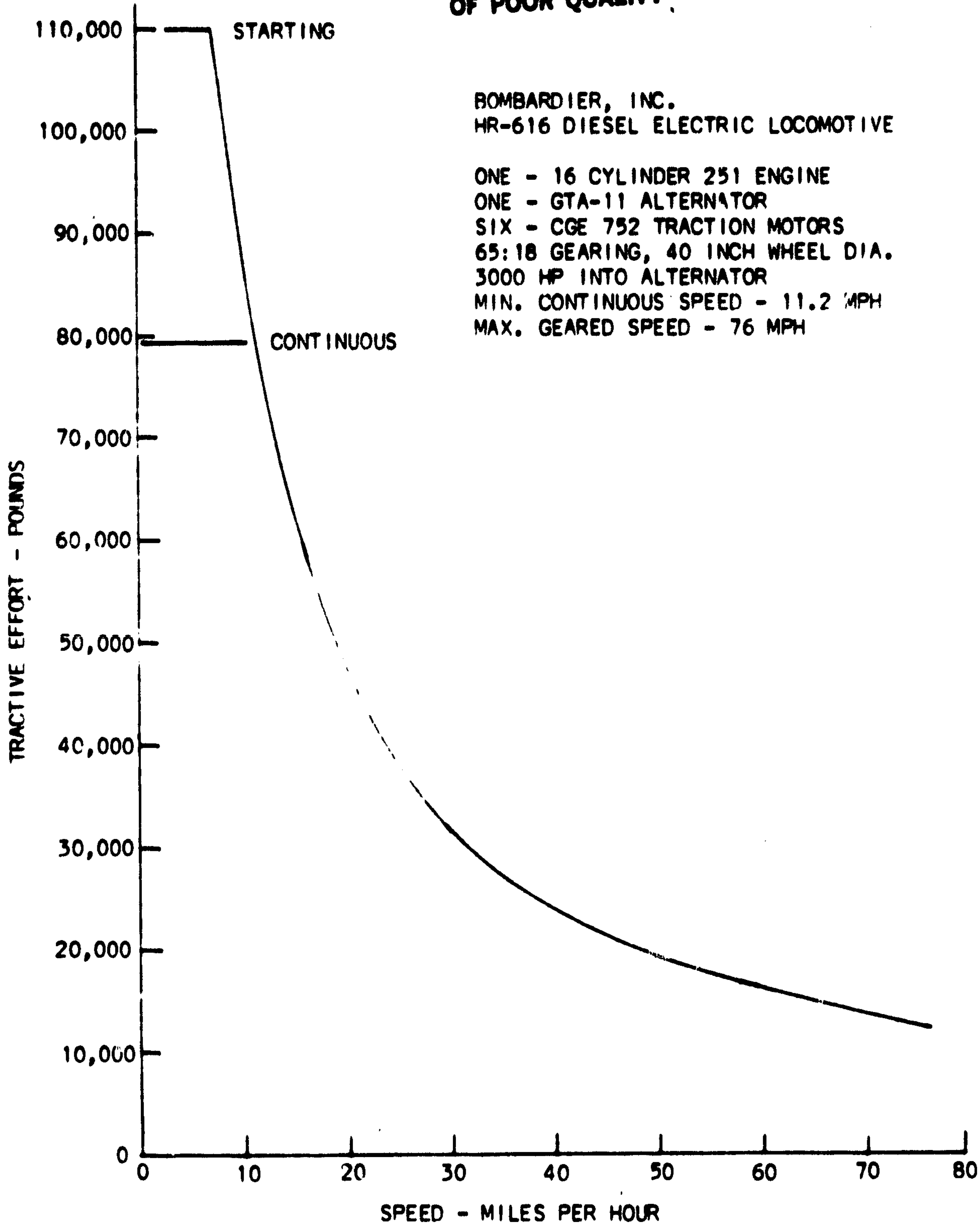


Figure 1-10. Tractive Effort for Bombardier HR-616 Locomotive (From Ref. 1-3)

rail power is ultimately limited by the output of the engine and the losses in the powertrain. The higher the available rail power, the higher the maximum speed and the lower the time necessary to complete a trip. Figure 1-11 (Ref. 1-5) shows the effect of power and of track speed limits on travel time. If the available power is low, the speed limits have little or no effect on the elapsed time of travel. The train simply cannot go fast enough to be affected by the limits. Increasing the power decreases the time in motion if the speed limits are high. If the speed limits are low, the engine is no longer a limiting factor and more power has little effect on elapsed time.

From an energy conservation standpoint, it is better to be power limited than speed limited; however, economics may favor a higher average speed. The optimum path for fuel savings is to use only those locomotives having enough power to just meet the schedule. Using engines with higher power ratings is wasteful.

DIESEL ENGINES

The railroad locomotives in the United States and Canada are primarily powered by Diesel engines designed and manufactured by four companies - General Motors (Electro-Motive Division), General Electric (Transportation Systems Division), Bombardier Inc., and Sulzer Bros. (Morrison-Knudsen). The basic difference in the designs is that GM manufactures two-stroke engines, whereas the other companies manufacture four-stroke engines. For a given engine speed, two-stroke engines deliver higher horsepower per unit weight of the engine and the four-stroke engines deliver more brake horsepower-hours per pound of fuel consumed. For example, at the same brake horsepower ratings, the two-stroke EMD engines are about 25% lighter than four-stroke GE engines but they consume 5% to 7% more fuel. The best brake specific fuel consumption (BSFC) of the present four-stroke engines is approximately 0.34 lb/bhp-hr. The BSFC is an important economic consideration, because a typical heavy-duty Diesel engine consumes enough fuel in one year to pay the cost of a bare engine. The fuel consumption of these engines is important but their contribution to air pollution must also be considered. Diesel engines, in general, are minor contributors to the overall pollution problem. Smoke, however, is troublesome especially for four-stroke engines at low speeds. The present day turbocharged locomotive Diesel has significantly reduced smoke compared to older engines even during transient conditions where it is particularly severe. The smoke from four-stroke engines may be reduced by going to higher engine speeds but fuel economy will be adversely affected. Under these conditions, two-stroke engines may yield better fuel economy than four-stroke engines if they have to meet the same exhaust gas emissions requirements.

The current Diesel engines used in locomotive applications are either Incline, 55° Vee, or 45° Vee engines having brake horse power ratings of 1000-4000 hp depending upon their use in passenger or freight service and also if they are in light, medium or heavy freight service. The number of cylinders in these engines varies from 8-20 (in steps of 4), with a 9 to 10.5 in. bore and 10 to 12 in. stroke. The engines employed for medium and heavy duty freight services are normally 12 or 16 cylinder engines having horsepower ratings of about 3300 bhp. The GM two-stroke, 16 cylinder, 3000 bhp engine weighs 36,250 lb and the GE four-stroke, 16 cylinder,

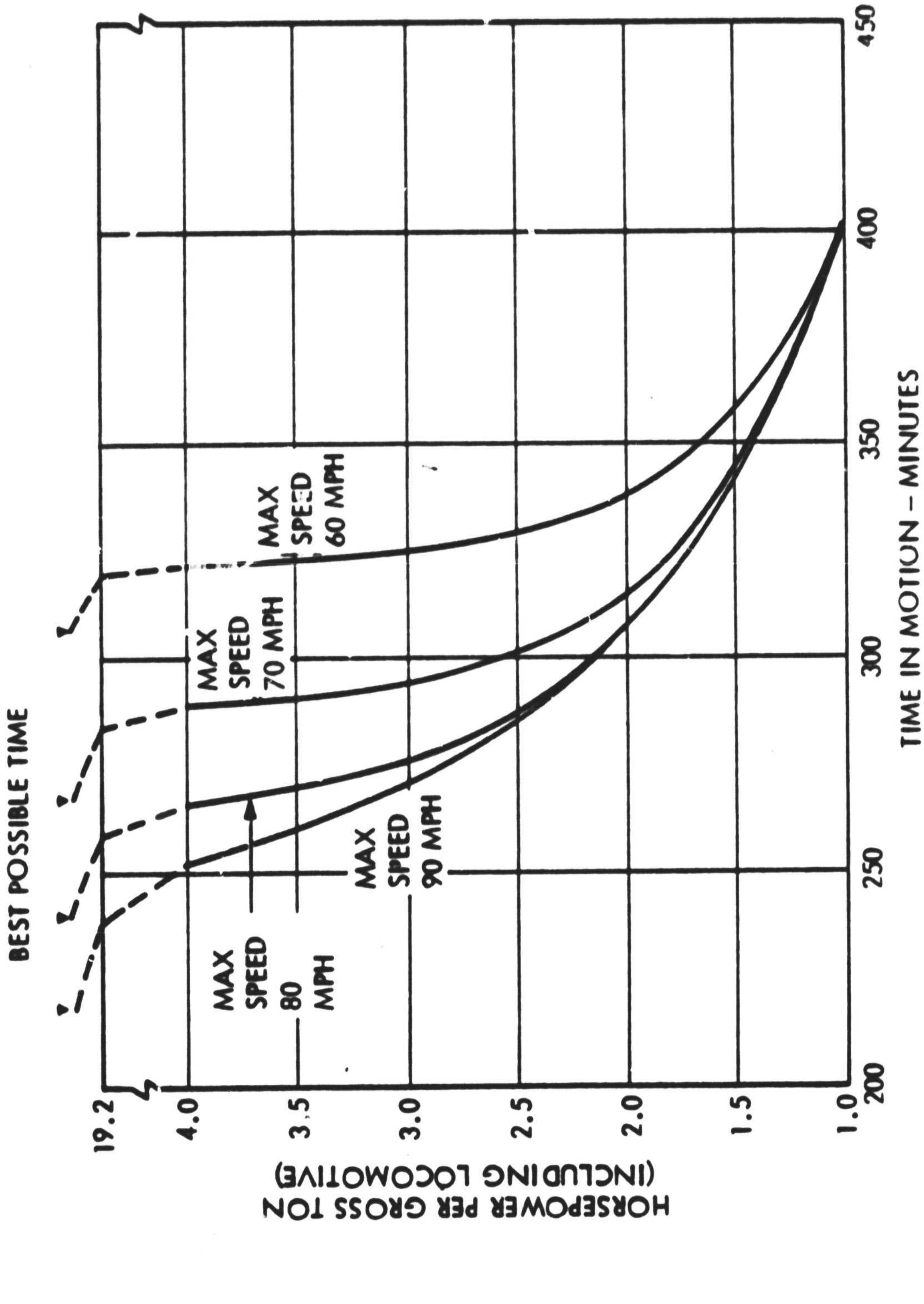


Figure 1-11. Travel Time Performance

engine delivering the same horsepower weighs 45,000 lb. Figure 1-12 shows the variation in weight for typical EMD, GE, and Sulzer engines as a function of number of cylinders (Refs. 1-2, 1-9, and 1-10).

Both the EMD and GE turbocharged engines are rated at 180 to 230 hp per cylinder depending on the number of cylinders, the boost pressure and the fuel injection system settings. The Sulzer engine is rated at 270 hp per cylinder. In general, the bore and stroke are the same for all the engines made by a manufacturer in one of his classes of engine. Table 1-3 is a list of engine parameters and their values for a number of typical turbocharged engines in locomotive use. This list is by no means complete but illustrates the range of values normally encountered.

Most of the present day engines used on line locomotives are turbocharged because turbocharging significantly augments the engine power density and also improves the fuel economy by using engine exhaust heat. The engine inlet pressure is increased to about 2 to 2.5 atm (absolute) through a compressor powered by the exhaust driven turbine. The power level and fuel economy of the engine is further increased by cooling the compressor outlet air before it enters the engine.

The compression ratios used in these turbocharged engines range from 12.7 to 14.5 and the ratios used in naturally aspirated and Roots blown type engines are typically 16.0. The maximum cylinder pressures are limited to the 1600 to 2000 psia range because of cost and reliability tradeoffs. Increasing the maximum cylinder pressures will improve the fuel economy but will require stronger components to withstand increased mechanial and thermal stresses. The higher stresses will, in turn, increase engine cost and weight.

The range of engine speeds from idle to maximum power is 255 to 1050 rpm. The engine speed is controlled by a series of notches or throttle positions. There are 8 power notches designated as notch 1 through notch 8 and an idle notch. The speed and power levels increase at each progressive notch and full power is attained at notch 8. In naturally aspirated or Roots blown Diesel engines, the highest efficiency or lowest BSFC is attained in notch 5 or 6. The turbocharged Diesel has its best BSFC in notch 8 or maximum power. Because main line locomotives operate most of the time in notch 8, the turbocharged engine has taken over this type of service. In switching applications where notch 7 and 8 are rarely used, the best fuel efficiency is achieved by the non-turbocharged engines.

Of more importance than the peak fuel efficiency is the average efficiency in actual operation. Poor part load efficiency can completely overshadow good peak efficiency. To estimate the actual fuel consumption, duty cycles which represent typical operating conditions are employed. The interaction of the engine and the duty cycles is the controlling factor in the fuel efficiency of the locomotive.

Figure 1-13 is the engine map of a typical two-stroke Diesel engine used in locomotive service (Ref. 1-11). On this figure, engine power is plotted against engine speed with contour lines of constant BSFC superimposed. The heavy line shows typical railroad service. The notch position would lie along this line with notch 8 at the top end near the

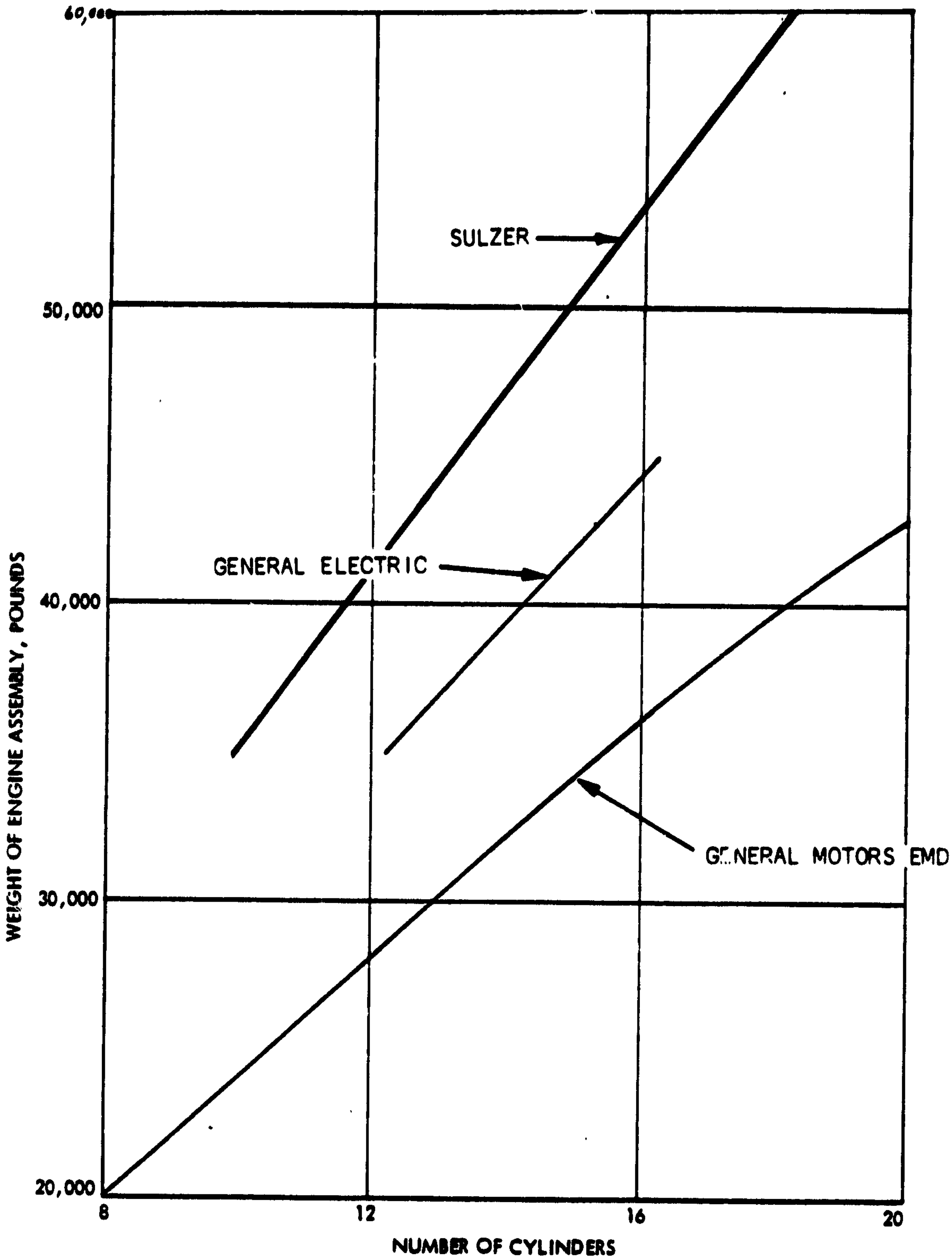


Figure 1-12. Engine Assembly Weights

Table 1-3. Base Engine Information of Diesel Engines for Application in Diesel-Electric Locomotives

Specifications	GM-EMD-645E3 Diesel Engine	GE-7FDL16 Diesel Engine
Cylinder Arrangement	45°V	45°V
Stroke Cycle	2	4
Bore x Stroke, Inches	9 1/16 x 10	9 x 10.5
No. Of Cylinders	16	16
Compression Ratio	14.5	12.7
Turbocharge Pressure, Atmospheres	2.4	2
Maximum Cylinder Pressure, psia	1,525	2,000
Idle Engine Speed, rpm	315	450
Maximum Engine Speed, rpm	900	1,050
Brake Mean Effective Pressure, psig	141	247
Gross Engine Power, (bhp)	3,300	3,300
Brake Specific Fuel Consumption, (lbm/bhp-hr)	0.351	0.345
Weight Of Engine Assembly, lb	36,300	45,000
Gross Power/Unit Engine Weight, (hp/lb)	0.0909	0.0733
Height (Overall-Including Stack), ft	N.A.[b]	9
Length (Overall-Including Generator), ft	N.A.	22
Width (Overall), ft	N.A.	6

Source: Ref. 1.9 and 1.0

Specifications	Sulzer[a] 125v25/30 Diesel Engine	Bombardier Diesel Engine
Cylinder Arrangement	55°V	V
Stroke Cycle	4	4
Bore x Stroke, Inch	9.84 x 11.81	N.A.
No. Of Cylinders	12	16
Compression Ratio	12.7	N.A.
Turbocharge Pressure, Atmospheres	N.A.	N.A.
Maximum Cylinder Pressure, psia	N.A.	N.A.
Idle Engine Speed, rpm	N.A.	400
Maximum Engine Speed, rpm	1,000	1,000
Brake Mean Effective Pressure, psig	235	N.A.
Gross Engine Power, (bhp)	3,240	3,450
Brake Specific Fuel Consumption, (lbm/bhp-hr)	N.A.	0.348
Weight Of Engine Assembly, lb	41,600	N.A.
Gross Power/Unit Engine Weight, (hp/lb)	0.0779	N.A.
Height (Overall-Including Stack), ft	N.A.	N.A.
Length (Overall-Including Generator), ft	N.A.	N.A.
Width (Overall), ft	N.A.	N.A.

[a] Used in Morrison-Knudsen TE70-45 Locomotive
[b] N.A. - Not Available
Sources: Refs. 1-2 and 1-3

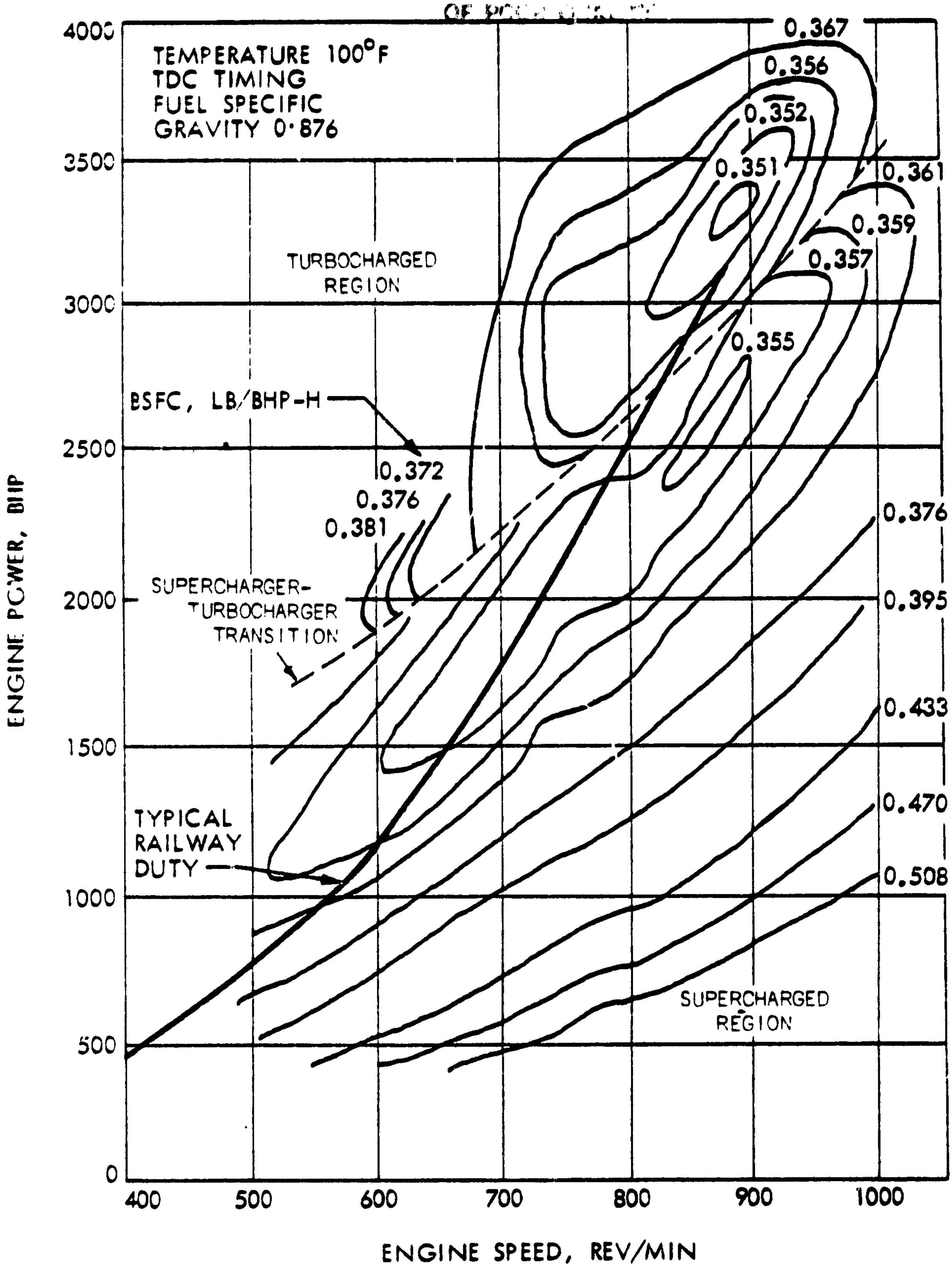


Figure 1-13. Two-Stroke Diesel Engine Map
(From Ref. 1-11)

best BSFC island. Notch 1 would be down near the 315 rpm mark and idle is also about 315 rpm but near zero engine power. In notch 1, the BSFC will be about 0.47 and the best BSFC is .34 at notch 8. This means that the BSFC is 38% higher at the low end of the range than at the top. This engine is a supercharged-turbocharged engine. The dashed line shows the dividing line between these two modes of operation. Above the line, the engine is turbocharged. Below it, the engine is supercharged with power from the crankshaft being used to drive the compressor. This relatively small change in efficiency is one of the factors that make Diesels so desirable for locomotive applications and for applications in other land vehicles. Another way to present the fuel usage of the engine is shown in Figures 1-14 and 1-15. Here, the fuel rate in pounds per hour is shown as a function of locomotive speed and notch position.

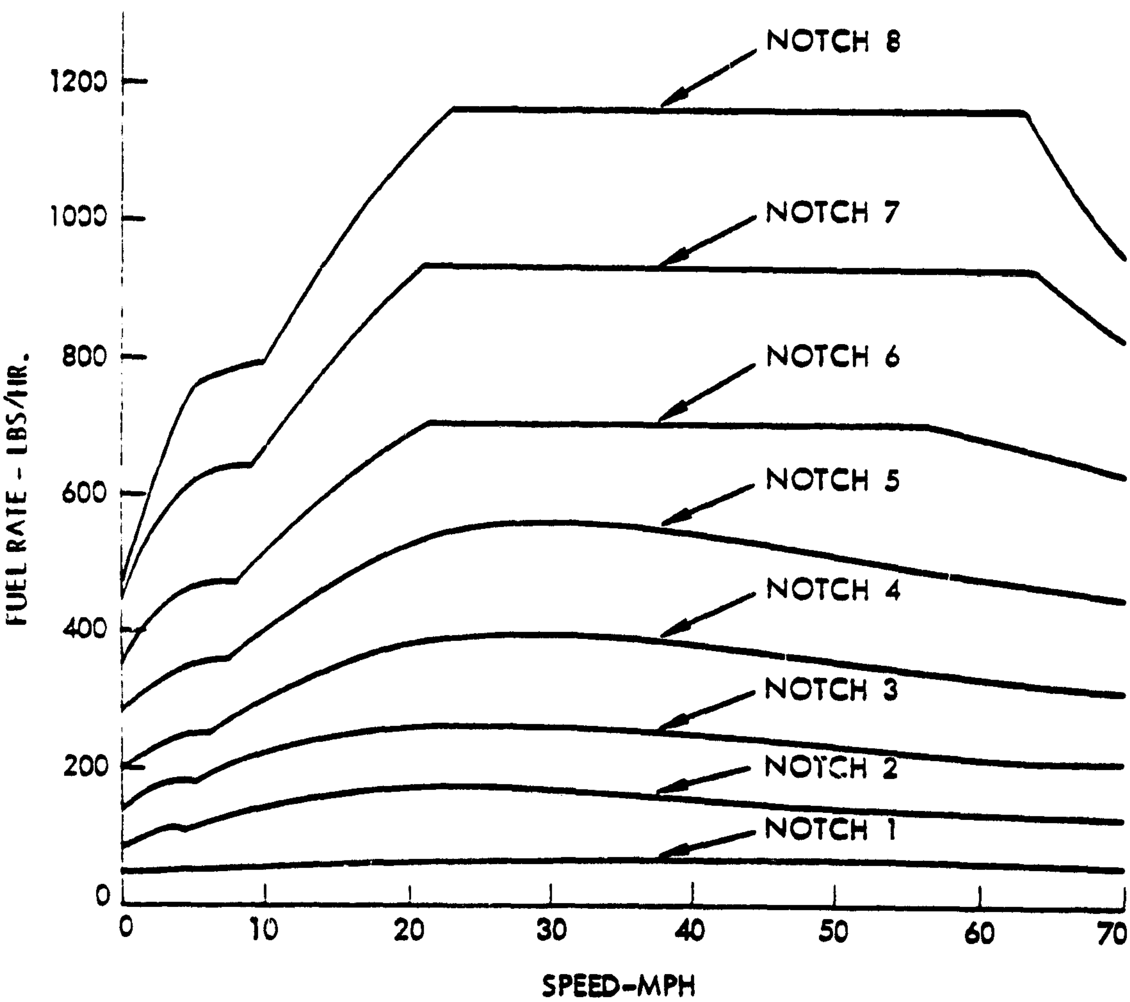


Figure 1-14. Fuel Consumption Rate vs. Speed and Throttle
EMD GP40-2 Locomotive (From Ref. 1-12)

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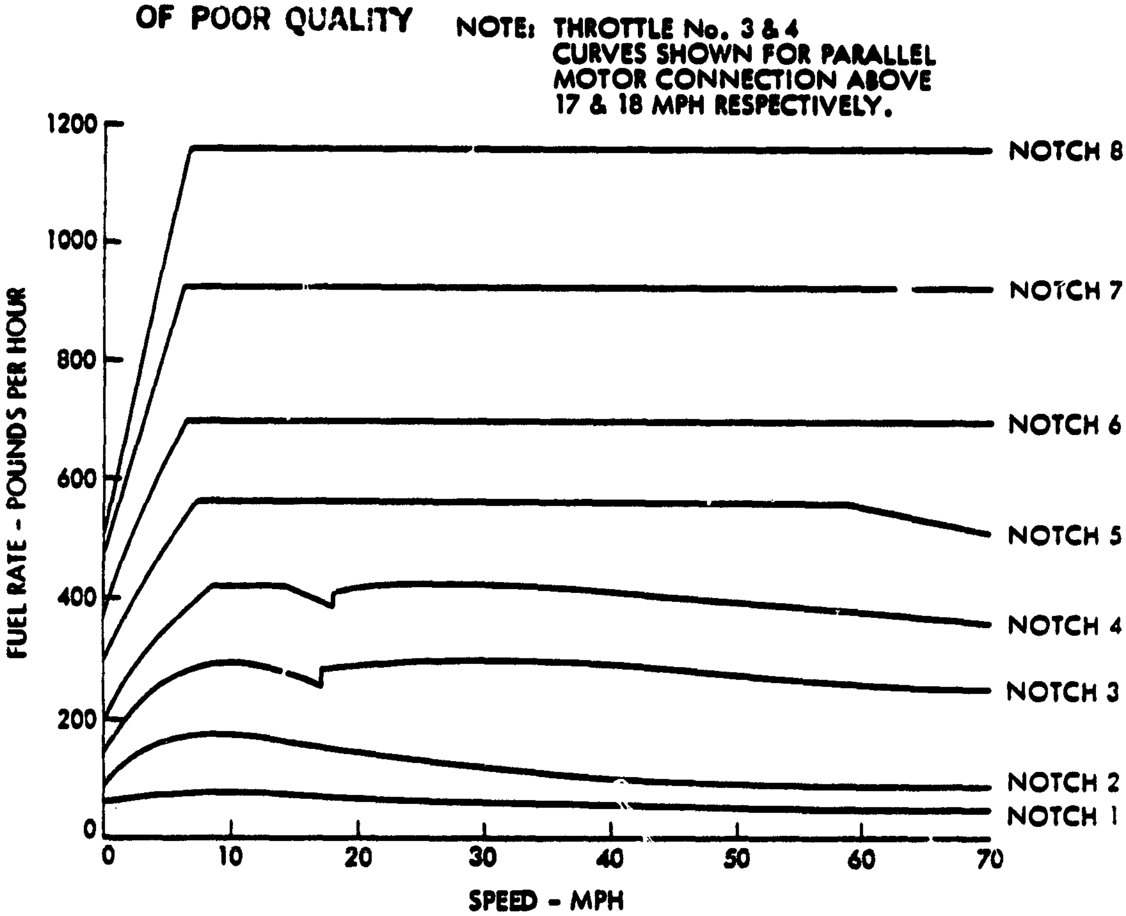

Figure 1-15. Fuel Consumption of SD40-2 Locomotive (From Ref. 1-13)

In Table 1-4, fuel consumption details for four engines are presented. Engine speed, engine power, accessory power, fuel flow rate, and gross brake specific fuel consumption (BSFC) are shown as functions of notch position. Gross brake specific fuel consumption is the fuel flow in pounds per hour for each gross horsepower output of the engine. Air-fuel ratio, where available, is also given.

The fuel consumption of a number of General Motors locomotives, both current production and older engines, are shown in Table 1-5. The numbers without parenthesis are the fuel flow rates in gallons per hour. The figures with parentheses are the gross BSFC in pounds per horsepower hour. The SD/GP39, SD/GP40, and SD45/SD45-2 engines are turbocharged, and the SD/GP38-2, SD/GP38, MP15/GP-1, and SW-1001 are Roots-blown engines. The minimum BSFC for the turbocharged engines occurs in notches 7 and 8. The minimum BSFC for the Roots-blown engines occurs in notch 5 and is about one to three percentage points higher. Low BSFC corresponds to high thermal efficiency, they are reciprocals of one another. The turbocharged Diesel engine shows a significant fuel consumption reduction when compared to the older non-turbocharged engines.

Table 1-4. Fuel Consumption Details of Typical Engines
General Electric C30-7 (From Ref. 1-14)

Notch or Throttle Position	Engine rpm	Gross Horsepower(1)	Accessory Horsepower	Net hp for Traction (2)	Fuel Flow Rate, lb/hr	Gross BSFC Fuel Rate, lb/bhp-hr(3)	Air-Fuel Ratio
1	450	125	25	100	60	.463	38
2	535	315	40	275	130	.406	24
3	705	650	75	575	255	.392	24
4	765	1055	95	960	385	.365	24
5	878	1580	140	1440	560	.355	28
6	878	2070	140	1930	715	.345	32
7	965	2680	180	2500	900	.336	32
8	1050	3230	230	3000	1085	.336	32
Dynamic Braking				102 lb/hr			
Idle	High (450 rpm)	25	25	31 lb/hr		1.24	76
	Low (385 rpm)			28 lb/hr			

(1) Gross horsepower corrected as follows:
Barometer 28.86 in Hg. (1000 ft altitude)
Ambient Air Temperature 60°F
Fuel in Temperature 60°F
Higher Fuel Heating Value - 138,000 Btu/gal (36° API) AAR Standard

(2) Net horsepower corrected as follows:
Air Compressor Unloaded
Normal C30-7 Fan and Gear Box Load
Battery Charger
Blower Load

(3) Specific Fuel Consumption corrected to inlet conditions:
Higher Heating Value - 19,350 Btu/lb (28° API) DEMA Standard

Table 1-4. Fuel Consumption Details of Typical Engines EMD GP40-2/SD 40-2 Engine[a] (cont'd) (From Ref. 1-13)

Notch or Throttle Position	Engine rpm	Gross Horsepower (1)	Accessory Horsepower	Net hp for Traction (2)	Fuel Flow Rate, lb/hr	Gross BSFC Fuel Rate, lb/bhp-hr (3)	Air-Fuel Ratio
1	316	114	17	97	65	.571	180
2	387	452	25	427	179	.395	75
3	488	824	41	783	310	.377	55
4	566	1148	58	1090	426	.371	47
5	650	1547	83	1464	560	.362	43
6	730	1964	112	1852	704	.359	38
7	820	2636	154	2482	936	.355	34
8	900	3295	199	3096	1156	.351	34
Idle	316	17	17	N.A.[b]	36	2.092	310
Low Idle	255	12	N.A.	N.A.	29	2.319	350
Dynamic Braking 4	566	145	N.A.	N.A.	130	.896	165
Dynamic Braking 1	316	103	N.A.	N.A.	63	.612	190

[a] Data pertain to locomotives equipped with current EMD Model E3B engines.
[b] N.A. - Not Available

Table 1-4. Fuel Consumption Details of Typical Engines
Morrison-Knudsen TE-50 (From Ref. 1-2)

Notch or Throttle Position	Engine rpm	Gross Horsepower(1)	Accessory Horsepower	Net hp for Traction (2)	Fuel Flow Rate, lb/hr	Gross BSFC Fuel Rate, lb/bhp-hr(3)	Air-Fuel Ratio
1	N.A.[a]	128	16	112	55	.437	N.A.
2	N.A.	241	65	175	96	.399	N.A.
3	N.A.	446	89	357	168	.377	N.A.
4	N.A.	667	129	538	237	.356	N.A.
5	N.A.	872	140	732	305	.350	N.A.
6	N.A.	1213	164	1049	413	.341	N.A.
7	N.A.	1447	176	1271	491	.339	N.A.
8	900	1644	192	1452	558	.339	N.A.
Idle		17	16	N.A.	17	.941	N.A.
Dynamic Braking							

[a]N.A. - Not available

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Table 1-4. Fuel Consumption Details of Typical Engines Bombardier HR-615 (From Ref. 1-3)

Notch or Throttle Position	Engine rpm	Gross Horsepower(1)	Accessory Horsepower	Net hp for Traction (2)	Fuel Flow Rate, lb/hr	Gross BSFC Fuel Rate, lb/bhp-hr(3)	Air-Fuel Ratio
1	400	224	19	205	97	.433	37
2	486	401	34	367	166	.414	30
3	571	653	56	597	264	.404	27
4	657	993	35	908	394	.397	26
5	743	1435	123	1312	549	.383	27
6	829	1939	169	1820	724	.364	29
7	914	2675	229	2446	940	.351	31
8	1000	3450	250	3200	1200	.348	33
Idle Dynamic Braking		N.A.[a]	N.A.	N.A.	24	N.A.	135

[a]N.A. - Not available

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Table 1-5. Fuel Consumption of GM-EMD Engines
Gallons per Hour (From Ref. 1-5)

Throttle	SD45[a] SD45-2	SD/GP40[a] SD/GP40-2	SD/GP39[a] SD/GP39-2	SD/GP38-2	SD/GP38	MP15/GP15-1	SW-1001
1	16.4 (.463)	9.3 (.571)	7.9 (.514)	7.6 (N.A.)	7.6 (N.A.)	N.A. (N.A.)	N.A. (N.A.)
2	2[illegible].2 (.417)	25.4 (.395)	23.1 (.385)	16.0 (.434)	15.4 (.427)	11.5 (.425)	6.0 (.420)
3	46.4 (.382)	44.1 (.377)	32.3 (.375)	31.3 (.380)	28.3 (.381)	25.2 (.394)	13.3 (.374)
4	67.8 (.372)	60.5 (.371)	47.1 (.369)	46.8 (.366)	47.1 (.364)	38.3 (.383)	22.1 (.367)
5	88.1 (.364)	79.7 (.362)	58.3 (.367)	63.8 (.365)	66.2 (.362)	52.4 (.381)	31.2 (.367)
6	117.3 (.360)	100.2 (.359)	78.7 (.363)	83.1 (.375)	85.7 (.373)	67.1 (.385)	39.8 (.369)
7	167.8 (.350)	133.0 (.355)	106.4 (.356)	102.8 (.385)	105.4 (.391)	79.4 (.391)	49.6 (.376)
8	195.8 (.352)	164.4 (.351)	126.2 (.350)	122.4 (.395)	128.4 (.413)[b]	92.[illegible] (.404)	60.0 (.388)
Idle	5.9	5.5	4.0	4.6	4.6	3.8	3.0
Low Idle	4.4	4.1	3.[illegible]	3.6	3.6	3.0	2.3
Dynamic Braking 4	18.6	18.5	11.9	15.0 (Thr. 5)	N.A.[c]	N.A.	N.A.
Dynamic Braking 1	9.5	8.9	6.4	N.A.	N.A.	N.A.	N.A.

Notes: ( ) Gross BSFC - lbs of fuel per gross hp-hr

[a]Data pertain to locomotives equipped with current EMD Model E3B engine.

Fuel Specific Gravity	.845
[b]Fuel Specific Gravity	.872
Barometric Pressure	29.92 in Hg
Air Intake Temperature	60° F

[c]N.A. - Not Available

F. ELECTRICAL EQUIPMENT

The primary function of the electrical traction equipment is to transfer the power developed at the output of the engine to the tractive effort at the wheel rims. Because the Diesel engine operates over a limited speed range, the power transfer equipment must provide the required wheel torque over the speed range from zero to the maximum running speed.

The electrical transmission system is ideally suited to the task. It is not the only possible transmission, however; hydraulic transmission locomotives have been used both in Germany and in the U.S. Hydraulic systems are widely used in the heavy construction industry where requirements are similar to railroad requirements. The electrical system is generally more efficient and is, by far, the most widely accepted system. For general heavy duty rail applications it has no effective competitor in the U.S.

Historically, direct current (dc) electrical equipment has been used. The speed and torque characteristics of the dc series motor and its short time overload capabilities are ideally suited to the tractive effort and speed requirements of general railroading. At one time, many other electrical systems have been successfully used although, in most cases, only on all-electric railroads with catenary or third rail power feed. These systems used single or polyphase, fixed frequency alternating current (ac) with synchronous, induction or commutator motors. The newest system, is briefly described in the sections on traction motors. It uses a combination of variable voltage and variable frequency with three-phase motors.

G. GENERATOR-ALTERNATOR

The first Diesel electric units used a dc generator feeding dc series motors that are connected in series, series-parallel, and parallel connections to reduce the current load on the generator. In recent years, the dc generator has been replaced by an alternator that generates three phase ac power at a frequency proportional to the Diesel speed. The ac power is fed to rectifier diodes to produce dc voltage and current.

The alternator-rectifier unit is a compact device that is connected directly to the Diesel crankshaft. An auxiliary alternator may be attached to the main alternator shaft. On a traction power basis, the main alternator is up to 25% lighter, smaller and more rugged than the comparable dc unit. It does not need a commutator because the output power is generated in the stator windings. Only one transition step may be needed on a six-axle locomotive. For example, the EMD SD45 unit starts with three groups of two motors in series and then goes to full parallel connections. Relatively low dc power is required for the alternator rotor field excitation which is typically supplied through simple slip rings. Figure 1-16 shows the voltage-load current limits for a series of alternators built by General Electric of Canada (Ref. 1-15). A set of similar curves are generated when the requirements for each notch position are taken into account. Figure 1-17 shows the effect of notch position on the voltage-load curves of the alternator. These voltage-current relationships are controlled by the excitation circuit in the electrical system. The excitation control system is discussed in detail in Section II.

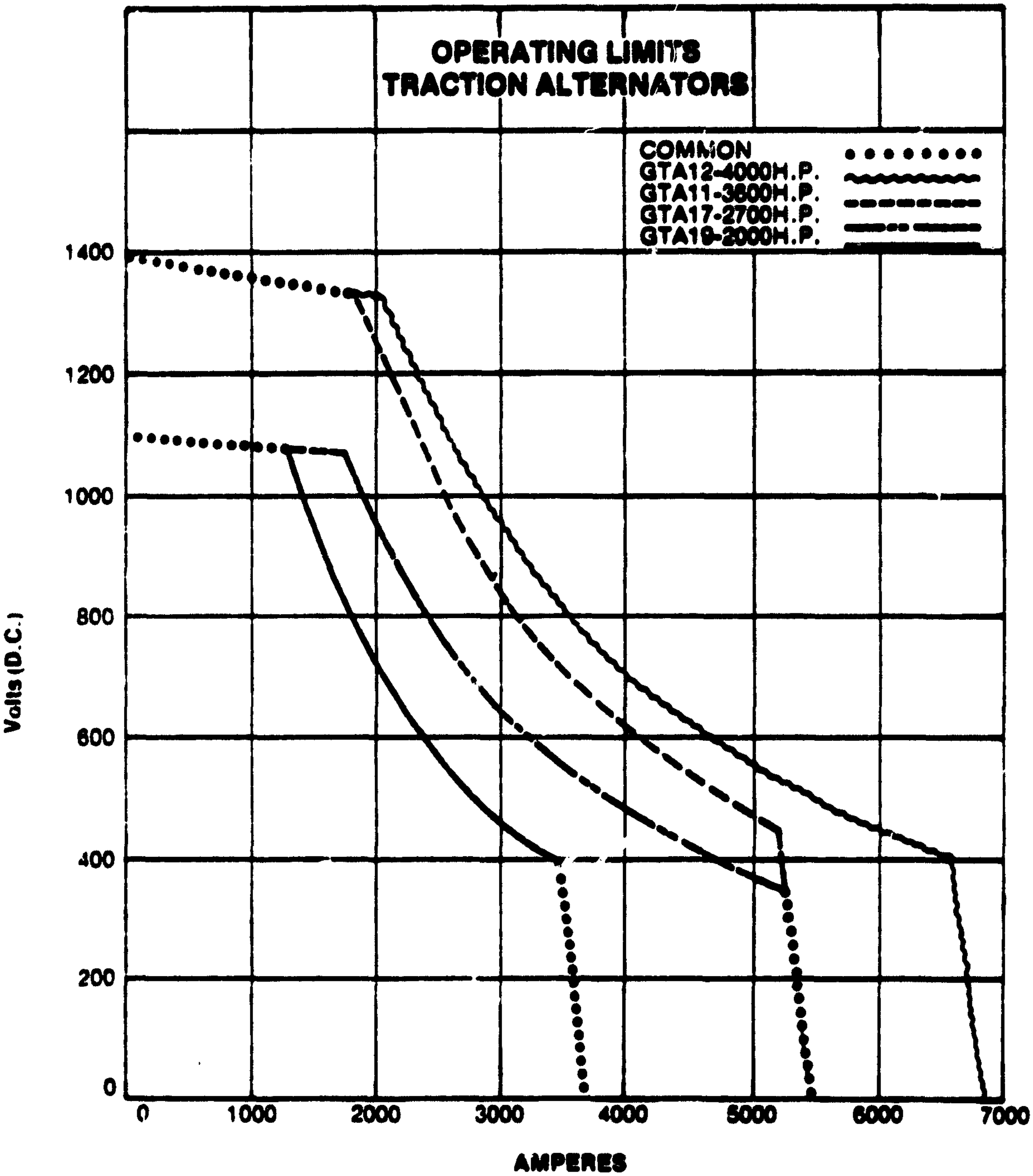

Figure 1-16. Operating Limits, Traction Alternators
(From Ref. 1-15)

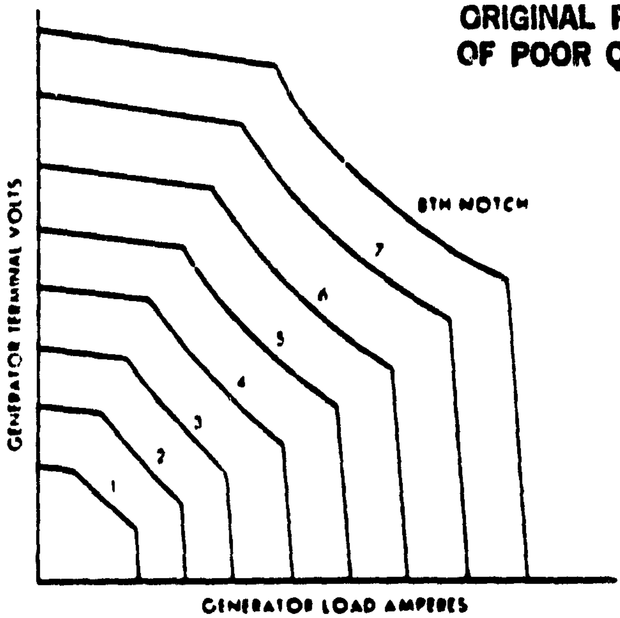

Figure 1-17. Voltage-Amperage-Notch Curves for Alternators (From Ref. 1-15)

H. TRACTION MOTORS

The time-tested, axle hung, dc series motor is used for all Diesel electric operations. There is usually one motor per axle with two or three axles per truck. The two truck locomotive is the conventional configuration. Truck frame mounted motors were used on some electric locomotives in the past. Motor to axle gear ratios are between 3 and 4 to 1 with motor top speeds ranging from 2000 to 2500 rpm.

The dc traction motor is not without limitations. At low speeds, the maximum torque and the available tractive effort are determined by the motor thermal ratings. There are a series of short term ratings and a continuous operational rating. Table 1-6 presents the various time-current ratings of some traction motors built by General Electric of Canada for Bombardier Inc. (Ref. 1-15). The rated current is a function of both time and the motor ventilation rate. In general, the current limit at the four minute rating is 24% to 31% greater than the continuous rated current limit. The increase in current limit because of improved cooling is less than 5% at the continuous rating and is as low as 2% for the four minute rating.

Although, like any dc motor, these motors can be overloaded for a short time, the overload capability is generally less than 30%. The high, short term current ratings result in high torques which can produce wheel slip. The wheel slip is corrected by using sand to increase the adhesion.

The Diesel power limit is reached at relatively low speeds and in some cases, motor thermal ratings will still exceed their continuous values. At the Minimum Continuous Speed (MCS), the Diesel may be operating at a reduced

Table 1-6. Traction Motor Ratings
(From Ref. 1-15)

GE-752 PC8 TRACTION MOTOR

The GE-752 is a series wound, four-pole, direct-current, commutating-pole railway motor, designed and insulated for operation with full or shunted field on track gauges of 56-1/2 inches or wider.

APPLICATION CURRENT LIMITS

Time	2300 cfm Ventilation Rate (4.8" H2O at Comm. Chamber) 0-750 HP	2300 cfm Ventilation Rate (4.8" H2O at Comm. Chamber) 751-900 HP	2600 cfm Ventilation Rate (6.1" H2O at Comm. Chamber) 0-750 HP	2600 cfm Ventilation Rate (6.1" H2O at Comm. Chamber) 751-900 HP
Continuous	1140 amps	1080 amps	1195 amps	1125 amps
50 Minutes	1165	1110	1220	1150
12 Minutes	1240	1195	1290	1235
6 Minutes	1345	1310	1380	1335
4 Minutes	1450	1420	1480	1445

MAXIMUM PERMISSIBLE SPEED ------------------------------- 2320 RPM

GE-785 PA1 and PB1 TRACTION MOTOR

The GE-785 is a series wound, four-pole, direct-current, commutating-pole railway motor, designed and insulated for operation with full or shunted field.

APPLICATION CURRENT LIMITS

Time	2300 cfm Ventilation Rate ($4.8''$ H_2O at Comm. Chamber) 0-650 HP	2600 cfm Ventilation Rate ($6.1''$ H_2O at Comm. Chamber) 0-650 HP
Continuous	1140 amps	1195 amps
50 Minutes	1165	1220
12 Minutes	1240	1290
6 Minutes	1345	1380
4 Minutes	1450	1480

MAXIMUM PERMISSIBLE SPEED -------------------------- 2320 RPM

GE-761PA14 TRACTION MOTOR

The GE-761 is a series wound, four-pole, direct-current, commutating-pole railway motor, designed and insulated for operation with full or shunted field.

APPLICATION LIMITS

Up to 450 engine horsepower input to generator motor.

MOTOR VENTILATION

TIME	1,680 cfm of air at 107.95 kilograms / square metre (4.25" H2O) at motor inlet	2,000 cfm of air at 152.39 kilograms / square metre (6.0" H2O) at motor inlet	2,500 cfm of air at 239.74 kilograms / square metre (9.4" H2O) at motor inlet
Continuous	645 amp	655 amp	665 amp
60 Minutes	655 amp	665 amp	680 amp
15 Minutes	695 amp	710 amp	720 amp
8 Minutes	740 amp	745 amp	765 amp
4 Minutes	830 amp	835 amp	845 amp

MAXIMUM PERMISSIBLE SPEED -------------------------- 3100 RPM* rating and the motors will deliver the required tractive effort continuously. The typical value of MCS is about 1/6 of maximum speed. Over the wide speed range above MCS, the Diesel power limits the tractive effort performance. Under some conditions, the electric motors can handle more power. For example, the 6-axle EMD SD40 and the 4-axle GP40 locomotives have the same tractive effort rating over the middle range of speeds. For the 4-axle EMD GP40 unit, the power limits of the motors determine the top speed. This does not occur on the 6-axle EMD SD40 locomotive.

Recently three-phase, variable-voltage, variable-frequency electronic drive equipment has been developed for traction applications. Relatively conventional three-phase induction or synchronous motors are used. A significant potential advantage is the elimination of the motor commutator with its maintenance and flash-over problems. The commutation function is performed by solid state inverters or cycleconverters which, it is expected, will be highly reliable and less subject to overload damage than the mechanical commutator. Either type of ac motor is inherently more rugged than the dc motor.

I. TRANSMISSION EFFICIENCY

The efficiencies of the individual alternators and traction motors in the electrical powertrain are not available. Values of the overall system efficiency, however, can be computed from the tractive effort curves if the input power is given. Figure 1-18 (Ref. 1-5) shows the transmission efficiency for a GM-EMD GP40-2 locomotive as a function of locomotive speed and notch position. Figure 1-19 (Ref. 1-14) shows the transmission efficiency for a GE U30C locomotive in notch 8 only. Both GM and GE locomotives have peak efficiencies in the 88% to 90% range over a wide speed range. Remembering that the overall efficiency is the product of four individual efficiencies, each of these efficiencies are obviously high. The gearing and rectifier efficiencies must be about 99%. The motors and alternators must be about 95% and 96% efficient, respectively. It will be difficult to make a significant improvement in efficiency on these components.

J. AUXILIARY POWER

In the typical Diesel powered unit there are five readily identifiable auxiliary loads that are coupled to the diesel engine:

(1) Turbocharger

(2) Air compressor for train brakes

(3) Traction motor blowers

(4) Auxiliary alternator

(5) Auxiliary generator

In the first three items, the power is used directly. The last two items supply electrical power to other devices.

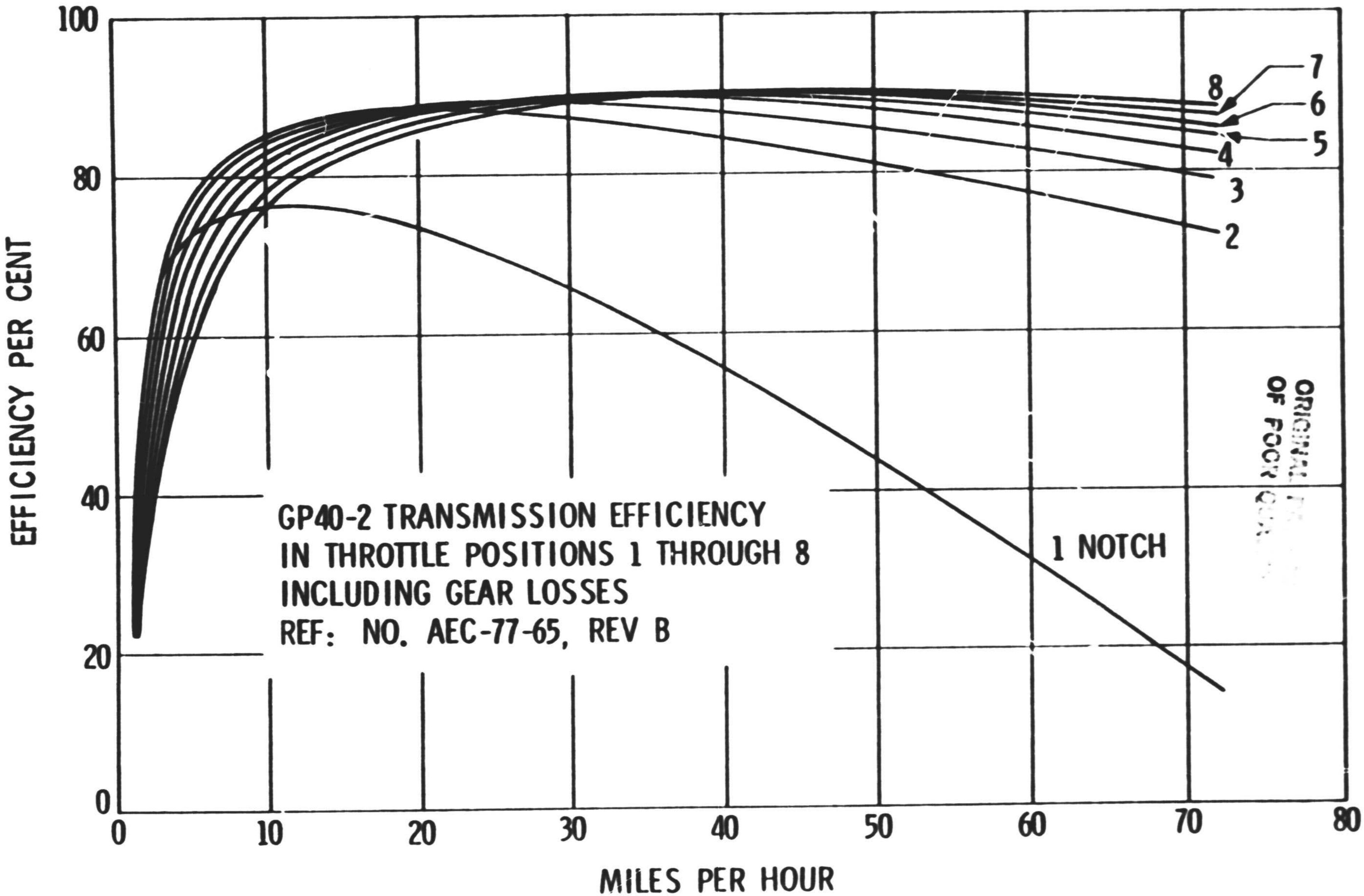

Figure 1-18. GP40-2 Transmission Efficiency
(From Ref. 1-5)

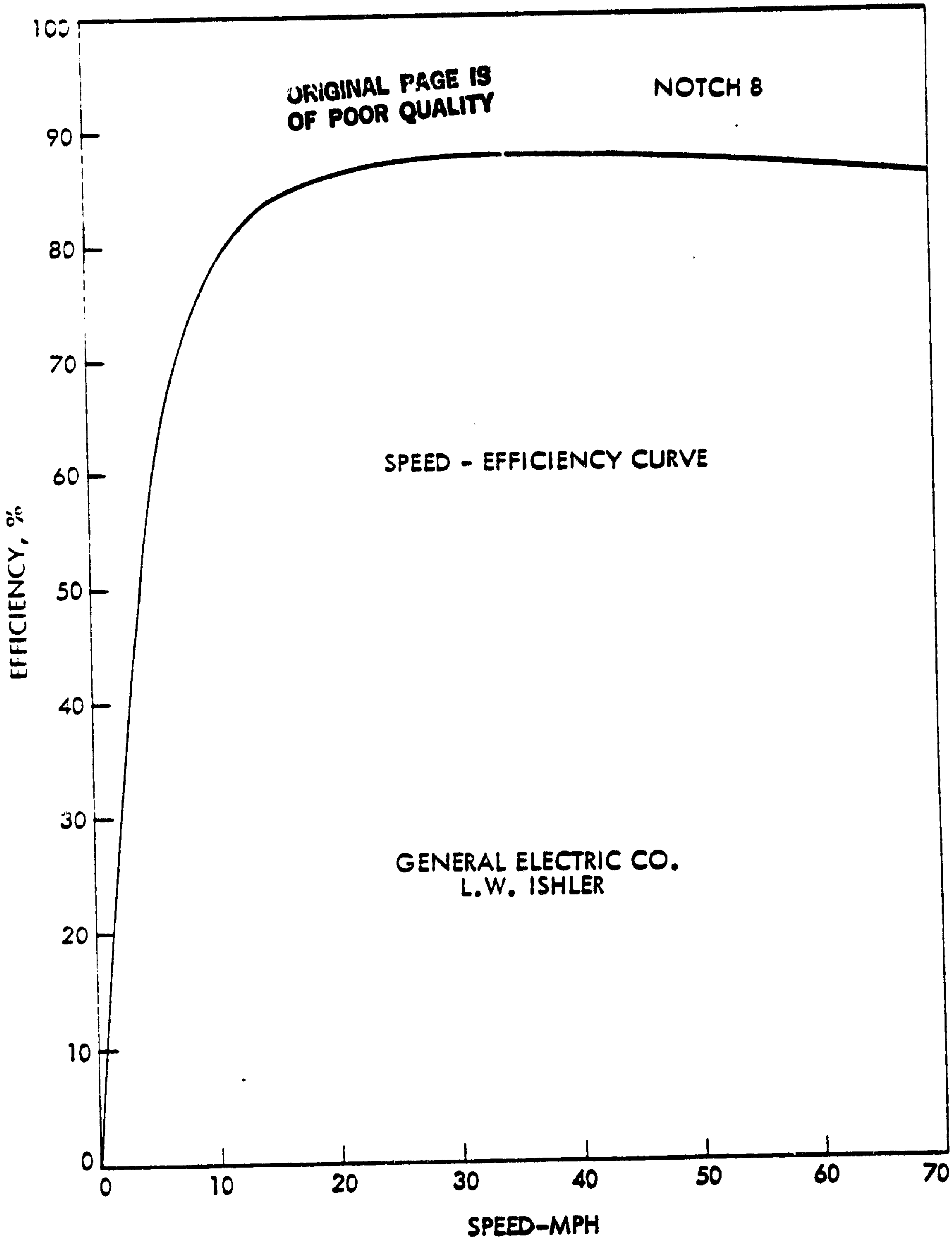

Figure 1-19. Speed vs. Efficiency Curve for a General Electric U30C Locomotive (From Ref. 1-14)

A partial list of other power using devices is given below. Some are mechanically connected to the Diesel; others use electrical power from the auxiliary alternator or generator.

(1) Engine Air Filter Blower

(2) Main Generator (Alternator) Blower

(3) Main Generator Excitation

(4) Cooling Water Pumps

(5) Radiator Cooling Fans

(6) Engine Lubrication Pump

(7) Turbocharger Auxiliary Lubrication Pump

(8) Diesel Fuel Pump

(9) Battery Charger

(10) Lights

(11) Electrical Control, Contactors etc.

(12) Cab Heating (from Diesel waste heat in some units)

The list is given as a general indication of the wide range and diversity of items for which auxiliary power is required. Typical values of the power absorbed by some of these auxiliaries are shown in Table 1-7 (Ref. 1-14). This table is by no means complete because separate load values for some of the accessories are not available. The total auxiliary power load varies with engine speed and, hence, notch position as shown in the second part of this table. In notch 1, the auxiliary power load is 20% of the gross engine output and in notch 8, it is 7% of gross output.

Although the traction drive uses most of the power, the auxiliary equipment uses a significant fraction. The effects of the auxiliary equipment on the fuel economy of the locomotive must be taken into account along with the effects of the main drive components.

Table 1-7. Power Required For Typical Auxiliary Equipment on 3000 HP Locomotives

	GM-EMD	Bombardier
Item	Auxiliary Power, hp	Auxiliary Power, hp
Equipment Blower	110	92
Radiator Fan	104	130
Air Compressor	12 (unloaded)	21
Auxiliary Generator	4	7
Total	230 in notch 8	250 in notch 8
	(Ref. 1-11)	(Ref. 1-3)

		GM-EMD	Bombardier
Variation in Power with Notch Position		Auxiliary Power, hp	Auxiliary Power, hp
Notch	1	24	19
	2	31	34
	3	42	56
	4	60	85
	5	88	123
	6	127	169
	7	175	229
	8	230	250
		(Ref. 1-11)	(Ref. 1-3)

K. SECTION I REFERENCES AND NOTES

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SECTION II
DIESEL ENGINE AND ELECTRIC TRANSMISSION CONTROLS

A. INTRODUCTION

The control function must meet two basic requirements: (1) operate the locomotive and move the train, and (2) protect the locomotive equipment. In practice, these two items are not always compatible. For example, the equipment is often required to perform beyond continuous thermal ratings in operation such as starting on a heavy grade. Even with extreme operational skill there is a chance of damage - typically to the traction motor. The implementation of the control function involves compromises during difficult operating conditions that regularly occur.

The EMD SD45 locomotive will be used as an example of a typical control system. The intent is to identify the control tasks that must be performed and not focus too strongly on the details of this particular implementation. The SD45 unit and other EMD units using similar control equipment are in wide use.

The presumption is that their capabilities satisfy a reasonable set of requirements for operation of modern railroad locomotives. A block diagram of the excitation system for the EMD SD45 locomotive is given in Figure 2-1. The later discussion that references this block diagram will be kept at the functional level.

B. TRAIN OPERATING MODES

There are two basic operating modes for each of the two main power-train components. First, Diesel engine operation at: (1) constant power loading, particularly in notch 8, and (2) variable power loading. Second, electrical transmission operation that is: (1) at or within the continuous thermal rating, and (2) above this continuous rating. All combinations of these modes occur in normal operation.

The electrical equipment is often thermally overloaded especially if the train is heavy or on a grade. Under this condition, the Diesel engine is usually operating at less than its maximum power rating and its power level is being increased. By using the classic series-parallel motor connections and alternator excitation control, thermal overloading of the alternator should not occur. It is the traction motors that are overheated. The control equipment and the operating procedures must limit this heating to acceptable levels.

When maximum acceleration tractive efforts are used, and again if the train is very heavy or on a grade, the maximum Diesel power condition can be reached and the motors are still thermally overloaded. The control function must now respond to the Diesel engine limitation. Both this mode and the mode previously described are transient modes. They must be time-limited so that the accumulating thermal overload does not damage the motor commutators or insulation.

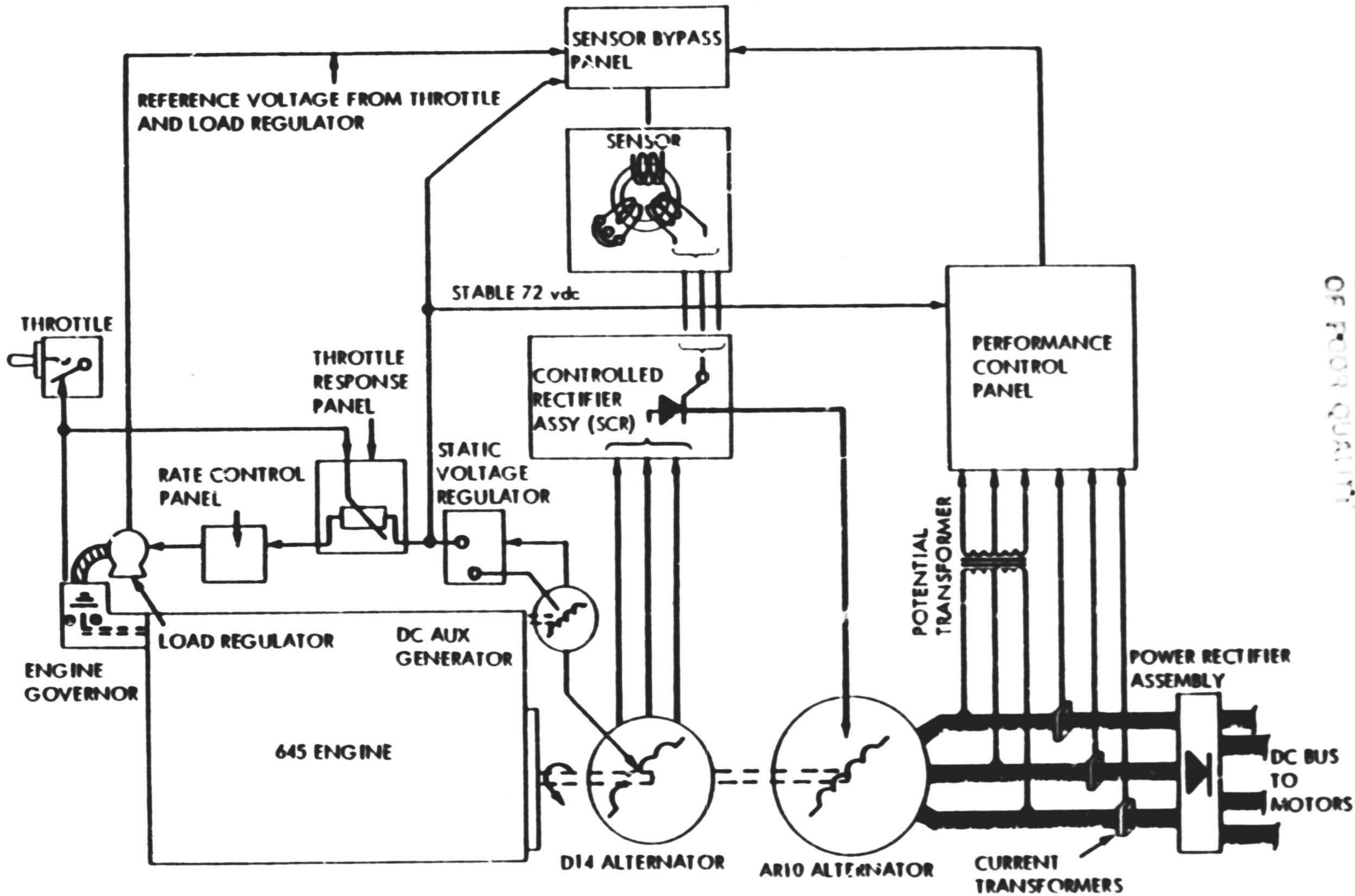

Figure 2-1. Excitation Control System for SD45 Locomotive

The above transient operations are terminated when the train has reached the highest minimum continuous speed (MCS) for any locomotive in the train. For the SD45 unit, the MCS is 11 mph. At this speed, the motors are at their continuous thermal rating.

Locomotive operation is usually power limited by the Diesel engine at speeds above the MCS. The tractive effort decreases as the speed increases because the Diesel engine power remains constant. The electrical equipment is not thermally overloaded and will probably be at its continuous rating. Because the locomotive operates most of the time in this mode, other considerations such as minimum specific fuel consumption and minimization of Diesel engine wear are important.

The last combination corresponds to the condition when the speed is above the MCS and the terrain does not require or permit operation at maximum Diesel power. The control system loads the Diesel to the power level specified by the selected throttle setting.

Any control equipment must function over the operating modes described above. These are not the only requirements. The control equipment must cope with wheel slip that can occur on badly worn rail or on rail that is contaminated with oil, ground-up tree leaves or water. It must also handle the requirements for Multiple Unit (MU) operation when locomotives of different types and manufacturers' must operate together in a single train. These two requirements are included in the following description of the control equipment.

C. CONTROL OF THE DIESEL ENGINE

The speed of the Diesel engine is controlled by an electro-hydraulic governor. Typically, there are 8-power speeds and an idle speed. They are selected by the position or "notch" of the engineer's throttle lever. When the engine speed does not match the selected speed, the action of the governor is to increase or decrease the amount of injected fuel. During this action, the position of the potentiometer in the load regulator is also changed. This potentiometer provides information to the electrical load control circuits discussed in the next section.

The selection of a power notch is the only Diesel engine control available to the engineer. In effect, the engineer is calling for a given load on the Diesel as its speed approaches the selected value. The load balancing is accomplished by adjusting of the alternator excitation and the speed is controlled by adjusting of the fuel injection rate. The Diesel is then operating in a steady state power mode at the selected speed. Table 2-1 lists data for the 16-cylinder engines used in EMD SD40, GP39, and GP40 units. The SD45 uses a 20-cylinder engine with corresponding higher horsepower.

The electro-hydraulic governor dominates the control function. It controls the Diesel, keeps the engine operation within prescribed limits, and provides appropriate signals for the electrical load control.

Table 2-1. Locomotive Characteristics

Model	MCS[a] mph	Tractive Effort, lb	Rail hp at MCS	Motors or Axles	hp/Motor	Nominal Rated hp at MCS	Transmission Efficiency at MCS in Notch 8
GP39	11.1	54,700	1,620	4	405	2,300	70%
GP40-2	11.1	56,000	1,658	4	414	2,350	70.5%
SD40-2	11.0	85,000	2,493	6	415	3,050	81.7%
SD45	11.0	82,100	2,400	6	400	3,600	67%

Note: [a]MCS-Minimum Continuous Speed

D. LOAD CONTROL TO THE LOCOMOTIVE WHEELS

The alternator and motors transfer the engine power to the drive wheels. In the EMD units, this action is controlled by the excitation system.

For starting and at low speeds, traction motors are typically connected in series-parallel. The SD45 uses a 2 series - 3 parallel configuration. At transition speed, the motors are connected in full parallel. The general purpose of this action is to modify the characteristics of the motors to limit the maximum voltage required.

As its name implies, the excitation system provides dc current to the rotor field windings of the alternator through slip rings and thereby determines the ac voltage generated in the stator windings. These voltages cause ac currents to flow through rectifiers that produce a dc current which delivers power to the motors.

A feedback comparison method is used to regulate the alternator excitation. In the SD45 unit, signals proportional to the ac voltage and current output from the alternator are rectified and summed into a single output value. This result is further modified (discussed under multiple unit operation) and the final output is a load power feedback signal that goes to the comparison device. This summation method recognizes that at low speeds the motors are current limited, independent of the power being delivered. The load power feedback signal is thus not strictly proportional to load power. It increases or decreases, however, with a similar change in load power and therefore, is a suitable feedback control signal. The potential and current transformers are shown in Figure 2-1. Their signals are processed by the performance control panel and the control output signal is compared in the sensor bypass panel.

The control signal for available power comes from the moving contact of the load regulator potentiometer. The total voltage across this device is controlled by the throttle setting (see Figure 2-1). In notch 8, when maximum Diesel power is available, this voltage has its maximum value. At lower notch settings, this voltage is appropriately reduced by inserting series resistance into the throttle response panel. The control action is the same for all throttle settings. The available power signal causes the excitation control to load the Diesel engine to the selected power level.

The rate control panel limits the rate at which the load power can be changed. Without this device, the excitation control circuits will react too fast compared with the mechanical fuel injection control and this will result in excessive loads being applied to the diesel engine.

E. MULTIPLE UNIT OPERATION

Table 2-1 gives the rating of several EMD locomotives. Their MCS's are virtually identical and, as a result, any combination of them may be used in multiple unit operation. To be able to do this, at or near the MCS, the 3600 hp SD45 locomotive is operated as a 3000 hp unit, the 3000 hp GP40 at 2000 hp and the 2300 hp GP39 at 2000 hp. These power restrictions for the SD45, GP40, and GP39 units are required because of motor limitations and are accomplished in the performance control panel in the EMD implementation. As the speed increases above the MCS, more power can be accepted by the motors in these units. Ultimately, the full Diesel power is reached and that is the power limit at high speeds.

F. WHEEL SLIP CONTROL

Automatic wheel slip control is typically installed in heavy, high powered locomotives. The EMD instantaneous detection and correction (IDAC) system is available. The IDAC senses differential changes in the currents going to the motors. The first system action is to insert voltage signals into the wire between the load regulator moving contact and the feedback comparison unit. The purpose is to rapidly and momentarily lower the power to all the traction motors and to attempt to recover adhesion for the slipping motor or motors. The engine doesn't see this action unless it is repeated often. If adhesion is not recovered by this action, the next step is to insert resistance in the load regulator potentiometer circuit. This step reduces the output power significantly and because the Diesel engine speed increases, the governor acts to reduce the injected fuel rate. Once rail adhesion is recovered, the inserted resistance is removed and normal control is resumed.

G. SUMMARY

The discussion of the control system has used EMD locomotives as examples. Different companies will implement the control functions in different ways. In the later work of this project, particularly the alternate engine studies, the control methods will be different but the end result must meet the general needs as outlined in this section.

SECTION III
LOCOMOTIVE MODELING

A. INTRODUCTION

The modeling effort provides a common basis for evaluating the effects of locomotive system modifications on fuel consumption. To determine the effect of changing a single component in a locomotive, whether it is an engine or a traction motor, it is necessary to know how both the new component and the original one interact with the rest of the system. The same is true for add-ons such as bottoming cycles or changes in fuel. There are two approaches used in this study. One approach is based on notch-time duty cycles such as those of General Motors and the other is using the locomotive simulation program (RAIL).

The method based on notch-time duty cycles is relatively simple and easy to use. With this method, it is first necessary to calculate the effect of the modification under study on fuel consumption for each of the eight notch positions, idle, and dynamic braking. The new fuel flow rate for each notch is multiplied by the fraction of time spent in that notch and the results summed over all of the notches, idle, and dynamic braking. Losses are then determined throughout the entire powertrain in the locomotive and the resistances of the train in each notch are calculated to obtain the new fuel flow rate resulting from a change in a component. The method is useful for screening changes when precise answers are are not required.

For those cases where a more precise value is needed, a simulation program, RAIL, has been developed to model the locomotive and its working environment. RAIL is a Fortran program written for the UNIVAC 1108 computer to support the analysis of the alternate locomotive propulsion systems program. The primary function of the program is to integrate the nonlinear differential equations which describe the behavior of the locomotive, train, and track. In developing the program, the principal objectives were:

(1) Flexibility in the program structure to allow different components and different arrangement of components to be used.

(2) Variability in the locomotive loading conditions and in duty cycles.

(3) Capability to perform parametric studies.

To meet these objectives, the locomotive system is divided into components (engine, alternator, etc) and node points. The node points are the interfaces between components. This method of program construction gives the program a wide spectrum of capabilities. For example, the locomotive component arrangements can be easily changed to handle a variety of systems. Systems included are those with a generator driven by the Diesel engine and using dc traction motors (the dc-dc system) typical of early Diesel-electric locomotives, systems with an alternator-rectifier and dc traction motors (the ac-dc system) typical of present-day locomotives, and the ac-ac system (alternator-rectifier-inverter-ac traction motors) which may be the trend of the future. The three systems are shown schematically in Figure 3-1.

In the schematics, all of the components are represented by individual subroutines that compute the output node conditions based on the input node data and user supplied data which describe the relationship between the input and output nodes. By changing subroutines or by using different sets of user supplied data, the whole range of locomotive systems can be simulated.

The user specified data are either taken from the data library or typed in at the computer terminal. The data library contains the component characteristics, locomotive loading data and the duty cycles. Duty cycles are defined and discussed in a later part of this section. This library is not stored during program execution thus allowing numerous sets of data to be "built-in" without increasing the cost of running the program. If only a few parameters are to be changed for the next run, they may be entered from the terminal at execution time. The program has the capability for the user to override any data in the data library. This system greatly simplifies the running of parametric studies.

There are three locomotive loading modes available: steady state at constant speed, maximum power acceleration, and duty cycle. In the steady speed mode, the program calculates the engine output power, the tractive effort, and the train resistance or drawbar load. In addition, the drawbar power and the fuel flow are calculated at different specified constant speeds. The second mode, maximum power acceleration, is what its name implies. The power to the wheels is limited by either the maximum engine power, the motor current limits or the wheel slip. The locomotive and consist are assumed to accelerate as rapidly as possible within the constraints mentioned above for either a given interval of time or until a specified speed is reached. The track is assumed to be straight and either flat or on a grade.

The duty cycle mode is the most flexible mode available. The cycle can be constructed by defining velocity, power, notch position or acceleration as a function of time. The cycle may also be defined by grade, degree of curvature or speed limits as a function of distance. The cycle could also be a combination of both time and distance as well. The program uses time as the independent variable. If the duty cycle is defined in terms of time, then as the value of time is changed during integration, the program goes to the duty cycle to get new values of the dependent variables. If a distance based duty cycle is specified, then the differential equation for acceleration is integrated first to velocity and, secondly, to distance. It is this value of distance which is then used to get the values of the dependent variables.

Two methods of integration are built into the program. The first is the fourth-order Runge-Kutta method commonly used for numerical integration because of its accuracy, stability and simplicity compared to other higher order methods. The other method is a constant slope or trapezoidal method.

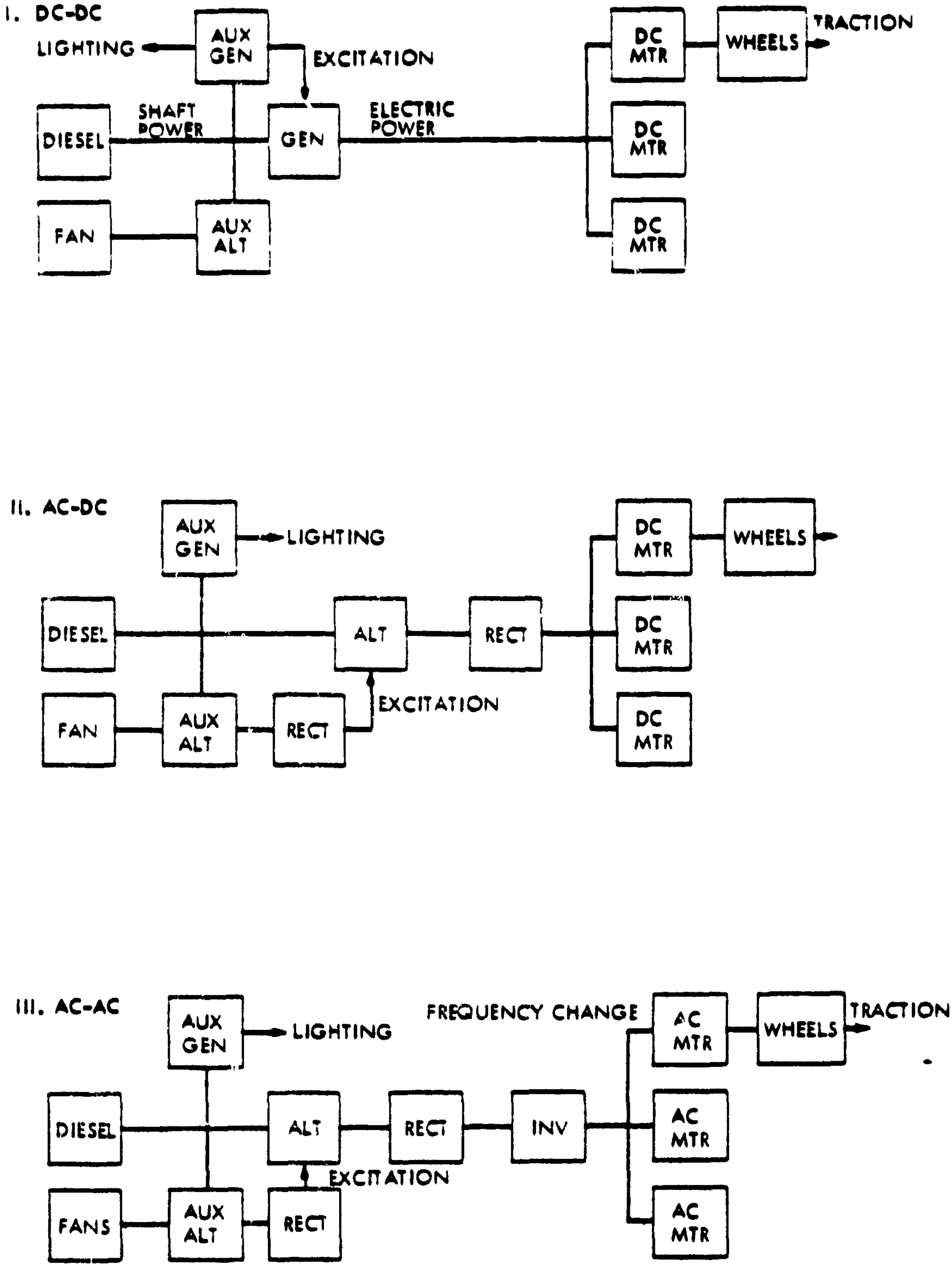

Figure 3-1. Locomotive Power Train Schematics

It is extremely simple but not nearly as accurate as Runge-Kutta. The reason for using the constant slope method is its low cost. The Runge-Kutta method requires the equations to be evaluated four times for each integration interval whereas the constant slope method needs only one evaluation per interval. A cost study of the two methods showed that the cost per interval for Runge-Kutta is three times as high as for the constant slope. However, for a given level of accuracy, the Runge-Kutta method can use fewer and larger integration intervals to complete this task. If the differentials are well behaved, the constant slope method can use quite large intervals. The point of having both methods is that solutions to a given problem can be computed to the required accuracy at the lowest possible cost.

B. DUTY CYCLES

To determine the fuel economy of a locomotive in service, the service must be characterized. This characterization is known as a duty cycle and can be defined in various ways. In the automotive field, the Federal Test Procedure for emissions and fuel economy defines urban and highway driving cycles. The fuel economy figures for these cycles are posted on new cars and are widely published. By testing all cars over the same driving cycle and under the same test conditions, relative fuel economy values can be obtained. These driving cycles which are defined as the vehicle speed at one second intervals for 1321 seconds on the urban cycle and 765 seconds on the highway cycle. These driving cycles are one type of duty cycle because they describe the way the car is being driven. The locomotive manufacturers use a different kind of duty cycle. Their cycles give the time or percent of time the locomotive operates in each notch position of the engine control. Some train simulation programs use a duty cycle based on the track profile. Information on track grade and the degree of the curve are given as a function of the distance traveled along the track. From this information and the characteristics of the cars in the consist, the train resistance can be calculated. When speed limit and time-table information is added, the acceleration and braking can be computed. The total drawbar power can be calculated and when added to the locomotive characteristics, the engine power, notch position and fuel flow can be computed. The fuel flow can be integrated with respect to time to get total fuel used. The end result of using any duty cycle is the total fuel consumed. If the duty cycle simulates an actual section of track and the actual fuel usage is known, it can then be compared to the computed value as a check on the accuracy of the program. If the program is reasonably accurate, it can be used to predict the effect on fuel consumption of a change in components, operating conditions, the control system or the type of fuel used.

However, the problem is: what duty cycle, what section of track, and what conditions are representative of typical railroad locomotive use? Obviously, a flat, straight section of track is not typical. Nor is the section of track over Donner Pass in the Sierra Nevada mountains. Even after deciding on a typical section of track, it may be several hundred miles long and take many hours of real time to traverse. Simulating such trips on the computer can be expensive especially for a parametric study where numerous trips are simulated with only small differences between them.

These are some of the reasons that the manufacturers prefer to use the much simpler notch-time duty cycle. Each notch represents a specific engine speed, output power level and fuel flow rate. The average time or percent of time in each notch is specified and therefore, the average fuel flow rate can be determined. As expected, there are numerous duty cycles of this type. Table 3-1 presents three that are used by GM-EMD. Table 3-2 shows four GE duty cycles and Table 3-3 presents three Santa Fe duty cycles and the Association of American Railroads (A.A.R.) cycle.

Another type of duty cycle is based on grade and curve information for a typical section of track. This duty cycle uses the distance, the degree of curvature, and the grade along the chosen section of track together with the block speed limits to define an operational profile. The profile is defined by a set of two extremes; one extreme is when the train speed is always right at the speed limit. This condition results in the minimum terminal-to-terminal time. The other extreme is when the train speed is so slow that it takes an unreasonably long time to make the trip. The

Table 3-1. General Motors Electro-Motive Division Notch-Time Duty Cycles (From Ref. 3-1)

	Percent of Time		
Throttle Position	E.M.D. Heavy Road Duty	E.M.D. Medium Road Duty	E.M.D. Light Road Duty
Notch 1	3%	4%	3%
Notch 2	3	4	3
Notch 3	3	4	3
Notch 4	3	4	3
Notch 5	3	4	3
Notch 6	3	4	3
Notch 7	3	4	3
Notch 8	30	17	9
Idle	41%	46%	66%
Dynamic Braking	8	9	5
Load Factor (Availability)	40.1%	30%	19.6%
Relative Duty Cycle Fuel Rate	100	75.0	48.8
Load Factor (Utilization)	54%	42%	32.6%
Utilization	72.7	69.3	56

Table 3-2. General Electric Notch-Time Duty Cycles (From Ref. 3-2)

	Percent of Time			
Throttle Position	Minimum Duty	Maximum Duty	First Average	Second Average
Notch 1	6.5%	2.5%	5.0%	5.1%
Notch 2	6.5	2.5	2.5	3.9
Notch 3	6.5	2.5	2.0	3.4
Notch 4	6.5	2.1	5.0	3.3
Notch 5	2.9	1.7	2.0	2.8
Notch 6	2.9	1.7	2.0	3.4
Notch 7	2.5	1.8	2.5	2.6
Notch 8	5.2	38.0	21.0	17.0
Idle	59.0%	40.0%	54.0%	53.0%
Dynamic Braking	1.5	7.0	4.0	5.5

Table 3-3. Atcheson, Topeka and Santa Fe and Association of American Railroads Notch-Time Duty Cycles (From Ref. 3-2)

	Percent of Time			
Throttle Position	A.T.S.F.			A.A.R.
	High	Medium	Switcher	
Notch 1	5%	5%	10%	3%
Notch 2	3	4	5	3
Notch 3	3	3	4	3
Notch 4	3	2	2	3
Notch 5	2	2	1	3
Notch 6	3	2	1	3
Notch 7	2	1	nil	3
Notch 8	24	20	nil	28
Idle	46%	59%	77%	43%
Dynamic Braking	9	2	*	8

* Switch engine not equipped with dynamic braking

railroad timetable defines one middle value. The program can, among other things, be used to simulate the effects of changes in travel time on fuel economy.

The plan has been to utilize at least two or possibly three of the notch-time duty cycles and one curvature-grade duty cycle. The West Colton to Yuma route has been used for this purpose. By providing several different types of duty cycles in the program, it is less likely to produce results having an unsuspected bias.

The curvature-grade duty cycle makes use of a section of existing track which has both hills and flat sections, and is long enough to provide a good test. Southern Pacific has provided the track profile data on their West Colton, California to Yuma, Arizona line. This line provides a variety of conditions over its nearly 200-mile length. By running a number of different types of trains and ones having different power-to-weight ratios, the effect of each locomotive variation can be assessed better than if just a single train or a single power-to-weight ratio is used.

C. PROGRAM LOGIC

Figure 3-2 shows the primary logic flow path of the program. This is just a skeleton and does not contain all of the details of the energy balance equations or of the other sections not relevant to the understanding of the overall logic of the program. Table 3-4 contains the definitions of the logic flow chart variables used in Figure 3-2.

The first step in the logic flow is to input data which specifies the locomotive system and the conditions for analysis. This step is followed by initializing the variables in the program. One of the input parameters is the loading mode. If the steady speed mode is selected, no integrating is required and the path on the left of Figure 3-2 is selected. If the maximum acceleration or the duty cycle mode are selected, integration is needed, and the path on the right is followed. If the steady speed mode is selected, the velocity or speed of the train is set to the initial value specified in the input data set. Because this is steady state, the acceleration and, hence, the inertial loads are set to zero. The heart of the program is the energy or power balance. With the speed specified and the train characteristics known, the drawbar power can be calculated. Similarly, the locomotive aerodynamic and rolling resistance losses can be calculated. The wheel power can be summed from the consist resistance and the locomotive losses. The motor, rectifier, alternator, and eventually, the engine power can be computed from the wheel power.

The notch position can be determined and the fuel flow calculated from the engine power. The results of the computations are printed out and the speed checked against the velocity limit to see if all the speeds desired have been computed. If they have, the program goes to the end and stops. If not, control is transferred back to the beginning of this leg of the program and a new speed computed by adding the incremental speed to the previous value.

If the maximum acceleration or duty cycle mode is selected, time rather than speed is incremented. Starting with time equal to zero, the

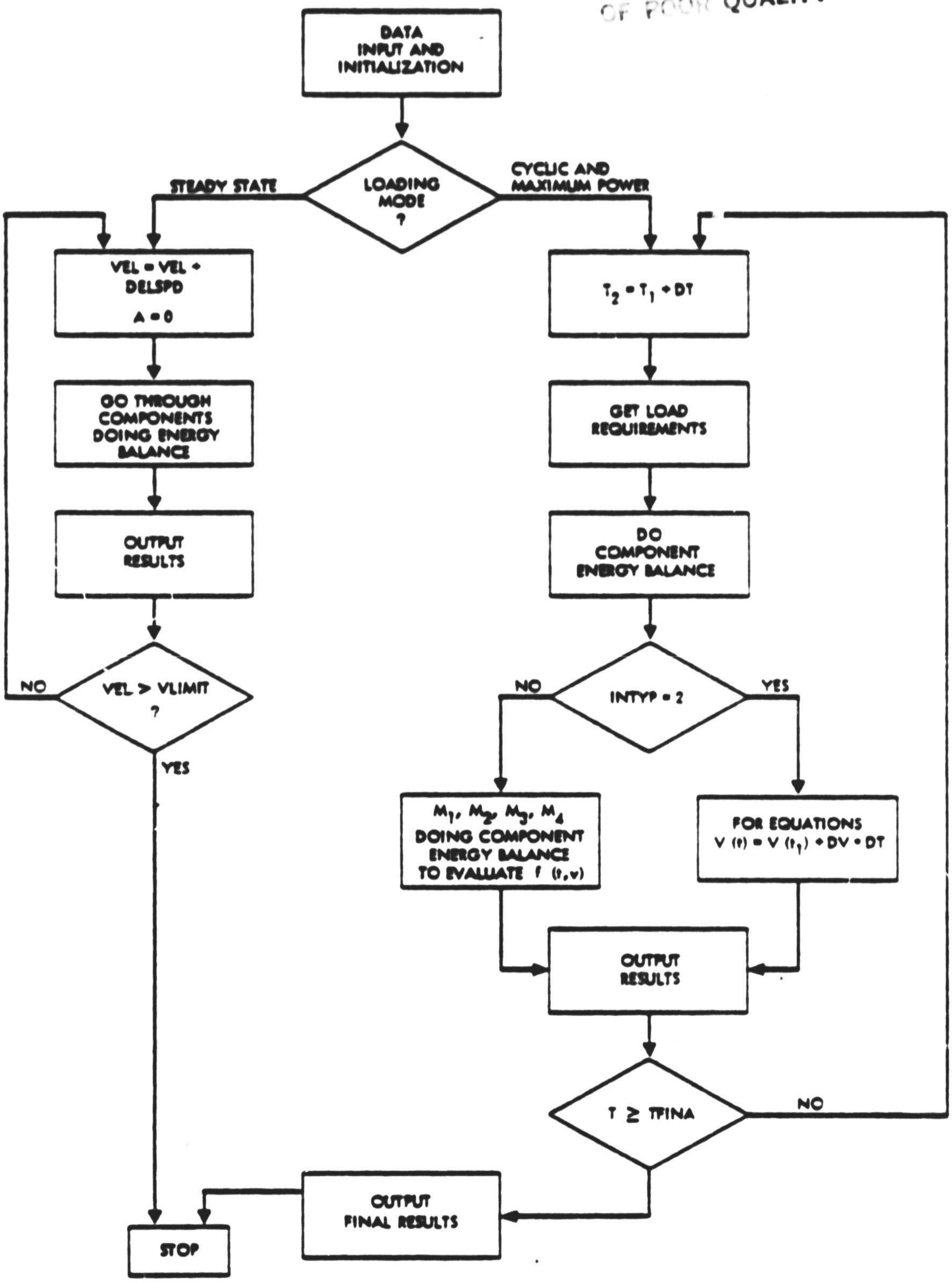

Figure 3-2. "Rail" Logic Flow Chart

Table 3-4. Logic Flow Chart Variables

Variable	Definition
VEL	Locomotive Velocity
V(t)	Integral of DV
DV	Derivative of V(t), Differential equation value
DELSPD	Speed Increment for Steady State Mode
A	Acceleration
VLIMIT	Max Velocity to Consider for Steady State
T	Time
TFINA	Max Time
DT	Time Increment
INTYP	Integration Method 1 = Runge-Kutta 2 = Constant Slope
M	Values of Runge-Kutta Subproducts

program integrates ordinary differential equations derived from the power balance of the locomotive and its consist. One of the differential equations, train acceleration, is integrated to give train velocity. Velocity is integrated to get distance traveled. The load requirements are calculated from the duty cycle using either the time or distance data. In the case of maximum acceleration, the acceleration is computed from maximum engine power. If the acceleration does not cause motor overload or wheel slip, it is used in the integration. If it does, a lower acceleration must be calculated which does not exceed either of these limits. Once satisfactory values of the integrands are computed, control is transferred to one of the integration methods discussed earlier and then the results are printed out. Time, distance or velocity is checked to determine if the end of the integration range has been reached. If it has not, control is looped back to get a new value of time and the computations are resumed. When the end of the range is reached, final results are printed out and the execution terminated.

D. SAMPLE OUTPUTS

Table 3-5 is an example of the output in the steady speed loading mode. The locomotive is an ac-ac configuration on B_o-B_o trucks. At the top of the output is the program name, version number, date and time of execution. Below that, the computer asks if there is any input for this run. In this case, three parameters have been changed from the default values built into the program. The following seventeen lines print out some of the characteristics of the locomotive and the train. The third line states that the locomotive is a B_o-B_o configuration and the consist has four cars. The weights in the following lines state that the total weight of the locomotive is 180 tons and the wheel loading is 45 tons per axle. The cars that are being used weigh 70 tons each and the wheel load is 17.5 tons per axle.

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Table 3-5. Steady Speed Operation with a B_o-B_o Locomotive using ac Traction Motors

```
                                          RAIL INFORMATION
 CASE  DCTYP  MODE  DT        INTEG  LOCO #LOCO  #CAR  CFIG
          1      1   1.00        1 BOBO      1     4     2
          WEIGHT (TONS)
  LOCO      LOCO     CAR         CAR
 TOTAL      PER AXLE  TOTAL     PER AXLE
  100.0       45.0      70.0       17.5
             ENG    PTO    GEN    ALT    RECT   INV    MTR
COMP TYPE     1      1      1      1      1      1      1
MAX POWER  3320.              0. 3300.  3300.  3300.  3300.
MAX SPEED  1050.              0. 1050.                1050.
IOFORM  ID1   ID2   ID3   NCP  TTPMX IDBUS NAXLEL
   0     1     1     0     7     8     0     4
ICRBRG ILRBRG IOFIDE MORE   MTP   NMPE  NWPM NAXLEC
   0     0     0     1     0     4     1     4
EDIFF   RDIFF   EFFPTO  RATALT  RATGEN  GRADE
  .950   2.679    .950   1.000   1.000    .000
```

TIME SEC	VEL MPH	ACC MPH/S	DIST MI	TTP	ENG-O PWR HP	TRACTIVE EFFORT LB	TRAIN RESIST LB/TON	DRAWBAR PWR HP	FUEL FLOW LB/HR
.0	.0	.00	.0	1	20.	8078.	8078.	0.	36.6
.0	10.0	.00	.0	1	123.	1316.	1316.	24.	87.5
.0	20.0	.00	.0	2	237.	1454.	1454.	53.	125.4
.0	30.0	.00	.0	3	357.	1592.	1592.	86.	161.8
.0	40.0	.00	.0	3	481.	1730.	1730.	124.	208.9
.0	50.0	.00	.0	3	549.	1868.	1868.	166.	234.6
.0	60.0	.00	.0	4	664.	2006.	2006.	213.	265.1

Below that is the maximum power and speed ratings of the engine, alternator, etc. The next listings are some of the input/ output control parameters. The last item in the third group down is the grade which in this example is zero (level track). The output of the calculations is shown at the bottom of the listing. Because this is steady state, time is not a factor and is printed as zero. In this example, the calculations were made for 10 mph intervals up to 60 mph. The interval and the maximum speed used can be specified by the user. Acceleration is naturally zero and the distance traveled is not computed and is therefore set to zero. TTP is the throttle position or "notch" setting. The engine output power is in the next column. At 60 mph, the engine output is 664 hp in notch 4. The tractive effort is 2006 lb which at steady speed is also the train resistance or drawbar pull. The drawbar power is 213 hp. The fuel flow to the engine at this speed is 265.1 lb/hr.

The second example is the operation of a train over a simple duty cycle which is 17.3 mi long and takes 50 min of real time to complete. The top speed is 30 mph. The data at the top of the Table 3-6 indicates that a C_0-C_0 locomotive is being used with a consist of 10 cars. The locomotive weighs 195 tons and has a 3000-hp Diesel engine. The cars weigh 70 tons each. The track is flat and straight.

The computed results are not printed out after each integration step but once every 100 s of computed time. The velocity, acceleration, and distance travelled are printed as well as the current values of notch position, engine output power, tractive effort, train resistance, drawbar power, and fuel flow. After the duty cycle is complete, the energy into, out of, and lost by each component is summarized. In this example, the engine produced 337 horsepower-hours (hp-hr) of energy over the driving cycle. Of this, 60 hp-hr were used for auxiliary equipment by means of the power take-off route. This means 277 hp-hr went into the alternator where 48 hp-hr of energy were lost. The alternator operated at an average efficiency of 83%. Twenty three hp-hr were lost in the rectifier which has an average efficiency of 90%. Each of the six motors received 34 hp-hr of which 9 hp-hr were lost and 26 transmitted to the wheels. Their average efficiency is 76%. The overall efficiency from engine to wheels is 46%.

The output of the program can be used in many ways. For one thing, the traction motors have the lowest average efficiency of any of the components. The program can pinpoint the areas of greatest energy loss. It can predict how much fuel can be saved if, for example, the motors were replaced with more efficient ones. If the cost differential between the original and the more efficient motors is known as well as the cost of fuel, then the results could be used to determine the life cycle costs and energy savings associated with each motor.

E. ADVANCED DIESEL ENGINES

Three advanced Diesel engines were considered for modeling. These are the turbocompound engine, the adiabatic engine, and the augmented engine. Several versions of the adiabatic engine were considered including the turbocompound adiabatic engine, the turbocompound minimum friction adiabatic engine, and a Rankine bottoming cycle added to this same engine.

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Table 3-6. Driving Cycle Operation using Runge-Kutta

```
 INPUT $IN)
) INPUT $IN)

                                          RAIL INFORMATION
 CASE  DCTYP  MODE  DT       INTEG  LOCO #LOCO  #CAR  CFIG
          1     2   1.00        1 COCO     1     10     3
           WEIGHT (TONS)
  LOCO       LOCO       CAR         CAR
 TOTAL       PER AXLE   TOTAL       PER AXLE
  195.0         32.5       70.0        17.5
             ENG      PTO     GEN     ALT    RECT    INV     MTR
COMP TYPE      1       1       1       1       1       1       2
MAX POWER   3000.               0.  3000.  3300.      0.    400.
MAX SPEED   1050.               0.  1050.                  2500.
IOFORM   IO1     IO2     IO3     NCP   TTPMX IDBUG MAXLEL
   0       1       1       1       6      8     0     6
ICRBRG ILRBRG IORIDE MORE    MTP   NMPE  NWPM MAXLEC
   0       0      0     0      0      6     1     4
EDIFF   RDIFF   EFFPTO  RATALT  RATGEN  GRADE
  .950  2.6?9    .950   1.000   1.000    .000
```

TIME SEC	VEL MPH	ACC MPH/S	DIST MI	TTP	ENG-O PWR HP	TRACTIVE EFFORT LB	TRAIN RESIST LB/TON	DRAWBAR PWR HP	FUEL FLOW LB/HR
.0	.0	.00	.0	1	20.	15923.	17.79	0.	36.0
100.0	3.0	.03	.0	2	128.	5076.	5.67	33.	79.6
200.0	6.0	.03	.2	2	219.	5016.	5.60	64.	118.1
300.0	8.7	.03	.4	3	297.	4951.	5.53	93.	138.7
400.0	11.3	.03	.7	3	362.	4882.	5.45	119.	163.4
500.0	13.7	.02	1.0	3	417.	4809.	5.37	142.	184.3
600.0	16.0	.02	1.4	3	464.	4730.	5.29	163.	202.1
700.0	18.1	.02	1.9	3	506.	4648.	5.19	181.	218.2
800.0	20.0	.02	2.4	3	541.	4561.	5.10	196.	231.7
900.0	21.7	.02	3.0	3	570.	4469.	4.99	209.	242.7
1000.0	23.3	.02	3.6	4	595.	4373.	4.89	220.	240.0
1100.0	24.7	.01	4.3	4	609.	4273.	4.77	228.	245.2
1200.0	26.0	.01	5.0	4	619.	4168.	4.66	234.	248.7
1300.0	27.1	.01	5.7	4	624.	4056.	4.53	238.	250.6
1400.0	28.0	.01	6.5	4	625.	3944.	4.41	239.	250.8
1500.0	28.7	.01	7.3	4	621.	3826.	4.27	238.	249.7
1600.0	29.3	.01	8.1	4	614.	3703.	4.14	236.	247.1
1700.0	29.7	.00	8.9	4	603.	3576.	4.00	231.	243.2
1800.0	30.0	.00	9.7	4	589.	3444.	3.85	225.	238.2
1900.0	30.1	.00	10.6	3	569.	3308.	3.70	217.	242.2
2000.0	30.0	-.00	11.4	3	548.	3167.	3.54	207.	234.4
2100.0	30.0	-.00	12.2	3	567.	3303.	3.69	216.	241.5
2200.0	30.0	-.00	13.1	3	567.	3303.	3.69	216.	241.5
2300.0	28.9	-.01	13.9	3	376.	2036.	2.27	131.	169.0
2400.0	27.0	-.02	14.7	2	259.	1362.	1.52	85.	134.8
2500.0	24.4	-.03	15.4	2	133.	668.	.75	41.	82.0
2600.0	21.1	-.04	16.0	1	20.	-46.	-.05	4.	36.0
2700.0	16.9	-.04	16.6	1	20.	-780.	-.87	-23.	36.0
2800.0	12.1	-.05	17.0	1	20.	-1535.	-1.71	-35.	36.0
2900.0	6.5	-.06	17.2	1	20.	-2309.	-2.58	-29.	36.0
3000.0	.1	-.07	17.3	1	20.	-3104.	-3.47	-1.	36.0
3100.0	.0	.00	17.3	1	20.	16259.	18.17	0.	36.0

COMPONENT ENERGY SUMMARY (HP-HR)

	WHL	MTR	RECT	INV	ALT	GEN	PTO	ENG
ENRGY IN	0.	34.	229.	0.	277.	0.	337.	0.
ENRGY OUT	0.	26.	206.	0.	229.	0.	277.	337.
ENRGY LOSS	0.	9.	23.	0.	48.	0.	60.	0.

To compare these engines using the program, their speed-power-fuel flow relationship must be established. Because some of these engines exist only on paper and the rest are experimental, a method of estimating their performance must be established. In Table 3-7, a typical 3000 hp locomotive Diesel engine is characterized. In each notch position, an engine speed and a gross output power are selected. From known accessory power values, a set of values for each notch position is derived. Subtracting the accessory power from the gross power gives the net power output BSFC is estimated from GE and GM data. The fuel flow is calculated from the BSFC and gross power. The turbocompound Diesel is characterized in Table 3-8. The effect of turbocompounding is to increase the output power of the engine. This increase in power is estimated from the amount of usable power available in the engine exhaust and the turbine efficiency. This change in power is added to the output power of the base engine to get the of the turbocompounded engine is the same as that of the base engine. Physically, the Diesel engine has been made smaller for the same output power rating. All of the engines examined in this section have been resized in the same way. A cycle analysis is used to estimate the change in thermal efficiency, engine friction, heat loss to the coolant and heat loss to the exhaust. The BSFC is derived from this analysis and the fuel flow rate calculated from the gross power and the value for BSFC. This process is repeated for each of the eight notch positions and for the idle and dynamic braking conditions.

The adiabatic engine analysis is handled in the same manner as the turbocompounded engine. Because there is no cooling system on an adiabatic engine, the radiator, fan, and water pump power must be subtracted from the accessory power. Because of this reduction, the fuel flow is down even more than the BSFC as shown in Table 3-9. The accessory power load of the minimum friction engine is two horsepower lower because of the elimination of the oil pump. This reduced load combined with the general effect of lower friction reduces the fuel flow still further as can be seen in Table 3-10. The addition of the Rankine bottoming cycle increases the accessory load but not to the original level. The effect of the bottoming cycle is to reduce the fuel flow still further. In notch 8 on Table 3-11, the fuel flow is 731 lb/hr, a reduction of 35% compared to the baseline engine. The augmented engine characterization is shown in Table 3-12. The reduction in fuel flow is not as dramatic as that of the adiabatic engine. The design of the engine, that is the matching of the gas turbine system with the Diesel engine, is critical to its efficiency and Table 3-12 reflects the most fuel efficient version.

F. BOTTOMING CYCLES

The bottoming cycles can be analyzed in a similar manner by using the exhaust gas temperature and flow rate of the Diesel engine as the input to a Rankine cycle analysis. The Rankine cycle analysis determines the amount of additional power available to the engine system. The addition of the bottoming cycle does not affect the fuel flow rate of the Diesel engine itself. The engine system is resized so that the total of the Diesel engine power and the bottoming cycle power is the same as the net power of the base

Table 3-7. Baseline Turbocharged Diesel Engine

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	125	25	100	.468	58
2	460	370	30	340	.412	152
3	550	725	40	685	.393	285
4	630	1120	60	1060	.372	417
5	720	1650	90	1560	.362	597
6	800	2195	125	2070	.356	781
7	890	2900	175	2725	.350	1015
8	925	3225	225	3000	.350	1129
Idle	380					35
Dynamic Braking						125

Note: BSFC is based on gross power

Table 3-8. Turbocompound Diesel Engine

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	125	25	100	.468	58
2	460	370	30	340	.408	151
3	550	725	40	685	.385	279
4	630	1120	60	1060	.361	404
5	720	1650	90	1560	.348	573
6	800	2195	125	2070	.338	742
7	890	2900	175	2725	.329	954
8	925	3225	225	3000	.326	1050
Idle	380					35
Dynamic Braking						124

Note: BSFC is based on gross power

Table 3-9. Engine Schedule for Turbocompound Adiabatic Diesel Engine

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	114	14	100	.426	49
2	460	356	16	340	.371	132
3	550	705	20	685	.350	247
4	630	1091	31	1060	.327	357
5	720	1606	46	1560	.315	506
6	800	2134	64	2070	.306	653
7	890	2814	89	2725	.298	837
8	925	3115	115	3000	.294	916
Idle	380					32
Dynamic Braking						112

Note: BSFC is based on gross power

Table 3-10. Engine Schedule for Minimum Friction Turbocompound Adiabatic Diesel

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	112	12	100	.295	33
2	460	354	14	340	.268	95
3	550	703	18	685	.263	185
4	630	1089	29	1060	.257	280
5	720	1604	44	1560	.257	412
6	800	2132	62	2070	.260	554
7	890	2812	87	2725	.262	737
8	925	3113	113	3000	.269	837
Idle	380					22
Dynamic Braking						81

Note: BSFC is based on gross power

Table 3-11. Engine Schedule for Minimum Friction Turbocompound Adiabatic Diesel with Rankine Bottoming Cycle

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	119	19	100	.276	33
2	460	363	23	340	.247	90
3	550	715	30	685	.240	172
4	630	1106	46	1060	.231	255
5	720	1627	67	1560	.228	371
6	800	2163	93	2070	.228	493
7	890	2855	130	2725	.228	651
8	925	3166	166	3000	.231	731
Idle	380					21
Dynamic Braking						75

Note: BSFC is based on gross power

Table 3-12. Engine Schedule for Augmented Diesel Engine

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	125	25	100	.468	58
2	460	370	30	340	.408	151
3	550	725	40	685	.385	279
4	630	1120	60	1060	.361	404
5	720	1650	90	1560	.348	574
6	800	2195	125	2070	.338	742
7	890	2900	175	2725	.329	954
8	925	3225	225	3000	.326	1051
9	925	3425	225	3200	.384	1315
10	925	3625	225	3200	.436	1580
Idle	380					35
Dynamic Braking						124

Note: BSFC is based on gross power

engine. The fan and pump power loads of the Rankine bottoming cycle must be included in the Diesel engine accesssory load. The final power and fuel flow numbers are those of the entire system and these numbers are used in the RAIL program. Tables 3-13 to 3-20 present engine schedules showing the relationship between engine power and speed for this group of engine variations. The first three, Tables 3-13 to 3-15, are for non-turbocompounded engines with Rankine or Stirling bottoming cycles. The thermal efficiency of the Stirling cycle is more sensitive to the difference betwee the heat source temperature and the heat sink temperature than is the Rankine cycle. When the temperature difference is small, the Rankine cycle is the more efficient of the two. The effects of fuel modification bottoming cycles are presented in Tables 3-16 to 3-20 for both base and adiabatic engines using methanol or Diesel fuels. The greatest improvement comes from using methanol in a direct decomposition mode. The gain is about 20% in all notches. These engine schedules have been put into the "RAIL" program for production running.

G. ALTERNATIVE FUELS

The same technique has been used to simulate the effects of alternative fuels on fuel consumpti . The effect of the fuel on engine power and fuel flow is estimated. The engine is resized so that the new output power level in notch 8 is the same as that of the base engine. A cycle analysis is used to compute the BSFC and thermal efficiency as well as the fuel flow rate. The brake specfic energy consumption (BSEC) is also computed. The BSEC is another way of stating the thermal efficiency and is preferred by some engineers when comparing fuels. The engine speed is unchanged. A series of engine schedules like those of Table 3-7 to 3-12 have been prepared (one for each candidate fuel) for use in the "RAIL" program. These schedules are shown as Tables 3-21 to 3-36. Some fuels such as the synthetic hydrocarbons, have little effect on engine operation. The alcohols show a greater effect because of the lower heat of combustion. The gaseous fuels lower the volumetric efficiency which affects the power levels for a given engine size. Hydrogen in the gaseous form shows a significant change because of both the volumetric efficiency and the effect of the heat of combustion.

H. ENGINE ANALYSIS

In preparing these schedules, a series of engine analyses were made using the baseline engine and the range of fuels of interest. The analytical approach was validated by conducting the analysis for Diesel fuel No. 2 and comparing these results to published engine data. These comparisons were found to agree. Using this same analytical approach, calculations were made for engines running on the alternative fuels.

These calculations are based on complete combustion of the fuel in excess air. All results are based on the lower heating value of the fuel. The results include the effects of the sensible heat and the latent heat of vaporization on the temperature drop in the fuel-air charge because of complete vaporization of the fuel. The effect of residual gases on engine performance is also taken into account.

Table 3-13. Engine Schedule for Rankine Bottoming Cycle on Base Engine

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	134	34	100	.431	58
2	460	379	39	340	.379	144
3	550	737	52	685	.358	264
4	630	1137	77	1060	.330	375
5	720	1673	113	1560	.317	530
6	800	2226	156	2070	.312	695
7	890	2943	218	2725	.306	901
8	925	3278	278	3000	.305	1000
Idle	380					32
Dynamic Braking						115

Note: BSFC is based on gross power

Table 3-14. Engine Schedule for Rankine Bottoming Cycle on Adiabatic Engine

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	119	19	100	.406	48
2	460	363	23	340	.357	130
3	550	715	30	685	.340	243
4	630	1106	46	1060	.310	343
5	720	1627	67	1560	.298	485
6	800	2163	93	2070	.290	628
7	890	2855	130	2725	.284	809
8	925	3166	166	3000	.281	890
Idle	380					30
Dynamic Braking						108

Note: BSFC is based on gross power

Table 3-15. Engine Schedule for Stirling Bottoming Cycle on Adiabatic Engine

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	121	21	100	.463	56
2	460	366	26	340	.375	137
3	550	719	34	685	.332	239
4	630	1111	51	1060	.301	334
5	720	1633	73	1560	.289	472
6	800	2170	100	2070	.281	610
7	890	2863	138	2725	.274	785
8	925	3175	175	3000	.271	861
Idle	380					35
Dynamic Braking						114

Note: BSFC is based on gross power

Table 3-16. Engine Schedule for Methanol Decomposition on Base Engine

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	125	25	100	.822	103
2	460	370	30	340	.724	268
3	550	725	40	685	.681	494
4	630	1120	60	1060	.646	724
5	720	1650	90	1560	.629	1037
6	800	2195	125	2070	.618	1356
7	890	2900	175	2725	.607	1760
8	925	3225	225	3000	.607	1958
Idle	380					61
Dynamic Braking						219

Note: BSFC is based on gross power

Table 3-17. Engine Schedule for Methanol Reforming on Base Engine

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	125	25	100	.893	112
2	460	370	30	340	.787	291
3	550	725	40	685	.741	537
4	630	1120	60	1060	.702	786
5	720	1650	90	1560	.682	1127
6	800	2195	125	2070	.671	1474
7	890	2900	175	2725	.661	1917
8	925	3225	225	3000	.659	2125
Idle	380					67
Dynamic Braking						238

Note: BSFC is based on gross power

Table 3-18. Engine Schedule for Partial Oxidation of Methanol on Base Engine with Rankine Bottoming Cycle

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	134	34	100	.958	128
2	460	379	39	340	.845	320
3	550	737	52	685	.800	590
4	630	1137	77	1060	.741	843
5	720	1673	113	1560	.718	1200
6	800	2226	156	2070	.705	1568
7	890	2943	218	2725	.692	2035
8	925	3278	278	3000	.689	2260
Idle	380					72
Dynamic Braking						256

Note: BSFC is based on gross power

Table 3-19. Engine Schedule for Partial Oxidation of Diesel Fuel on Adiabatic Engine with Rankine Bottoming Cycle

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	134	34	100	.424	57
2	460	379	39	340	.374	142
3	550	737	52	685	.356	262
4	630	1137	77	1060	.328	373
5	720	1673	113	1560	.317	530
6	800	2226	156	2070	.310	690
7	890	2943	218	2725	.303	893
8	925	3278	278	3000	.301	988
Idle	380					32
Dynamic Braking						113

Note: BSFC is based on gross power

Table 3-20. Engine Schedule for Partial Oxidation of Diesel Fuel on Base Engine with Rankine Bottoming Cycle

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	134	34	100	.442	59
2	460	379	39	340	.389	147
3	550	737	52	685	.369	272
4	630	1137	77	1060	.342	389
5	720	1673	113	1560	.331	554
6	800	2226	156	2070	.325	723
7	890	2943	218	2725	.319	939
8	925	3278	278	3000	.318	1044
Idle	380					33
Dynamic Braking						118

Note: BSFC is based on gross power

Table 3-21. Engine Schedule for Diesel No. 2

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.468	8.70	58
2	460	370	340	.412	7.66	152
3	550	725	685	.393	7.31	285
4	630	1120	1060	.372	6.92	417
5	720	1650	1560	.362	6.73	597
6	800	2195	2070	.356	6.62	781
7	890	2900	2725	.350	6.51	1015
8	925	3225	3000	.350	6.51	1129
Idle	380					35
Dynamic Braking						125

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 18,600 Btu/lb

Table 3-22. Engine Schedule for Naphtha

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.466	8.77	58
2	460	370	340	.411	7.72	152
3	550	725	685	.392	7.37	284
4	630	1120	1060	.371	6.97	415
5	720	1650	1560	.361	6.78	596
6	800	2195	2070	.355	6.67	779
7	890	2900	2725	.349	6.56	1012
8	925	3225	3000	.349	6.56	1126
Idle	380					35
Dynamic Braking						124

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu / hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 18,800 Btu/lb

Table 3-23. Engine Schedule for Liquid Methane

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.408	8.77	51
2	460	370	340	.359	7.72	133
3	550	725	685	.343	7.37	249
4	630	1120	1060	.324	6.97	363
5	720	1650	1560	.315	6.78	520
6	800	2195	2070	.310	6.67	681
7	890	2900	2725	.305	6.56	885
8	925	3225	3000	.305	6.56	984
Idle	380					30
Dynamic Braking						108

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 21,500 Btu/lb

Table 3-24. Engine Schedule for Coal Derived Distillate

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.488	8.74	61
2	460	370	340	.430	7.70	159
3	550	725	685	.410	7.34	297
4	630	1120	1060	.388	6.95	435
5	720	1650	1560	.378	6.76	623
6	800	2195	2070	.372	6.65	815
7	890	2900	2725	.365	6.54	1060
8	925	3225	3000	.365	6.54	1178
Idle	380					36
Dynamic Braking						130

Note: BSEC - Brake Specific Energy Consumption in thousands of BTU/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 17,900 Btu/lb

Table 3-25. Engine Schedule for Oil Shale Distillate

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.483	8.74	60
2	460	370	340	.425	7.70	157
3	550	725	685	.406	7.34	294
4	630	1120	1060	.384	6.95	430
5	720	1650	1560	.373	6.76	616
6	800	2195	2070	.367	6.65	806
7	890	2900	2725	.361	6.54	1047
8	925	3225	3000	.361	6.54	1165
Idle	380					
Dynamic Braking						128

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 18,100 Btu/lb

Table 3-26. Engine Schedule for Methanol

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	1.048	9.00	131
2	460	370	340	.923	7.92	341
3	550	725	685	.881	7.56	639
4	630	1120	1060	.834	7.15	934
5	720	1650	1560	.811	6.96	1338
6	800	2195	2070	.798	6.84	1751
7	890	2900	2725	.784	6.73	2275
8	925	3225	3000	.784	6.73	2530
Idle	380					76
Dynamic Braking						271

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 8,580 Btu/lb

Table 3-27. Engine Schedule for Ethanol

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (hp/hr)
1	380	125	100	.773	8.93	97
2	460	370	340	.681	7.86	252
3	550	725	685	.649	7.50	471
4	630	1120	1060	.615	7.10	689
5	720	1650	1560	.598	6.91	987
6	800	2195	2070	.588	6.79	1291
7	890	2900	2725	.578	6.68	1677
8	925	3225	3000	.578	6.68	1865
Idle	380					56
Dynamic Braking						201

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 11,550 Btu/lb

Table 3-28. Engine Schedule for Liquid Hydrogen

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.174	8.98	22
2	460	370	340	.153	7.91	57
3	550	725	685	.146	7.55	106
4	630	1120	1060	.138	7.14	155
5	720	1650	1560	.135	6.95	222
6	800	2195	2070	.132	6.83	291
7	890	2900	2725	.130	6.72	378
8	925	3225	3000	.130	6.72	420
Idle	380					13
Dynamic Braking						45

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 51,600 Btu/lb

Table 3-29. Engine Schedule for Liquid Ammonia

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	1.122	8.98	140
2	460	370	340	.989	7.91	366
3	550	725	685	.943	7.55	684
4	630	1120	1060	.893	7.14	1000
5	720	1650	1560	.869	6.95	1434
6	800	2195	2070	.854	6.83	1875
7	890	2900	2725	.840	6.72	2436
8	925	3225	3000	.840	6.72	2710
Idle	380					81
Dynamic Braking						291

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 8,000 Btu/lb

Table 3-30. Engine Schedule for Coal-Diesel Fuel Slurry (20%/80%)

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.503	8.90	63
2	460	370	340	.443	7.84	164
3	550	725	685	.423	7.48	306
4	630	1120	1060	.400	7.08	448
5	720	1650	1560	.389	6.89	642
6	800	2195	2070	.383	6.77	840
7	890	2900	2725	.376	6.66	1091
8	925	3225	3000	.376	6.66	1213
Idle	380					37
Dynamic Braking						131

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 17,700 Btu/lb

Table 3-31. Engine Schedule for Gasoline-Lube Oil Blend (15%/85%)

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.495	8.81	62
2	460	370	340	.436	7.75	161
3	550	725	685	.416	7.40	301
4	630	1120	1060	.394	7.01	441
5	720	1650	1560	.383	6.81	632
6	800	2195	2070	.376	6.70	826
7	890	2900	2725	.370	6.59	1074
8	925	3225	3000	.370	6.59	1194
Idle	380					37
Dynamic						130

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 17,800 Btu/lb.

Table 3-32. Engine Schedule for Gasoline-Lube Oil Blend (70%/30%)

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.468	8.77	58
2	460	370	340	.412	7.72	152
3	550	725	685	.393	7.37	285
4	630	1120	1060	.372	6.97	417
5	720	1650	1560	.362	6.78	597
6	800	2195	2070	.356	6.67	781
7	890	2900	2725	.350	6.56	1015
8	925	3225	3000	.350	6.56	1128
Idle	380					35
Dynamic Braking						124

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 18,750 Btu/lb

Table 3-33. Engine Schedule for Water-Diesel No. 2 Emulsion (10%/84%)

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.538	8.45	67
2	460	370	340	.474	7.44	175
3	550	725	685	.452	7.10	328
4	630	1120	1060	.428	6.72	479
5	720	1650	1560	.416	6.53	687
6	800	2195	2070	.409	6.43	899
7	890	2900	2725	.403	6.32	1167
8	925	3225	3000	.403	6.32	1298
Idle	380					41
Dynamic Braking						148

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 15,700 Btu/lb
Balance of fuel is surfactant

Table 3-34. Engine Schedule for Methanol-Diesel No. 2 Emulsion (10%/86%)

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.494	8.39	62
2	460	370	340	.435	7.39	161
3	550	725	685	.415	7.05	301
4	630	1120	1060	.393	6.68	440
5	720	1650	1560	.382	6.49	630
6	800	2195	2070	.376	6.39	825
7	890	2900	2725	.369	6.28	1071
8	925	3225	3000	.369	6.28	1191
Idle	380					38
Dynamic Braking						137

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 17,000 Btu/lb
Balance of fuel is surfactant

Table 3-35. Engine Schedule for Ethanol-Diesel No. 2 Emulsion (20%/72%)

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.500	8.35	63
2	460	370	340	.440	7.35	163
3	550	725	685	.420	7.02	305
4	630	1120	1060	.398	6.64	446
5	720	1650	1560	.387	6.46	638
6	800	2195	2070	.381	6.36	835
7	890	2900	2725	.374	6.25	1085
8	925	3225	3000	.374	6.25	1207
Idle	380					39
Dynamic Braking						139

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 16,700 Btu/lb
Balance of fuel is surfactant

Table 3-36. Engine Schedule for Heavy Aromatic Naphtha-Diesel No. 2 (75%/25%)

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	380	125	100	.470	8.79	59
2	460	370	340	.414	7.74	153
3	550	725	685	.395	7.39	286
4	630	1120	1060	.374	6.99	419
5	720	1650	1560	.364	6.80	600
6	800	2195	2070	.358	6.69	785
7	890	2900	2725	.352	6.58	1020
8	925	3225	3000	.352	6.58	1135
Idle	380					35
Dynamic Braking						125

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 18,700 Btu/lb

Cycle analysis is corrected for frictional losses and heat transfer losses to the coolant. When using fuels with low cetane ratings (e.g., alcohols, hydrogen, etc.) a small amount (about 5%) of liquid Diesel fuel is injected at the end of the compression stroke to start ignition. The pilot charge properties are assumed to be the same as those of the main fuel.

I. ALTERNATIVE ENGINES

Nine alternative engines have been analyzed to determine the fuel consumption for the eight notch positions, idle and dynamic braking. The schedules for these engines are shown in Tables 3-37 through 3-45. These schedules differ from the earlier ones for the Diesel engines in that the engine speeds are not included. The engine speeds range from very high for the steam and gas turbines to relatively low for the Stirling engines. The speeds themselves cannot be directly compared to that of the Diesel engine, and have not been included for that reason.

A schedule has been included for the first generation reciprocating drive steam engine even though the RAIL program was designed for the electric transmission. The first generation steam locomotive and other locomotives using non-electric transmissions can be analyzed using modified versions of the engine schedules specifically formulated for them. The fuel flow rates are changed to compensate for the differences in the transmission efficiencies. This approach is not strictly accurate but it does enable the RAIL program to be used for all locomotives and the results can be directly compared.

Because of the wide range of thermal efficiencies involved in these engines and the range of fuels with their associated costs, it is not possible to make meaningful comparisons solely on the basis of the energy consumption of the various engines. The life-cycle costs are required to provide the detail needed for the ranking of the engines.

Table 3-37. Engine Schedule for Steam Turbine-Electric Using Bituminous Coal

Notch Position	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	128	100	1.064	13.09	136
2	373	340	.882	10.85	329
3	747	685	.831	10.22	621
4	1154	1060	.812	9.99	937
5	1699	1560	.785	9.66	1334
6	2267	2070	.771	9.48	1748
7	3021	2725	.762	9.37	2302
8	3364	3000	.749	9.21	2520
Idle					92
Dynamic Braking					154

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 12,300 Btu/lb

Table 3-38. Engine Schedule for Phosphoric Acid Fuel Cell Using Methanol

Notch Position	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	128	100	.729	6.25	101
2	362	340	.628	5.39	227
3	716	685	.600	5.15	430
4	1103	1060	.588	5.04	648
5	1619	1560	.581	4.98	941
6	2147	2070	.587	5.04	1260
7	2834	2725	.599	5.14	1698
8	3125	3000	.605	5.19	1891
Idle					32
Dynamic Braking					90

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 8,580 Btu/lb

Table 3-39. Engine Schedule for Open Cycle, Internal Combustion Regenerative Gas Turbine Using Oil Shale Distillate

Notch Position	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	128	100	1.155	21.14	148
2	362	340	.617	11.29	223
3	716	685	.466	8.53	334
4	1103	1060	.402	7.36	443
5	1619	1560	.364	6.66	589
6	2147	2070	.339	6.20	728
7	2834	2725	.331	6.06	938
8	3125	3000	.331	6.06	1034
Idle					103
Dynamic					145

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 18,300 Btu/lb

Table 3-40. Engine Schedule for Open Cycle, External Combustion Regenerative Gas Turbine Using Bituminous Coal

Notch Position	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC kBtu (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	121	100	2.235	[illegible]	270
2	365	340	1.030	[illegible]	376
3	721	685	.754	9.27	544
4	1108	1060	.652	8.02	722
5	1629	1560	.589	7.24	959
6	2162	2070	.564	6.94	1219
7	2870	2725	.564	6.94	1619
8	3175	3000	.544	6.69	1727
Idle					112
Dynamic					276

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 12,300 Btu/lb

Table 3-41. Engine Schedule for Closed Cycle, External Combustion Regenerative Gas Turbine Using Bituminous Coal

Notch Position	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	126	100	.712	8.76	90
2	371	340	.643	7.91	238
3	726	685	.581	7.15	422
4	1119	1060	.551	6.78	616
5	1651	1560	.544	6.69	898
6	2196	2070	.537	6.60	1179
7	2899	2725	.533	6.56	1545
8	3225	3000	.530	6.52	1709
Idle					106
Dynamic					89

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 12,300 Btu/lb

Table 3-42. Engine Schedule for Advanced Stirling Engine Using Oil Shale Distillate

Notch Position	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	121	100	.388	7.10	47
2	369	340	.345	6.31	127
3	722	685	.330	6.04	238
4	1110	1060	.320	5.86	355
5	1628	1560	.312	5.71	508
6	2166	2070	.305	5.58	661
7	2872	2725	.303	5.54	870
8	3180	3000	.302	5.53	960
Idle					12
Dynamic					48

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 18,300 Btu/lb

Table 3-43. Engine Schedule for Advanced Stirling Engine Using Bituminous Coal

Notch Position	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	128	100	.885	10.88	113
2	372	340	.668	8.22	248
3	725	685	.662	7.65	451
4	1114	1060	.594	7.31	662
5	1637	1560	.572	7.04	936
6	2176	2070	.556	6.84	1210
7	2895	2725	.546	6.72	1581
8	3210	3000	.544	6.69	1746
Idle					53
Dynamic					112

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 12,300 Btu/lb

Table 3-44. Engine Schedule for Stratified Charge Rotary Engine Using Diesel No. 2

Notch Position	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	126	100	.952	17.71	120
2	371	340	.589	10.96	218
3	726	685	.494	9.19	359
4	1119	1060	.453	8.43	507
5	1651	1560	.425	7.90	702
6	2196	2070	.409	7.61	898
7	2899	2725	.401	7.46	1162
8	3225	3000	.397	7.38	1280
Idle					89
Dynamic					119

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 18,600 Btu/lb

Table 3-45. Engine Schedule for First Generation Steam Engine Using Bituminous Coal

Notch Position	Gross Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Fuel Flow (lb/hr)
1	128	100	2.65	31.8	339
2	380	340	1.51	18.2	575
3	740	685	1.25	15.0	923
4	1133	1060	1.15	13.8	1305
5	1660	1560	1.12	13.4	1852
6	2210	2070	1.09	13.0	2403
7	2937	2725	1.07	12.8	3145
8	3260	3000	1.06	12.7	3456
Idle					228
Dynamic Braking					228

Note: BSEC - Brake Specific Energy Consumption in thousands of Btu/hp-hr
BSFC and BSEC are based on gross power
Lower heat of combustion is 12,300 Btu/lb

J. SECTION III REFERENCES AND NOTES

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SECTION IV
LOCOMOTIVE MODIFICATIONS

A. INTRODUCTION

Locomotive modifications are those modifications which can be made to present locomotives or that could be incorporated into new locomotives in the next five to ten year period. The subject of regenerative energy storage systems which could be incorporated within this five to ten year time frame are discussed in a separate chapter because they involve more than just a change to the locomotive. The retrofitting of turbochargers is discussed in this chapter. A more detailed discussion of turbocharging is presented in Section V as an introduction to turbocompounding and advanced Diesel engine designs.

The near-term modifications discussed in this section are combustion chamber and fuel system changes, bottoming cycles, electric transmission improvements, improved accessories and more efficient dynamic braking. These modifications could be introduced into the locomotive fleet either as field modifications by maintenance personnel or when the engines are rebuilt in the central railroad shops. In some cases, specific engines may be scheduled for a major changeover.

B. ENGINE MODIFICATIONS

Locomotive Diesel engines are highly developed but there appears to be room for improvement in their combustion chamber, valves, and the fuel injection systems. These changes are more evolutionary than revolutionary. The parameters which are examined are maximum cylinder pressures, compression ratio, piston speed, fuel injection timing, injection pressure and rate, fuel spray formation, spray pattern, air/fuel mixture, and turbocharger boost pressure.

These areas have been, and are, under continuous development by the manufacturers. The present engines are close to optimum for Diesel No.2 fuel. With the advent of alternative fuels, however, the engine system will have to be re-optimized. Modifications to some of the parameters mentioned above have been examined for their effect on engine efficiency. These are:

(1) Increase turbocharge boost pressure to increase mean effective pressure.
(2) Increase maximum cylinder pressures to increase indicated thermal efficiency.
(3) Reduce piston speed to increase mechanical efficiency.
(4) Lean-out fuel-air mixture to increase indicated thermal efficiency.

Based on cost and performance tradeoffs, turbocharger pressures are currently limited to about 2.7 atm (absolute). Above this limit, the combustion pressure will exceed the maximum cylinder pressure limit of 1500 to 2000 psia, depending on the particular engine design. If this maximum pressure limit is relaxed, the turbocharger boost pressure can be increased even further. The energy limit (exhaust energy required to drive the turbocharger) is reached at about 4 atm. The analysis assumes that turbocharger efficiencies will be maintained at the higher boost pressures. By increasing the boost pressure of the baseline engine from 2 to 3 atm, the mean effective pressure is increased by 37%. This increases the engine efficiency by 2 to 3 percentage points. However, the thermal efficiency will decrease by one percentage point if fuel injection retardation is necessary to limit the maximum cylinder pressure to 2000 psia. The brake thermal efficiency is improved by only 1 to 2 percentage points in this case.

The thermodynamic cycle efficiency of a Diesel engine increases with an increase in maximum cylinder pressures. An increase in the pressure, however, will either adversely affect the engine durability or will require a redesign of the engine because of increased mechanical and thermal stresses. An analysis was made to assess the potential efficiency gains that will result from increasing the cylinder pressure. These results indicate that an additional 4 percentage point improvement in thermal efficiency is possible if the pressure limit is raised to 3000 psia.

The brake thermal efficiency can be further increased if the piston speed is reduced to keep the output power constant. Because the piston friction power loss is proportional to mean piston speed, there could be a measurable improvement in the mechanical efficiency. There may be a slight penalty in indicated thermal efficiency because of poorer fuel-air mixing. The present calculations indicate a possible net improvement of one percentage point in the brake thermal efficiency for a 37% reduction in piston speed. The engines would require basic redesign, however, to maintain existing power ratings.

Another alternative for increasing engine efficiency is to increase turbocharge boost pressure and to lean out the fuel-air mixture while maintaining constant power. The leaned out mixture reduces the combustion temperature and the exhaust gas specific heats. Because of lower specific heats, there is a higher temperature rise per unit mass of fuel. This effect increases indicated efficiency and because the mechanical efficiency remains essentially unaffected, the net result is an increase in brake thermal efficiency. The locomotive Diesel engines, however, already operate very lean and, hence, there is not much potential for further leaning out of the mixture.

In summary, engine modifications could improve the brake thermal efficiency of an engine from one to three percentage points without adversely affecting the existing engine and from three to four percentage points if the engine is redesigned.

C. BOTTOMING CYCLES

In a heat engine, a large portion of the heat energy loss because of thermodynamic limitations leaves the engine through the exhaust system. If this comparatively low grade heat energy could be recovered and converted to useful work, the overall system efficiency would increase.

A broad spectrum of waste heat recovery concepts are evaluated in this study. These concepts are grouped into three main categories: power generating cycles, charge heating cycles and fuel reforming cycles.

Of these, the power generating cycles have received maximum attention. The engine exhaust heat is converted to useful power through options such as turbocharging, turbocompounding, Rankine engine compounding and Stirling engine compounding. The power thus generated can be added to the base engine crank shaft or can be used to drive the auxiliary equipment as shown in Figure 4-1.

In the case of charge heating cycles, the engine exhaust energy is used to heat the air, fuel, or fuel-air charge, before or after compression as shown schematically in Figure 4-2. The options falling under this category are preheating and regeneration. Preheating is not attractive for Diesel engines and regeneration is discussed under the heading of "Other Diesel Engines" in Section V.

In fuel reforming cycles, the exhaust energy is used to convert a liquid fuel into a hydrogen-rich gas, which is then used to fuel the engine as shown in Figure 4-3. The heating value of the resultant gaseous fuel is increased by the energy absorbed from the exhaust gas and, hence, the overall thermal efficiency of the system is increased. The output power of the engine, however, is reduced because the gaseous fuel displaces some of the intake air. The options in this category are: direct decomposition, steam reforming, and partial oxidation. Each of these options is only suitable for certain fuels. These options can not be used on two-stroke engines because of problems in the crankcase and with the scavenging air.

Any discussion of bottoming cycles necessitates an evaluation of the amount of energy available in the exhaust gases of the engine. Table 4-1 compiled from References 4-1 through 4-4 lists the exhaust flow rate and temperature for the two baseline Diesel engines. The exhaust temperature of the two-stroke engine is substantially below that of the four-stroke engine because of the scavenging air used to clear the cylinders. Some of this air goes out with the exhaust and cools it. As a result, the use of bottoming cycles with General Motors engines will be less effective than their use on General Electric or Morrison-Knudsen engines.

The most attractive options and their possible application to specific engines are shown in Table 4-2. Some of the options simply can not be used with a particular engine because of the exhaust gas temperature or the type of fuel system. Of the items on the table, turbocharging is discussed both in this chapter and in Section V and turbocompounding is discussed only in Section V. The Rankine cycle compounding or bottoming cycle is one of the main topics in this chapter as are all of the fuel decomposition and reforming cycles. The Stirling engine bottoming cycle is only applicable to adiabatic engines and is discussed in Section V.

D. RANKINE ENGINE COMPOUNDING

In Rankine engine compounding, the exhaust energy is used to vaporize a low-boiling point liquid (usually an organic liquid) which drives an expander and the power generated is transmitted to the base engine crank-

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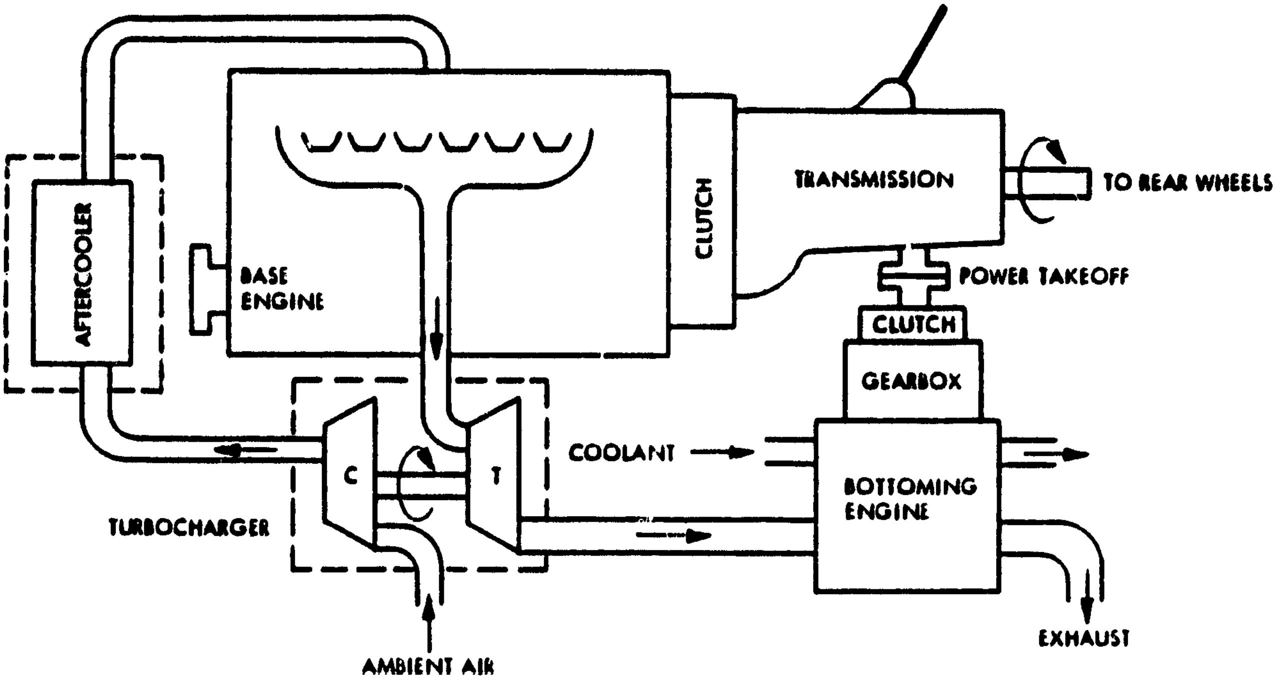

Figure 4-1. Schematic of Rankine Engine Bottoming Cycle

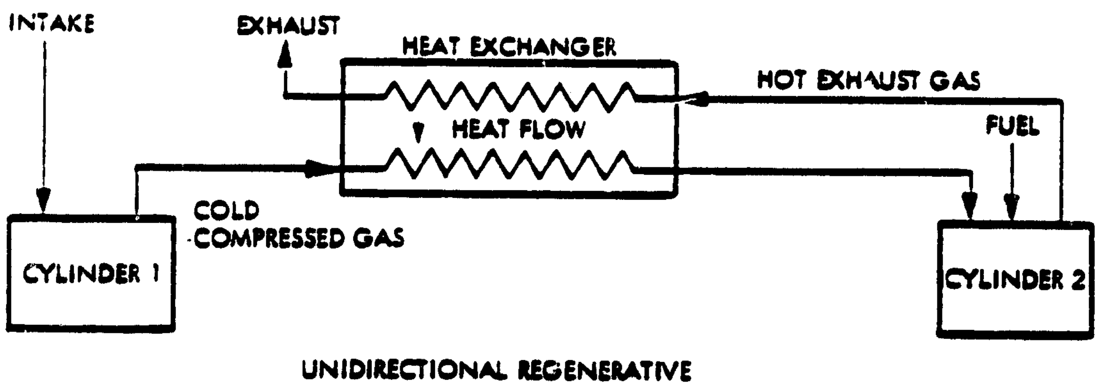

Figure 4-2. Charge Heating Bottoming Cycle Schematic

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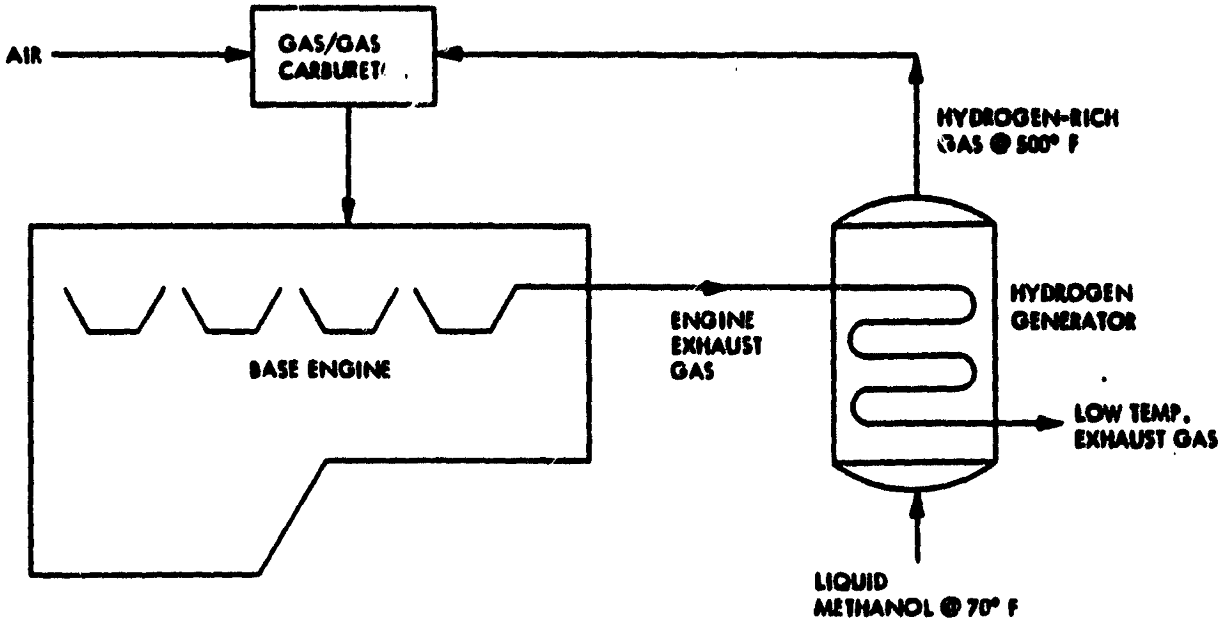

Figure 4-3. Schematic of Methanol Decomposition Bottoming Cycle System

shaft through a speed-matching gear box. The expander can be a turbine, a piston engine, or a rotary vaned machine. Figure 4-1 shows a simplified schematic of one such system. The flow diagram of this Rankine cycle is shown in Figure 4-4. The exhaust gases of the base engine pass through the vapor generator where the working fluid is vaporized. The fluid then goes to the expander where its energy is converted to work. The expanded fluid passes through a recuperator, releasing part of its heat, and then through a condenser for further cooling. In this configuration, the purpose of the recuperator is to reduce the size of the condenser and to conserve energy. After flowing through the condenser, the liquid is pumped through the recuperator again, where it is heated, and then into the vapor generator to complete the circuit. Rankine engine compounding has significant advantages in its use of low-quality heat and its minor effect on engine back pressure. Because of these advantages, the Rankine engine bottoming cycle is receiving increased attention. Thermo-Electron Corporation (TECO) (Ref. 4-5), General Electric (GE) (Ref. 4-6), JPL/Caltech (Ref. 4-7), and others have proposed or developed Rankine cycles with different organic fluids. The Garrett Corporation is proposing water as a Rankine engine fluid rather than the organic liquids. In the present analysis; Fluorinol-50, Fluorinol-85, and water were evaluated for use in the Rankine cycle. Thermo-Electron Corporation is using Fluorinol-50 and GE is using Fluorinol-85.

Table 4-1. Energy Distribution in Baseline 3000-hp Engines

Notch	Distribution of Fuel Energy, % of Total: Shaft	Exhaust	Coolant	Exhaust Gas Flowrate, lb/hr	Downstream Exhaust Temperature, °F
Two-Stroke Diesel Engine					
1	28.9	26.3	44.8	11700	304
2	32.7	18.3	49.0	13430	362
3	34.6	25.2	40.2	17050	491
4	36.0	25.2	38.8	20020	554
5	37.0	29.5	33.5	24080	633
6	37.3	29.0	33.7	26750	699
7	37.7	29.0	33.3	31640	727
8	37.5	31.3	31.2	39420	720
Idle	7.8	19.7	72.5	11160	200
Low Idle				10150	195
Dyn. Br. 4	19.2	20.4	60.4	21450	255
Dyn. Br. 1				11970	239
Four-Stroke Diesel Engine					
1	29.5	29.0	41.5	2350	690
2	33.7	21.8	44.5	3250	675
3	34.9	28.7	36.4	6400	800
4	37.5	28.7	33.8	9600	880
5	38.5	33.5	28.0	16200	915
6	39.7	33.7	26.6	23600	860
7	40.7	33.7	25.6	29700	800
8	40.7	36.0	23.3	35800	775
Idle	7.8	30.7	61.5	2700	350
Dynamic Brake	[illegible]	29.7	51.5	8200	550

Table 4-2. Waste Heat Utilization Configuration Matrix

Waste Heat Utilization Option	Two Stroke	Four Stroke	Adiabatic
Rankine Engine Compounding	X	X	X
Stirling Engine Compounding	N/A	N/A	X
Methanol Decomposition	N/A	X	X
Steam Reforming of Methanol	N/A	X	X
Steam Reforming of Ethanol	N/A	N/A	X
Steam Reforming of Diesel Fuel	N/A	N/A	X
Preheating	X	X	X
Regeneration	X	X	X

N/A - not applicable

During the study, total exhaust energy, usable exhaust energy, ideal Rankine cycle efficiency, expected Rankine cycle efficiency and fan horsepower requirements were evaluated. The results at the maximum power point indicate a maximum expected efficiency improvement of 13% with Fluorinol-50, 10% improvement with Fluorinol-85, and a 9% improvement with water as the working fluid. These improvement numbers are for the four-stroke engine. For the two-stroke engine, these improvements are lower by about four percentage points because of lower exhaust temperatures. The results of the analysis are summarized in Table 4-3. The driving cycle improvement numbers presented here are for the GM-EMD heavy duty cycle. In medium or light duty service, the improvements will be slightly less. The percent improvements calculated here are slightly lower than the General Electric test results (Ref. 4-6). This is partly because of slightly different component efficiencies and partly because this analysis does not take advantage of the heat rejected by the engine aftercooler. The General Electric system takes advantage of aftercooler heat with Fluorinol-50 as the Rankine fluid and this increases the improvement by more than one percentage point.

Although water yields less improvement than Fluorinol-50, it substantially reduces the requirement of heat transfer areas. First order heat transfer estimates indicate that with water as the working fluid, the heat transfer area needed would be only half as much as that for Fluorinol-50. This advantage is attributed to the high density and high heat required for vaporizing water. Water is less expensive than the organic fluids and results in a more compact system. The use of water, however, requires some protection from freezing in cold climates.

The Rankine bottoming cycle is applicable to both existing and future engines. There are considerable modifications which must be made to incorporate this system in an existing locomotive but with a savings of 8% to 12%, the fuel saved amounts to some 32,000 to 35,000 gallons per year for a 3000-hp locomotive. Because the Rankine bottoming cycle is more effective in the higher notches than in the lower notches, it is most useful on locomotives in heavy duty service.

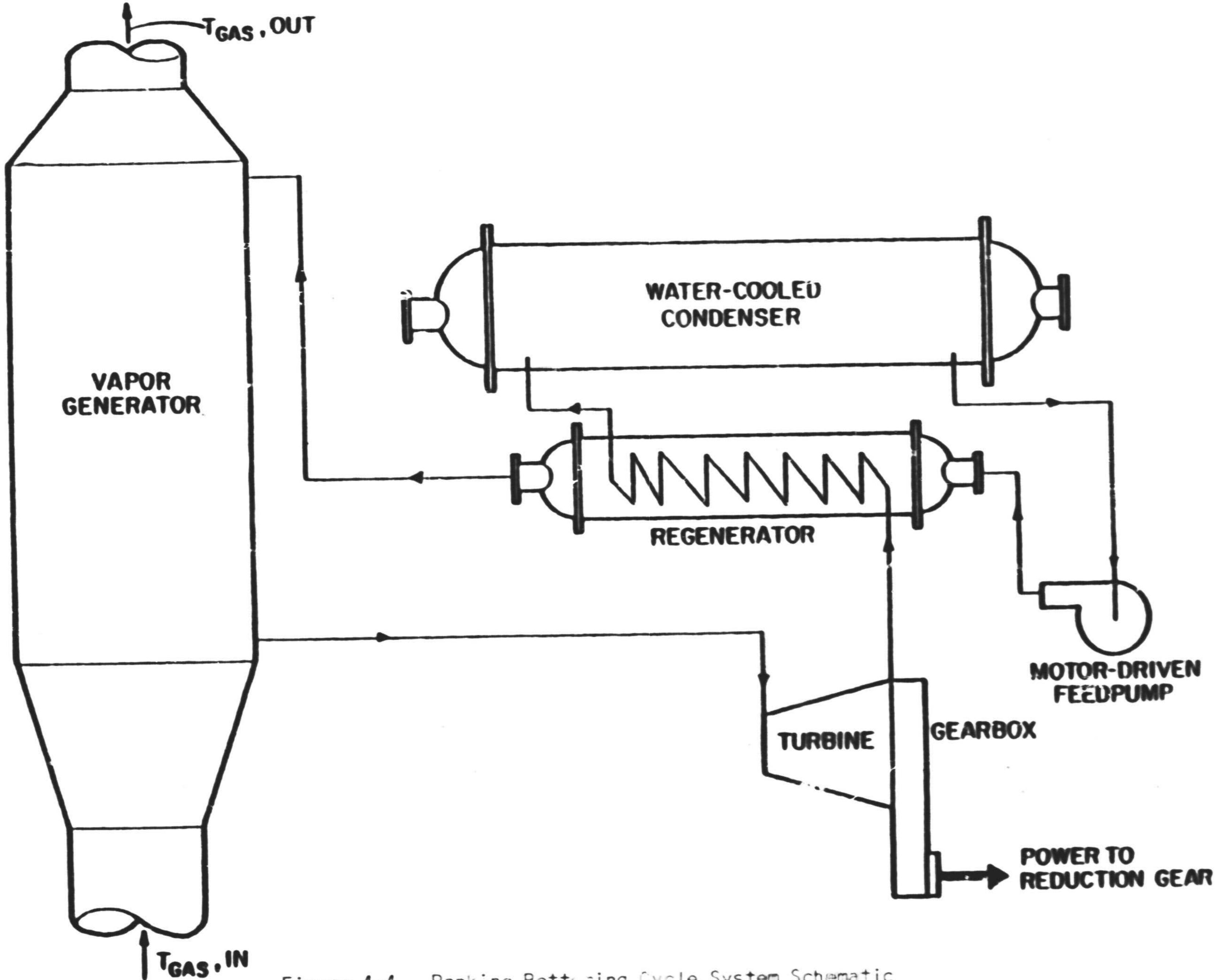

Figure 4-4. Rankine Bottoming Cycle System Schematic

Table 4-3. Rankine Bottoming Cycle Improvements

Base Engine	Rankine Fluid	Improvement in Fuel Economy, %		
		Maximum Theoretical	Maximum Expected	Expected over Driving Cycle[a]
Four Stroke				
	Fluorinol-50	22	13	12
	Fluorinol-85	17	10	9
	Water	16	9	8
Two Stroke				
	Fluorinol-50	16	9	8
	Fluorinol-85	12	7	6
	Water	11	6	5

[a]GM-EMD Heavy Duty Cycle

E. FUEL REFORMING CYCLES

Fuel decomposition and reforming cycles are based on the concept of on board hydrogen generation from liquid fuels (Refs. 4-8 and 4-9). In addition to improving the system efficiency, the reforming cycles markedly reduce exhaust emissions. The cycles represent a transition from a conventional engine to a hydrogen engine.

Although a hydrogen engine has advantages in terms of efficiency and emissions, it is difficult to store hydrogen on board a vehicle, either as a compressed gas, metal hydride, or a cryogenic liquid. Storage of hydrogen in a metal hydride is promising but it has an inherent high weight penalty associated with it. Another solution to the hydrogen storage problem is to generate the hydrogen on board the vehicle from a storable liquid.

Two chemical processes can be employed as bottoming cycles to produce hydrogen from liquid fuels. These are direct decomposition and steam reforming.

In direct decomposition, the liquid fuel is decomposed into a hydrogen rich gas through an external heat source. If the decomposition energy and temperature are lower than the engine exhaust energy and temperature, then the exhaust gas makes an ideal heat source. During decomposition, the engine exhaust energy is absorbed by the fuel, and the heating value of the resulting hydrogen-rich gas is increased by a corresponding amount. This, in turn, increases the thermal efficiency of the system.

In steam reforming, the fuel reacts with steam to produce a mixture of hydrogen, carbon monoxide, and carbon dioxide, in varying ratios depending upon the fuel. Like direct decomposition, this reaction is

C-2

endothermic and an external heat source is required. In steam reforming, the heating value of the resultant gaseous fuel is increased by the amount of energy supplied to the reaction. If this energy can be supplied from the engine exhaust heat, the system efficiency will be correspondingly increased.

Direct decomposition is the least complicated of the processes. Steam reforming requires a source of on board water but partial oxidation does not. Obviously, there are some choices in on board hydrogen generation with regard to both fuels and chemical processes. For instance, each process is suitable only for particular fuels. As an example, direct decomposition of hydrocarbon fuels and ethanol produces free carbon and therefore, is not satisfactory with these fuels. On the other hand, steam reforming of hydrocarbon fuels requires high reaction temperatures and is not suitable for use with conventional engines because the exhaust temperatures are too low. However, the exhaust temperature of an adiabatic engine may be adequate for these reactions. Both of the processes are suitable for methanol but direct decomposition is simpler, easier to implement, and yields the greatest improvement in system efficiency. Table 4-4 presents the various processes and fuels that have been evaluated.

F. DIRECT DECOMPOSITION

The direct decomposition of one mole of methanol into two moles of hydrogen and one mole of carbon monoxide represents a convenient cycle for generating hydrogen-rich gas from liquid methanol:

$$CH_3OH_{(l)} \xrightarrow[\dot{Q}]{} 2H_{2(g)} + CO_{(g)}$$

This decomposition can be carried out in a catalytic reactor at about 500°F and requires the input of energy because the reaction is endothermic. This energy input is equivalent to 20% of the lower heating value of the liquid methanol and appears as an increased heating value for the resultant gaseous fuel. Fortunately, the energy required for decomposition can be

Table 4-4. Processes for Fuel Reforming

Fuel	Process: Direct Decomposition	Process: Steam Reforming
Diesel No. 2	N/A	X
Methanol	X	X
Ethanol	N/A	X

N/A - not applicable

supplied from engine exhaust gases which are typically in the temperature range of 600 - 850°F. The major advantages of this approach are:

(1) Decomposed methanol has a 20% higher heat of combustion than liquid methanol. Because the energy required for decomposition is extracted from the engine exhaust, the thermal efficiency of the engine is increased by a corresponding amount.

(2) The presence of carbon monoxide in the gaseous fuel increases the ignition energy and lowers the flame speed in the gaseous fuel. This reduces the explosive tendencies of pure hydrogen, and smooths out engine operation.

A schematic diagram of a direct decomposition bottoming cycle is shown in Figure 4-3. In this cycle, liquid methanol is vaporized and the vapor flows through the hydrogen generator where a catalyst heated by the engine exhaust gases decomposes the methanol. Decomposition could be carried out in a non-catalytic system, but a catalyst increases decomposition efficiency and prevents the formation of soot. After passing through a cooler, the gaseous fuel is mixed with air in the desired proportions and aspirated into the engine.

Unlike conventional Diesel fuels, the hydrogen-rich gas is not ignited by the heat of compression alone. A small amount of liquid Diesel fuel has to be injected slightly before top dead center to initiate ignition. One disadvantage of a gaseous fuel is a reduction in the maximum power of the engine because the gaseous fuel displaces some of the air during the intake stroke. Another disadvantage with this system is that during engine startup and during operations where the exhaust gas temperatures are too low to decompose the fuel, liquid methanol must be injected into the engine. The results of the analysis of the methanol direct decomposition bottoming cycle indicate that a 20% improvement in the system efficiency can be expected. Further details are given in Ref. 4-8.

G. STEAM REFORMING

In case of methanol decomposition, the molar ratio of hydrogen to carbon monoxide is 2 to 1. To increase the concentration of hydrogen and to eliminate the carbon monoxide, it is possible to steam reform methanol.

In steam reforming, the fuel is first vaporized and mixed with steam. The resultant mixture goes through a catalytic reactor where a reaction takes place to yield hydrogen and carbon dioxide. Theoretically, one mole of methanol reacts with one mole of steam to yield three moles of hydrogen and one mole of carbon dioxide according to the reaction:

$$CH_3OH + H_2O \xrightarrow[\dot{Q}]{} 3H_2 + CO_2$$

This reaction takes place at about 500° F and the heat of reaction can be supplied by the engine exhaust gases. The chief advantage of the method is that, as a result of this reaction, the heating value of the gaseous fuel is increased by 13% as compared to the liquid fuel. There is a corresponding gain in the system efficiency. A simplified schematic is shown in Figure 4-5. In addition to supplying heat to the catalyst bed, the exhaust energy is required to vaporize the fuel and water. Nearly as much energy is required for these purposes as is required for the reaction itself. For a given improvement in system efficiency, twice as much exhaust energy is required than is needed for direct decomposition. The biggest disadvantage is that an on board source of water is necessary and this increases the system complexity. This option does not look nearly as attractive as methanol direct decomposition.

Steam reforming of ethanol is represented by the following equation:

$$C_2H_5OH + H_2O \xrightarrow[\dot{Q}]{} 4H_2 + 2CO$$

It is estimated that this reaction would take place at a temperature of 800 - 1000°F. The endothermic energy could be supplied from the exhaust of an adiabatic engine. The supplied energy is equivalent to 23% of the heating value of the liquid fuel and, therefore, the system efficiency will be improved by the same amount. In addition to reaction heat, another 5% is required to form steam and to vaporize the fuel. In the steam reforming of ethanol, the concentration of hydrogen is higher than for any of the other fuels considered.

Steam reforming of Diesel fuel is represented by the following:

$$CH_{1.86} + H_2O \xrightarrow[\dot{Q}]{} 1.93H_2 + CO$$

The reaction normally takes place at a temperature of 1200 - 1700°F over a nickel catalyst. Because of the high reaction temperatures, even the exhaust heat of an adiabatic engine may be barely sufficient to sustain this reaction. As a result, this scheme does not appear attractive for bottoming cycles. An analysis was conducted, however, to evaluate its potential. The results indicate that if the heat required for the reaction can be extracted from the exhaust of an adiabatic Diesel, the heating value of fuel and, therefore, the system efficiency can be improved by about 24%. The results of the analysis of all three of the methods of fuel reforming are summarized in Table 4-5.

H. PARTIAL OXIDATION FUEL REFORMING

There are several methods of fuel reforming. Some of the methods such as direct decomposition and steam reforming are endothermic and energy must be supplied to them. In this study, the energy is supplied by the engine exhaust gases. These methods are described and discussed in the section on bottoming cycles. There are also methods of fuel reforming which are exothermic, that is, energy is released by the reaction. One of these methods is partial oxidation.

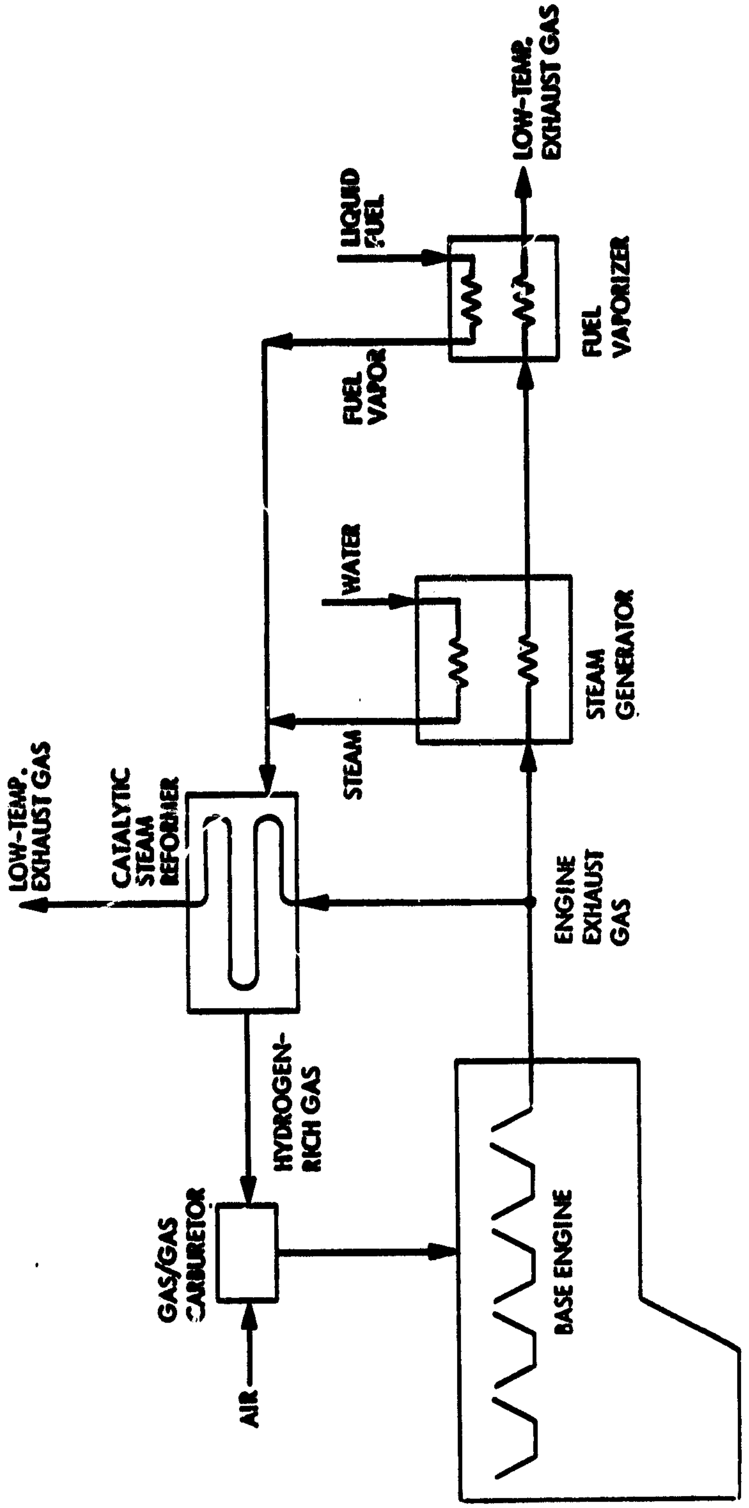

Figure 4-5. Schematic of Methanol Steam Reforming System

Table 4-5. Fuel Reforming Cycle Summary

Fuel	Process	Type of Reaction	Reaction Temperature, °F	Reaction Energy	Vaporization of Fuel and Water	Improvmenet in System [c] Efficiency, %	Feasibility
			Fraction of Exhaust Energy, (Percentage)				
Methanol	Direct Decomposition	Endothermic	500	20	N/A	20	Yes
	Steam Reforming	Endothermic	500	13	13	13	Yes
	Partial Oxidation	Exothermic	1600	N/A	N/A	10[b]	No
Ethanol	Direct Decomposition	Endothermic	N/A	N/A	N/A	N/A	N/A
	Steam Reforming	Endothermic	1200-1400[a]	23	5	23	Yes
	Partial Oxidation	Exothermic	1600	N/A	N/A	10[b]	Yes
Diesel No. 2	Direct Decomposition	Endothermic	N/A	N/A	N/A	N/A	N/A
	Steam Reforming	Endothermic	1200-1700[a]	24	8	24	Yes
	Partial Oxidation	Exothermic	1600	N/A	N/A	10[b]	Yes

[a]Application limited to adiabatic Diesel only
[b]Needs Rankine engine heat recovery unit for recovering gasifier heat and engine exhaust heat
N/A - Not applicable
[c]Improvement in percentage of the base efficiency, not percentage points

In partial oxidation, the fuel reacts with air either on a catalytic surface or in a flame front to yield a mixture of hydrogen and carbon monoxide. Because the process is exothermic, the heating value of the resultant gaseous fuel is decreased by the amount of heat evolved during the reaction. Unless this heat is recovered, the system thermal efficiency is adversely affected.

Direct decomposition of Diesel fuel is undesirable because it produces free carbon and steam reforming is not feasible in this application because the reaction temperatures exceed the available exhaust gas temperatures. Partial oxidation of Diesel fuel is possible. The partial oxidation concept employs two-stage combustion. During the first stage, liquid Diesel fuel is vaporized, mixed with air and partially oxidized in a catalytic reactor. The partial oxidation converts the fuel and air into a mixture consisting primarily of hydrogen, carbon monoxide, and nitrogen along with small amounts of methane, carbon dioxide and water vapor. To prevent soot formation during the reaction, the proper air-fuel ratio must be maintained in the catalytic reactor. The second stage of combustion takes place in the engine cylinder. The hydrogen-rich gas produced during the first stage of combustion is inducted into the intake manifold along with enough air to maintain the desired overall equivalence ratio, and it is then burned to completion.

The major advantages of this concept are:

(1) Because the first stage reaction is carried out at a suitable air-fuel ratio, no soot is produced.

(2) The presence of carbon monoxide and nitrogen in the hydrogen-rich gas lowers the flame speed, increases the ignition energy relative to pure hydrogen, and results in good engine operation.

As the conversion of liquid Diesel fuel into a hydrogen-rich gas is exothermic, energy equivalent to 16% of the lower heating value of the liquid fuel is released. The product gas temperature is about 1600°F and must be cooled before it is inducted into the cylinders. The main disadvantage aside from the complexity of the system is that unless this heat is recovered, the fuel economy will be penalized. A single Rankine bottoming cycle can be used to recover the heat from the product fuel gas and from the engine exhaust. A schematic diagram of the system is shown in Figure 4-6. Analytical and experimental details are given in Ref. 4-10.

Partial oxidation of methanol is similar to that of Diesel fuel. The stoichiometric air-fuel ratio for methanol is about 6.4 to 1. As the ratio of air to fuel is reduced, hydrogen starts to form and its concentration increases with further reductions in the air-fuel ratio of 1.4 to 1. Partial oxidation of methanol is not an attractive option because direct decompositic and steam reforming are both possible, much simpler, and yield signi[illegible]tly higher improvements in system efficiency.

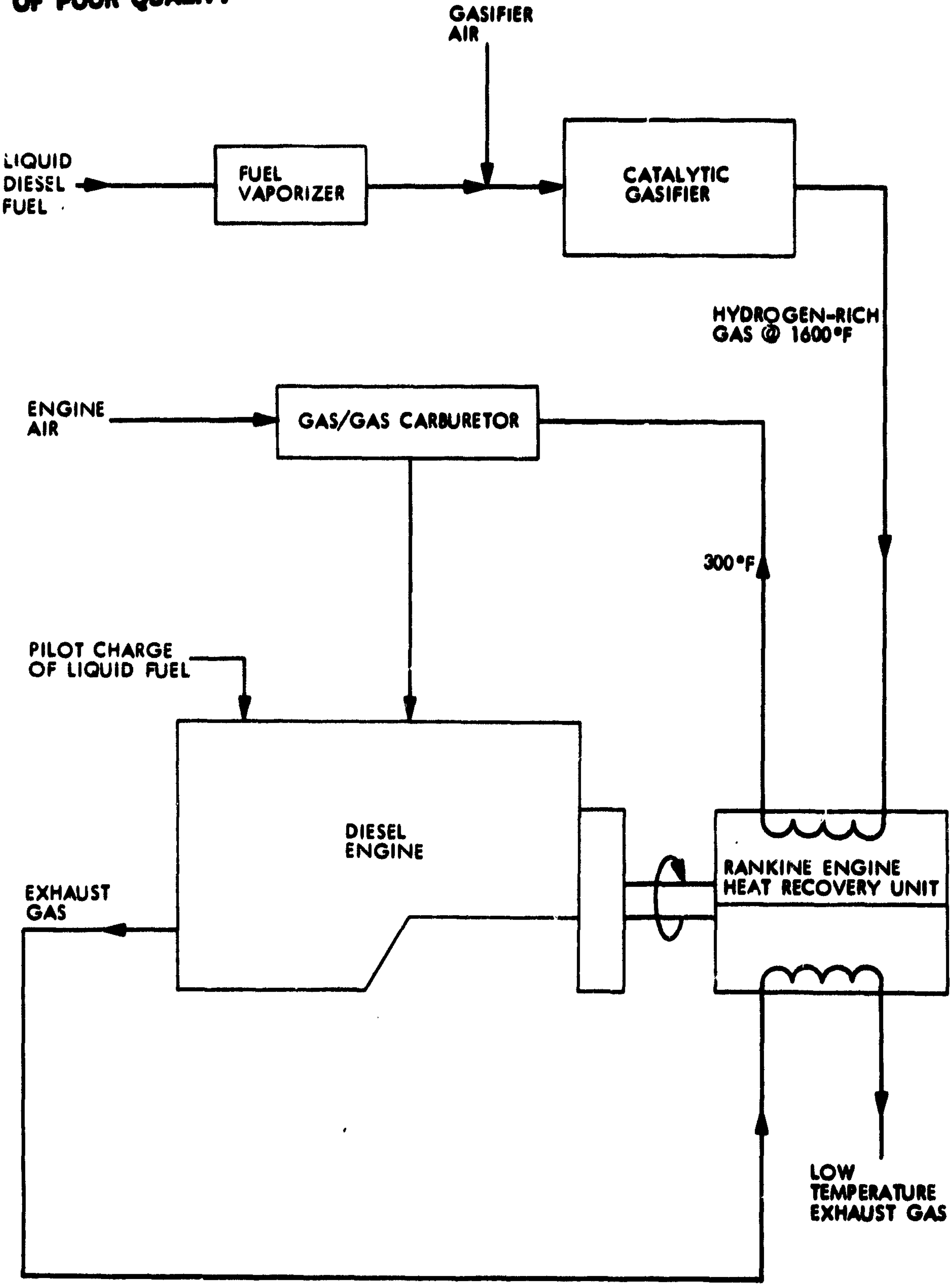

Figure 4-6. Schematic Diagram of the Partial Oxidation Fuel Reforming System

I. RETROFITTING OF TURBOCHARGERS

Although all of the engines now being installed in new locomotives are turbocharged, there are many non-turbocharged engines on the rails. For example, at least 90% of the GM-EMD GP-7 locomotives built are still in service. The retrofitting of turbochargers to these older engines can have a positive effect on fuel savings provided that the engines are in medium to heavy duty service. If the engines are in light duty service, there is no reason to do it. In notches 4 and 5, the naturally aspirated or Roots-blown Diesel is more efficient than the turbocharged Diesel. In notch 6, they are about equal and in notches 7 and 8, the turbocharged Diesel is more efficient by some 6 to 9%.

The decision to retrofit turbochargers onto engines must be made by the railroads and the manufacturers on the basis of the type of service the locomotive is expected to perform. Years ago the cost of a turbocharger was so high and the cost of fuel so low that the 6 to 9% fuel savings would never have paid off the cost. Today the situation is quite different and retrofitting can be economically feasible. To establish the economics of retrofitting turbochargers, information must be available on the inventory of units by age (and normal assignment to heavy or medium duty service) and on the proportion of time in the various notch positions. If the savings are dramatic (e.g. payback period of 2 years, for example, on the retrofit cost) then age distribution would be less important, because more units would be worth converting. If the cost of fuel continues to climb, it might cost more not to retrofit.

J. AUXILIARY AND ACCESSORY POWER

The auxiliary or accessory power required on a typical 3000 hp locomotive ranges from a high of about 25% of the gross engine output in notch 1 to about 6% in notch 8. Table 4-6 shows the breakdown of the accessory load for a typical 3000 hp locomotive by component and by notch position. This table is derived from data in References 4-1 and 4-2 but is not a direct copy of either. In terms of the actual flow of fuel, 80 lb/hr is used for the accessories in notch 8 and the rate decreases to 13 lb/hr in notch 1. An improvement of 10% of the average efficiency of the accessories would save about 3.5 lb/hr of fuel or about 3000 gal/yr/locomotive in medium duty service. The savings would be more for heavy duty service and less for light duty service.

It appears possible to improve the accessory efficiency by 10%. Most of the accessories were designed and developed years ago when energy was considerably cheaper. They were made to be reliable and relatively inexpensive with only moderate concern for their efficiency. The big power items are the equipment blowers and radiator fans. For example, in notch 8, they absorb some 215 hp. Improved aerodynamics can reduce this figure by about 8%. The use of demand controls can reduce the power by another 5 to 7%. A demand control could sense the temperatures and regulate the blower or fan speed to provide the necessary cooling. Because the power absorbed by a fan varies as the cube of the speed, even a small change in speed, particularly in notch 8, can result in a much larger change in power. By keeping the speeds as low as possible yet still providing the necessary cooling, considerable energy can be saved.

Table 4-6. Power Required for Typical Auxiliary Equipment

Item	Auxiliary Power, hp
Equipment Blower	110
Radiator Fan	104
Air Compressor	12 (unloaded)
Auxiliary Generator	4
TOTAL:	230 in notch 8

Variation in Power with Notch Position	Auxiliary Power, hp
Notch 1	24
Notch 2	31
Notch 3	42
Notch 4	60
Notch 5	88
Notch 6	127
Notch 7	175
Notch 8	230

At General Motors and General Electric a great deal of effort is currently going into reducing the auxiliary power loads (Ref. 4-11). There will be improvements in this area over the next several years.

K. DYNAMIC BRAKING

There are two braking systems on a train. One is the air brake system which is operated off the compressed air system. These brakes act on the wheels of each car upon command from the locomotive or if there is a loss of air pressure in the system. The other system is the dynamic braking system which uses the dc traction motors as generators to convert the kinetic energy of the train into electrical power. This electrical power is dissipated in a bank of air cooled resistors. Dynamic braking is provided on most new main line locomotives but there are many older locomotives on the tracks which do not have it or have it in a less efficient version.

There are problems associated with air brakes including overheating of the wheels with the possibility of failure, brake locking on empty cars, and run-in which can buckle the train. Run-in is usually prevented by "stretch" braking. The locomotive applies power at the front of the train as the air brakes are applied. The pull at the front and the drag on the cars keeps the train stretched out. This technique tends to overheat the wheels because they are absorbing both the kinetic energy of the train and the power generated by the locomotive. This method requires the locomotive to consume a substantial amount of fuel during braking.

Most railroads now make maximum use of dynamic braking to avoid the problems associated with using the air brakes. Some engine power is required even for dynamic braking. During braking, the engine is operated at a condition approximating notch 2. For a 3000 hp locomotive, the fuel used is 125 to 150 lb/hr and the locomotive will be in the dynamic braking mode 1.5% to 9% of the time depending on the type of service. The power generated by the engine is used to drive the auxiliary equipment especially the motor blowers. The high currents generated during braking produces heat which must be dissipated. Ideally, all of the auxiliary loads could be supplied by the power generated in the motors. Unfortunately, it is not always in an usable form. The voltages vary with motor speed and do not match the requirements of the blower motors. If the power could be conditioned using modern solid state electronic devices so that it could be used, then the engine could be operated at idle. For a 3000 hp engine, the difference in fuel flow between idle and dynamic braking is from 70 to 120 lb/hr. Assuming a savings of 90 lb/hr and that 8% of the time is spent in dynamic braking, then the savings in fuel is about 9000 gal/yr.

Some progress has already been made in this area (Ref. 4-11). General Motors has announced a 1.5% savings in fuel because of improved dynamic braking on the GP-50 locomotive. Better dynamic braking contributes to part of the 7% to 7.2% improvement in fuel consumption claimed by General Electric for its C30-7 and B36-7 locomotives. Although these gains are significant, more work in this area is needed.

L. ELECTRIC TRANSMISSION MODIFICATIONS

The present electric transmission has reached a very high state of development and is rugged, reliable and efficient. The level of efficiency can be seen in Figure 1-18 in Section I. The overall efficiency of the alternator, rectifier, dc motor and gearing is about 90% over a wide range of locomotive speeds. As the overall efficiency is the product of the individual efficiencies,the alternator and motors must operate at or near the 95 to 96% efficiency level. The rectifier and gearing must be at or above 99% efficiency. There is little incentive and little potential for improving the electric transmission. However, the electric transmission is the object of much research and considerable development effort. The current effort on the dc motors is to develop thinner insulation and more copper to improve heat dissipation and their ratings. The latest research is on the ac traction motor and on better wheel slip controls.

Three phase, variable frequency-variable voltage ac motors have been and are being used in railroad and rail rapid transit operations. Additional applications are either planned or in the construction phase (see Tables 4-7 and 4-8 for a partial listing from Ref. 4-12). There are potential benefits from using ac motors in some areas but their use is not without problems in others. The intent here is to highlight a few of the items that can have significant influence in the ultimate acceptance and use of a three-phase ac system.

Table 4-7. Three-Phase Traction Drive Locomotives

Date	Electrical-Equipment Manufacturer	Railway	Class + Number	Status	Inverter	Supply	Power kW	Quantity
1965	Brush	BR	Ex DE loco. "HAWK"	D	VFI	D.E.+Alt.	1100	1
1968	Nowotscherkassk	USSR	Ex WL80 K (Induction M)	D	VFI	25kV AC	4800	1(1/2)
1968	Nowotscherkassk	USSR	Ex WL80 B (Synchronous M)	D	VFI	25kV AC	4000	1(1/2)
1971	Nowotscherkassk	USSR	Ex WL80 V (Synchronous M)	P	VFI	25kV AC	8000	1
1971	BBC	DB	"DE2500" No. 1	P	PWM	D.E.+Alt.	1840	1
1972	BBC	SBB	Be4/4 12001 Ex Parceis Van	D	PWM	15kV AC	1000	1
1972	T. H. Leningrad	USSR	Shunting Locomotive	D		D.E.+Alt.	340	1
1973	BBC	DB	"DE2500" No. 2 + No. 3	P	PWM	D.E.+Alt.	1840	2
1974	BBC	DB	"DE2500" No. 1 rebuilt	D	4-Q + PWM	15kV AC	1300	1+T
1975	Nowotscherkassk	USSR	WL80a	P	VFI	25kV AC	9600	1
1976	Jevmont-Schneider	SNCF	Ex3 phase Converter Locomotive CC14003	D	PWM	25kV AC	500	1+T
1976	BBC	Ruhrkohle	A.g.	S	PWM	15kV AC	1500	6
1976	BBC	SBB	Am 6/6 Class	S	PWM	D.E.+Alt.	1840	6
1977	BBC	NS	1600P - 'DE2500' No. 1 rebuilt	P	CH-PWM	1500V DC	1400	1
1977	BBC	Eisenbahn, Hafe, Duisburg	EDE 100/500 Class Dual System	P	PWM	600V DC D.E.+Alt.	1000 473	6
1977	FS	FS	E323 Shunter	D	PWM	3000V DC	280	1
1978	BBC	SBB	Ee 6/6 Class	G	4Q-PWM	15kV AC	1100	0
1979	BBC	DB	120 Class (Universal locomotive)	S	4Q-PWM	15kV AC	5600	5
1980	Siemens	Stuttgart Stadtbahn		P	CFI	750/600V DC	840	[illegible]

Table 4-8. Three-Phase Traction Drives
Rapid Transit

Date	Electrical-Equipment Manufacturer	Location	Status	Inverter	Supply	Power kW	Quantity
1969	Wabco (USA)	Cleveland	D	PWM	600V DC	N/A	1
1973	Wabco (USA)	Cleveland	P	PWM	600V DC	N/A	3
1975	Siemens	Nuremburg (Tram)	D	CFI	600V DC	250	1
1976	Aeg-Telefunken	Berlin	P	PWM	750V DC	560	1
1977	Oy-Stromberg	Helsinki	P	PWM	750V DC	1000	3
1978	Siemens	Vienna	P	CFI	750V DC	1140	1
1979	Oy-Stromberg	Helsinki	S	PWM	750V DC	1000	39
1979	Siemens	Mulheim (Tram)	P	CFI	600V DC	380	1
1980	Siemens/Aeg	Berlin	S	CFI	750V DC	700	6
1980	Siemens/BBC	• Dusseldorf (Tram)	S	CFI	750/600V DC	600	12
1980	GEC	BR conversion of 314 P unit	D	CFI	25kV/750V DC	648	1
1980-1	Siemens	Nuremburg (underground)	S	CFI	750V DC	840	14
1981	Siemens/BBC	Munich	P	CFI	750V DC	1080	6

D = Development - Conversion of Existing Unit
P = Prototype - New Design of Equivalent
S = Series Production of Prototype

In a general comparison of dc (commutator) drives to multiphase ac systems, the desirable features of the ac motor are partially offset by the increased complexity required in the control and drive system. Some of the desirable features are: (1) no commutator with an absence of flash over and bar-to-bar voltage limitations, (2) no rotor connection and higher voltage-lower current stator connection, (3) higher motor speeds which can be used to decrease motor size or increase power per axle (a double gear reduction can be an added complexity), (4) steep torque-speed curve which facilitates wheel slip control (either parallel or separate excitation), and (5) the ac motor can be more rugged and less costly than its dc counterpart. The complexity of the ac driver system is well recognized and the system is the focus of an extensive development program. The dc commutator, a conceptually simple device, has probably been brought close to its fundamental physical limits over the last 100 years of development. In contrast, the ac inverter drive can be expected to improve with more analytical and developmental engineering efforts.

Energy transfer efficiency is one system aspect that merits special attention here. The present standard system alternator, rectifier, dc motor - approaches an efficiency of 90% under full load and in medium to high speed operations. Any ac system must not be significantly less efficient - a few percent less may still be acceptable. In fact, efficiency considerations may influence the final choice between two ac system options, i.e., induction and synchronous, because the inverter requirements and characteristics are significantly different for the two cases.

Another aspect of the ac versus dc traction motor debate is related to the possibility of improved wheel slip control of the ac motors. An important consideration in dispatching trains is whether to use four-axle or six-axle locomotives. Figure 4-7 shows the performance curves for two locomotives with four and six axles each having a 3000-hp Diesel engine. As can be seen, above 18 mph, there is no difference in locomotive performance because of the limited engine output capability. Therefore, at speeds above 18 mph, it is not important whether the locomotive has four or six axles. At lower speeds, the higher tractive effort of the six axle is desirable especially in drag service. The penalties of using six-axle locomotives are increased initial cost, increased track maintenance, increased weight, and higher locomotive cost.

If a four-axle locomotive could be developed which could match the six-axle locomotive in tractive effort and in the assumable adhesion level by using improved slip control, then this locomotive would be especially attractive in drag service. The adoption of ac motors and the attendent wheel slip control might make it possible to assume a higher level of adhesion in dispatching trains than is now possible with dc motors for the same risk of wheel slip. The elimination of the commutator allows the ac motor to operate at a higher speed and power than the dc motor and, therefore, the adhesion advantage could be used. The combination of higher power and speed results in a motor which develops more torque than a dc motor over a wide speed range. Figure 4-8 shows schematically one proposed (Reference 4-13) ac traction system.

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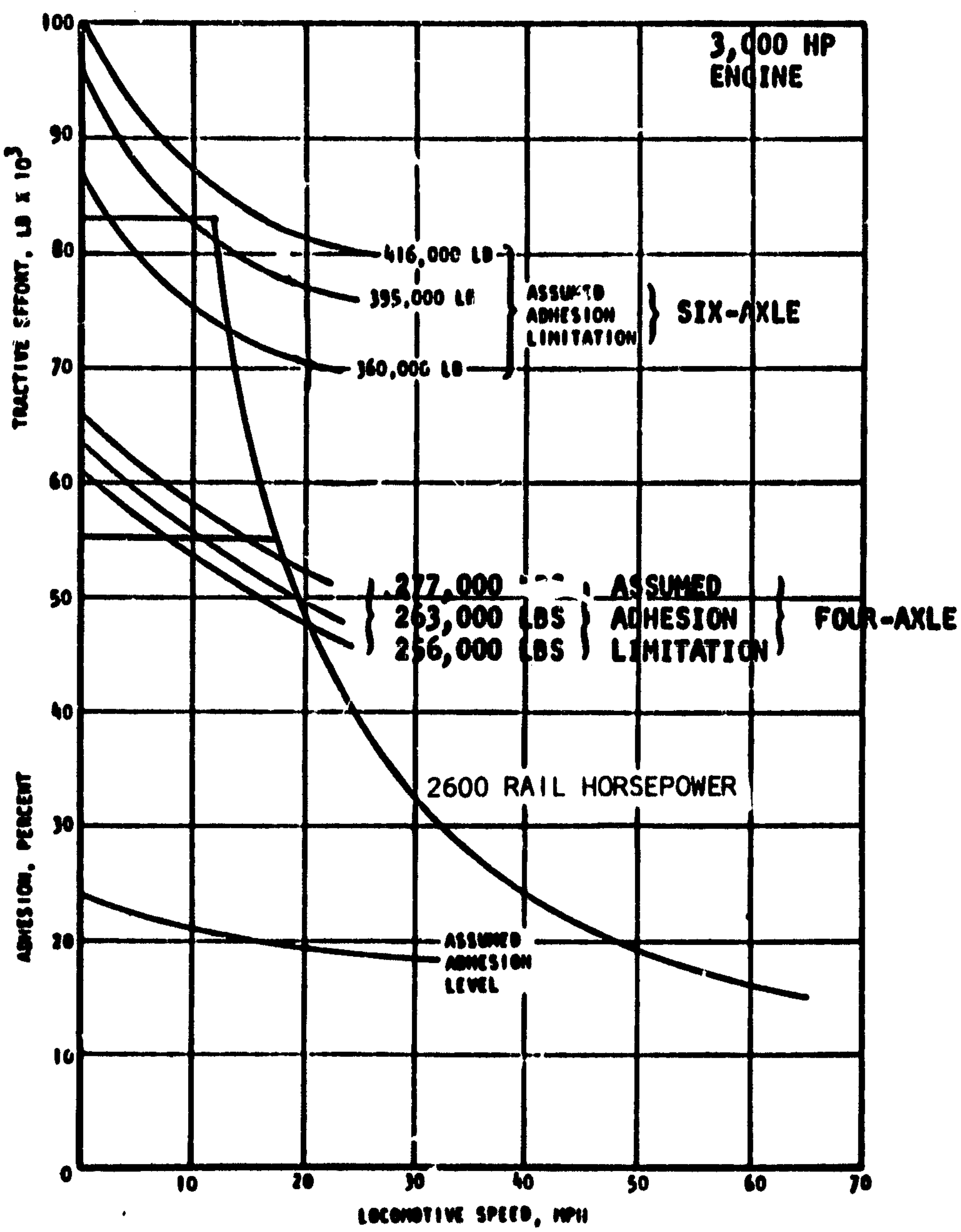

Figure 4-7. Performance of Four- and Six-Axle dc Motored Locomotives (From Ref. 4-1)

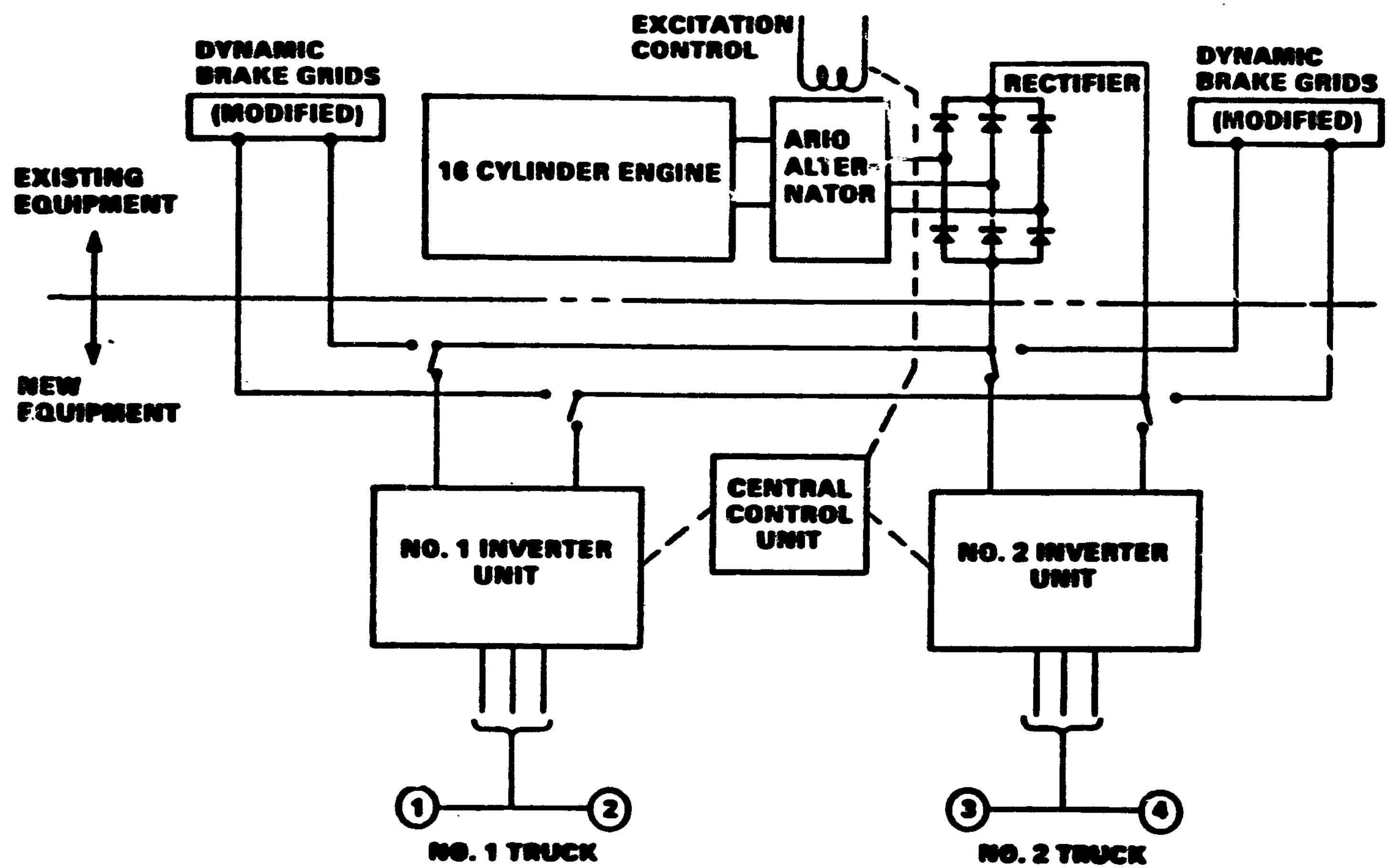

Figure 4-8. Proposed ac Motor System for Diesel-Electric Locomotive (From Ref. 4-1)

The energy consumption of the four-axle locomotive compared with the six-axle locomotive will be reduced because of the reduced dead weight of the train. Locomotive weight can account for as much as 10% of the total train weight and, therefore, the reduction in locomotive weight can represent a fuel savings of up to 4%.

The biggest gain in energy efficiency can come from the Rankine bottoming cycle. The next highest would be from higher turbocharger boost pressures and peak cylinder pressures. The ability of the present Diesel engines to use these modifications is limited by the exhaust temperature. The four-stroke engine, with its higher exhaust temperature, will benefit more than the present two-stroke engine as it now exists.

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SECTION V
NEW DIESEL ENGINES

A. INTRODUCTION

The Diesel engine has been under development for nearly a century and has become an efficient prime mover. The engine may be combined with other thermodynamic cycle engines to produce even more efficient systems. These combinations were reviewed in Section IV in the discussion of bottoming cycles. This section will describe how other multiple cycle engines can achieve efficiencies far in excess of today's engines. Improvements in BSFC (brake specific fuel consumption) may be as high as 30% in the long term, while gains of 5 to 10% could be achieved in less than 10 years.

The first discussion concerns turbochargers, which are not new and are on a majority of the production locomotives today. The discussion is presented as an introduction to the concept of turbocompounding which is important to the more advanced engines. Turbocompounding is a concept dating back several decades and is a means of recovering exhaust gas energy to supplement the engine output power. Turbocompounded spark ignition aircraft piston engines were built in the 1940s. Turbocompounded truck Diesel engines are currently under development. Turbocompounding has not yet been applied to railroad locomotives as far as is known. The discussion of turbocompounding leads to methods of increasing the exhaust temperature which increases the effectiveness of exhaust heat utilization devices and leads to improved overall overall system efficiency. Two possible means of doing this are: use of the adiabatic engine and use of the augmented engine. The adiabatic engine eliminates the cooling system and diverts that heat to the exhaust system. The augmented engine has a combustor, like that of a gas turbine, in the exhaust duct and injects fuel to raise the temperature. The last pages of this chapter discuss briefly some other Diesel engines which may have some application to locomotives but, for one reason or another, were not selected for a more detailed analysis in this project.

B. TURBOCHARGED ENGINES

Before the late 1950s, most of the railroad Diesel engines were either naturally aspirated or Roots-blown. About one-third of the fuel energy went into useful work, one-third went to the cooling system, and one-third went into the exhaust gases. The turbocharger uses some of the exhaust gas energy which would otherwise be wasted. In a turbocharger, a turbine driven by exhaust gases from the Diesel engine, in turn, drives a compressor in the intake manifold. Because the efficiency of a Diesel engine increases with pressure ratio, a high pressure ratio is desired. Fuel ignition is also enhanced by a high pressure ratio. Structural considerations put a limit on the maximum cylinder pressure that can be used. This high pressure can be achieved either entirely within the cylinder, as in a naturally aspirated engine, or partly external, as in the turbocharged engine. If it is noted that the compression of a gas is usually less efficient than the expansion, then it is desirable to keep the compression ratio to a minimum. On the other hand, the efficiency increases with pressure ratio. There is then an optimum value of the pressure ratio for a naturally aspirated Diesel engine. If part of the work of compression could be obtained at no cost, or at least at a low energy cost, the overall efficiency of the engine could be improved. The turbocharger does this by using exhaust energy to compress the intake air to about two atmospheres. The compression ratio of the reciprocating portion of the cycle is reduced with its associated losses being lowered in proportion. The net effect is an increase in power without exceeding the structural pressure limits or using more fuel. The addition of a turbocharger to a locomotive Diesel results in a reduction in BSFC of 6 to 7% in Notch 8 and 3 to 4% in Notch 1. Other improvements can add another 2 to 3 percentage points to those gains.

The performance of a turbocharged engine depends heavily on the matching of the turbocharger to the engine. The flow characteristics of the turbine, compressor, and Diesel engine are not the same over the normal operating range of the engine. All the power developed in the turbine goes either to the compressor or to accelerating the turbocharger's rotating parts. The turbine power is a function of the Diesel engine exhaust gas temperature and pressure, and the characteristics of the turbine itself. The power absorbed by the compressor is a function of its design and the third power of its speed. The turbocharged boost pressure is a function of the square of the compressor speed and is limited by the structural stresses in the Diesel engine. There is an operating line where the compressor, engine and turbine work together to produce the maximum boost pressure. By suitable selection of turbocharger components, this operating line can be made to fall anywhere in the Diesel engine's operating range. If the engine is operated at a condition below the operating line, the turbocharger will run slower with a corresponding reduction in the boost pressure. If the engine runs above the operating line, the compressor will run faster with a higher boost pressure. Because the boost pressure must be limited, one approach is to bypass or "waste-gate" part of the exhaust gas around the turbine to reduce its power and limit the boost pressure. Another approach is to set the operating line such that maximum boost pressure occurs at maximum engine output.

Locomotive manufacturers match their turbochargers to the notch 8 level, so no waste-gate is needed. The low boost pressure, which would be expected in the low notch positions, can be overcome by supercharging, i.e. driving the compressor from the crankshaft. An overriding clutch disconnects the turbocharger from the crankshaft at about the notch 7 operating condition. In this manner, a reasonable boost pressure is achieved in the low notch positions without the need for waste-gating in the higher notch positions.

C. TURBOCOMPOUNDED ENGINES

Turbocompounding is another means of using the energy in the exhaust gases of a Diesel engine. In this concept, a turbine, geared to the crankshaft of the engine through a torsional isolator, is used to extract energy from the exhaust gases of the engine. The idea is not new, nor is it limited to Diesel engines. One of the most notable commercial turbocompounded powerplants was the Curtiss Wright R3350 spark ignited aircraft piston engine, which appeared just before the jet age. In the discussion of turbocharging, it was mentioned that if the operating line is below the maximum operating condition, it is necessary to waste-gate the turbine.

It is possible to avoid waste-gating by directing the excess turbine power, via a geartrain, to the crankshaft and supplementing the engine output. In this type of installation, the turbocharger is always geared to the engine which may be advantageous in some cases and not in others. With this method, power is supplied to the crankshaft when the turbocharger is operating above match-point, and has power supplied to it by the crankshaft when operating below match-point. Another method is to use a turbine separate from the turbocharger. This turbine is geared only to the crankshaft. This method allows the system to be more flexible and permits it to be optimized for either maximum performance or maximum fuel economy.

Figure 5-1, taken from Reference 5-1, shows a base engine of 500 hp with a BSFC of 0.34. This engine is turbocharged and aftercooled with a conventional cooling system. This figure is based on a four-stroke truck or heavy industrial engine. The BSFC quoted is near the .336 specified by General Electric for their four-stroke 3000 hp engine and lower than the .351 of General Motors for their two-stroke model EB engine of the same rating. The BSFC in Figure 5-1 is based on gross horsepower, which is the basis of the General Electric and General Motors ratings. The energy balance shown on the right side of the figure is close to that of locomotive Diesel engines. Except for the power rating, the base engine is similar to current locomotive engines. Below the base engine on Figure 5-1 is the turbocompound Diesel, and on the left is a diagram showing the power turbine, the reduction gear, the torsional isolator and how it is connected to the crankshaft via the flywheel. Figure 5-2 is a schematic of the system. The gross horsepower has increased to 581 hp with 14% of it supplied by the power turbine. The BSFC is now only 0.31 lb/hp-hr, a reduction of 9%. This value of BSFC has been confirmed on the test stand.

Because of relatively low upstream manifold pressures, only 15 to 30% of the exhaust energy is available to a power turbine. As the power turbine operates at an efficiency of 60 to 80%, it can effectively use the exhaust energy available to it. One drawback of turbocompounding is that it increases the engine back pressure. Higher back pressure increases the engine pumping losses, reduces volumetric efficiency and power, and can cause reverse flow.

The conversion of exhaust energy to mechanical power is better explained in Figure 5-3. The figure represents the exhaust energy distribution on a pressure-volume diagram for a four-stroke, turbocharged engine. The solid dark line in the diagram (4-5-5'-6) represents the exhaust process. The hot gases at the end of the expansion stroke (point 4) suddenly expand into a low exhaust manifold pressure (point 5). This expansion process, from 4 to 5, is called the blowdown process. Because of the very short time interval, it is difficult to capture the kinetic energy acquired during this process. When the expanding gases stagnate, the kinetic energy is converted to heat (process 5-5'). If the process is adiabatic, no energy will be lost, but the thermodynamic availability will be decreased. The hot gases at point 5' are then expanded to point 6 in two steady-flow turbines. The first turbine drives the turbocharger compressor and the second generates power that is transmitted to the crankshaft.

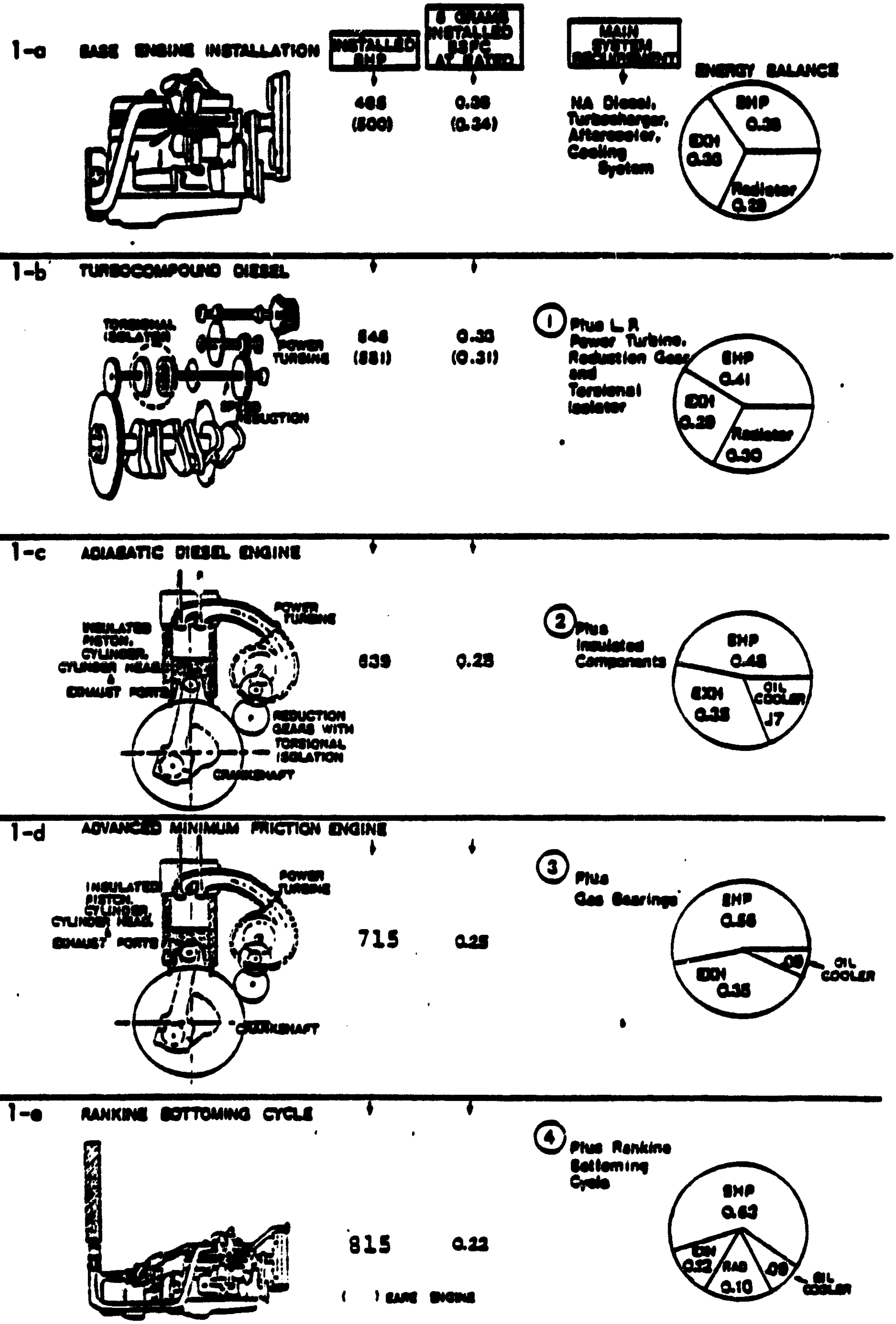

Figure 5-1. Advanced Diesel Base Power Plants
(From Ref. 5-1)

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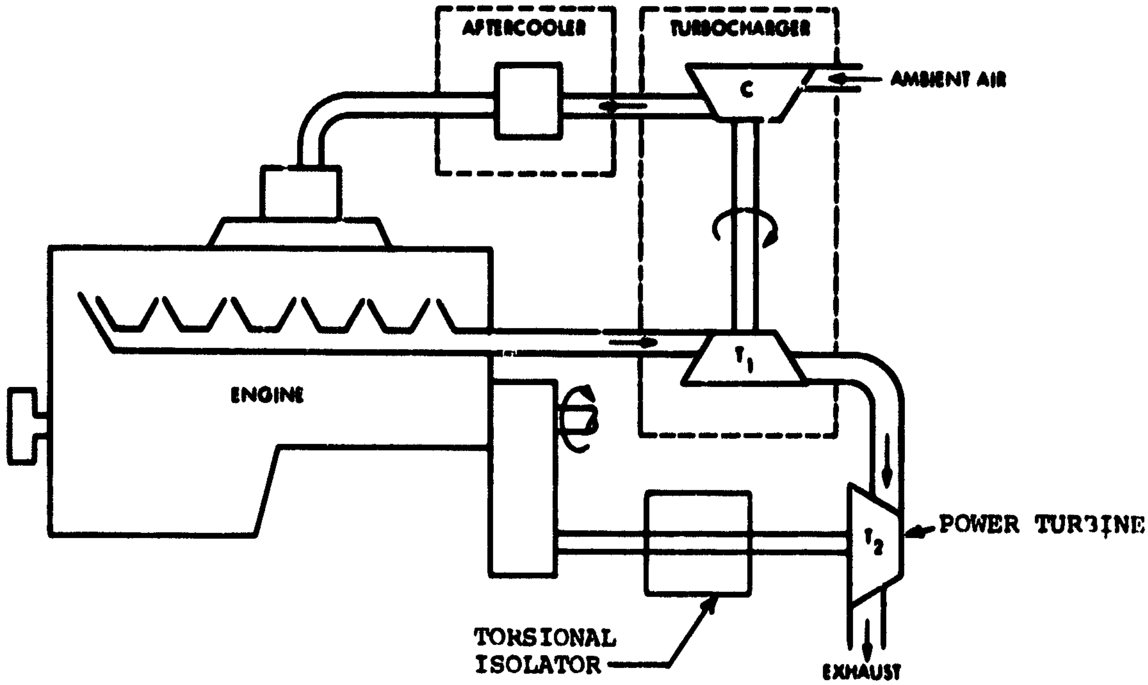

Figure 5-2. Schematic of Turbocompounding

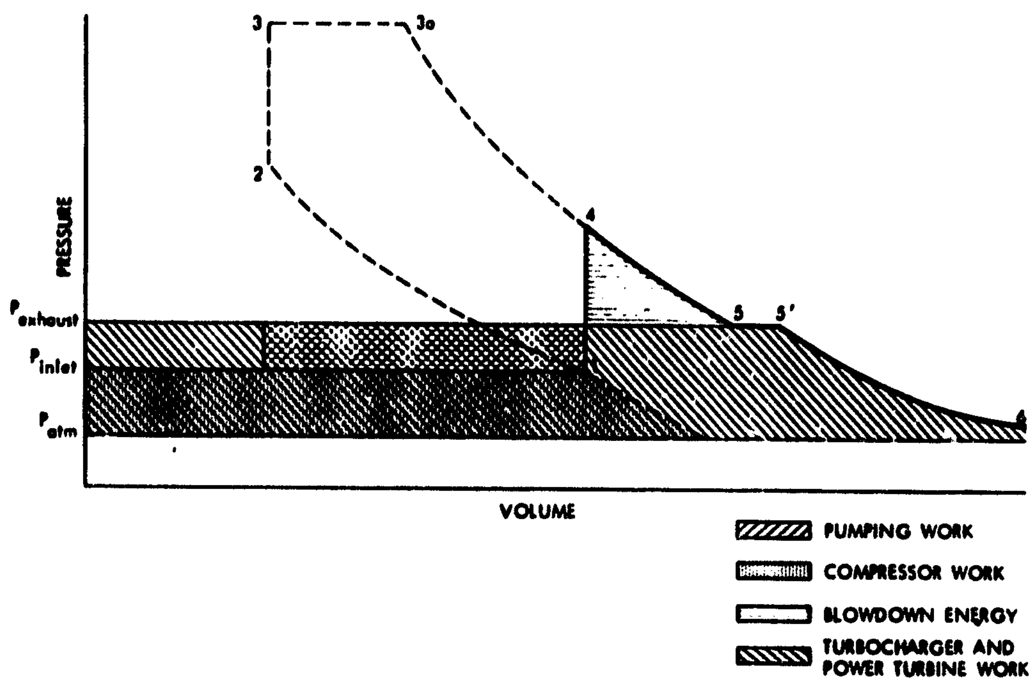

Figure 5-3. Energy Available from Ideal Exhaust Process

Figure 5-3 also identifies the engine pumping work, the energy required to drive the compressor, and the turbine output. The pumping work represents the work done by the engine during the inlet and exhaust strokes. If the exhaust pressure is higher than the inlet pressure, the pumping work is negative and detracts from the engine output.

The turbocompounding alternatives considered in this study are:

(1) Allow the engine back pressure to increase
(2) Maintain a constant pressure differential across the engine by increasing the turbocharge ratio.

In the first case, the analysis assumes that inlet pressure is held constant as the back pressure is increased. The effects of pumping work, volumetric efficiency, and power turbine efficiency are included in the analysis. The results indicate that a maximum improvement in system efficiency of 10% occurs at a pressure differential of -5 to -10 psia across the engine. In actual service, the maximum expected efficiency improvement is about 7% and the average improvement will be from 4 to 6%, depending upon the type of service.

In the second case, the turbocharger boost pressure ratio is increased to maintain constant pressure drop across the engine. There are several reasons for increasing the turbocharge ratio. Higher boost pressures can alleviate problems associated with high back pressures, improve base engine performance, and increase the amount of energy available to the power turbine.

In this analysis, it is assumed that the turbocharge pressure is increased to 3 atm without any decrease in compressor or turbine efficiency. The results in this case indicate that the maximum improvement in system efficiency is about 17%. The expected gain over a duty cycle in service is 6 to 9%. The installation of a turbocompounding system on a Diesel test engine is shown in Figure 5-4. Figure 5-5 is an engine map of the test engine and shows a best BSFC point of 0.316, about 2% higher than predicted. This value of 0.316 is about 5% better than current GE engines and 8.7% better than the GM-EMD engines. The latest information from Reference 5-3 shows that for the two engines now being tested, the minimum BSFC is 0.313 (Table 5-1). The effect of turbocompounding decreases with load, or in the case of locomotives, with notch position as shown in Figure 5-6.

Assuming that a similar gain could be made in a turbocompounded locomotive engine, what kind of fuel savings would be possible over a duty cycle? Taking a relatively conservative improvement in fuel consumption of 7% in notch 8 and decreasing it 1% or 2% for each lower notch position, the fuel consumption in each notch position can be calculated. Using the GM-EMD duty cycles, it is possible to compute the fuel usage for baseline and turbocompounded locomotives. The results of these calculations are shown in Tables 5-2 and 5-3. A savings of 5.1% overall is possible, with a 7% BSFC improvement in notch 8. The amount of fuel used is reduced by 22,600 to 23,100 gal/yr. The economic feasibility depends on the development, production, inflation, and maintenance costs of the added equipment. The system is technically feasible.

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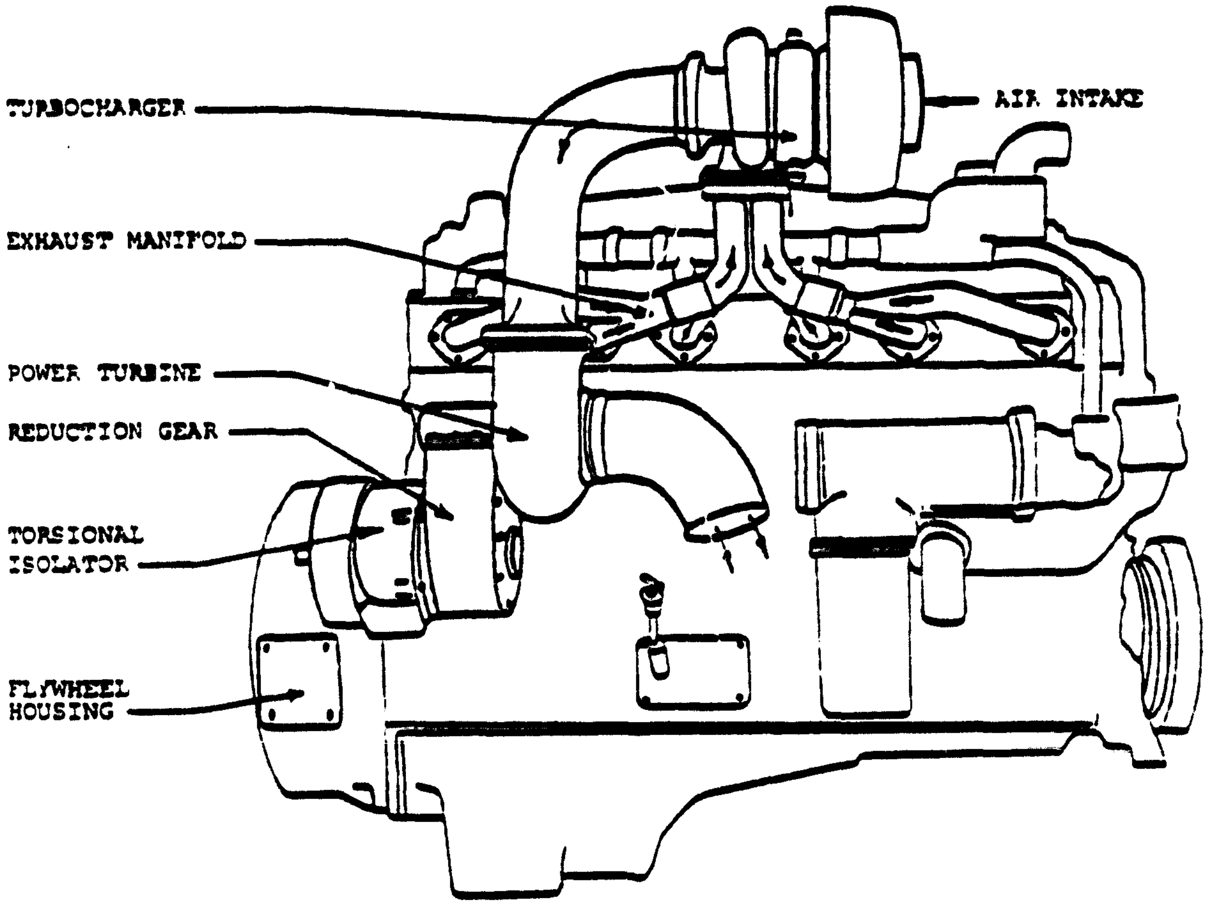

Figure 5-4. Turbocompound Engine with Low Pressure Power Turbine

D. ADIABATIC TURBOCOMPOUND DIESEL

With most of the energy saving devices centering on the use of exhaust gas heat, it is logical to try to increase the heat at the expense of the cooling system. Elimination of the cooling system with the waste heat going into the exhaust gases would be desirable. The adiabatic Diesel engine is designed to do just this. The system is shown on Figure 5-1c. Instead of a cooling jacket, the cylinder is insulated as shown in Figure 5-7 to prevent heat from flowing away from the combustion chamber. Similarly, the cylinder head, valves and piston crown are insulated. The amount of heat lost to cooling is reduced 13 percentage points to only 17% of the total input energy. The exhaust gases now contain about 42% of the total heat and an increase of 400°F is realized in the exhaust temperature. In addition, the elimination of the cooling system means that there is no

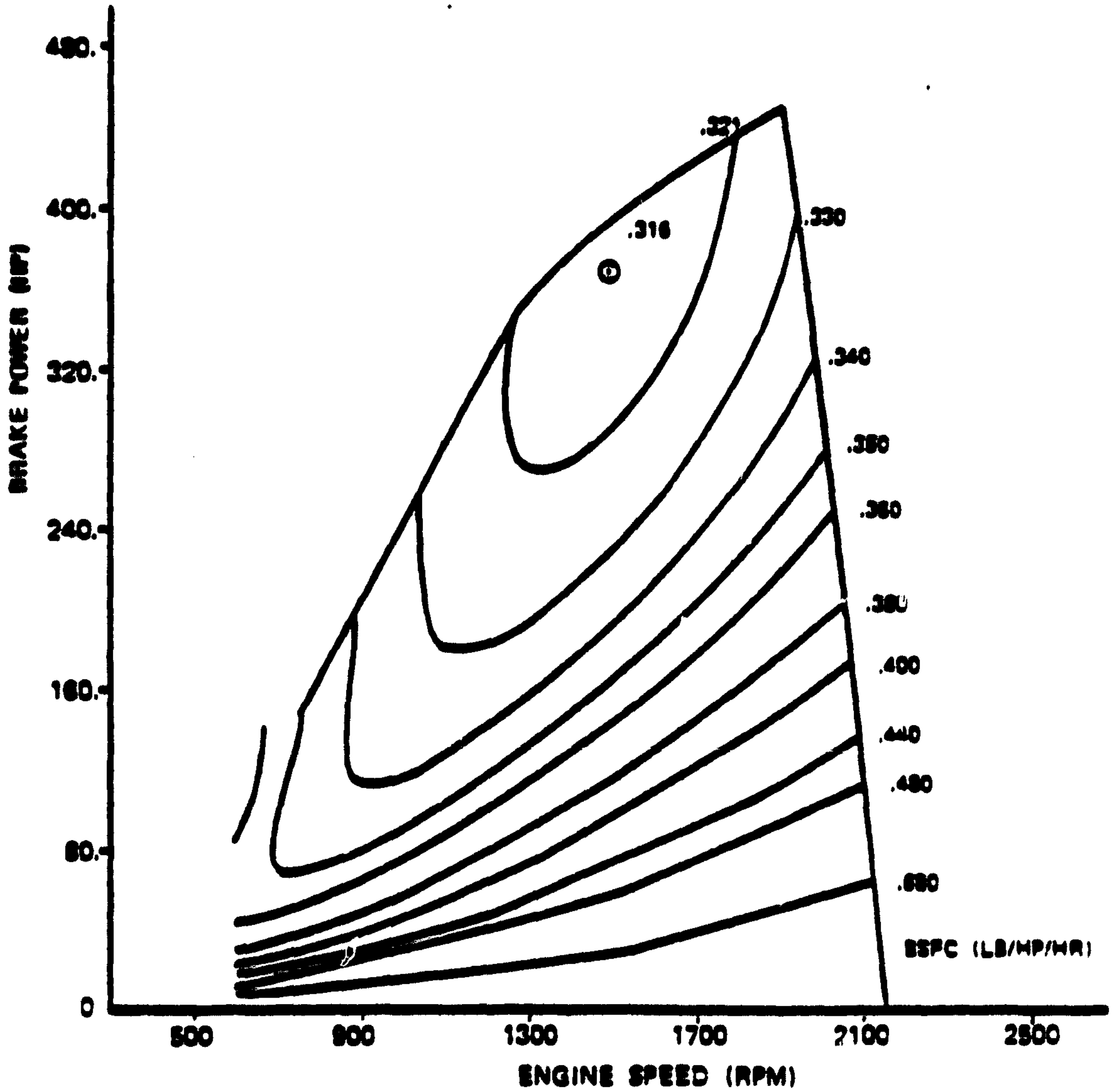

Figure 5-5. Turbocompound Test Engine Operating Speed and Load Range

water pump, no radiator, and no reason for long periods of idling to prevent freezing in winter. The adiabatic engine does not necessarily include turbocharging, turbocompounding, or other devices and it will work without them. The high exhaust energy, however, means that these systems will be more effective on this engine than on the more conventional type. Turbocharging is highly desirable and should be used on the adiabatic Diesel. The remaining exhaust heat can be recovered by turbocompounding, a bottoming cycle, or both. The Cummins engine plan as shown in Figure 5-1 uses turbocompounding in one version and turbocompounding with bottoming cycle in another. The adiabatic engine has such a high exhaust temperature

Table 5-1. Comparison of Two Turbocompounded Engines

	Engine #1	Engine #2
Rated Speed (RPM)	1900	1900
Rated Output (HP)	455	451
Rated BSFC (LB/BHP-HR)	.323	.325
Torque Peak Speed (RPM)	1300	1300
Torque Rise	15%	15%
Torque Peak BSFC (LB/BHP-HR)	.319	.319
Minimum BSFC	.313	.313
Combined Gaseous Emissions	5.778	5.939
NOx	5.482	5.507
UHC	.296	.432
Federal Smoke Cycle "A" Value	12.32	13.02
Truck	Kenworth	Kenworth
Configuration	Conventional	Conventional
Body	Van, Sq. Rib	Van, Sq. Rib
Transmission	Fuller RTO-12513	Fuller RTO-12513
Drive Axle Ratio	4.11	3.70
Geared Speed	64.0	74.4

that even after turbocompounding there is an appreciable amount of energy left which could be recovered with a bottoming cycle. This version is discussed in more detail later in this Section.

The engine power for the adiabatic engine shown in Figure 5-1 is 630 hp. The increase in power comes from the recovery of exhaust gas energy and from the elimination of the cooling system. The BSFC is 0.28, about 10% better than for the turbocompound engine and nearly 18% better than the base engine. The engine efficiency is up to 48%.

The impressive gains shown here do not come free or even easily. Without a cooling system, the hot section runs even hotter. The test engines actually run red-hot. The high temperature causes severe material problems both in the combustion chamber and in the lubrication. The temperatures are well above the coking point for natural oils and even the best synthetic oils are not completely satisfactory.

The details of the hot section are shown in Figure 5-8. The exhaust port is insulated to reduce heat loss. The hot plate is a ceramic plate across the cylinder head. The valves are insulated and the exhaust valve may be ceramic. The injector body is insulated to reduce heat flow to the fuel before it enters the chamber. The cylinder liner is ceramic as is the piston crown. Details of two experimental pistons are shown in Figure 5-9.

Ceramics are an integral part of the adiabatic engine design. No other material will stand up to the high temperatures encountered in this type of

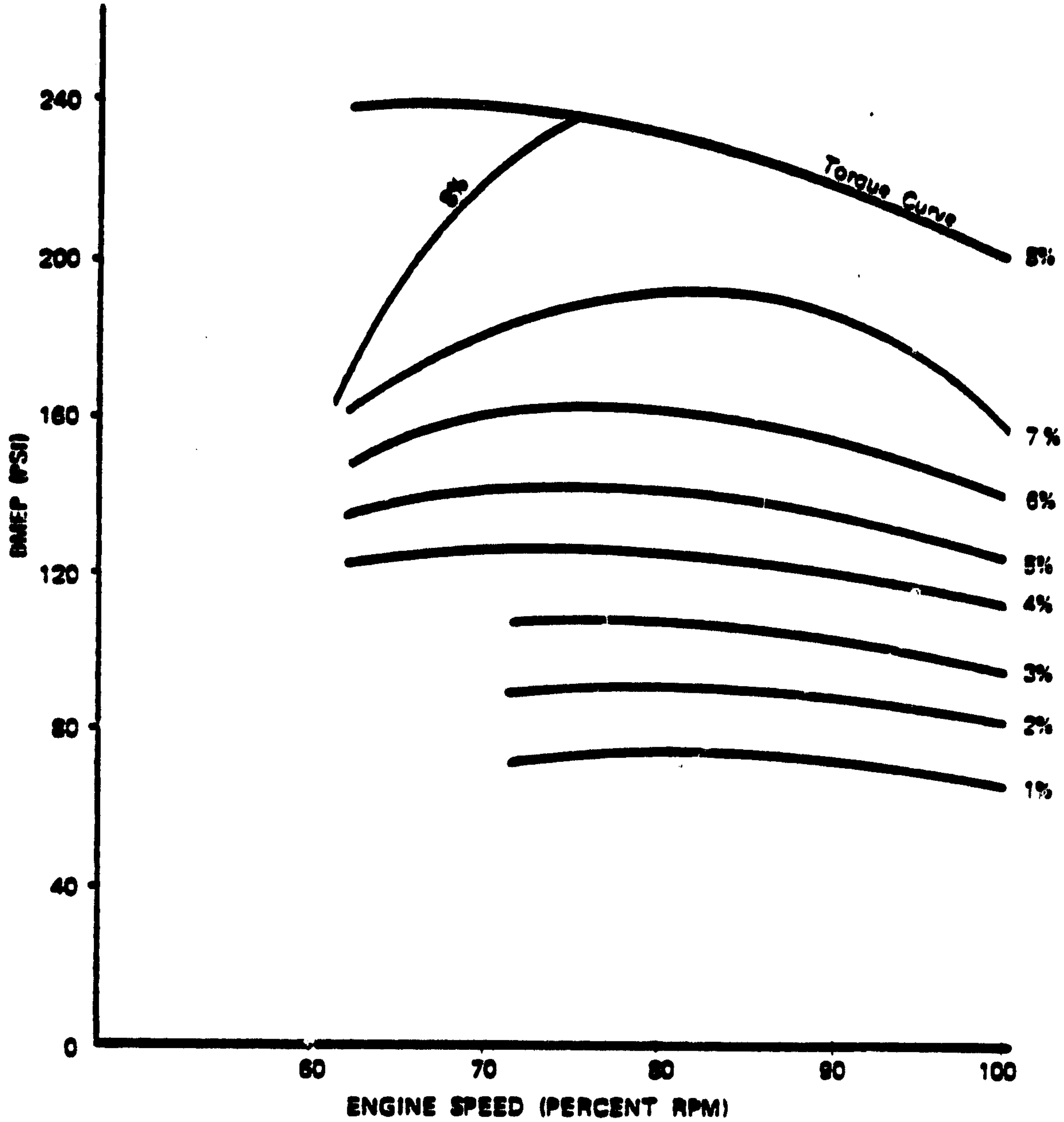

Figure 5-6. Percent Improvement in Fuel Consumption
Turbocharged vs. Turbocompound Engine

engine. However, designing reliable engine components with ceramics is considerably more difficult and unquestionably different than designing with metallic materials. The probability of failure of a metallic design is quite predictable. The failure of ceramic designs is less predictable. To avoid failures, ceramic components must be stressed low and ceramics of good quality and strength must be used. Some of the ceramics which may be used are:

(1) Hot pressed Silicon Nitride
(2) Sintered Silicon Nitride

Table 5-2. Fuel Consumption of Turbocompounded Diesel Locomotives

Throttle Position	Gross Horsepower	Base Engine BSFC (lb/hp-hr)	Base Engine Fuel Flow (lb/hr)	Turbocompounded Engine Percent Improvement	Turbocompounded Engine Fuel Flow (lb/hr)
Four-Stroke Diesel Engine					
1	125	.463	58	0	58
2	315	.406	128	1	127
3	650	.392	255	2	250
4	1055	.365	385	3	373
5	1580	.355	561	4	539
6	2070	.345	714	5	678
7	2680	.336	900	6	846
8	3230	.336	1085	7	1009
Idle			31		31
Dynamic Braking			102		101
Two-Stroke Diesel Engine					
1	106	.473	50	0	50
2	410	.418	171	0	171
3	705	.395	278	1	275
4	1057	.380	402	2	394
5	1519	.370	562	3	545
6	2056	.367	755	4	725
7	2762	.363	1002	5	952
8	3261	.365	1190	7	1107
Idle			38		38
Dynamic Braking			147		146

Table 5-3. EMD Medium Duty Fuel Consumption

Four-Stroke Diesel Engine

Throttle Position	Percent of Time in Notch	Base Engine Fuel Flow (lb/hr)	Base Engine Fuel Used (lb/hr)	Turbocompounded Engine Fuel Flow (lb/hr)	Turbocompounded Engine Fuel Used (lb/hr)
1	4	58	2.32	58	2.32
2	4	128	5.12	127	5.08
3	4	255	10.20	250	10.00
4	4	385	15.40	373	14.92
5	4	561	22.44	539	21.56
6	4	714	28.56	678	27.12
7	4	900	36.00	846	33.84
8	17	1085	184.45	1009	171.53
Idle	46	31	14.26	31	14.26
Dynamic Braking	9	102	9.18	101	9.09
	Total Fuel Used		327.93		309.72
	Fuel Used per Year		2,872,667 lb		2,713,147 lb
	Savings per Year		159,520 lb		
			22,659 gal	5.5%	

Two-Stroke Diesel Engine

Throttle Position	Percent of Time in Notch	Base Engine Fuel Flow (lb/hr)	Base Engine Fuel Used (lb/hr)	Turbocompounded Engine Fuel Flow (lb/hr)	Turbocompounded Engine Fuel Used (lb/hr)
1	4	50	2.00	50	2.00
2	4	171	6.84	171	6.84
3	4	278	11.12	275	11.00
4	4	402	16.08	394	15.76
5	4	562	22.48	545	21.80
6	4	755	30.20	725	29.00
7	4	1002	40.08	952	38.08
8	17	1190	202.30	1107	188.19
Idle	46	38	17.48	38	17.48
Dynamic Braking	9	147	13.23	146	13.14
	Total Fuel Used		361.81		343.29
	Fuel Used per Year		3,169,456 lb		3,007,220 lb
	Savings per Year		162,236 lb		
			23,038 gal	5.1%	

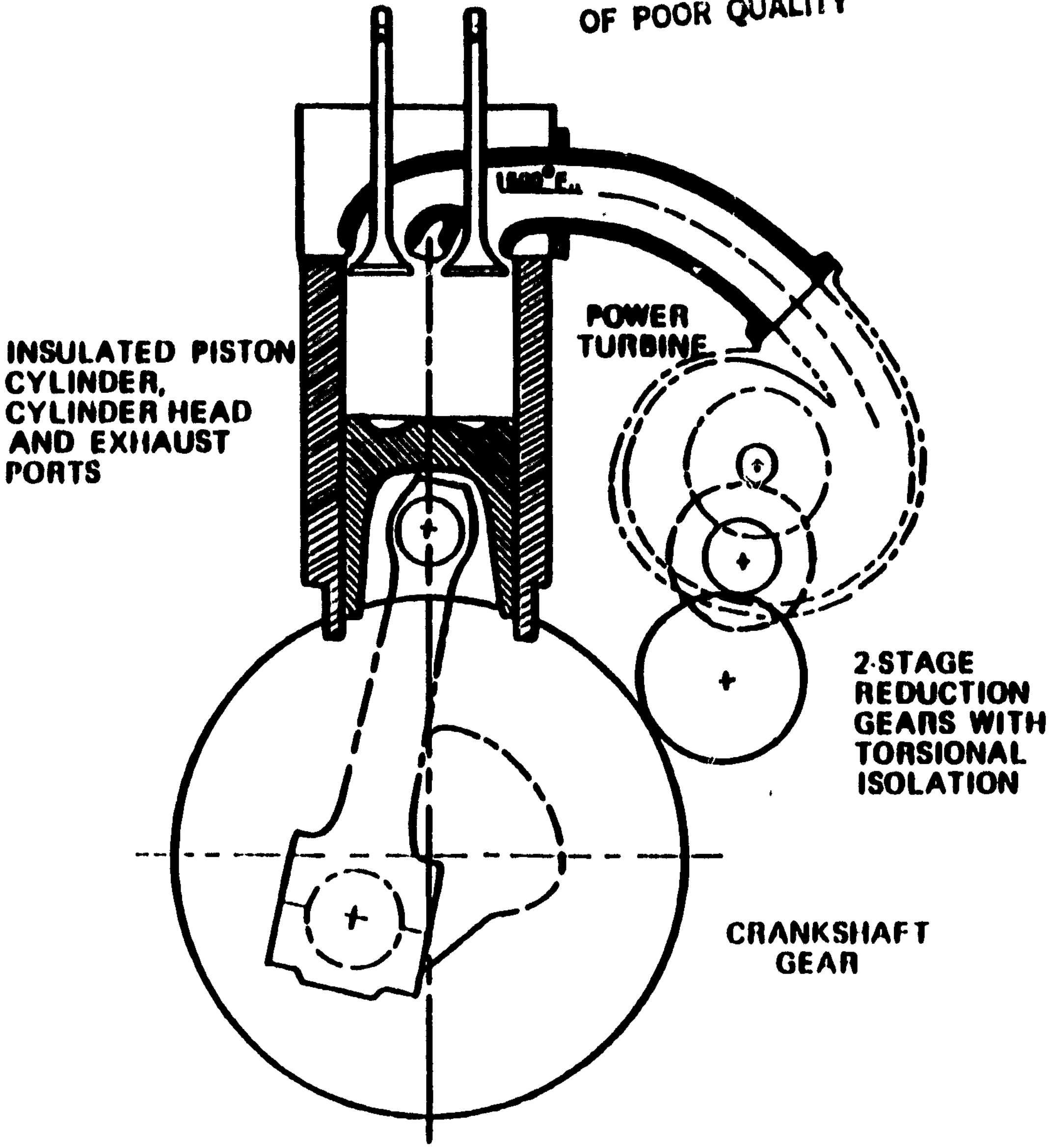

Figure 5-7. Total Energy Recovery via Cummins Adiabatic Turbocompound Engine

(3) Reaction Bonded Silicon Nitride
(4) Lithium Alumina Silicate
(5) Alumina Silicate
(6) Silicon Carbide

In an adiabatic engine, the ceramic combustion chamber material must possess:

(1) High strength at high temperatures
(2) Good insulating properties

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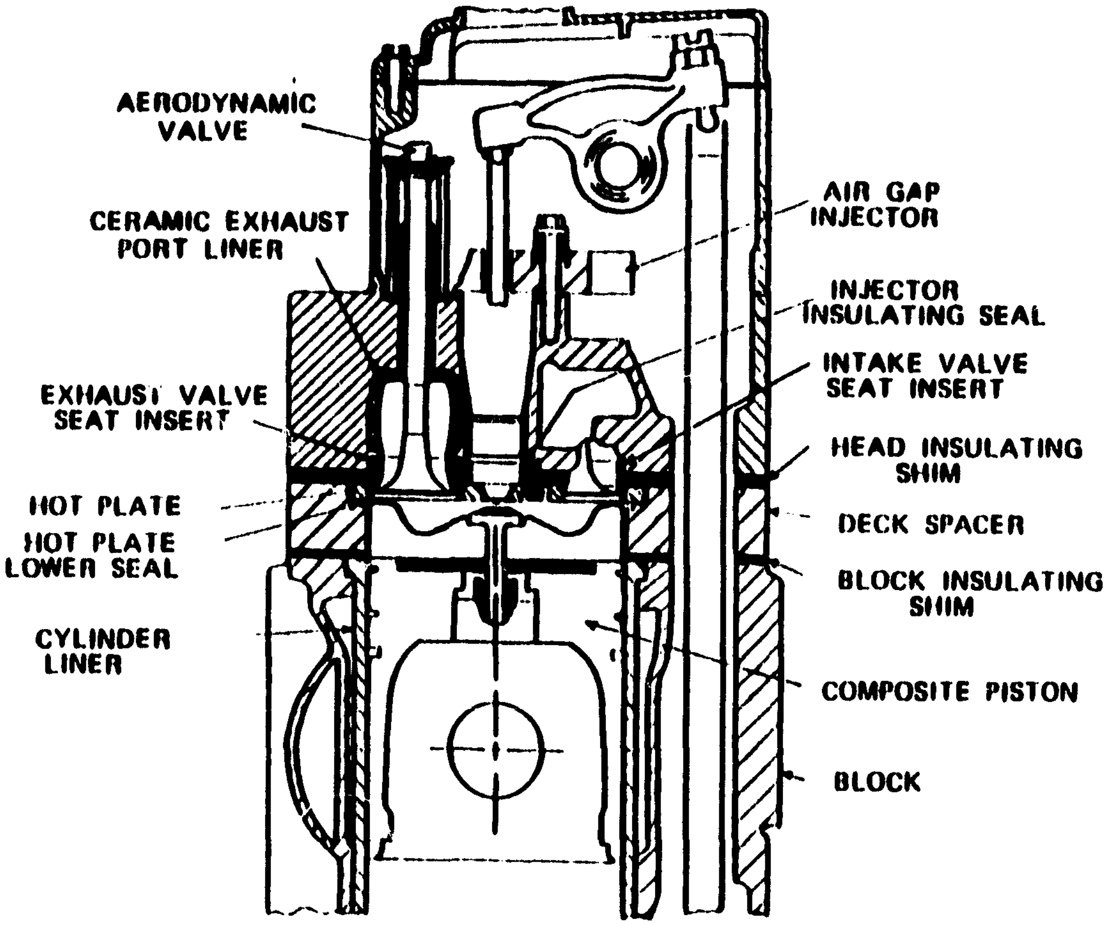

Figure 5-8. Cross Section of Cummins Basic Adiabatic (Insulated) Diesel Engine

(3) Low coefficient of expansion
(4) Low cost

None of the available materials possess all of these properties and the research in ceramics is being expanded in an attempt to develop one.

The lack of high temperature lubricants is one of the major stumbling blocks in the development of the adiabatic turbocompound Diesel engine. There are, however, ways around this problem. One way is to eliminate lubricants entirely. This approach, illustrated in Figure 5-10, would use gas

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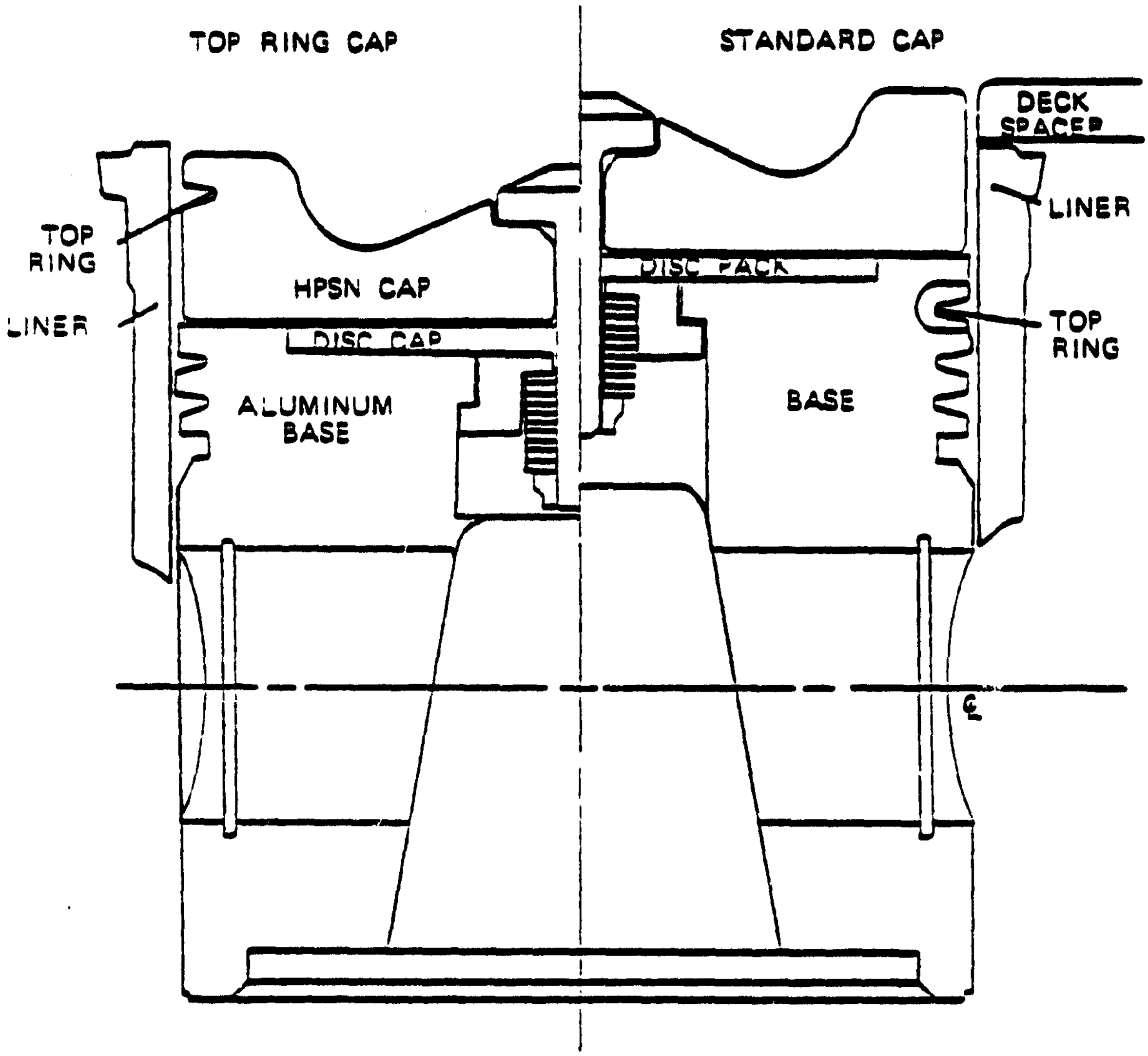

Figure 5-9. Two Types of Composite Pistons with HPSN Caps Under Investigation

bearings in the piston-cylinder liner interface, ceramic roller bearings for the wrist pins, crank pins, and main crankshaft bearing. Solid lubricants would be used for gears, valve guides, rocker arms and push rod assemblies. A minimum friction engine is illustrated in Figure 5-1. The reduction in friction would boost the thermal efficiency to 56% (Ref. 5-1). This version has the gas bearing only at the piston so there is still an oil system for the remaining bearings. This way 9% of the heat is going to the oil cooler. In a totally unlubricated engine, no oil cooler is needed. The engine power has increased from 500 hp for the base engine to 715 hp and the BSFC has dropped to 0.25 lb/hp-hr. This is a decrease in BSFC of 26% compared to the base engine.

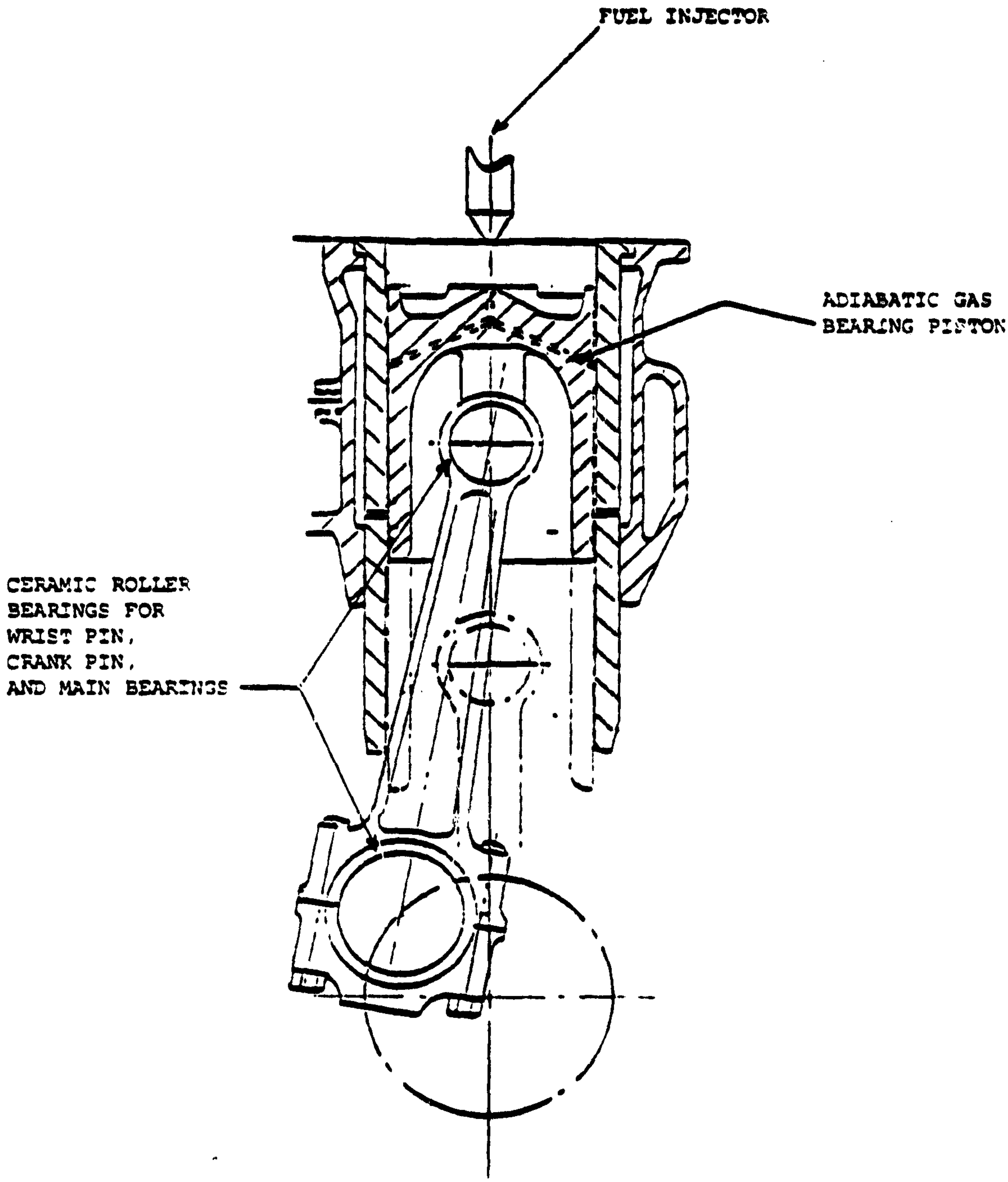

Figure 5-10. An Approach to an Unlubricated Adiabatic Engine

An even greater reduction in BSFC is possible if a bottoming cycle is added as shown in Figure 5-1e. The addition of the bottoming cycle increases output power to 815 hp while decreasing BSFC to 0.22 lb/hp-hr, a 63% increase in power and a 35% decrease in BSFC as compared to the base engine. This system is shown in Figure 5-11 and is discussed in more detail

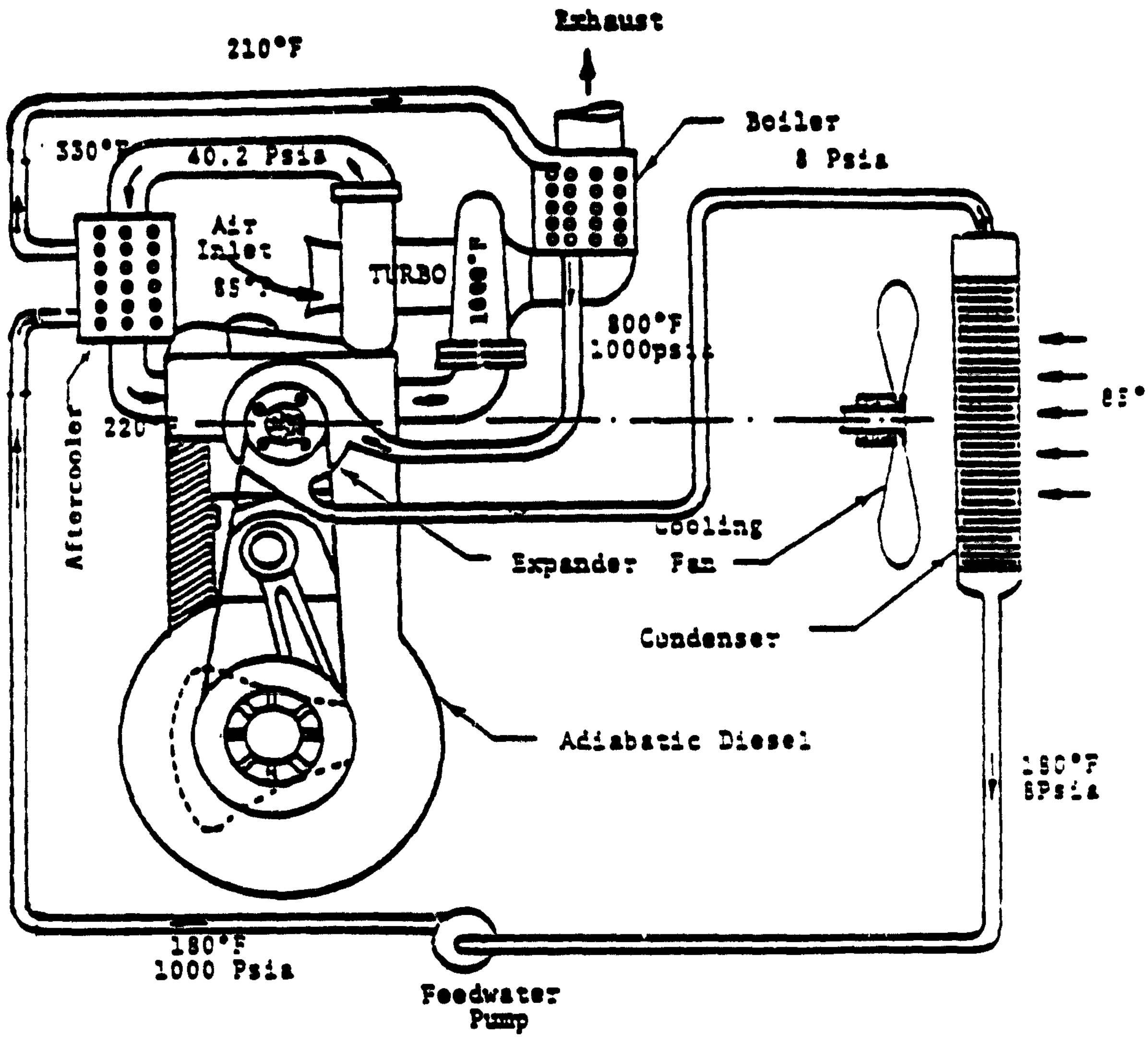

Figure 5-11. Schematic of a Possible Rankine Bottoming Cycle on a Diesel Engine Showing Thermodynamic State Points and Numbers

Although only the Rankine engine has received much interest as a bottoming cycle, theoretically any external combustion engine may be used as a bottoming engine provided that the exhaust temperatures and energy are adequate. It appears that the exhaust gas temperatures of the adiabatic Diesel are high enough to be the heat source for a Stirling engine. Because the thermal efficiency of a Stirling engine is potentially higher than that of a Rankine engine, it may produce greater fuel economy improvements. Mechanical Technology Inc. has studied the use of a free-piston Stirling engine/linear alternator for this application. Much of the complication, size and cost of a Stirling engine is in the combustion system (the fuel nozzle and associated combustion chamber are not unlike those of a gas turbine). The combustion air pre-heater arrangements are also complex and costly. Other engine components are: an atomizing air compressor, temperature controller, fuel pump, combustion air blower, spark ignition and the associated parts. By using the adiabatic Diesel's exhaust heat, these components can be eliminated, greatly simplifying the Stirling engine's design. However, a means of cooling will have to be provided for the Stirling's cooler tubes. The lower the temperature of these tubes, the higher the Stirling engine efficiency. It is, therefore, necessary that an adequate radiator be provided for this engine although the Diesel itself does not require one. The BSFC of an adiabatic turbocompound engine with minimum friction and the Stirling engine bottoming cycle is estimated at 0.20 lb/hp-hr, a gain of about 10% as compared to the Rankine bottoming cycle.

How does this engine impact the railroads? The single cylinder adiabatic engine at Cummins has been run for some time testing ceramic components and the multicylinder (six cylinder) engine is due to be tested in early 1980. Assuming that the tests are successful, then a 3000 hp adiabatic locomotive engine might be ready for testing in about 10 years, probably no less, and on the rails in limited service in about 15 years. The same engine with the bottoming cycle could be ready in about the same time frame. This engine can not be considered a near term alternative for the industry.

Lacking actual data, it is necessary to make some assumptions about the effect of the modifications on the BSFC in each notch position. Table 5-4 shows the percent improvement in fuel consumption which is assumed to be achievable in each notch. These figures are more conservative than the estimates that Cummins shows in Figure 5-1. These lower figures are the result of assessing the effects of the changes on the engine itself and on the affected accessories (radiator fan, condenser fan, pumps etc.). The turbocompounding effect is the same here as described in the earlier discussion. The percentage effect from the use of the adiabatic engine is assumed to be constant in all notches. The percent of the heat in the exhaust increases with decreasing notch position but the temperature of the exhaust decreases. In the engine, it was thought that these factors tend to compensate so that the improvement is constant. The bottoming cycle is more sensitive to temperature and the improvement declines with decreasing notch position. Friction power, on the other hand, decreases at a slower rate than engine power so that in the low notch positions, it is a significant fraction of the losses. A reduction in friction shows up as a greater relative improvement at part load than it does at full load. The bottoming cycle improvements shown here are for use with a turbocompound engine. If the cycle was used with a non-turbocompound engine, the exhaust gas temperature available to the cycle would be higher and would increase the percent improvement in all notches.

To determine the effect of these changes on the locomotive fuel consumption in operation, a typical medium-duty train was selected and analyzed for its performance. Two base engines, one General Electric and one General Motors were selected, both with 3000 hp engines. The same train

Table 5-4. Percentage Improvements in BSFC for Advanced Diesel Engines

Throttle Position	Turbo-Compounding	Adiabatic Engine	Minimum Friction	Bottoming Cycle[a]
1	0	9	28	4
2	1	9	25	5
3	2	9	22	6
4	3	9	19	7
5	4	9	16	8
6	5	9	13	9
7	6	9	10	10
8	7	9	7	9
Idle Fuel Flow (%)	0	9	28	3
Dynamic Braking	1	9	25	5

[a]For use with turbocompounding, differs from values for no turbocompounding (Section IV).

was simulated with both locomotives over the same duty cycle. The changes to the engines were progressively introduced and the effects on fuel usage compiled. Table 5-5 shows the fuel used by each base engine and each variation of the engine. There are basic differences between the two engines but the percent fuel savings is nearly identical. The reductions in fuel consumption are impressive. A 30% savings in fuel would amount to over a billion gallons a year. It should be remembered that the gallons per year savings shown here are for particular size engines on a specific duty cycle. Extrapolating this data can be very misleading. The effect of the duty cycle on the fuel savings is shown in Table 5-6. The GP40-2 locomotive is run on the GM-EMD light, medium, and heavy duty cycles and the fuel usage and savings are shown. Although the fuel consumption varies by a factor of 2 to 1, the percent savings for this particular advanced engine is nearly constant, less than a 4% variation between the medium duty and either the light or heavy duty cycles.

If the adiabatic engine can be adapted to locomotive service, the fuel savings will be impressive. The problems in developing the engine are equally impressive and challenging. The results of the current work at Cummins will have a strong influence on its possible use by the railroads.

E. AUGMENTED DIESEL ENGINE

The augmented engine shown in Figure 5-12 is in effect a hybrid between a turbocharged Diesel engine and a gas turbine. It could also be described

Table 5-5. Effect of Progressive Changes in the Diesel Engine (3000 hp) on Fuel Consumption on Medium Duty Cycle

Engine	Fuel Used (lb/hr)	Fuel Savings Compared to Base Engine (lb/hr)	(gal/yr)	(%)
Four-Stroke Diesel Engine				
Base	328.0			
Base with Turbocompounding	309.8	18.22	22,674	5.6
Adiabatic Engine with Turbocompounding	280.3	47.74	59,404	14.6
Adiabatic Engine with Turbocompounding and Minimum Friction	243.0	84.95	105,709	25.9
Adiabatic Engine with Turbocompounding, Minimum Friction and Bottoming Cycle	211.9	116.1	144,510	35.4
Two-Stroke Diesel Engine				
Base	361.8			
Base with Turbocompounding	342.0	19.8	24,653	5.5
Adiabatic Engine with Turbocompounding	309.4	52.4	65,165	14.5
Adiabatic Engine with Turbocompounding and Minimum Friction	267.86	93.95	116,904	26.0
Adiabatic Engine with Turbocompounding, Minimum Friction and Bottoming Cycle	233.6	128.2	159,574	35.4

Table 5-6. Effect of Duty Cycle on Fuel Consumption

Duty Cycle	Base Engine	Advanced[a] Engine	Fuel Savings lb/hr	gal/yr	%
Light	236.2	184.3	51.8	64,500	21.9
Medium	361.8	279.3	82.5	102,700	22.8
Heavy	481.0	367.2	113.8	141,700	23.7

Base engine is GM-EMD GP40-2
[a]Advanced Engine is adiabatic Diesel with turbocompounding and bottoming cycle, conventional lubrication.

as a Turbocompound Diesel with a combustor in the exhaust duct upstream of the power turbine. This description is really an oversimplification. The system is quite complex and its performance depends heavily on the way the two engines are matched. The Diesel engine can be operated without adding fuel to the combustor, in which case, it is simply a turbocompound engine. Fuel could be added to the combustor at all engine speeds boosting the output power across the speed range. On the other hand, it could be added only at the top end as a boost. In this case, the engine is operated as a turbocompound engine over most of its range with the fuel being added after maximum engine power is reached. This extra power extends the maximum power rating of the engine.

The first method of adding fuel would result in a relatively small Diesel engine with a high output rating but would not increase its thermal efficiency. In fact, at the lower power and speed conditions, it would probably be worse. For use on locomotives, the second method, top end power boosting, would be preferable. The engine would operate as a turbocompound Diesel up through notch 8. In notch 8, a second control system would be used. This system would control the fuel flow to the combustor and could have three positions; off, half, and full. With the Diesel engine in notch 8, the output of the engine would increase from 3225 hp to 3425 hp in the half flow notch and 3625 hp in the full flow notch. In effect, the engine operates on a 10 notch control rather than the conventional 8 notch control. The fuel flow increases to 1314 lb/hr in the full flow notch.

Obviously, the augmented engine using the combustor is not even as good as the present engine system. It would only be usable if an additional 200 to 400 hp per locomotive were needed and if the only alternative was to add another locomotive. The amount of fuel saved would be marginal. It must be concluded, from an energy standpoint, that the augmented Diesel engine is of very limited value unless equipped with a Rankine or Stirling bottoming cycle. The analysis of this system has not been made.

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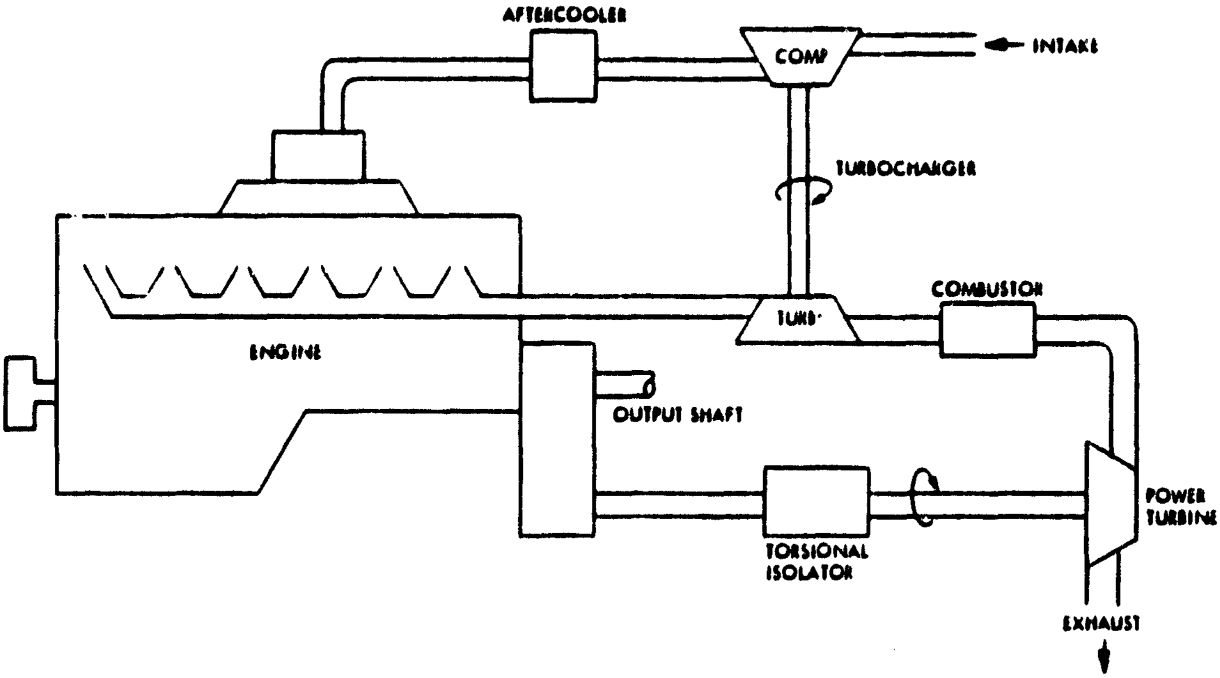

Figure 5-12. Schematic of the Augmented Engine

F. HYPERBAR DIESEL ENGINE

The use of a turbocharger can be advantageous to a Diesel engine from both a performance and a fuel economy standpoint. However, the turbocharger boost pressure is currently limited to 3 to 4 atm by the amount of usable energy in the engine exhaust and by the peak combustion pressures in the cylinders. The peak pressures can be reduced by aftercooling of the air between the turbocharger and the intake manifold of the engine. This approach is limited by the bulk of the coolers and, at low temperatures, by the problem of its icing-up. Another approach is to increase the boost pressures while reducing the compression ratio at the same time so as not to exceed the combustion pressure limit. Unfortunately, this approach may aggravate cold weather starting problems and may even reduce the energy in the exhaust to a point where the higher boost pressures could not be attained. Some of these problems could be solved by using variable compression ratio engines but these engines present mechanical and combustion problems of their own.

Another possible solution to these problems is the use of the Hyperbar system. In this method, an auxiliary combustion system is installed downstream of the exhaust manifold. Additional fuel is burned to raise the exhaust temperature and the increased exhaust energy is used to drive a special turbocharger which can boost the intake pressure up to as high as 10 atm. A simplified schematic of the system is shown in Figure 5-13.

Although not shown in the figure, a multi-stage turbine and compressor will be required. To keep the maximum cylinder pressures down to the current design limit of 2000 psia, the compression ratio must be reduced and this adversely affects the thermal efficiency. This reduction in the Diesel engine compression ratio just balances out the gain due to the more efficient use of the exhaust energy so that the thermal efficiency and, hence, the fuel consumption is the same for a conventional engine and a Hyperbar engine. The advantage of the Hyperbar system is the reduction in engine size required to produce a given level of power. The weight of a Hyperbar engine is estimated at 35% to 43% less than a conventional engine with the same rated output. If the peak combustion pressure limit is raised, the engine efficiency would also increase.

In summary, the hyperbar engine system, is not more efficient than a conventional Diesel and may be less efficient depending on the design details. A promising variation is the addition of a Rankine bottoming cycle but the analysis of this system has not been undertaken. Further details on the Hyperbar engine along with test results are available in Reference 5-12.

G. OTHER DIESEL ENGINES

There are a number of other variations of the basic Diesel engine. A few years ago, Rolls-Royce announced a two rotor-in-series rotary Diesel engine based on the Wankel engine. The Fairbanks-Morse opposed piston Diesel is another variation which has been in commercial production. The

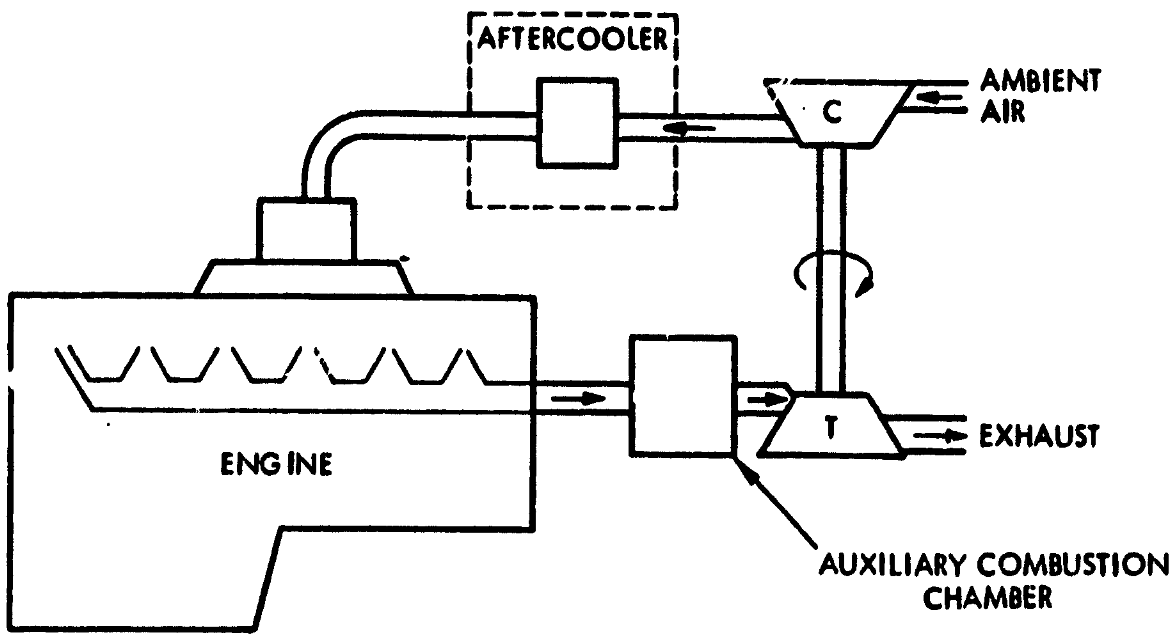

Figure 5-13. Schematic of the Hyperbar Engine

Rolls-Royce K60 is another engine of this type. Radial Diesel engines such as the Nordberg 12 cylinder engine have also been built. The patent offices and the textbooks are full of engine variations. However, few show any real potential for fuel savings. Of the few that do, the mechanical complexity or additional losses, incurred tend to negate the potential savings.

As an example, the concept of the regenerative Diesel engine shows a possible savings in fuel of 8%. In regeneration, part of the engine exhaust heat is transferred to the compressed air charge prior to the injection of the fuel. The mechanical implementation of this concept is not trivial. Heat must be transferred into the regenerator from the hot gases after expansion and at constant volume. The heat must then be transferred to the intake air after compression but before fuel is injected. This process must also be at constant volume. By coupling two cylinders together, a number of variations are possible as shown in Figure 5-14. The simplest version is the unidirectional flow engine where one piston is used for compression with the other being used for combustion and expansion. The heat exchanger is a fixed boundary counterflow or crossflow unit. The air flow is always in one direction. It is possible to use both cylinders for compression, combustion and expansion alternately in the two systems shown in the middle and bottom of Figure 5-14. The middle figure uses the fixed boundary heat exchanger. The upper half of the unit is the low pressure, hot gas side. The lower half is the high pressure, cold gas side. Heat is transferred across the boundary between them. Four three-way valves are needed to direct the gases through the system. The bottom figure uses two regenerators. The hot and cold gases pass through the units alternately, but always in the same direction. The matrix is heated as the hot exhaust gases pass through it and deliver the heat to the cold gas as it goes through. Again four three-way valves are needed but the locations are different than those in the middle figure.

The mechanical problems of all of these approaches plus the additional heat and pressure losses incurred by moving the air around in the engine negates the small gain in thermal efficiency which is theoretically possible. Cummins Engine Company developed a regenerative engine for Otto cycle operation. Their effort included actual hardware as well as a theoretical analysis. No actual Diesel engine applications are known.

H. "MATURE" DIESELS

The term "mature" refers to a system which utilizes as many of the energy saving features as possible together with the "best" alternative fuel. These "mature" locomotives are used to see what the limits are in saving fuel. One "mature" system utilizes the energy savings that can be made with a water cooled conventional Diesel engine. Another one will use an adiabatic Diesel engine with all of the energy saving features possible. These two systems represent the limits of the near term and the long term Diesel-electric locomotives.

I. NEAR-TERM SYSTEM

The features incorporated in the near-term mature Diesel-electric locomotive are higher allowable peak combustion pressures (over 2000 psia), three atmospheres of turbocharger boost, improved accessories, Rankine

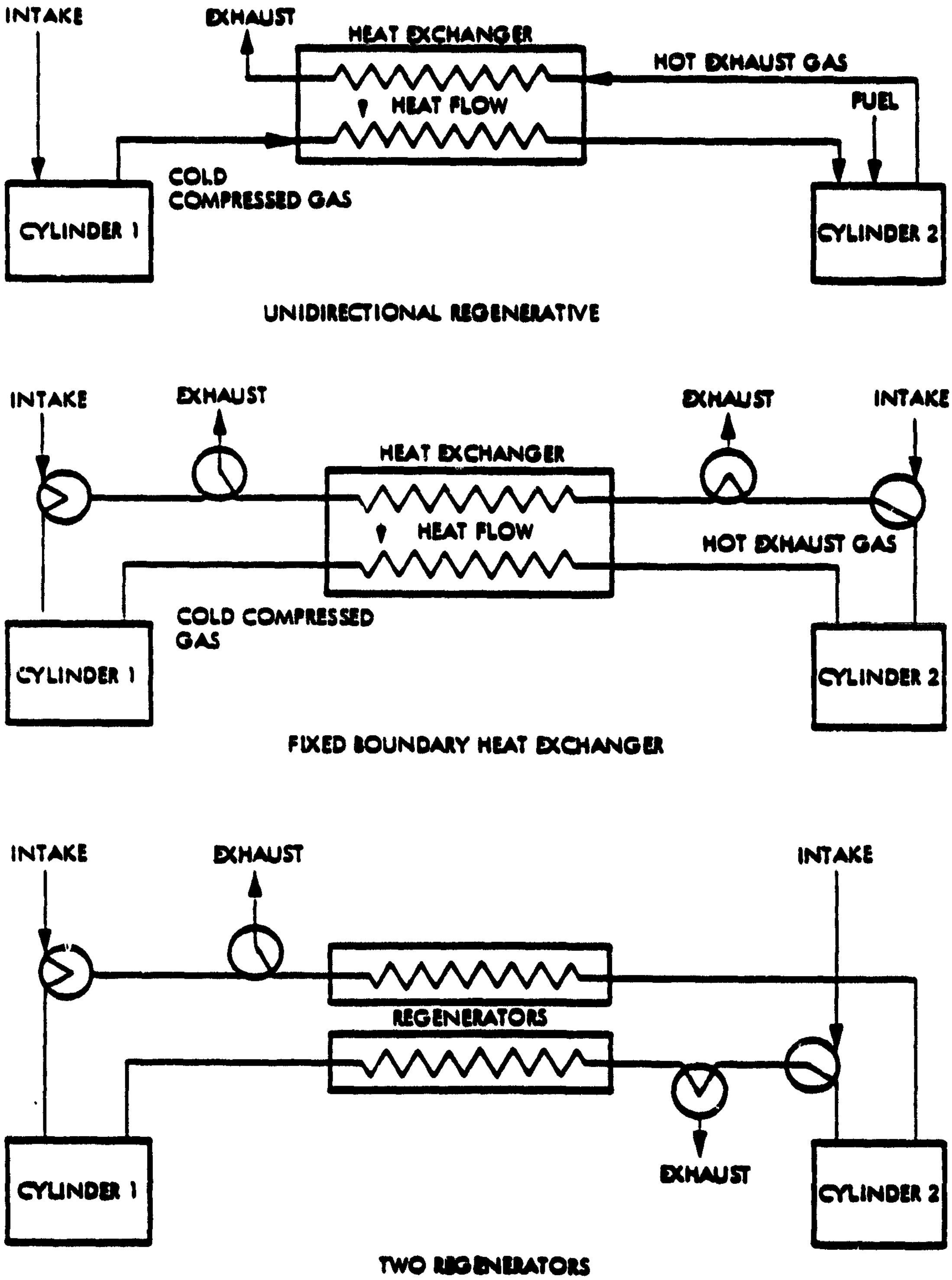

Figure 5-14. Regenerated Diesel Engine Schematics

bottoming cycle, dynamic braking modification to eliminate the need for engine power and wayside energy storage with a dual mode locomotive. The fuel used by this system is oil shale derived distillate supplemented by electricity on hills. While it is unlikely that all of these features would ever be on a real locomotive, this "kitchen sink" approach defines the theoretical limit for conventional locomotive systems. This approach starts with a 3000 hp locomotive and adds all of these features. The output power, after this is done, will be well above the 3000 hp base so that the whole engine system must be resized back to the original output.

What are the efficiency or BSFC gains that can be realized? Table 5-7, lists each of the features and the expected efficiency and power gains that result. The overall gain in power is 21%. The 3000 hp base engine will now produce 3630 hp or 227 hp per cylinder. Hence, a 12 cylinder engine would be able to do what it now takes 16 cylinders to do. The improvement in thermal efficiency is 21% plus the savings over the duty cycle from the dynamic braking modifications and wayside energy storage. These savings are estimated to be an additional 5%. The effects of these changes at each notch position are shown in Table 5-8.

The amount of fuel saved on the EMD medium duty cycle is 85 lb/hr per locomotive or 22.6% of the fuel. Over the course of a year, it could amount to 105,800 gal per locomotive. Assuming a 15-year life for a locomotive, this amounts to a 1.5 million gal savings.

J. LONG-TERM SYSTEM

The long-term, "mature", Diesel-electric locomotive has all of the features that the near-term version has with two changes and two additional features incorporated. The two changes are the use of the adiabatic Diesel in place of the conventional cooled engine and the use of a Stirling bottoming cycle instead of the Rankine. The two additional features are the minimum friction option and turbocompounding before the bottoming cycle. Three methods of utilizing the exhaust heat are used; turbocharging, turbocompounding and the bottoming cycle. The result is a very efficient and very complex engine system. The expected efficiency and power gains for this "mature" locomotive are listed in Table 5-9. These gains are considerable and show that the Diesel engine system is still capable of improvement. The increase in power is over 2000 hp. The original 16 cylinder engine will now produce 5250 hp. It is necessary to reduce the number of cylinders to 8 or 10 in order to get back to 3000 hp. The gain in efficiency is 61% and the decrease in BSFC is 37%.

Table 5-10 shows the changes by notch number. The differences between Tables 5-8 and 5-10 are dramatic. A 200 lb of fuel per hour decrease in notch 8 and a 10 lb per hour decrease at idle are shown. The fuel flow rates given in Table 5-10 are only about 60% of those of the base engine shown in Table 3-7. In notch 8, the difference between these two sets of data is about 430 lb of fuel per hour. Using the EMD medium duty cycle, the hourly flow rate for the "mature" adiabatic Diesel-electric locomotive is 219 lb/hr as compared to 374 lb/hr for the base engine. The savings is 155 lb/hr or 22 gal. Extrapolating to an entire year, the savings is 192,870 gal per year per locomotive. If it is possible to build the adiabatic Diesel in the size needed by the locomotives and to equip it with

Table 5-7. "Mature" Near-Term Diesel-Electric Locomotive Gains

Feature	Power Gain (%)	Efficiency Gain (%)
3 atm Boost Pressure	4	4
Higher Allowable Peak Pressures	3	3
Improved Accessories	1	1
Rankine Bottoming Cycle	12	12
Dynamic Braking Modification	0	2.3[a]
Wayside Electrical Energy Usage	N/A	2.5[a]
Total Gain	21	24

Note: This table applies to notch 8 only
N/A - Not applicable
[a]Applies only to overall fuel usage, not to notch 8 specifically

Table 5-8. Effect of "Mature" Near-Term Diesel-Electric Locomotive Gains on Fuel Consumption by Notch

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	122	22	100	.426	52
2	460	366	26	340	.367	134
3	550	720	35	685	.342	246
4	630	1113	53	1060	.316	352
5	720	1639	79	1560	.300	492
6	800	2180	110	2070	.292	636
7	890	2879	154	2725	.284	818
8	925	3198	198	3000	.280	895
Idle	380					32
Dynamic Braking						32

Note: BSFC is based on gross power
Fuel is oil shale distillate with LHV of 18,000 Btu/lb

Table 5-9. "Mature" Adiabatic Diesel-Electric Locomotive Gains

Feature	Power Gain (%)	Efficiency Gain (%)
3 atm Boost Pressure	4	4
Higher Allowable Peak Pressures	3	3
Improved Accessories	1	1
Stirling Bottoming Cycle	13	14
Turbocompounding	16	8
Minimum Friction	12	8
Adiabatic Engine	10	11
Dynamic Braking Modification	0	2.3[a]
Wayside Electrical Energy Usage	N/A	2.5[a]
Total Gain	75	61

Note: This table applies to notch 8 only
[a]Applies only to overall fuel usage, not to notch 8 specifically
N/A - Not applicable

Table 5-10. Effect of "Mature" Adiabatic Diesel-Electric Locomotive Gains on Fuel Consumption by Notch

Notch Position	Engine Speed (rpm)	Gross Power (hp)	Accessory Power (hp)	Net Power (hp)	BSFC (lb/hp-hr)	Fuel Flow (lb/hr)
1	380	119	19	100	.290	35
2	460	363	23	340	.255	93
3	550	715	30	685	.239	171
4	630	1106	46	1060	.223	247
5	720	1627	67	1560	.221	360
6	800	2163	93	2070	.221	478
7	890	2855	130	2725	.217	620
8	925	3166	166	3000	.220	697
Idle	380					22
Dynamic Braking						22

Note: BSFC is based on gross power
Fuel is oil shale distillate with
LHV of 18,100 BTU/lb

most of the features, the savings in fuel for the entire locomotive fleet would amount to about 1.7 billion gal annually. Even at today's price for Diesel fuel the savings amount to over $1.1 billion dollars. The fuel that is expected to run these engines will be an oil shale derived distillate or a blend of oil shale and petroleum distillates. These fuels have the best combination of physical and combustion properties, lowest costs, and good availability.

When could all this take place? The best guess for the short-term "mature" Diesel operating on a petroleum based-oil shale based blend fuel is five to ten years. The first locomotives using this near-term system could be on the tracks in about five years and full production in ten years. The long-term "mature" locomotive would probably enter limited service on an experimental basis in about fifteen years, and could be in production some five years later, in the year 2000.

K. SECTION V REFERENCES AND NOTES

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SECTION VI
ALTERNATIVE FUELS FOR DIESEL ENGINES

A. INTRODUCTION

Part of this study has been concerned with saving petroleum by making the locomotive more efficient. A complementary approach to locomotive efficiency is the use of fuels other than the Diesel No. 2 currently used by the railroads in their locomotives. Diesel No. 2 is a kerosene-like petroleum distillate that comes from the same part of the crude oil barrel as home heating oil, aviation jet fuel, and the Diesel fuel used in highway vehicles. The competition for this fuel in recent years has caused its price to escalate so that it is now nearly as expensive as gasoline. More important, its availability has become a serious problem. For these reasons, it is necessary to consider the use of alternative fuels in locomotives. These alternative fuels may be petroleum products, other than Diesel No. 2, and non-petroleum fuels such as the alcohols, coal and liquids derived from coal, oil shale liquids, vegetable oils, and inorganic fuels such as hydrogen. In addition, there are the blends slurries and emulsions that can be made from these fuels. There are approximately twenty to thirty candidate fuels and an even larger number of mixtures and blends.

The use of alternative fuels by the railroads in Diesel engines is not new. Several railroads, including Southern Pacific and Union Pacific, have used residual oils on certain portions of their lines at various times. Residual oils are petroleum based, however, and their prices have also risen sharply in recent years.

Natural gas and sewage gas have been used for years in stationary Diesel engines as have the liquified petroleums (propane and butane). The technology is already developed for these fuels to a point where they might be used by the railroads. The main problems are their availability, safety considerations, and the storage of the fuel in either gaseous or liquid form on the locomotive.

A list of candidate fuels for use in either pure form or as part of a fuel blend is presented in Table 6-1. The fuels are divided into groups based on their source except for hydrogen and ammonia which are classed just as inorganic fuels. The list of non-petroleum fuels is short and some of them such as the powdered coal and the vegetable oils have proven to be poor fuels.

In addition to the pure fuels, the emulsions, solutions, slurries and blends listed in Table 6-2 might be used. Because there are a wide variety of fuels which can be mixed, two or more at a time, and because the proportions can be varied widely, the list of such fuels becomes almost limitless. References 6-1, 6-2, and 6-3 list a large number of mixes which have been tested in Diesel engines and discuss some of their special problems.

B. FUEL COMPARISON METHODOLOGY

The use of an alternative fuel in a locomotive Diesel affects far more than just the combustion chamber of the engine. Of comparable

Table 6-1. Candidate Fuels for Diesel Engines

Petroleum	Coal	Alcohols
Diesel No. 2	Powdered Coal	Methanol
Gasoline	Coal Derived Gasoline	Ethanol
Light Distillate	Coal Derived Distillate	
Naphtha	Solvent Refined Coal	**Vegetable Oils**
Lube Stock		
Broadcut Fuel Oil	**Oil Shale**	Cottonseed Oil
Jet A		Corn Oil
Kerosene	Oil Shale Distillate	Peanut Oil
Methane	Oil Shale Gasoline	Olive
Butane		
Propane	**Inorganic**	**Tar Sands**
No. 4 Fuel Oil		
No. 5 Fuel Oil	Hydrogen	Syn-crude
No. 6 Fuel Oil	Ammonia	

Table 6-2. Slurry, Emulsion and Blend Fuels

Slurry Fuels	Petroleum Blend Fuels
Coal in Diesel Fuel	Gasoline in Lube Oil
Carbon Black in Diesel Fuel	Heavy Aromatic Naphtha in Diesel Fuel
Petroleum Coke in Diesel Fuel	Gasoline in Diesel Fuel
Flour in Diesel Fuel	
Cornstarch in Diesel Fuel	
Wood Fibers in Diesel Fuel	
Coal in Water	

Emulsion Fuels
Water in Diesel Fuel
Methanol in Diesel Fuel
Ethanol in Diesel Fuel

importance to the cetane number or ignitability of a fuel are such wide ranging properties as viscosity, specific gravity, cost, toxicity and stability. The storage of a fuel on a locomotive or at a refueling terminal must be considered and evaluated in comparison to all other candidate fuels. In order to evaluate the wide range of fuels with grossly different properties that might be used by the railroads, a method of comparison had to be devised.

The method that was implemented is based on the concept of functional properties rather than the physical and chemical properties normally used in evaluating fuels. Functional properties are defined as the properties of the fuel that are related to some function of the engine, of the locomotive, or of the railroad system as a whole. For example, cetane number is a measure of a fuel's ignitability. This is a functional property. The thermochemical property that is the counterpart to the cetane number is auto ignition temperature. Usually, the lower the auto ignition temperature, the higher the cetane rating. Some of the functional properties may depend on two or more physical or chemical properties. They may even depend on completely different physical phenomena. Material compatibility would be one such case where corrosion, elastomeric compatibility, hydrogen embrittlement and low temperature strength of materials are included in a single functional property.

The functional properties are not used on an absolute basis in this fuel ranking method but a relative basis has been selected. Since locomotives now use Diesel No. 2 as their main fuel, it is important to know how the alternative fuels compare to Diesel No. 2 functionally. A scale of -3 to +3 has been used to measure the relative value of a candidate fuel as compared to Diesel No. 2. A value of zero means that the two fuels are functionally similar. A rating of +3 means the candidate fuel is very much better than Diesel No. 2 for a specific functional property. On the other hand, a -3 rating means it is very much worse. The intermediate numbers between -3, 0 and +3 provide for fuels that are only slightly better or worse, or much better or worse than Diesel No. 2.

The functional properties are not all equally important. Some properties such as future availability and cetane no. are more important in this application than is the water reactivity of the fuel. A weighing factor of 1 through 8 is used to rate the relative importance of the functional properties. A similar method is used in Ref. 6-4 to rate various fuels. Since the purpose of the method is to get a relative ranking of the fuels, the numerical values of the functional properties rating is not critical because they apply to all fuels equally. If a quantitative rating of each fuel was needed, a more elaborate functional property rating system would be needed. It would be necessary to establish just how much more important the future cost of the fuel is than its cetane no. or its material compatibility.

For each combination of a fuel and a functional property, the relative value is multiplied by the weighing factor to get an individual score. The individual scores of a fuel are added up to get the final score. The ranking of all the fuels is based upon these scores. The numerical values of the final scores have no significance other than relative ranking.

C. PHYSICAL AND CHEMICAL PROPERTIES OF ALTERNATIVE FUELS

The physical and chemical properties of a fuel are important to its engine performance, its acceptability to the user, its impact on the environment, and its handling and distribution. A good alternative fuel should work well with the engine subsystems, like the fuel system, the lubrication system, and the combustion and exhaust related systems. The fuel should allow easy engine starting at ambient temperatures, provide rapid warm-up, and trouble-free operation, have good thermal efficiency, and keep engine and equipment maintenance to a minimum.

The fuel must also be safe and easily handled. Both the fuel and the exhaust must be free from objectionable odors. Environmental considerations involve not only the quantity of exhaust emissions but also fuel emissions encountered in handling, storage, and refueling. Storage and handling considerations also require that large quantities of fuel can be contained and moved without serious deterioration or loss for periods up to 6 months or longer. Again this must be done safely, without serious environmental damage and without significant contamination with rust, water, or dust. Thus, it can be seen that the physical and chemical properties of the fuel have a significant impact in many important areas.

The properties of the petroleum based fuels are presented in Table 6-3. The number of properties listed is small. Only those properties which appear to be of special significance are included. The properties of the alcohols, ethanol and methanol, are shown in Table 6-4. The list of properties for coal and coal derived synthetic hydrocarbons are shown in Table 6-5. The oil shale derived liquid fuels are presented in Table 6-6. More information is needed on the physical and chemical properties of the synthetic hydrocarbon fuels although it is expected that synthetic hydrocarbons will be similar to their petroleum based counterparts.

The inorganic fuels are hydrogen and ammonia. Their properties are shown in Table 6-7. These two fuels mark the high and low values of the heat of combustion for all the fuels. Vegetable oil properties are shown in Table 6-8 with cottonseed oil as the representative fuel.

In addition to the fuels already presented, slurries, emulsions and blends must be considered. The composition and properties of some slurry fuels are listed on Table 6-9 which was taken from Ref. 6-2. Coal in Diesel fuel slurry is not included in this reference but it is very similar to a petroleum coke slurry. The properties of water-Diesel fuel emulsions are presented in Table 6-10. A variety of both macro- and micro-emulsions are shown in this table for three different water and emulsifier ratios.

Alcohol emulsions are also included in both macro- and micro-emulsion form. The properties of these emulsions are shown in Table 6-11. Actually, in concentrations up to 20%, dry ethanol will dissolve in Diesel No. 2 but if water is present in concentrations of 0.5% or more, the solution will tend to separate and an emulsifier is needed. Methanol does not mix well at any concentration and emulsifiers are needed to form stable mixtures.

Table 6-3. Properties of Petroleum Products

Property	Diesel No. 2	Gasoline	Light Distillate	Naphtha	Lube Stock	Broad-Cut Oil	Jet A
Cetane No.	51.1	17.2	54.8	39.9	N.A.	54	46.1
Viscosity, CS 104° F	3.13	0.55	1.66	0.89	121.6	3.0	1.47
Specific Gravity	.845	.727	.830	.810	N.A.	.850	.835
Heat of Combustion (Lower) Btu/lb	18,600	19,300	18,700	18,800	17,500	18,500	18,900
Flashpoint, °F	183	-45	165	75	241	100	122
Pour Point, °F	7	-94	-44	-72	-21	[illegible]	-58
Distillation Range, °F							
10%	473	121	418	248	N.A.	300	37
90%	603	275	459	338	N.A.	620	48
EP	685	405	503	378	N.A.	700	55
Lubricity (LFW-1 Wear Test)	1.17	1.99	1.15	1.15	N.A.	N.A.	1.38
Accelerated Stability mg/100 ml	3.51	0.4	0.1	0.23	N.A.	N.A.	0.0

N.A. - Not available

Table 6-3. (cont'd)

Property	Kerosene	Methane	Propane	Butane	No. 4 Fuel Oil	No. 5 Fuel Oil	No. 6 Fuel Oil
Cetane No.	53	N.A.	N.A.	N.A.	54.4	41.0	26.2
Viscosity, CS 104° F	1.46	N.A.	N.A.	N.A.	2.89	185.9	137.9
Specific Gravity	.811	.424(L)	.510(L)	N.A.	.876	.920	.972
Heat of Combustion (Lower) Btu/lb	19,100	21,500	19,930	19,670	18,600	19,200	17,160
Flashpoint, °F	135	N.A.	-156	N.A.	160	227	150
Pour Point, °F	-47	-296	N.A.	N.A.	37	70	27
Distillation Range, °F							
10%	380	---	---	---	419	N.A.	N.A.
90%	452	---	---	---	750	N.A.	N.A.
EP	509	-259	-44	31	774	N.A.	N.A.
Lubricity, (LFW-1 Wear Test)	1.15	2+	2+	2	1.36	1.41	1.09
Accelerated Stability Mg/100 ml	0.3	N.A.	N.A.	N.A.	18.9	N.A.	N.A.

Note: N.A. - not available
L - Liquid

Table 6-4. Properties of Alcohols

Property	Methanol	Ethanol
Chemical Formula	CH_3OH	C_2H_5OH
Boiling Point, °F	148	172
Specific Gravity	.796	.794
Heat of Combustion (Lower) Btu/lb	8,580	11,550
Flash Point, °F	52	55
Lubricity	Poor	Poor
Auto Ignition Temp., °F	878	738
Flammability Limits	6.72-36.50%	3.28-18.95%

Table 6-5. Properties of Coal and Coal Derived Synthetic Hydrocarbons

Property	Powdered Coal	Coal Distillate[a]	Coal Gasoline[a]
Cetane No.	N.A.[b]	40	5
Viscosity, CS	N/A [c]	3	0.5
Specific Gravity	0.70 to 0.94	0.91	0.80
Heat of Combustion (Lower) Btu/lb	6,700 to 12,300	17,900	18,200
Flash Point, °F	N.A.	148	N.A.
Pour Point, °F	N/A	N.A.	-40

[a]Based on TOSCOAL process with subsequent hydrogenation and conventional refining
[b]N.A. - Not Available
[c]N/A - Not Applicable

D. DIRECT USE OF COAL IN DIESEL ENGINES

This subject arises more than for any other alternative fuel. Therefore, it will be discussed in more detail than for the other fuels. The idea of burning coal in a Diesel engine is as old as the engine itself. Rudolph Diesel, in his patent, considered the use of powdered coal as the fuel in his engine. Over the last 80 years, there have been numerous experimental coal-fired Diesel engines. None of the engines have been successful for three main reason including slow combustion rate, wear due to ash, and problems associated with the injection of the powdered coal.

Table 6-6. Properties of Oil Shale Liquids[a]

Property	Distillate	Gasoline
Cetane No.	28	<22
Viscosity, CS @ 77° F	3	0.5
Specific Gravity	.87	.83
Heat of Combustion (Lower) Btu/lb	18,300	18,200
Flash Point °F	135	
Auto Ignition Temp., °F	500	900

[a] Based on TOSCO II Hot Solids retort extraction followed by upgrading at mining site to synthetic crude quality

Table 6-7. Properties of Inorganic Fuels

Property	Hydrogen	Ammonia
Chemical Formula	H_2	NH_3
Boiling Point, °F	-432	-28
Specific Gravity (Liquid)	.071	0.674
Heat of Combustion (Lower) Btu/lb	51,600	8,000
Flammability Limits	4.0-74.2%	15.5-26.6%
Ignition Temperature, °F	1,065	1,204

Table 6-8. Properties of Vegetable Oils

Property	Cottonseed Oil
Cetane No.	
Melting Point, °F	23
Specific Gravity	.912
Heat of Combustion (Lower) Btu/lb	16,113
Flash Point	486
Ignition Temperature, °F	650

Table 6-9. Composition and Properties of Slurry Fuels
(From Ref. 6-2)

PREPARATION NO.	COMPONENT	WT. % OF COMPONENT	DIESEL(1) FUEL	WT. % DIESEL FUEL	ADDITIVE	WT. %(1) ADDITIVE	PREP. TIME (HRS.)	GRINDING ELEMENTS	DENSITY	ΔH COMBUSTION (BTU/LB)	BROOKFIELD(2) VISCOSITY	RELATIVE(3) STABILITY
1	Carbon Black	1.3	Diesel Fuel (A)	97.4	Lecithin	1.3	2.5	No	0.8579		6	U
2	Carbon Black	10.0	Diesel Fuel (A)	89.0	Lecithin	1.0	2.5	No	0.9083		36	U
3	Carbon Black	20.0	Diesel Fuel (A)	79.0	Lecithin	1.[illegible]	6.0	No	0.9516		294	U
4	Carbon Black	20.0	Diesel Fuel (A)	79.0	Lecithin	1.0	5.0	Yes	0.9501	18,120	763	G
5	Carbon Black	20.0	Diesel Fuel (A)	79.0	Lecithin	1.0	2.5	Yes	0.9465		205	F
6	Carbon Black	20.0	Diesel Fuel (B)	79.0	SOA	1.0	4.5	No	0.9367		152	U
7	Carbon Black	40.0	Diesel Fuel (A)	59.0	Lecithin	1.0	4.0	No	1.1110	17,065	700	G
8	Petroleum Coke	20.0	Diesel Fuel (A)	79.0	Lecithin	1.0	5.0	No	0.9497	18,530	8	U
9	Petroleum Coke	20.0	Diesel Fuel (B)	79.0	SOA	1.0	1.0	No	0.8968		6	U
10	Petroleum Coke	20.0	Diesel Fuel (B)	74.0	EA-37	6.0	3.0	Yes	0.9119		735	E
11	Petroleum Coke	20.0	Diesel Fuel (B)	79.0	Lecithin	1.0	3.5	Yes	0.8985		265	E
12	Petroleum Coke	20.0	Diesel Fuel (B)	79.0	Lecithin	1.0	5.0	Yes	0.9180		1080	E
13	Petroleum Coke	20.0	Diesel Fuel (B)	79.0	Lecithin	1.0	5.0	Yes	0.9177		2300	E
14	Petroleum Coke	20.0	Diesel Fuel (B)	74.0	SOA	6.0	3.5	No	0.9069		7	U
15	Petroleum Coke	40.0	Diesel Fuel (B)	59.0	SOA	1.0	3.0	No	0.9854		16	U
16	Petroleum Coke	40.0	Diesel Fuel (A)	59.0	Lecithin	1.0	2.5	No	0.9967		24	E
17	Flour	20.0	Diesel Fuel (A)	79.0	Lecithin	1.0	3.0	No	0.9192	15,875	8	U
18	Flour	40.0	Diesel Fuel (B)	59.0	Lecithin	1.0	3.0	No	1.004		14	U
19	Cornstarch	20.0	Diesel Fuel (B)	79.0	SOA	1.0	2.5	Yes	0.9150		7	-
20	Cornstarch	40.0	Diesel Fuel (B)	59.0	SOA	1.0	2.8	Yes	1.0156		10	U
21	Wood	20.0	Diesel Fuel (B)	79.0	SOA	1.0	2.5	Yes	0.9079		700	F

(1) Diesel Fuel A (AI 6766F)
Diesel Fuel B (AI 8064F)

(2) Due to the shear sensitivity of these fuels, these viscosities are listed here only to give indication of relative magnitude and should not be construed as precise numbers.

(3) Stability Code
E = Excellent - No Settling on 68 hrs.
G = 6% or less settling in 68 hrs.
F = 20% to 6% settling in 68 hrs.
U = 21% or Greater settling in 68 hrs.
- = No measurement

Table 6-10. Properties of Water-in-Diesel Fuel Emulsions (From Ref. 6-2)

Fuel and Composition	Sp. Gr. @ 15.6°C	Viscosity @ 40°C, cs	Cetane Number	Net Heat of Combustion,(3) Mjoule/kg. (BTU/lb)
Neat fuel(1)				
Referee grade II diesel	0.8443	2.17	47.4	42.38 (18,222)
Macro-emulsions(2)				
10% water/1% emulsifier/89% diesel	0.8611	2.54	41.5	36.99 (15,906)
20% water/2% emulsifier/78% diesel	0.8828	4.40	34.7	31.79 (13,669)
30% water/3% emulsifier/67% diesel	0.8944	5.71	28.6	26.85 (11,544)
Micro-emulsions(2)				
10% water/6% emulsifier/84% diesel	0.8795	3.53	40.1	36.51 (15,697)
20% water/8% emulsifier/72% diesel	0.8887	12.84	30.0	31.34 (13,477)
30% water/10% emulsifier/60% diesel	0.9088	12.83	24.9	26.43 (11,362)

(1) Diesel fuel used to prepare the various water-in-diesel emulsions.

(2) The various components of the emulsions are given in volume percents.

(3) The net heat of combustion was calculated from the net heat of combustion of the various components in the fuel.

Table 6-11. Properties of Alcohol-in-Diesel Fuel Emulsions (From Ref. 6-2)

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Fuel and Composition	Sp. Gr. @ 15.6°C	Viscosity @ 40°C, cs	Cetane Number	Net Heat of Combustion, Mjoule/kg. (BTU/lb)
Neat fuel				
Referee grade II diesel	0.844	2.17	47.4	42.38 (18,222)
Macro-emulsions				
20% Ethanol 3.25% surfactant 1.75% H_2O	0.833	-	33	38.07 (16,367)
20% Methanol 3.25% surfactant 1.75% H_2O	0.835	-	30	36.72 (15,789)
Micro-emulsions				
10% Ethanol 4% surfactant	0.836	-	42.4	40.62 (17,465)
20% Ethanol 8% surfactant	0.828	-	37.8	38.83 (16,695)
30% Ethanol 12% surfactant	0.820	-	31.8	36.99 (15,906)
10% Methanol 10% surfactant	0.833	-	41.8	39.47 (16,972)
20% Methanol 20% surfactant	0.823	-	35.4	36.49 (15,689)
30% Methanol 30% surfactant	0.812	-	28.5	33.43 (14,373)

There are several other minor problems encountered with powdered coal such as the pulverizing equipment, fire hazards, and the pumping of the powder.

For any fuel to react with the oxygen in the air, it must be in a gaseous state. Gaseous fuels react quickly and have high combustion rates. Liquid and solid fuels must first be vaporized before the oxidation reaction can take place. There are a number of factors which effect the rate of vaporization including the heat of vaporization, the air temperature, wall temperatures, size of droplets or particles, turbulence in the combustion zone, thermal radiation from other burning particles, and the chemical properties of the fuel itself. Regardless of the cause, slow combustion can result in Diesel knock, high exhaust temperatures, burned valves, and piston damage.

Coal is not a readily vaporized fuel even though many coals have a high volatile matter content. The volatile matter is usually made up of heavy oils and tars with high boiling points. As a result of its low combustion rate, coal is suitable only for low speed Diesels with a rated speed of 200 rpm or less. Since a typical locomotive Diesel engine operates at 900 to 1100 rpm rated speed, the displacement of a coal-fired engine would have to be at least five times that of present engines to produce the same output power. The size and weight of such an engine would rule out its use in a locomotive.

The use of pulverized raw or unprocessed coal in a Diesel engine results in very high wear rates for the piston rings, cylinder, crankshaft, and bearings. The composition of coal ash varies widely but is primarily composed of silicates which are usually harder than most metals. The fine ash released during combustion infiltrates the gap between the piston and the cylinder. Blowby will carry the ash into the oil sump where it can mix with the oil. Where two metals of different hardness are in contact such as at the rings and in the bearings, the ash becomes embedded in the softer metal and literally grinds away the harder metal. The ash and the sulfur in coal can be removed prior to combustion by chemical and physical processing but this usually doubles or triples its price depending on the degree of ash removal. This increase in fuel cost significantly reduces the economic benefits of using coal.

Powdered coal, by itself, is not capable of being pumped or being injected into a Diesel engine. It is necessary to provide a fluidizing agent, typically air, steam, oil, or water depending on the application. Electric utility boilers normally use air or steam as the fluidizing agent The use of oil or water as a fluidizing agent is normally in the form of a slurry. For a locomotive, the slurries are probably better than the use of air or steam with the coal since the overall fuel supply system is simpler. Depending on the stability of the slurry, the system may not be anymore complex than the present Diesel fuel system.

When all of the limitations are considered, the direct use of coal in a locomotive Diesel engine is not as attractive as most of the other alternative fuels. Highly processed coal in an oil slurry may be used with some Diesel engines but at a cost approaching that of Diesel No. 2. The gains achieved by the use of coal are negated by the problems associated with the preparation of the slurry.

The use of coal in an adiabatic Diesel engine may be possible although actual tests have not been conducted. The ceramic walls of this engine are at a red heat and the thermal radiation should insure rapid ignition and combustion. Since the walls are ceramic, they are harder than the ash from the coal so wear is not likely to be a problem. However, ash could collect in the clearance space between the head and the piston and may cause problems. The use of a powdered, physically processed coal in a water slurry would keep the cost down while providing an injectable fuel. Theoretically, such a fuel is attractive and tests of it should be made.

E. ENGINE ANALYSIS

In preparation for ranking the fuels, a series of engine analyses were made using both two-stroke and four-stroke engines typical of existing locomotive Diesels. The operation of these locomotives was simulated for a variety of duty cycles. The analytical approach was validated by conducting the analysis for Diesel fuel No. 2 and comparing these results to published engine data. These comparisons were found to agree. Using this same analytical approach, calculations were made for engines running on the alternative fuels.

The analysis is based on the assumption of isentropic compression and expansion. Combustion is assumed to be complete and adiabatic. It occurs partly at constant volume and partly at constant pressure. The idealized cycle, called a limited pressure cycle, is shown on the pressure-volume diagram in Figure 6-1. Specific heats and specific heat ratios that are functions of temperature and equivalence ratio are used. All results are based on the lower heating value of the fuel, and include the effects of the sensible heat and the latent heat of vaporization on the temperature drop in the fuel-air charge due to complete vaporization of the fuel. The effect of residual gases on engine performance is also taken into account.

Cycle analysis is corrected for frictional losses and heat transfer losses to the coolant. When using fuels with low cetane ratings (e.g. alcohols, hydrogen, etc.) a quantity (about 5 to 30% of the total energy) of liquid Diesel fuel is injected at the end of the compression stroke to start ignition. The combustion properties of the pilot charge are assumed to be the same as those of the main fuel. The inlet air is assumed to be turbocharged and intercooled.

As an example, the method for calculating the combustion properties of methanol-air mixtures is shown. These calculations are based on complete combustion of the fuel in the presence of air and the 1% argon present in the air is treated as if it is nitrogen. This results in a very small error in molecular weight and does not affect engine performance calculations.

The combustion equation for methanol is:

$$2CH_3OH + a(O_2 + 3.76N_2) \longrightarrow 4H_2O + 2CO_2 + (a-3)\ O_2 + (3.76a)N_2$$

In this equation, "a" represents the number of moles of air involved in the reaction. The value of "a" can be chosen to obtain the desired equivalence ratio. For example, for a stoichiometric reaction the equivalence ratio is one and the value of "a" is three. The molar values of the specific heats

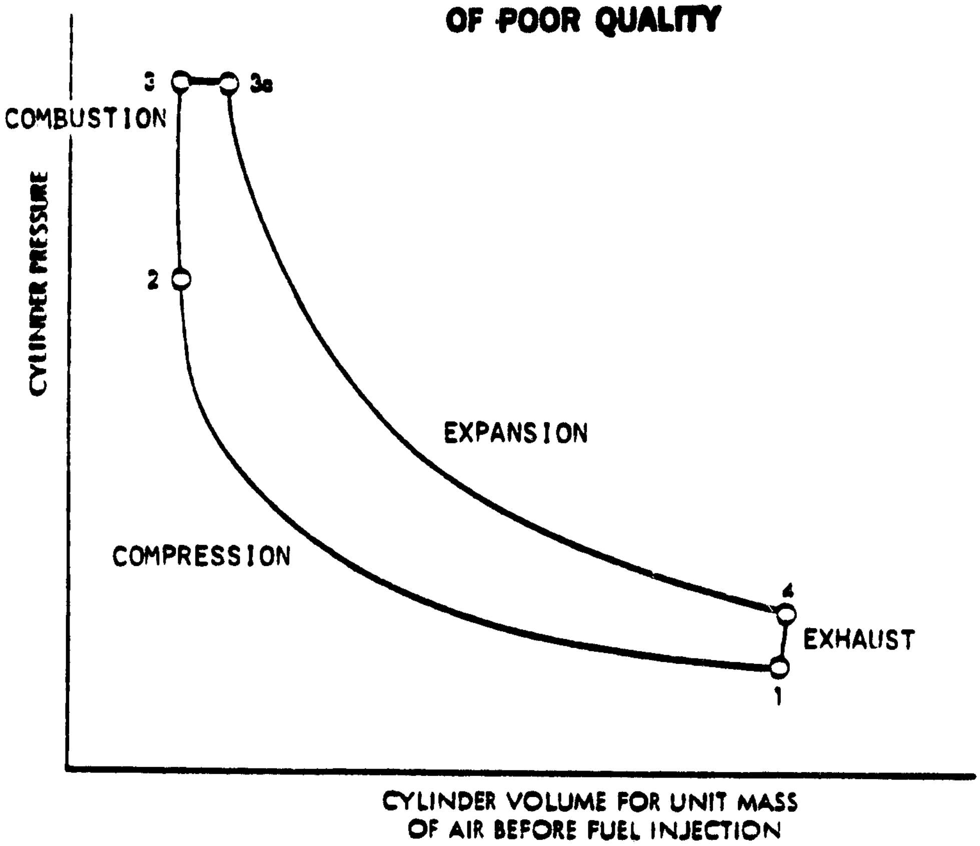

Figure 6-1. Limited Pressure Cycle on a Pressure-Volume Diagram

(Cp and Cv) for each gas on the right-hand side of the above equation are given in Ref. 6-5 as a function of temperature. Using this information the properties of the combustion products are calculated for different equivalence ratios.

Unless stated otherwise, the analysis of the alternative fuels is made for the same mass flow rate of air and the same equivalence ratio as that of the baseline engine. Table 6-12 shows that results of the fuel-air cycle analysis for a few representative fuels ranging from Diesel No. 2 to methane and methanol. The difference between a low energy per pound fuel like methanol and a high one like methane is only two percentage points. There are performance differences between fuels but the thermal efficiency of an engine depends primarily upon its design and operating conditions.

The peak values of brake thermal efficiency with the corresponding values of brake specific fuel consumption and brake specific energy consumption are shown in Table 6-13 for a number of fuels. The effect of notch position on brake specific energy consumption is shown in Section III, Tables 3-15 to 3-30, for the same fuels as in Table 6-13. These tables present the engine schedules used in the computer simulation of the locomotive to determine fuel consumption in service.

Table 6-12. Results of Fuel-Air Cycle Analysis for a Diesel Engine

Parameter	Diesel No. 2	Methanol	Ethanol	Methane
Stoichiometric Air-Fuel Ratio	14.7	6.45	9.1	17.2
Analysis Equivalence Ratio	0.5	0.5	0.5	0.5
Specific Heat Ratio-Compression	1.371	1.371	1.371	1.371
Specific Heat for Constant Volume Combustion	.2368	.2485	.2442	.2436
Specific Heat for Constant Pressure Combustion	.3231	.3380	.3318	.3330
Specific Heat Ratio-Expansion	1.2831	1.2767	1.2790	1.2790
Indicated Horsepower	4127	4000	4049	4001
Indicated Thermal Efficiency	51.2	48.7	50.1	50.7

Because thermal efficiency is not significantly influenced by the type of fuel used by the engine, it has not been included in the list of functional properties, instead its effect is included in the power density functional property.

F. FUNCTIONAL PROPERTIES

The functional properties differ from the physical and chemical properties in that they are specifically related to the impact of the fuel on the operation of the rail system. One or more physical and chemical property may be involved in a single functional property. As an example, tank volume is a function of specific gravity and the heat of combustion. The tank volume is the volume of fuel that contains a specified amount of energy. Tank volume is important to the size of the storage facilities both at the refueling terminal and on the locomotive.

Twenty-two functional properties are used in this study. In some cases where the function depends only on a single physical and chemical property, either the name of the functional property or the physical and chemical property may be used. For example, ignitability is used rather than cetane no. Similarly, pumpability is used rather than viscosity though they are very closely related. When discussing the slurry fuels,

Table 6-13. Peak Thermal Efficiency of Diesel Engines Using Different Fuels

Fuel	BSFC (lb/hp-hr)	BSEC (kBtu/hp-hr)	Brake Thermal Efficiency (%)	Lower Heat of Combustion (Btu/lb)
Diesel No. 2	.350	6.51	39.1	18,600
Naphtha	.349	6.56	38.8	18,800
Liquid Methane	.305	6.56	38.8	21,500
Coal Derived Dist.	.365	6.54	38.9	17,900
Oil Shale Dist.	.361	6.54	38.9	18,100
Methanol	.784	6.73	37.8	8,580
Ethanol	.578	6.68	38.1	11,550
Liquid Hydrogen	.130	6.72	37.9	51,600
Liquid Ammonia	.840	6.72	37.9	8,000
Coal-Diesel No. 2 Slurry (20%/80%)	.376	6.66	38.2	17,700
Gasoline-Lube Oil Blend (15%/85%)	.370	6.59	38.6	17,800
Gasoline-Lube Oil Blend (70%/30%)	.350	6.56	38.8	18,750
Water-Diesel No. 2 Emulsion (10%/84%)	.403	6.32	40.3	15,700
Methanol-Diesel No. 2 Emulsion (10%/85%)	.369	6.28	40.5	17,000
Ethanol-Diesel No. 2 Emulsion (20%/72%)	.374	6.25	40.7	16,700
Heavy Aromatic Naphtha-Diesel No. 2 (75%/25%)	.352	6.58	38.7	18,700

pumpability is a more meaningful term than viscosity. These twenty-two properties are divided into four groups:

(1) Combustion related properties.
(2) Engine operation related properties.
(3) Logistics and storage related properties.
(4) Maintenance and safety related properties.

These twenty-two properties are listed by groups in Table 6-14. Some of the properties could be put in one group just as well as another. Also property in one group may be related to two or more in other groups. The grouping is just a convenient way of relating similar properties for purposes of discussion.

G. COMBUSTION RELATED PROPERTIES

The functional properties related to combustion are ignitability, flame speed, power density, combustion rate, and smoke. In a Diesel engine, ignitability is defined by the cetane number which is a measure of the ignition delay period in a compression ignition engine. Current Diesel

Table 6-14. Functional Properties

Combustion Related Properties	Engine Operation
Ignitability	Energy Density
Flame Speed	Pumpability
Power Density	Boiling Point
Combustion Rate	Pour Point
Smoke	Vaporization
	Lubricity
Logistics and Storage	**Maintenance and Safety**
Future Cost	Special Equipment
Future Availability	Compatibility
Water Reactivity	Lubricant Reactivity
Tank Volume	Toxicity
Special Tankage	Fire Hazard
Stability	

fuel No. 2 has an average cetane number of 48 while a minimum cetane number of 40 is recommended by some Diesel engine manufacturers. Except for distillates from petroleum, oil shale and coal, the cetane numbers of most other fuels are too low for use in a compression ignition engine unless auxiliary ignition sources are provided. In addition to auxiliary ignition sources such as glow plug, pilot injection, and spark plug, ignition can be initiated by blending in a high cetane number fraction or by use of very high compression ratios. The ignitability of the slurries and emulsions depends heavily on the presence of a high cetane number fuel component. In the slurries and emulsions in this study, the use of Diesel No. 2 insured good ignitability.

Current engines have been designed to operate best with hydrocarbon fuels having a flame speed of about 1.1 feet per second. A wide variation in flame speed from this value would probably require that the engine operation, specifically the injection rate, be modified for best results. For a given injection rate, a high flame speed can cause injector tip burning and a low flame speed can result in the accumulation of fuel in the combustion chamber. Therefore, the high flame speed of hydrogen and the low speed of ammonia suggests that caution should be used with these fuels in reciprocating engines. The use of any fuel with a flame speed different from that of Diesel No. 2 may require engine modifications.

The functional property, called power density, deals with the changes in power in a given size engine or the size of an engine required for a given output power level. Some fuels, particularly gaseous hydrogen in an inducted fuel system, will reduce the output power of the engine as compared to the same engine using Diesel fuel. The hydrogen displaces some of the air in the cylinder so that the mass of fuel and air is less than in a liquid fueled engine. The gaseous fueled engines are the worst engines in this respect but some liquid fuels can also have a significant effect on the output power. This functional property also includes the effects of fuel on thermal efficiency of the engine.

The combustion rate property is related to the volume burning rate of the fuel-air mixture. This property is similar to but not the same as the flame speed. The combustion rate is related to the heat of vaporization, atomization of the fuel spray, combustion chamber temperature and the boiling point of the fuel at the pressure existing in the cylinder. For a fuel to burn, it must first be converted to a gaseous state and the combustion rate is a measure of this preparatory state as well as the actual rate of burning. A heavy hydrocarbon fuel such as No. 5 fuel oil has a low combustion rate although its flame speed is comparable to that of other hydrocarbon fuels. A low combustion rate can result in incomplete burning before the exhaust valve opens. Burnt valves, overheated exhaust system, and low efficiency are the consequences.

Smoke or particulate emissions have not been a major problem in the past but they can be expected to be more closely regulated in the future. Different fuels have widely differing amounts of smoke ranging from none for hydrogen to a marked amount for coal slurries and heavy fuel oils. The weighing factor for this functional property is largely dependent on the expected level of regulation imposed on Diesel engine particulate emissions. The use of coal produces "smoke" in the form of fly ash. The term "smoke" in this functional category includes all ash, soot, smoke and other solid particulate matter produced by the combustion of fuel.

The rankings of the 41 fuels, slurries, emulsions and blends are shown in Tables 6-15 and 6-16. The rankings of the 18 slurries, emulsions and blends are separate from the rest of the fuels. They can be cross-ranked by using the combustion scores if it is necessary. Of the 23 fuels on Table 6-15, the best are propane and butane with scores of 5 and 3. At the other extreme is powdered coal with a rating of -60. Any two fuels with scores that are within ten points of each other can be considered to be equals. A fuel with a score between -5 and 5 is equal to Diesel No. 2 within this functional property group. Of the non-petroleum fuels, only hydrogen ranked in the top ten fuels. Among the slurries, emulsions and blends, in Table 6-16, the best fuels were the petroleum blends and the worst were the slurries. The gasoline-Diesel fuel blend is comparable to the best fuels on Table 6-15. Because all of the slurries, emulsions and blends are petroleum based, they tend to score higher than most of the non-petroleum fuels.

H. ENGINE OPERATION RELATED PROPERTIES

The functional properties in the engine operation are extensions of the combustion group but are oriented towards the whole engine more than just the combustion process itself. The properties in this group are energy density, pumpability, boiling point, pour point, vaporization and lubricity.

Table 6-15. Ranking of Fuels Relative to Diesel No. 2 in Combustion Performance

	Ignit-ability	Flame Speed	Power Density	Combustion Rate	Smoke	Score	Rank
Weighing Factor	6	5	4	6	5		
Gasoline	-2	+1	0	-1	0	-13	15
Naphtha	-1	0	0	0	0	-6	8
Light Distillate	0	0	0	0	0	0	3
Lube Stock	0	-1	0	-1	-3	-26	19
Broadcut Fuel Oil	0	0	0	0	0	0	3
Jet A	0	0	0	0	0	0	3
Kerosene	0	0	0	0	0	0	3
Methane	-3	+1	-1	+1	+2	-1	7
Butane	-2	+1	-1	+1	+2	3	2
Propane	-2	+1	-1	+1	+2	5	1
No. 4 Fuel Oil	0	0	0	-1	-1	-11	10
No. 5 Fuel Oil	-1	-1	0	-2	-2	-33	20
No. 6 Fuel Oil	-2	-2	0	-2	-2	-44	22
Methanol	-3	-1	0	-1	+3	-14	16
Ethanol	-3	-1	0	-1	+3	-14	16
Powdered Coal	-3	-3	0	-2	-3	-60	23
Coal Derived Distillate	-1	0	0	0	-1	-11	10
Coal Derived Gasoline	-2	+1	0	0	-1	-12	13
Oil Shale Distillate	-1	0	0	0	-1	-11	10
Oil Shale Gasoline	-2	+1	0	0	-1	-12	13
Hydrogen	-3	+3	-1	-3	+3	-10	9
Ammonia	-3	0	-1	0	+1	-17	18
Cottonseed Oil	0	-2	0	-3	-3	-37	21

Table 6-16. Ranking of Fuel Slurries, Emulsions and Blends Relative to Diesel No. 2 in Combustion Performance

	Ignit-ability	Flame Speed	Power Density	Combustion Rate	Smoke	Score	Rank
Weighing Factor	6	5	4	6	5		
Diesel Fuel Slurries							
20% Carbon Black	0	0	0	-2	-2	-22	18
20% Petroleum Coke	0	0	0	-1	-1	-11	9
20% Flour	0	0	0	-1	0	-6	2
20% Cornstarch	0	0	0	-1	0	-6	2
20% Wood Fibers	0	0	0	-1	-1	-11	9
20% Coal	0	0	0	-1	-2	-16	16
Diesel Emulsions							
10% Water-6% Emulsifier	-1	0	0	0	0	-6	2
30% Water-10% Emulsifier	-2	0	+1	-1	+1	-9	7
10% Methanol-10% Surfactant	-1	0	0	-1	0	-12	12
30% Methanol-30% Surfactant	-1	0	0	-1	+1	-7	5
10% Ethanol-4% Surfactant	-1	0	0	-1	0	-12	12
30% Ethanol-12% Surfactant	-1	0	0	-1	+1	-7	5
Petroleum Blends							
30% Gasoline in Lube Oil	0	0	0	0	-2	0	8
70% Gasoline in Lube Oil	-1	0	0	0	-1	-11	9
64% Heavy Aromatic Naphtha in Diesel	-1	0	0	-1	0	-12	12
35% Heavy Aromatic Naphtha in Diesel	-2	0	0	0	0	-12	12
50% Gasoline in Diesel No. 2	0	0	0	0	0	0	1
80% No. 6 Fuel Oil in Diesel No. 2	0	0	0	-1	-2	-6	16

Energy density is the product of the lower heat of combustion and the density of the fuel. It is a measure of the energy per unit volume of the fuel. This functional property determines the size of the fuel lines, pumps, valves and injectors on the engine. The use of a low energy density fuel such as hydrogen, ammonia, or methanol necessitates extensive engine fuel supply system modifications if it is to be used in an existing engine. Differences in energy density is the main problem area in making true multi-fuel Diesel engines. For a given injector orifice size, the injection velocity varies directly with the energy density of the fuel. Injection velocity influences the spray atomization and penetration, hence, the efficiency of the combustion process. The use of a single injector for fuels with widely differing energy densities generally results in poorer performance for all fuels except the one for which the injector was designed.

Pumpability is related to viscosity, but in the case of fuels like the slurries, it involves more than just viscosity. However, viscosity is a prime concern in Diesel engines since pressure fuel injection is used to introduce the fuel into the combustion chamber. For pressure injection, the degree of atomization and the spray penetration becomes poorer as viscosity increases. The kinematic viscosity is important in the low temperature handling of a fuel. Pumping difficulties increase with viscosity until the freezing or pour point is reached. One of the pumpability problems of slurries will be touched upon in the discussion on stability. This is the separation of the fuel components either in the tank or in the fuel system. Fuels such as the wood fiber-Diesel fuel or the carbon black-Diesel fuel slurries are poor candidate fuels because of their poor pumpability. Some problems can be alleviated by the use of fuel additives or by using different types of pumps.

The boiling point of a fuel has several effects on engine operation. A high boiling point reduces the combustion rate which can result in engine damage. A very low boiling point is indicative of a cryogenic fuel. A cryogenic fuel is one which is a gas at room temperature but has been liquified by reducing its temperature below its boiling temperature. For example, methane is a cryogenic fuel when it has been liquified by reducing its temperature to below -259° F. A boiling point near ambient temperatures can result in vapor lock and fuel component separation. Ideally, the boiling point should be in the 300 to 600° F range. In the case of a blend or other fuel mixture, all of the components should have similar boiling points.

Pour point is closely related to the freezing point, to the viscosity and to the cloud point of the fuel. At low temperatures, many fuels will freeze or become so viscous that they cannot be used in the engine. In the hydrocarbon fuels, some components such as the waxes, may freeze out even at relatively mild temperatures. The temperature at which this occurs is known as the cloud point. These waxes or any other solid material freezing in the fuel will cause problems such as filter and injector clogging. Some fuels, such as the cottonseed oil, freeze at such a high temperature that some means of warming the fuel is necessary even in only moderately cold climates. All of the low temperature problems are included in the pour point functional property.

The vaporization functional property includes both the effects of the heat of vaporization and the vapor pressure. The problems areas associated with this functional property are vapor lock, tank venting and combustion rate. A fuel with a high heat of vaporization and a low vapor pressure can cause the combustion rate of a fuel to be low. It can also reduce the output power of the engine because the heat absorbed to vaporize the fuel is not available to produce power. A high vapor pressure fuel will tend to evaporate with time because the tank must be vented to prevent a pressure build-up which could structurally damage it. The loss of fuel can be significant especially for the cryogenic fuels such as liquid hydrogen.

The functional property, lubricity, refers to the ability of a fuel to lubricate the equipment through which it flows. The pumps and injectors are the main problem areas. Some fuels with low lubricity such as gasoline and the alcohols are also excellent solvents and will remove any trace of oil, grease or solid lubricant used in the fuel system. These fuels will require different pumps and injectors than are now used for the medium and heavy hydrocarbon fuels. The cryogenic fuels have extremely low lubricity and their low temperatures complicate the design of the equipment used to handle it.

The fuels, slurries, emulsions and blends are ranked in Tables 6-17 and 6-18 according to their relative merit in engine operation. Lube stock is ranked highest mainly because of its excellent lubricity but most of the other medium to heavy hydrocarbons are also ranked quite high. Of the non-petroleum fuels, oil shale distillate and coal derived distillate are highest ranked but they are ranked ninth overall. Two blends and two emulsions tied for the first ranking in their category. The 20% wood fiber in Diesel oil slurry is ranked last primarily because of its extremely poor pumpability rating.

I. LOGISTICS AND STORAGE RELATED PROPERTIES

Regardless of what fuel is used, it must be purchased, distributed to fueling terminals, stored and loaded into the locomotives. The ideal fuel must be readily available at a low price. It needs to be transported easily without special precautions and stored for extended periods without significant loss or degradation. It must be able to be loaded onto locomotives without difficulty by the crews handling the fuel. Its energy density must be high enough so that a good supply could be carried on the locomotive to minimize the frequency of refueling. For many years, this ideal was closely matched by Diesel fuel. Even today, only the first two items, cost and availability, are problems. However, these two items are of such importance, that some unlikely fuels are receiving serious consideration.

The future cost of a fuel is the first functional property in this group. This cost is not only the direct cost of the fuel itself but also the cost of the modifications of the engine and locomotive, of any added maintenance required and of new storage and distribution at the railroad's refueling terminals. The future prices of all fuels are conjecture and even their relative prices may change drastically in the next two decades. However, the existence of synthetic fuels industry can have a braking

Table 6-17. Ranking of Fuels Relative to Diesel No. 2 in Engine Operation

	Energy Density	Pump-ability	Boiling Point	Pour Point	Vapori-zation	Lubricity	Score	Rank
Weighing Factor	4	5	4	3	3	5		
Gasoline	0	+1	-1	+1	0	-1	-1	7
Naphtha	0	+1	-1	+1	0	0	+4	2
Light Distillate	0	0	0	+1	0	0	+4	2
Lube Stock	0	-1	+1	0	0	+2	+9	1
Broadcut Fuel Oil	0	0	0	0	0	0	0	5
Jet A	0	0	0	0	0	0	0	5
Kerosene	0	0	0	+1	0	0	3	4
Methane	-2	+1	-2	+3	+1	-3	-14	18
Butane	-1	+1	-1	+3	+1	-3	-6	13
Propane	-1	+1	-1	+3	+1	-3	-6	13
No. 4 Fuel Oil	0	0	+1	-1	-1	0	-2	8
No. 5 Fuel Oil	0	-1	+1	-2	-2	0	-13	17
No. 6 Fuel Oil	+1	-2	+1	-1	-2	0	-11	15
Methanol	-2	+1	-1	+1	-2	-2	-20	19
Ethanol	-2	+1	-1	+1	-2	-2	-20	19
Powdered Coal	-1	-3	0	0	0	-3	-34	21
Coal Derived Distillate	0	0	0	-1	0	0	-3	9
Coal Derived Gasoline	0	+1	-1	0	0	-1	-4	11
Oil Shale Distillate	0	0	0	-1	0	0	-3	9
Oil Shale Gasoline	0	+1	-1	0	0	-1	-4	11
Hydrogen	-3	-1	-3	0	+1	-3	-41	22
Ammonia	-3	-1	-3	0	+1	-3	-41	22
Cottonseed Oil	0	-2	0	-2	0	+1	-1	15

Table 6-18. Ranking of Fuel Slurries, Emulsions and Blends Relative to Diesel No. 2 in Engine Operation

	Energy Density	Pump-ability	Boiling Point	Pour Point	Vapori-zation	Lubricity	Score	Rank
Weighing Factor	4	5	4	3	3	5		
Diesel Fuel Slurries								
20% Carbon Black	0	-3	0	0	0	0	-15	16
20% Petroleum Coke	-1	-1	0	0	0	0	-9	10
20% Flour	-1	-2	0	0	0	0	-14	14
20% Cornstarch	-1	-2	0	0	0	0	-14	14
20% Wood Fibers	-1	-3	0	0	0	0	-19	18
20% Coal	+1	-1	0	0	0	0	-1	5
Diesel Fuel Emulsions								
10% Water-6% Emulsifier	0	0	0	0	-1	0	-3	7
30% Water-10% Emulsifier	0	0	-1	-1	-1	-1	-15	16
10% Methanol-10% Surfactant	0	0	0	+1	-1	0	0	1
30% Methanol-30% Surfactant	-1	0	-1	+1	-1	-1	-13	12
10% Ethanol-4% Surfactant	0	0	0	+1	-1	0	0	1
30% Ethanol-12% Surfactant	-1	0	-1	+1	-1	-1	-13	12
Petroleum Blends								
30% Gasoline in Lube Oil	0	0	0	0	-2	+1	-1	5
70% Gasoline in Lube Oil	0	0	-1	0	-2	0	-10	11
64% Heavy Aromatic Naphtha in Diesel	0	0	0	0	0	0	0	1
35% Heavy Aromatic Naphtha in Diesel	0	0	0	0	0	0	0	1
50% Gasoline in Diesel No. 2	0	0	0	0	0	-1	-5	8
80% No. 6 Fuel Oil in Diesel No. 2	0	0	0	-1	-1	0	-6	9

effect on the prices of petroleum products. It is very likely, therefore, that the price of Diesel No. 2 and the major synthetic fuels will be competitive.

The second functional property is future availability. The particular fuel does not have to be available now but there must be a reasonable probability of it being available when needed and in the volume needed in the future. The availability depends on the source and if the volume of the container is also considered, the ratio would be essentially the same except for the cryogenic fuels where a larger volume is required because of the insulation. The weight of these tanks will also increase. Previous studies of the storage of cryogenic fuels in automotive applications that included the weight of the container, show an increase in the weight ratio from the 0.36 value shown in Table 6-19 to 2.6 for liquid hydrogen, from 0.86 to a value of 1.78 for liquid methane, and from 2.3 up to a value of 3.4 for liquid ammonia. The requirement for special tankage is a significant problem and must be considered. This functional property would also include the need for stirring devices for slurries and any other similar equipment needed by the fuel. A ranking of zero for this functional property means that tankage suitable for Diesel fuel is also suitable for the alternative fuel as well. This can also be assumed to mean that existing fuel tanks and distribution systems will be compatible.

The functional property defined as stability, includes several different physical and chemical properties. With the present fuel distribution system, it is possible that a period of 6 months or more may elapse between the time a fuel is produced and it is burned in an engine. Thus, it is important that a fuel have the ability to retain its desirable characteristics in storage over this period. Instability in storage may take a number of forms such as loss of volatile fuel components, formation of gum or sediment, and change in color.

Storage of slurries is expected to cause problems because of the settling of the solid phase from the liquid. Close control of particle size is necessary and the use of stabilizing agents have proven helpful. In some cases, a means of continuous mixing of the slurry may be required. The method of preparing a slurry is extremely important to its stability and on its injector spray pattern. Even the type of material can have a strong effect on the slurry performance. The carbon black slurries have a different consistency than the coke slurries. During some pump and injector tests it was found that the carbon black formed a porous structure. The Diesel fuel separated and streamed out of the injector while the carbon was extruded like a black wire. The coke slurries, on the other hand, appear to behave more like normal Diesel fuel.

The slurries of cornstarch or flour are not particularly stable even when a stabilizer, lecithin, is added. Constant mixing is necessary to use them in an engine. The slurry of wood is very fibrous and difficult to handle. The Diesel fuel separates from the fiber which then remains in the fuel tank.

The rankings of the various fuels are shown in Tables 6-20 and 6-21. In this group of logistics and storage, powdered coal is the first ranked

Table 6-19. Volume and Weight Ratios of Fuels for the Same Total Energy Content

Fuel	Weight Ratio[a]	Volume Ratio[b]
Petroleum		
Distillate	1.0	1.0
Jet	0.99	1.04
Ammonia	2.3	4.2
Coal Liquids		
Diesel	1.01	1.0
Fuel Oil	1.06	0.92
Jet	0.99 - 1	1.04 - 1.05
Coal/Methanol Slurry	1.64 - 1.84	1.9
Coal/Oil Slurry	1.08 - 1.16	0.86 - 0.92
Hydrogen (l)	0.36	4.23
Ethanol	1.59	1.7
Methanol	2.13	2.26
Higher Alcohols	1.59	1.7
Methane (l)	0.86	1.62
Shale Oil		
Raw	1.01	0.96
Syncrude	0.99	1.05
Vegetable Oil	1.11	1.01

Note: [a] Weight of fuel divided by weight of Diesel fuel having same energy content
[b] Volume of fuel divided by volume of Diesel fuel having same energy content

fuel. In the combustion group, it was ranked last. Cottonseed Oil ranked last principally because of its high cost and its limited availability. Many of the fuels have high positive scores indicating that they are better than Diesel No. 2 in this functional group. Of the non-petroleum fuels, oil shale distillate is the highest ranked fuel primarily because of anticipated favorable future price and availability. From a logistics and storage standpoint, the 30% water-10% emulsifier emulsion ranked first and the 20% coal in Diesel No. 2 slurry ranked second. The lowest ranked fuel was the 30% methanol-30% surfactant emulsion because of the relatively high prices for both the methanol and the surfactant.

J. MAINTENANCE AND SAFETY RELATED PROPERTIES

An alternative fuel should not result in any increase in maintenance and, ideally, might even decrease it. At present, none of the fuels investigated appear to have any chance of decreasing maintenance but some do not appear to increase it significantly. Because the maintenance of a locomotive is labor intensive and, therefore, expensive, it is necessary to consider the maintenance required in the use of any particular fuel.

Table 6-20. Ranking of Fuels Relative to Diesel No. 2 in Logistics and Storage

	Future Costs	Future Avail-ability	Water Reac-tivity	Tank Volume	Special Tankage	Stability	Score	Rank
Weighing Factor	7	8	2	3	4	3		
Gasoline	0	0	0	-1	0	+1	0	11
Naphtha	0	0	0	0	0	+1	+3	7
Light Distillate	0	0	0	0	0	+1	+3	7
Lube Stock	0	0	0	0	0	-1	-3	13
Broadcut Fuel Oil	+1	+1	0	0	0	0	+15	3
Jet A	0	0	0	0	0	+1	+3	7
Kerosene	0	0	0	0	0	+1	+3	7
Methane	0	0	+1	-2	-3	+3	-7	19
Butane	0	0	+1	-1	-1	+3	+4	5
Propane	0	0	+1	-1	-1	+3	+4	5
No. 4 Fuel Oil	0	0	0	0	0	-1	-3	13
No. 5 Fuel Oil	0	0	0	0	0	-1	-3	13
No. 6 Fuel Oil	0	0	0	+1	0	-1	0	11
Methanol	-1	+2	-3	-3	-1	+2	-4	16
Ethanol	-2	+1	-3	-2	0	+2	-12	20
Powdered Coal	+3	+3	+2	-2	-2	+3	+44	1
Coal Derived Distillate	0	+2	0	0	0	-1	+13	4
Coal Derived Gasoline	-1	+1	0	-1	0	-1	-5	17
Oil Shale Distillate	+1	+2	0	0	0	-1	+20	2
Oil Shale Gasoline	-1	+1	0	-1	0	-1	-5	17
Hydrogen	-2	+1	+3	-3	-3	+2	-13	21
Ammonia	-3	+1	-1	-2	-3	+2	-27	22
Cottonseed Oil	-3	-3	0	0	0	-2	-51	23

Some of the fuels being considered have cetane numbers so low that they will not ignite reliably in a compression ignition engine. For these fuels, an auxiliary ignition system is needed. Some of the ignition systems that have been used are glow plugs, spark plugs, pilot injection, fuel blending and the use of a high compression ratio. Except for the last two items, these auxiliary ignition systems increase the complexity of the engine and will also increase the maintenance requirements. Pilot injection is the injection of a small quantity of Diesel fuel into the cylinder to initiate combustion of the main fuel. A tank, pump, injectors and filters are required for this system in addition to the main fuel supply system. Since the pilot fuel flow rate is small (about 5% of the main fuel supply rate), the tank, lines, pumps and injectors are all smaller than regular Diesel fuel equipment and are more sensitive to dirt and gum than the normal fuel system. All these added components and their special requirements increase the maintenance of the engine. The functional property known as special equipment is used to evaluate this additional fuel system.

Material compatibility refers to the reactivity (chemical, dissolution, etc.) between a fuel and the materials of the fuel system in which it is used. A reaction may cause engine malfunctions due to the formation of fuel leaks, the distortion of fuel system components, and the development of undesirable deposits in sensitive areas of the controls and the pumps. It is important, therefore, that a fuel be compatible with the fuel system in which it is used. The degree of compatibility must be very high since the expected service life is long (more than 10 years) and relatively small changes in sensitive parts (e.g. metering devices) can be very serious.

Another type of material compatibility problem is encountered when hydrogen is used as a fuel. Hydrogen can cause metal embrittlement at the high pressures encountered in Diesel engine fuel supply systems. Its high diffusivity makes it difficult to contain in any type of equipment and is especially difficult when non-metallic parts are used. Each fuel has its own material requirements thus complicating the fuel system if multi-fuel capabilities are needed.

The lubricant reactivity property of a fuel refers to sludging, additive precipitation or component separation in the crankcase of the Diesel engine. This usually occurs because some of the unburned or partially burned fuel has leaked past the piston rings and into the crankcase. The petroleum and synthetic hydrocarbons are expected to react in a manner similar to Diesel No. 2. Clean burning fuels such as hydrogen or methane are not expected to have any adverse effects, but nitrogen-rich fuels may have unusually severe effects. The nature and severity of lubricant reactivity must be considered when evaluating any non-hydrocarbon fuel.

All fuels must be able to be transported to a refueling terminal, stored there and eventually loaded into the locomotive without undue danger to either the equipment or the crews handling the fuel. Some of the fuels do present special safety problems. The functional properties in the safety area are toxicity and fire hazard.

The toxicity of a fuel is extremely important in that it determines the precautions that must be taken in the safe handling and use of the fuel. The three main types of toxicity are: vapor inhalation, ingestion and skin contact. Vapor toxicity determines the care that must be taken to avoid breathing the fumes. Ingestion toxicity indicates the hazard in case of accidental swallowing of the vapor or liquid. Skin toxicity refers to the hazard connected with fuel coming in contact with skin. Ammonia can be very hazardous because of its vapor toxicity. Its strong smell, however, will usually provide ample warning of its presence. With liquified ammonia, hydrogen, and methane, skin contact must be avoided. Coal liquids are expected to contain a high concentration of aromatic hydrocarbons, such as benzene. In view of the toxic nature of benzene and aromatics in general, the coal derived fuels may require much more careful handling than the current petroleum fuels.

From a fire hazard standpoint, Diesel No. 2 is a relatively safe fuel, considerably safer, for instance, than gasoline. Some of the alternative fuels present special problems with respect to fire which are not only a danger to the refueling crews and the train crews but to the locomotive and refueling terminal as well. The vapor pressure of the fuel, the flammability limits, the ignition temperature, flash point and electrical conductivity are all involved in the evaluation of the fire hazard. The last item, electrical conductivity, can be used as an example of the rating of the fire hazard. A static electric charge can accumulate on a liquid when it flows at high speeds through lines, pipes and tubes. Under certain conditions, the charge may jump to the conductor such as a grounded tank wall as a spark capable of igniting the fuel. Static electric generation varies inversely with electrical conductivity and is at a maximum with materials having conductivity in the range of 10^{14} to 10^{18} ohm-centimeters. The electrical conductivities of liquid methane and liquid hydrogen are below this range while the conductivity of ammonia is above this range. However, many of the fuels, particularly the petroleum and synthetic hydrocarbons, are within this range.

The fuels are ranked in Tables 6-22 and 6-23. The highest ranked fuel on Table 6-22 is cottonseed oil. The second ranked fuel is lube stock. Ammonia is ranked in last place. Among the slurries, emulsions and blends, on Table 6-23, there is a twelve-way tie for the top ranking. In this group, there is a narrow range of scores indicating that most of these fuels behave similarly as far as maintenance and safety is concerned.

K. FINAL RANKING OF THE FUELS

The scores from each of the four functional groups are totaled for all of the fuels in Tables 6-24 and 6-25. The highest ranked fuel in Table 6-24 is the petroleum based broadcut fuel oil. The second highest ranked fuel and the highest ranked non-petroleum fuel is the oil shale distillate for the second place in the overall ranking. The lowest ranked fuel in this comparison is ammonia. Among the slurries, emulsions and blends, in Table 6-25, 50% gasoline in Diesel No. 2 is ranked highest with 30% gasoline in lube oil ranked second. The highest ranking emulsion is 10% water-6% emulsifier in Diesel No. 2 with an overall ranking of third. The best slurry fuel was 20% coal in Diesel No. 2 with a seventh place ranking overall. The lowest ranking in this class of fuels went to the 30% methanol-30% surfactant emulsion.

Table 6-21. Ranking of Fuel Slurries, Emulsions and Blends Relative to Diesel No. 2 in Logistics and Storage

	Future Costs	Future Avail-ability	Water Reac-tivity	Tank Volume	Special Tankage	Stability	Score	Rank
Weighing Factor	7	8	2	3	4	3		
Diesel Fuel Slurries								
20% Carbon Black	0	0	0	0	0	-2	-6	11
20% Petroleum Coke	0	0	0	0	0	-1	-3	10
20% Flour	0	0	0	-1	0	-3	-12	14
20% Cornstarch	0	0	0	-1	0	-3	-12	14
20% Wood Fibers	+1	0	0	-1	-1	-2	-6	11
20% Coal	+1	0	0	0	0	-1	4	2
Diesel Fuel Emulsions								
10% Water-6% Emulsifier	0	0	+3	0	0	-1	3	3
30% Water-10% Emulsifier	+1	0	+3	-1	0	-1	7	1
10% Methanol-10% Surfactant	-1	0	-1	0	0	-1	-12	14
30% methanol-30% Surfactant	-2	0	-1	-1	0	-1	-22	18
10% Ethanol-4% Surfactant	-1	0	-1	0	0	0	-9	13
30% Ethanol-12% Surfactant	-2	0	-1	-1	0	0	-19	17
Petroleum Blends								
30% Gasoline in Lube Oil	0	0	0	0	0	0	0	4
70% Gasoline in Lube Oil	0	0	0	0	0	0	0	4
64% Heavy Aromatic Naphtha in Diesel	0	0	0	0	0	0	0	4
35% Heavy Aromatic Naphtha in Diesel	0	0	0	0	0	0	0	4
50% Gasoline in Diesel No. 2	0	0	0	0	0	0	0	4
80% No. 6 Fuel Oil in Diesel No. 2	0	0	0	0	0	0	0	4

Table 6-22. Ranking of Fuels Relative to Diesel No. 2 in Maintenance and Safety

	Special Equipment	Compati-bility	Lube Reactivity	Toxicity	Fire Hazard	Score	Rank
Weighing Factor	5	2	2	5	2		
Gasoline	-2	0	0	0	-2	-14	15
Naphtha	0	0	0	0	-1	-2	7
Light Distillate	0	0	0	0	0	0	3
Lube Stock	0	0	0	0	+1	+2	2
Broadcut Fuel Oil	0	0	0	0	-1	-2	7
Jet A	0	0	0	0	-1	-2	7
Kerosene	0	0	0	0	-1	-2	7
Methane	-3	+1	+1	0	-2	-15	16
Butane	-3	+1	+1	0	-2	-15	16
Propane	-3	+1	+1	0	-2	-15	16
No. 4 Fuel Oil	0	0	0	0	0	0	3
No. 5 Fuel Oil	-1	0	0	0	0	-5	12
No. 6 Fuel Oil	0	0	0	0	0	0	3
Methanol	-2	-1	-1	-2	-1	-26	22
Ethanol	-2	0	-1	-1	-1	-19	19
Powdered Coal	-3	+3	-1	-2	0	-21	20
Coal Derived Distillate	0	0	0	0	0	0	3
Coal Derived Gasoline	-2	0	0	0	0	-10	13
Oil Shale Distillate	0	0	0	0	-1	-2	7
Oil Shale Gasoline	-2	0	0	0	0	-10	13
Hydrogen	-3	-3	-2	-2	-2	-39	21
Ammonia	-2	-3	-3	-3	-1	-38	23
Cottonseed Oil	0	0	0	+2	+2	+14	1

Table 6-23. Ranking of Fuel Slurries, Emulsions and Blends Relative to Diesel No. 2 in Maintenance and Safety

	Special Equipment	Compat-ibility	Lube Reactivity	Toxicity	Fire Hazard	Score	Rank
Weighing Factor	5	2	2	5	2		
Diesel Fuel Slurries							
20% Carbon Black	0	0	0	0	0	0	1
20% Petroleun Coke	0	0	0	0	0	0	1
20% Flour	0	0	0	0	0	0	1
20% Cornstarch	0	0	0	0	0	0	1
20% Wood Fibers	0	0	0	0	0	0	1
20% Coal	0	-3	0	0	0	-6	17
Diesel Fuel Emulsions							
10% Water-6% Emulsifier	0	-1	-1	0	0	-4	16
30% Water-10% Emulsifier	0	-1	-1	0	0	-4	16
10% Methanol-30% Surfactant	0	0	0	0	0	0	1
30% Metahnol-30% Surfactant	C	-1	-1	-1	-1	-11	18
10% Ethanol-4% Surfactant	0	0	0	0	0	0	1
30% Ethanol-12% Surfactant	0	0	0	0	-1	-2	13
Petroleum Blends							
30% Gasoline in Lube Oil	0	0	0	0	0	0	1
70% Gasoline in Lube Oil	0	0	0	0	-1	-2	13
64% Heavy Aromatic Naphtha in Diesel	0	0	0	0	0	0	1
35% Heavy Aromatic Naphtha in Diesel	0	0	0	0	0	0	1
50% Gasoline in Diesel No. 2	0	0	0	0	-1	-2	13
80% No. 6 Fuel Oil in Diesel No. 2	0	0	0	0	0	0	1

Table 6-24. Final Ranking of Fuels Relative to Diesel No. 2

Fuel	Combustion	Engine Operation	Logistics and Storage	Maintenance and Safety	Total Score	Final Rank
Gasoline	-13	-1	0	-14	-28	12
Naphtha	-6	+4	+3	-2	-1	6
Light Distillate	0	+4	+3	0	7	2
Lube Stock	-26	+9	-3	+2	-18	11
Broadcut Fuel Oil	0	0	+15	-2	13	1
Jet A	0	0	+3	-2	1	5
Kerosene	0	+3	+3	-2	4	3
Methane	-1	-14	-7	-15	-37	15
Butane	+3	-6	+4	-15	-14	9
Propane	+5	-6	+4	-15	-12	8
No. 4 Fuel Oil	-11	-2	-3	0	-16	10
No. 5 Fuel Oil	-33	-13	-3	-5	-54	16
No. 6 Fuel Oil	-44	-11	0	0	-55	17
Methanol	-14	-20	-4	-26	-64	18
Ethanol	-14	-20	-12	-19	-65	19
Powdered Coal	-60	-34	+44	-21	-71	20
Coal Derived Distillate	-11	-3	+13	0	-1	6
Coal Derived Gasoline	-12	-4	-5	-10	-31	13
Oil Shale Distillate	-11	-3	+20	-2	+4	3
Oil Shale Gasoline	-12	-4	-5	-10	-31	13
Hydrogen	-10	-41	-13	-39	-103	22
Ammonia	-17	-41	-27	-38	-123	23
Cottonseed Oil	-37	-11	-51	+14	-85	21

L. WAYS OF USING THE ALTERNATIVE FUELS

In this section, the use of alternative fuels is concerned not with how these fuels react in the engine but rather how the railroads themselves can use them. There are three basic ways to use these fuels; as emergency, extender and replacement fuels.

Emergency fuels are intended for short term usage in the absence of the regular fuel. The regular fuel supply is expected to resume in the near future and something is needed to tide the railroads over. Work on emergency fuels is underway at Southwest Research Institute. Bailey and Russell (Reference 6-1) report on progress in this area. Work by Tyler et al, also at Southwest Research, on hybrid fuels is applicable to emergency fuels (Ref. 6-2).

In general, emergency fuels will be blends of whatever fuels are available at the time and in proportions which will yield acceptable cetane ratings, viscosity and lubricity. A cetane rating greater than 30 is necessary for satisfactory operation. The viscosity should be less than 30 centistokes at 100° F and the lubricity should be high enough to prevent significant wear in the pumps and injectors. A small amount (10%) of lubricating oil (new or used, but clean) or 20% Diesel No. 2 will provide adequate lubrication. The use of alcohols, gasoline, naphtha or Jet A fuel will lower the viscosity of heavy oils to a level where they can be pumped and injected in a satisfactory manner.

Because of its short term usage, little or no engine or locomotive modification should be required for an emergency fuel. The cost and time required to convert to the emergency fuel and to convert back again once the regular fuel is again available is unwarranted in all but the most serious conditions. Generally, proper blending of the fuel will eliminate any need for modifications. Experience with residual fuels in locomotives indicates that combustion chamber damage may occur if a heavy fuel is used for any great length of time. These heavy fuels can be used for the base of an emergency fuel but their use for longer periods of time must be carefully monitored. The new Sulzer Diesel engines used by Morrison-Knudsen were designed for heavier fuels and, in these engines, the use of residual based fuels for longer periods is warranted.

Petroleum blends like those in Table 6-25 are well suited to use as emergency fuels. The gasoline-Diesel fuel blend is the highest ranked with the gasoline-lube oil blend next. There are a number of other blends which could be tried such as No. 6 fuel oil-gasoline blends and medium naphtha with No. 5 or No. 6 fuel oil or lube stock.

The purpose of an extender fuel is to stretch the supply of Diesel No. 2 when it is inadequate. In general, the use of extenders may be either short- or long-term. One requirement of an extender fuel is that it can be mixed with Diesel No. 2 and that a stable mixture results. In some cases, such as methanol, an emulsifier is needed. All of the petroleum products will mix readily as will the coal derived and oil shale derived synthetic hydrocarbon fuels. The cetane rating and the lubricity will be adequate because of the presence of the Diesel No. 2. Unless the extender is a very heavy oil, viscosity should not present any serious problems.

Table 6-25. Final Ranking of Fuel Slurries, Emulsions and Blends Relative to Diesel No. 2

Fuel	Combustion	Engine Operation	Logistics and Storage	Maintenance and Safety	Total Score	Final Rank
Diesel Fuel Slurries						
20% Carbon Black	-22	-15	-6	0	-43	17
20% Petroleum Coke	-11	-9	-3	0	-23	10
20% Flour	-6	-14	-12	0	-32	13
20% Cornstarch	-6	-14	-12	0	-32	13
20% Wood Fibers	-11	-19	-6	0	-36	15
20% Coal	-16	-1	+4	-6	-19	6
Diesel Fuel Emulsions						
10% Water-6% Emulsifier	-6	-3	+3	-4	-10	2
30% Water-10% Emulsifier	-9	-15	+7	-4	-19	6
10% Methanol-30% Surfactant	-12	0	-12	0	-24	12
30% Metahnol-30% Surfactant	-7	-13	-22	-11	-53	18
10% Ethanol-4% Surfactant	-12	0	-9	0	-21	8
30% Ethanol-12% Surfactant	-7	-13	-19	-2	-41	16
Petroleum Blends						
30% Gasoline in Lube Oil	-10	-1	0	0	-11	3
70% Gasoline in Lube Oil	-11	-10	0	-2	-23	10
64% Heavy Aromatic Naphtha in Diesel	-12	0	0	0	-12	4
35% Heavy Aromatic Naphtha in Diesel	-12	0	0	0	-12	4
50% Gasoline in Diesel No. 2	0	-5	0	-2	-7	1
80% No. 6 Fuel Oil in Diesel No. 2	-16	-6	0	0	-22	9

Because extenders may be used for a longer period of time than the emergency fuels, the requirements for engine compatibility are greater. Combustion chamber damage cannot be tolerated but more extensive modifications to accommodate the fuel can be permitted. Changes to fuel tanks, injectors, pumps, controls and fueling facilities may be made. The use of extenders would pave the way for the changeover to oil shale or coal derived liquid fuels. In the early years, the replacement fuel would be used as an extender to Diesel No. 2 and as its supply increases and that of petroleum decreases, its percentage in the blend would increase until it dominates. The Diesel No. 2 can now be thought of as the extender. The amount of Diesel No. 2 could decrease until it is no longer used at all and the conversion to the alternate replacement fuel is complete. This scenario allows engine and locomotive changes to be made as necessary as the amount of the replacement fuel is increased.

The two petroleum blends in Table 6-25 that use heavy aromatic naphtha are similar to an oil shale distillate-Diesel fuel blend that could be used as an extended fuel. The emulsions that use 10% water, methanol or ethanol are also examples of extended fuels which show some promise.

A replacement fuel is a non-petroleum fuel that is used as a total replacement in the locomotives for Diesel No. 2. The changeover may take place over a period of time as was just described or it could take place abruptly. The changes necessary to the engine and locomotive are extensive for alcohols, hydrogen, or ammonia. These fuels will not permit a gradual changeover. Fueling equipment and storage facilities must be provided even before the locomotives are converted. Because it is not possible to change all of the locomotives in a railroad fleet at once, duplicate fueling equipment will be needed. One approach in this case would be to convert on a route-by-route basis.

The most attractive alternative replacement fuel for the medium-speed Diesel is the oil shale distillate. Its properties are similar to those of Diesel No. 2. It requires a minimum of engine or locomotive modification and can be used as an extender before it replaces the petroleum fuels entirely. The other fuel that is a likely replacement is the coal derived distillate. This fuel, like the oil shale distillate, can be used in a gradual replacement program by using it initially as an extender fuel.

In general, the greater the differences between the properties of Diesel No. 2 fuel and those of an alternative fuel, the more extensive the changes that will be required to adapt the railroad system to its use. The Diesel engine, however, must be considered as a multi-fuel engine and the range of fuels that can be used is very broad. The railroads are not limited to any one fuel although the use of a single fuel has provided the them with high flexibility, safety, and performance levels.

M. SECTION VI REFERENCES AND NOTES

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SECTION VII
RANKINE CYCLE (STEAM) ENGINES

A. INTRODUCTION

In 1804, Richard Trevithick demonstrated his crude steam locomotive on a plate-way in England. In the next 25 years, there were many attempts to produce a practical steam locomotive. In 1829, George and Robert Stephenson produced their locomotive, the "Rocket", which is the ancestor of all later steam locomotives. Many of the features of the "Rocket" are found on locomotives produced as late as 1950. In the U.S., the steam era started with the "Tom Thumb" in 1829. Regular railroad service in the U.S. began in 1831 on the South Carolina Railroad and, later the same year, on the Mohawk and Hudson Railroad (Ref. 7-1). Steam dominated the railroads for the next 120 years but by 1951, the Diesel-electric locomotive was rapidly displacing it. Within a few more years, steam was essentially extinct.

Today, with the rapidly escalating cost of petroleum, it is time to re-evaluate the use of coal. The steam engine is an obvious means of using coal as a fuel for transportation. There have been a number of changes in steam power since the demise of the old steam locomotive. The Rankine cycle is characterized by the fact that the working fluid undergoes phase changes during the cycle and any fluid with a suitable boiling temperature could be used. Halogenated hydrocarbon (Freon-type) working fluids have been used for some experimental "steam" automobiles and for Rankine bottoming cycles to recover exhaust heat on trucks. Two of the halogenated working fluids as well as water were investigated earlier in this project in connection with exhaust heat recovery for locomotives.

Modern steam utility powerplants operate at much higher temperatures and pressures than the old steam locomotives. Even at the end of the steam era, the pressures were only 300 psi and most locomotives operated at steam pressures nearer to 200 psi. Early locomotives operated near the saturation line and it was not until the 1920s that superheating became common. Towards the end of the steam era, 300° to 350° F of superheating was being used (Ref. 7-1). At 300 psi, the steam temperature is 775° F for 350° F of superheat. At 200 psi and 300° F of superheat, the steam temperature is 680° F. When these figures are compared to modern electrical powerplant steam conditions of 1000° F and 1800 to 2500 psig (or up to 3500 psig for supercritical units), it is obvious that the steam locomotives were running at very conservative conditions. Further, the electrical powerplant is exhausted to a sub-atmospheric condenser while the steam locomotive was exhausted to the atmosphere. Therefore, the thermal efficiency of the locomotive was only 5 to 8% as compared to the 32 to 35% of the utility steam systems. Another characteristic of old steam locomotives that seriously degraded their efficiency was the incomplete utilization of the coal. It was not unusual for 40% to 50% of the coal to be blown out of the furnace and lost up the stack.

There were good reasons for the low temperatures used on locomotives. The slide valves and cylinders used on these engines had to be lubricated and most lubricants break-down if the temperatures are too high. The poppet steam valve came into use because of the tendency for slide valves to score at high temperatures. Very late in the steam era, the turbine was tried but it was not sufficiently developed to stop or delay the advent of the Diesel-electric locomotive with its 30% thermal efficiency rating. Historically, the cost of Diesel fuel has been about three times that of coal on an energy basis ($/million Btu). The steam locomotive would have needed a thermal efficiency of at least 10% to just match the fuel cost of the Diesel. Today, with the cost of Diesel fuel energy being over four times that of coal energy as shown in Figure 7-1, the steam engine would still have to be at least 10% efficient since the thermal efficiency of the modern Diesel engine is about 40%. Therefore, the old style steam locomotive is still not economically justified in today's world.

The cost per unit of energy and the availability of coal make it an attractive fuel. The question is whether or not the Rankine cycle, using steam or another working fluid, is a viable locomotive engine for use with coal. The answer might very well be "yes" if a number of conditions can be met. These are:

(1) The system efficiency of the locomotive must be at least 15%.

(2) Smoke, soot and cinder emissions from the engine must be no more than the smoke from present-day Diesel engines.

(3) The locomotive must be compatible with current railroad operating practices including multiple unit operation.

(4) Support and maintenance requirements of the locomotive should be comparable to current Diesel engines.

(5) New facilities such as water tanks, coaling depots, and ash disposal sites should be minimized and should be adjacent, whenever possible, to existing railroad facilities.

(6) Technical and economic feasibility

Actually, this list is applicable to all alternative engines, not just Rankine cycle engines. The minimum value for the thermal efficiency (Item 1) will vary with the fuel. An alternative engine that uses Diesel No. 2 fuel will have to be at least as efficient as a Diesel engine to be able to compete economically. The 15% value in Item 1 is the minimum for coal based on national average prices for 1979 (Ref. 7-2).

The thermal efficiency of even a very advanced, high technology Rankine cycle engine is not likely to be greater than 25% based on this analysis. This is due to the physical limitations imposed by the locomotive and its operating environment. The fuel cost, therefore, must be quite low as compared to Diesel No. 2. The list of fuels is quite limited and includes coal, lignite, petroleum coke, high sulfur heavy oils, and crude oil. The most likely candidate is coal and the analysis of the Rankine cycle engine made for this project assumes its use.

Coal is a complex fuel whose physical and chemical properties vary greatly, depending on the region in which it is mined. Of particular interest is the heating value and the ash and sulfur content. Most U.S.

coals have a sulfur weight-percent content in the 0.5 to 4.5% range. The weight-percent ash content is in the 4 to 15% range for bituminous coals and up to 20% for Virginia anthracite. Heating values are in the 12,000 Btu/lb range (Ref. 7-3). The sulfur normally appears as SO_2 in the exhaust gas of conventional furnaces when coal is burned with excess air. For coal with a 1% sulfur content, 0.02 lb of SO_2 is produced for every pound of coal burned. The ash can end up as a molten slag, an agglomerated clinker, or as a dry powder depending on the type of furnace used.

Emission standards that would apply specifically to a coal-fired locomotive do not yet exist. However, the emission standards for new stationary sources can serve as a guide to possible locomotive standards of the future. These standards are available in the Code of Federal Regulations (Ref. 7-4), and apply specifically to electric utility steam generating

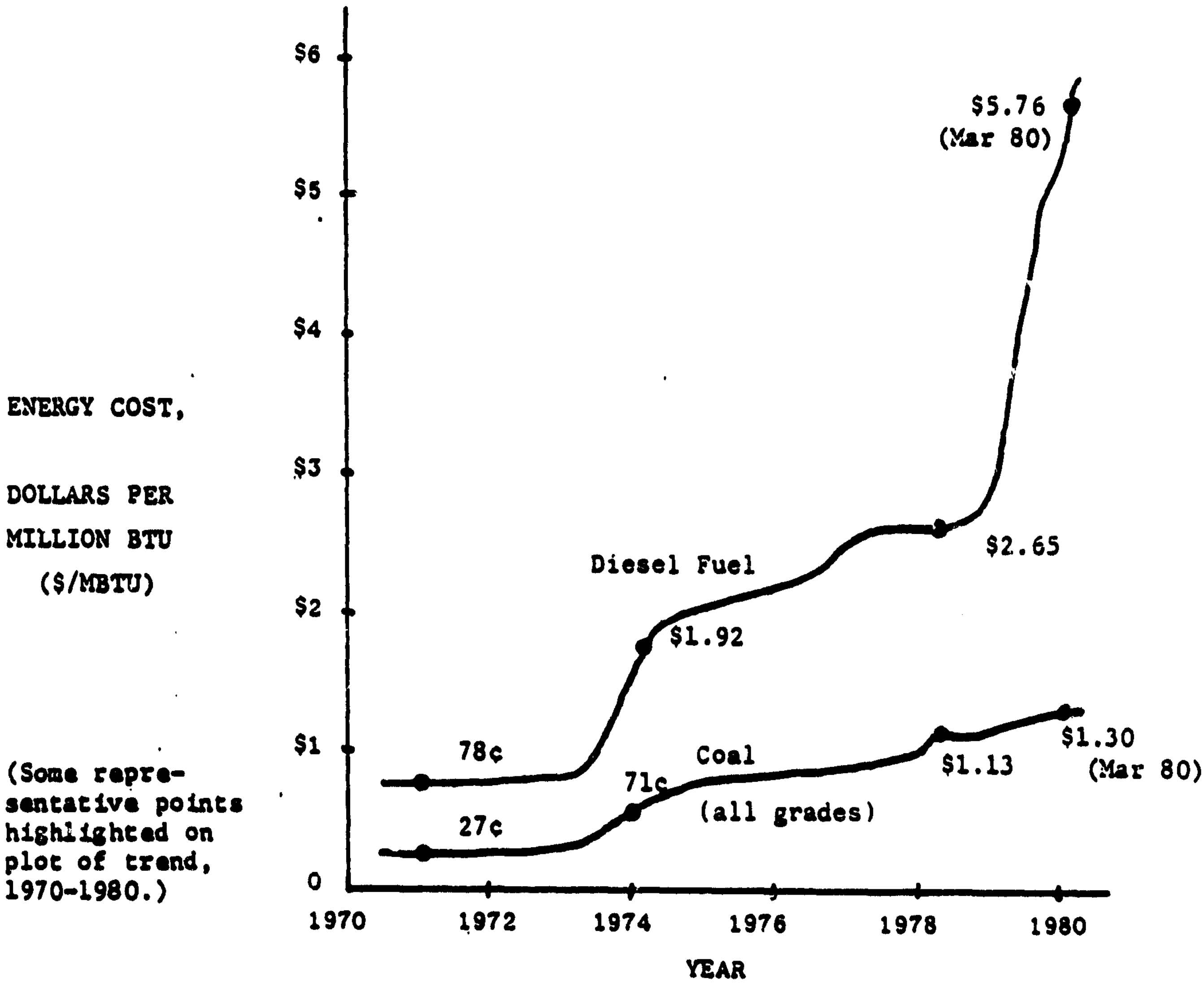

Figure 7-1. Average Energy Cost Comparison for Coal and Diesel Fuel (From Ref. 7-2)

units capable of firing at least 250 lb per million Btu/hr of fossil fuel input. By comparison a 3000 hp locomotive would have an input of approximately 50 lb/MBtu/hr. The emission standards in Ref. 7-4 are:

(1) SO_2 Standards for Coal-Fired Plants

(a) 1.2 lb per million Btu maximum emission, <u>and</u> a 90% reduction in potential emission (70% reduction is satisfactory if total emission is less than 0.6 lb per million Btu).

(b) Emission computed by 30-day rolling average (Anthracite coal is exempt from the percent-reduction requirement).

(2) Particulate Matter; 0.03 lb per million Btu

(3) NO_x Standards; 0.6 lb per million Btu emissions computed by using 30-day rolling average.

The design of the furnace has a strong influence on the emissions. Although specific standards are not in force now, it must be expected that if a significant number of coal fired locomotives are in use, then standards will be imposed on them.

The basic steam system for an advanced locomotive is shown in Figure 7-2. There are a number of variations due to the use of different components. Several different furnace/boilers could be used as well as at least two types of expanders. Before analyzing the system as a whole, each of the major components will be discussed.

B. FURNACES

There are three basic forms of furnaces considered in this study. They are the pulverized fuel furnace, the stoker-grate furnace, and the fluidized bed furnace. All three are used in stationary power plants. The stoker-grate furnace with a fire tube boiler is the one that was used on the old steam locomotives. The choice of the furnace type depends not only on their combustion characteristics but also on the effects of the ash within the furnace and its method of disposal.

The pulverized fuel furnace is widely used in thermal electric power generating plants where pulverized coal is injected into the boiler using a steam jet. The coal burns in a flame much as an oil spray would. Often these furnaces can be converted from coal to oil with a minimum of effort. The coal flame is luminous and heat transfer to the water tubes is accomplished partly by convection and partly by radiation. In the radiant portion of a conventional pulverized fuel furnace, a portion of the ash collects on furnace walls forming a slag which runs down the walls and is collected at the bottom of the furnace. The remaining portion is entrained in the flue gas and is cooled below the ash fusion temperature as it passes into the convective region. Fouling of heat transfer assemblies can occur if the ash is still sticky or when non-combustible volatile coal components condense and capture the ash. Furnace volume, spacing of tubes, and

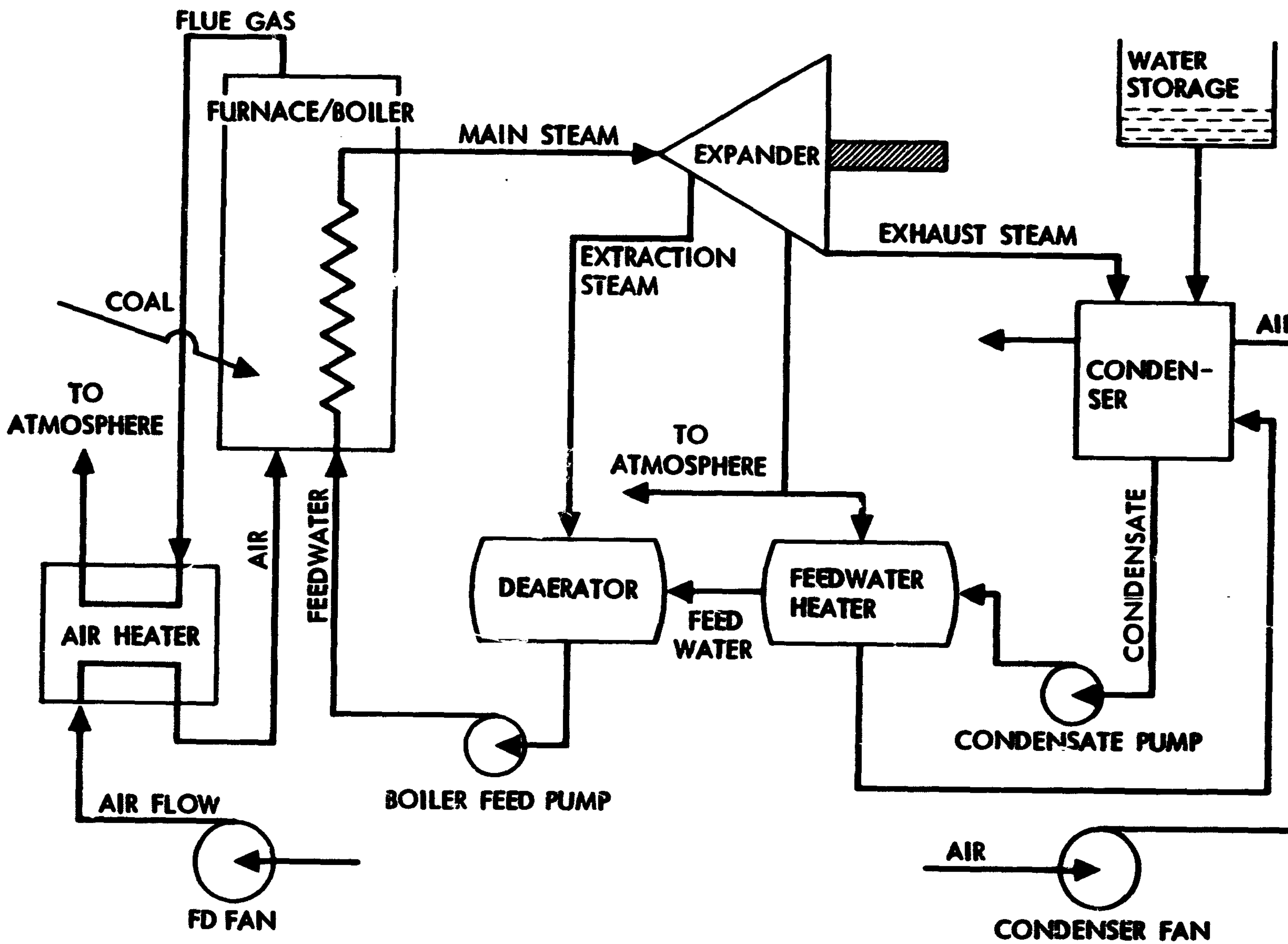

Figure 7-2. Steam Cycle for Coal-Fired Locomotive
(From Ref. 7-5)

the fraction of ash collected as slag or entrained in flue gas are all determined in design trade-offs related to specific coal properties.

Table 7-1 summarizes the ash content, fusion temperatures, and ash analysis for a number of typical coals (Ref. 7-3). The chemical composition of the coal ash varies widely with the location of the mine. However, the eastern coal ashes are generally high in silica and ferric oxides, and the western coal ashes are high in silica and limestone. Aluminum oxide is common in both areas of the country. The ash softening and the fluid temperatures vary with the type of coal. They are lowest for the lignites and highest for the low volatile bituminous coal. The amount of ash and sulfur found in the coal varies widely. No. 6 Illinois coal contains about 17% ash and 4% sulfur, and the Utah high volatile bituminous coal has only 6.6% ash and 0.5% sulfur. The volume and type of ash must be taken into consideration in designing the disposal system for a locomotive.

In a pulverized fuel furnace, the ash is released from the coal in mid-air and in or near the flame. Fly ash is particularly severe in this type of furnace and extensive exhaust gas clean-up is necessary. The ash disposal problem does not make this type of furnace attractive for locomotives. In addition, it is necessary to either provide pre-pulverized coal to the tender or to have a pulverizer on board. Neither arrangement is attractive.

A second type of furnace is the stoker-grate. On the grate of a well-controlled, stoker furnace burning larger pieces of coal in a medium to thick fuel bed, the temperature is adjusted so that the ash tends to stick together to form large agglomerates. These agglomerates sink to the bottom of the bed and are carried away by a traveling grate or they fall through the grate into the pit below. Agglomeration of the ash simplifies the problem of ash disposal considerably and is, therefore, highly desireable. The fuel bed temperature is controlled by adjusting the amount of combustion air. If the combustion air flowing through the bed is at or near the amount necessary for stoichiometric combustion, the bed temperature is high and the ash melts. Increasing the air supply will lower the bed temperature. When the temperature is low enough, the ash will form agglomerates and combustion efficiency will be quite low. There will also be a considerable amount of fly ash and unburned carbon particles in the exhaust gas.

Another means of controlling bed temperature is to reduce the air flow through the bed to less than the stoichiometric amount. The bed acts as a gasifier in this case and combustion is completed by injecting air over the top of the bed. The relatively low air velocity through the bed also tends to minimize the fly ash and soot problems. The fouling of the heat transfer surfaces is minimized and the exhaust gas cleanup necessary for environmental reasons is greatly reduced when compared to any of the combustion systems discussed earlier. A bed temperature of about 2000° F is needed to meet ash agglomeration requirements. The thick bed system operating in a gasifier mode is the most attractive of the stoker-grate furnaces.

The fluidized bed combustion system or furnace has been discussed in earlier chapters but is treated in more detail here. The average temperature of the fluidized bed combustion system can be maintained

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Table 7-1. Ash Content and Fusion Temperatures (From Ref. 7-6)

Ash content and ash fusion temperatures of some U.S. coals and lignite

Rank:	Low Volatile Bituminous	High Volatile Bituminous				Sub-bituminous	Lignite
Seam	Pocahontas No. 3	No. 9	Pittsburgh	No. 6			
Location	West Virginia	Ohio	West Virginia	Illinois	Utah	Wyoming	Texas
Ash, dry basis, %	12.3	14.10	10.87	17.36	6.6	6.6	12.8
Sulfur, dry basis, %	0.7	3.30	3.53	4.17	0.5	1.0	1.1
Analysis of ash, % by wt							
SiO_2	60.0	47.27	37.64	47.52	48.0	24.0	41.8
Al_2O_3	30.0	22.96	20.11	17.87	11.5	20.0	13.6
TiO_2	1.6	1.00	0.81	0.78	0.6	0.7	1.5
Fe_2O_3	4.0	22.81	29.28	20.13	7.0	11.0	6.6
CaO	0.6	1.30	4.25	5.75	25.0	26.0	17.6
MgO	0.6	0.85	1.25	1.02	4.0	4.0	2.5
Na_2O	0.5	0.28	0.80	0.36	1.2	0.2	0.6
K_2O	1.5	1.97	1.60	1.77	0.2	0.5	0.1
Total	98.8	98.44	95.74	95.20	97.5	86.4	84.3
Ash fusibility							
Initial deformation temperature, F							
Reducing	2900+	2030	2030	2000	2060	1990	1975
Oxidizing	2900+	2420	2265	2300	2120	2190	2070
Softening temperature, F							
Reducing		2450	2175	2160		2180	2130
Oxidizing		2605	2385	2430		2220	2190
Hemispherical temperature, F							
Reducing		2480	2225	2180	2140	2250	2150
Oxidizing		2620	2450	2450	2220	2240	2210
Fluid temperature, F							
Reducing		2620	2370	2320	2250	2290	2240
Oxidizing		2670	2540	2610	2460	2300	2290

below the ash softening point by the removal of heat from the bed using immersed heat exchangers. Thus, the ash that is entrained in the flue gas is dry. It is normally collected in cyclones, and can be recycled through the bed to improve the efficiency of the carbon utilization. A portion of the ash can also remain in the bed to form ash agglomerates which are removed with spent bed material. By keeping the ash dry, fluidized bed furnaces result in much cleaner coal combustion without slagging and fouling. Another advantage of fluidized bed combustion is that most of the sulfur dioxide is absorbed by the limestone or dolomite bed material.

Fluidized bed combustion technology promises a method of coal combustion which will comply with all EPA emission standards for large stationary plants and which is suitable for burning all types of coal. Combustion in a thick fuel bed operating in a gasifier mode with significant over-bed burning is also an attractive method of coal combustion that will offer environmentally acceptable performance. The relative merits of each approach will be greatly influenced by emission standards. If there is some relaxation in the sulfur dioxide standards, the gasifier fuel bed approach would be very practical. If the emission of fly ash and other fine particles proves to be a problem, the gasifier fuel bed may also be more advantageous because of its ash agglomerating feature (Ref. 7-5).

C. EXPANDER

The traditional expander for steam locomotives is the reciprocating drive system. The majority of the U.S. steam locomotives utilized two simple outside mounted cylinders. In the 1800s, these cylinders usually had slide valves. Piston valves replaced the slide valves in the 1900s. Early attempts at compound arrangements were not generally accepted because of their increased complexity and maintenance. In the 1920s, there was brief interest in three-cylinder simple drives but valve complexity kept them from being accepted. The four-cylinder compound system was used on the Mallet type locomotives. The four-cylinder simple system was used on many of the largest locomotives while the two-cylinder simple system was typical of the smaller engines (Ref. 7-1).

Toward the end of the steam era, several locomotives were built using turbines as the steam expander. Steam turbines are widely used in marine and in electric utility applications. Their relatively small size, high reliability, and simplicity have made them the prime choice for high power level steam systems. Steam turbines for low power applications such as locomotives are not as well developed as those for utilities. They are not even as highly developed as gas turbines are for the same power level. The turbine for a 3000 hp gas turbine engine would have an adiabatic efficiency of 85 to 90% at design point. A modern steam turbine for the same 3000 hp output would be only 50 to 60% efficient. There has been little motivation in the past to develop highly efficient small steam turbines.

A steam turbine that is driving the alternator of an electric transmission is a more familiar type of operation for the railroads than a steam cylinder system would be. In fact, the alternator, rectifier, and dc motors would be the same as those used on the present Diesel-electric locomotives. However, when remembering that turbines tend to drop significantly in efficiency at part load, and that the peak efficiency is only 50 to 60%, it is likely that the reciprocating drive system is a better alternative until a steam turbine with a peak efficiency of at least 75% and preferably 85% is developed. The steam cylinder expander must be lubricated. If lubricant is applied at the piston rings in a manner similar to diesel engine practice, lubricant temperature will be equal to cylinder wall temperature (typically 400 to 500° F), and steam temperatures up to 900° F can be used. Use of dry lubrication (graphite rings) is being explored for both high temperature Diesel engines, as well as steam engines. This development will permit reciprocating expanders to operate at the same steam temperatures as steam turbines.

D. CONDENSER

American steam locomotive practice utilized an open steam cycle with the exhaust steam from the cylinders used in an air ejector system to provide furnace draft. This system provides a simple feedback control between the steam flow and the furnace heat production rate. Because about 8 lb of water was required per pound of coal burned, the tender was designed with a large "U" shaped water tank surrounding the fuel space. Even with this arrangement, the common practice was for two water stops for every fuel stop, except where water could be scooped from a track pan as was common on some of the busier lines.

After the conversion of the locomotive fleet to Diesel engines, the watering facilities were dismantled or simply abandoned. If the railroads were to return to non-condensing steam engines, these facilities would have to be rebuilt. If a working fluid other than water is used, then a condenser must be used. The cost of all other working fluids is too high to use them. The use of a condenser with a steam system would eliminate having to replace the watering facilities except at the fueling depots which could be in existing yards. The condenser also allows the expander to exhaust to a vacuum, thus increasing the expansion ratio and, hence, the thermal efficiency. Several examples presented later will show how much of an effect the condensing system has on thermal efficiency.

On a locomotive, the waste heat has to be rejected to the ambient air. There are several means of accomplishing this transfer of heat from the condensing steam to the outside air. Two methods appear promising. One is a simple condensing liquid-to-air surface condenser of finned tube construction. This type can be used for steam and for any other working fluid. The second method uses a jet condenser with the water cooled in conventional radiators.

The use of condensers on steam locomotives has not had a very successful history. They have been tried but the added maintenance, weight, and space problems have prevented their general acceptance. Most railroads already had watering facilities along their tracks so that condensers did not really provide any significant advantages. Today, the situation is different. If steam locomotives are expected to operate with Diesels in multiple unit operations, they will need to be equipped with partial or full condensing systems.

The other components of the steam system shown in Figure 7-2 are essentially stock items found in nearly all steam systems, large and small. The two fans use a lot of power and should be chosen for high efficiency. The forced draft (FD) fan uses nearly 2% of the engine output power and the condenser fan uses nearly 10% of the output. The fans could be variable speed so that the fan load would vary with engine output in order to keep part load system efficiency as high as possible.

E. ADVANCED STEAM LOCOMOTIVES

A new steam locomotive could take a variety of forms depending on the choice of the components, on the temperature and pressures of the steam, and on the logistics of the fuel and water supply. The water supply is important because steam is likely to be the working fluid for at least the first two generations of new Rankine cycle locomotives. The halogenated hydrocarbon (Freon-type) working fluids appear to provide higher efficiencies in many cases but they are expensive and the amount needed in a 3000 to 5000 hp locomotive would be about 1000 lb.

The first generation of new steam locomotives will rely heavily on proven technology, particularly in the area of industrial coal burning furnaces. It may use the thick bed coal gasifier furnace with a moving grate. The boiler could be a fire tube type of welded construction. The expander may be either a two or four-cylinder reciprocating design depending on the size of the locomotive. It is likely to be a partial or full condensing cycle with surface condensers; however, a jet condenser is possible. This coal-fired locomotive will be used initially for unit-train service in either the western coal fields or in the West Virginia area. A prototype locomotive could be built in two to three years with commercial production in about five years.

A new coal-fired steam locomotive has been proposed by the American Coal Enterprises which contains most of these features. It also contains some special features such as modularized coal and ash handling. The furnace is stoker fed and uses the thick bed gasifier type of combustion. The system efficiency of the A.C.E. 3000 locomotive is expected to be about 15% using a fully condensing steam cycle. It is designed to be fully compatible with Diesel locomotives in multiple unit operation.

The next generation of steam locomotives will require considerable research and development. Rather than take a broad look at many alternatives, it is more appropriate to take a detailed engineering look at just one design. This design embodies the following elements:

(1) Atmospheric fluidized bed combustor and boiler

   (a) Natural circulation, drum-type

   (b) Steam conditions 615 psia, 900° F

   (c) Forced draft fan

   (d) Electrically driven feedpump, condensate pump

(e) multicyclones for stack gas clean-up and fly ash reinjection

(f) forced air circulation for condenser cooling

(2) Steam turbine

(3) Electric transmission

(4) Condensation of steam

With respect to steam condensation, three variations will be considered: minimum condensation (recycling of drains from the feedwater heater only), partial condensation (approximately half of the steam flow is condensed, the other half vented to the atmosphere), and total condensation.

For the maximum probability of penetrating the locomotive market, which at present is totally dominated by Diesel-electrics, the coal-fired steam locomotive must be at least compatible with present operating and maintenance procedures. Major acceptability issues associated with performance and operational compatibility include:

(1) Operation in multiple unit (MU)

(2) Operation in pusher/helper service, or in a mixed locomotive train

(3) Double-ended (bi-directional)

(4) Size: height 15 ft 6 in, width 10 ft, and length 70 ft

(5) Rapid start-up/shut-down

It is not suggested in any way that the proposed atmospheric fluidized bed combustor, steam turbine-electric locomotive is an optimal design, nor one that is superior to other coal-fired locomotive designs which could incorporate other boiler configurations, gas turbine engines, hydraulic or reciprocating transmissions, and other components. Indeed, there are several fluidized bed combustor steam turbine-electric design concepts that can readily be explored and compared.

The key component in this design is the fluidized bed combustion system. It has been discussed before in connection with the Stirling engine and the closed cycle gas turbine. In this chapter, it will be covered in greater detail. This description is taken primarily from Ref. 7-5.

F. ATMOSPHERIC FLUIDIZED BED COMBUSTION

The components of a fluidized bed steam generator are shown schematically in Figure 7-3. The bed material is a granular solid, with a particle size that is typically in the range of 0.02 to 0.12 in. It could be an inert solid, such as sand, or as in the case of atmospheric fluidized bed combustion of medium or high sulfur coal, it could advantageously be chosen as a sorbent for sulfur dioxide, such as limestone or dolomite.

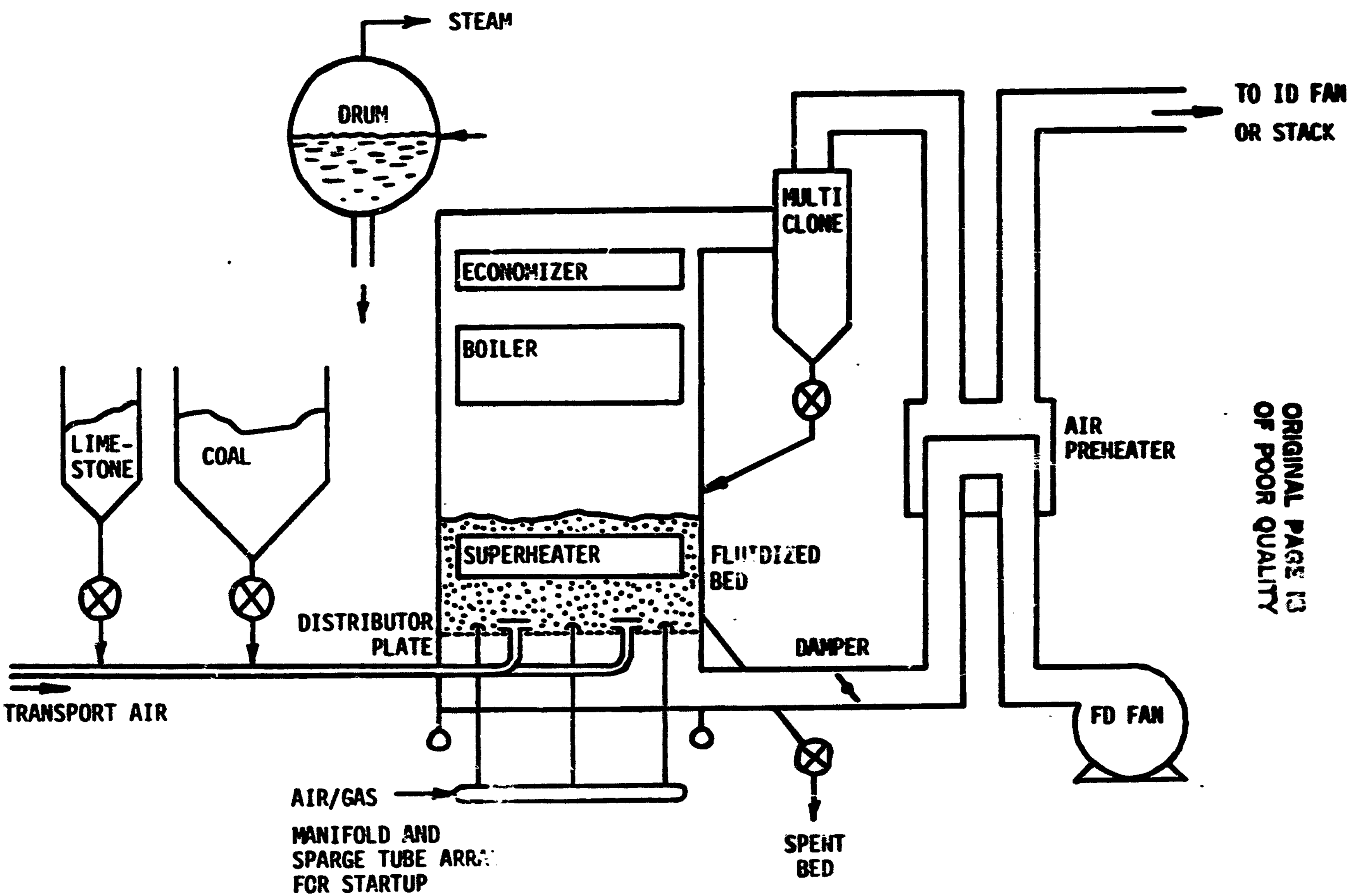

Figure 7-3. Simplified Schematic of an Atmospheric Fluidized Bed Combustion Steam Generator (From Ref. 7-5)

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The bed is fluidized by a gas stream flowing up through the bed at a flow rate sufficient to support the weight of the bed. Figure 7-4 is a typical fluidization characteristic in which the ratio of the pressure drop across the bed to the bed weight per unit area is plotted against superficial gas velocity. Superficial gas velocity is defined as the volume flow rate divided by the bed area. At the minimum fluidizing velocity, which is typically about 1 ft/sec, the static, granular bed "unlocks", the void fraction increases somewhat, and the bed assumes properties similar to a turbulent fluid. For further increases in superficial velocity, the pressure drop across the bed stays relatively constant, the void fraction increases slowly, larger voids tend to form and "float" to the surface much like bubbles, and the bed is in the so-called bubbling regime. At high superficial velocities, approximately 10 or 20 ft/sec, entrainment of bed particles occurs and the bed surface is no longer well defined (Ref. 7-7).

The fluidized bed is a medium with excellent contact between the gas and the solids (Ref. 7-7). This property can be exploited in order to improve the combustion of a solid fuel, for improving the rate of heat transfer to a metal surface, and for the chemical reaction of the gaseous and solid species.

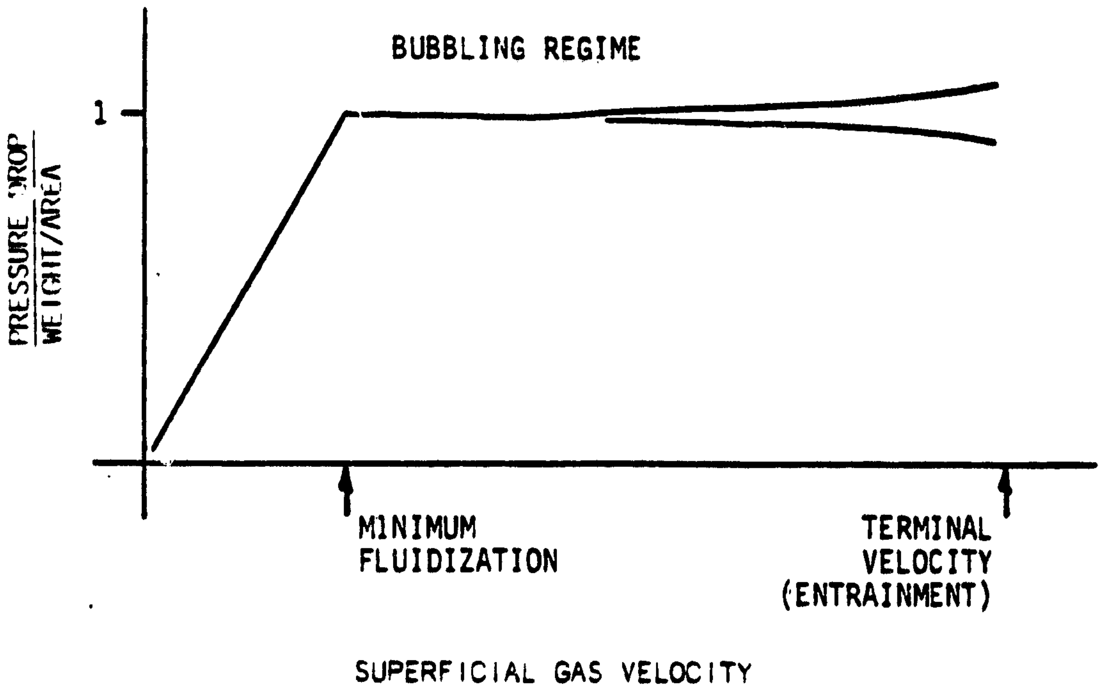

Figure 7-4. Typical Fluidization Characteristic
(From Ref. 7-5)

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In the fluidized bed combustion of coal, the bed is the combustion medium into which coarse coal is injected. The fluidizing gas is just the air required for combustion. Coal particles, which can be up to 1/2 in. or larger, reside in the bed until consumed. At any instant of time, the mass of coal in the bed is only 1 to 2% of the total bed mass. In continuous operation, the bed temperature is normally in the 1500 to 1600° F range, which is high enough for freshly injected coal to ignite readily. It is also low enough (below the ash softening temperature) for the ash to stay dry Therefore, much of the ash elutriates. A portion of it is carbon-bearing and can be collected and stored or reinjected into the bed for additional combustion.

The bed is well-mixed, however, care must be taken in the design and construction of the air distributor plate and furnace enclosure to guard against the channeling of air up through the bed or the creation of reducing regions. The air distributor can be simply a metal plate with an array of 1/16 in. holes on 1 in. centers. Other designs incorporating water-cooled furnace tube walls and button caps instead of holes will avoid problems of differential heating, plugging of distributor plate holes, and non-uniform air distribution.

Coal can be mechanically spread on top of the bed, or it can be injected into the bed pneumatically from the side or up through the distributor plate. Both systems have been used successfully. Limestone can be introduced separately or added into the coal stream. Typically one coal feed point is suitable for feeding 18 sq ft of fluidized bed (Ref. 7-8). Larger beds require multiple feed points and a suitable means for dividing the coal flow.

The bed temperature is selected primarily to optimize the conditions for the sulfation reaction in which sulfur dioxide that is generated during combustion combines with oxygen and calcined limestone (calcium oxide) to form calcium sulfate. This mechanism for sulfur dioxide absorption is most effective in the 1500° to 1600° F bed temperature range. The bed material is initially limestone which calcines to a porous matrix of calcium oxide. As sulfur dioxide absorption occurs, the bed particles are coated with calcium sulfate and any further absorption requires diffusion of sulfur dioxide and oxygen to the unreacted shrinking core surface. Thus, the bed particles become less effective with age, and eventually, are spent without all of the available calcium being fully utilized. Fresh limestone is added with coal in such an amount that the calcium-to-sulfur mole-ratio is approximately 3. This results in about an 85 to 90% reduction in the sulfur dioxide flue gas concentration in steady state operation. This corresponds to about 6.5 lbm of limestone for 100 lbm of coal for each percentage point of sulfur content by weight.

A typical sorbent particle will spend many hours in the bed and its absorbing effectiveness continually decreases in that time. At any instant of time, therefore, the bed consists primarily of spent bed particles. As fresh material and coal are added to the bed, the spent material is withdrawn. The so-called "dump" rate for the spent material is adjusted to maintain the bed mass or bed level at the desired value.

Spent bed material is a waste product of the combustion process. Several possible uses of it are being explored. Since it is withdrawn at bed temperature, it represents a small energy loss that could be partially recovered by preheating a water, air, or solid stream. There are also circumstances where reinjection into the bed is desirable. It is assumed that on-board storage will be provided for the spent bed, and that when coal, fresh limestone, and water are taken on, the spent bed will be removed.

To maintain the bed in the 1500 to 1600° F temperature range, it is necessary to continually remove heat energy. The heat transfer surface is immersed in the bed for raising steam and/or superheating. It could also be used for heating a gas. The gentle scrubbing action of the bed particles against the metal tube surfaces, combined with radiative and convective heat transfer, results in very high in-bed heat transfer coefficients - typically 40 Btu/(hr-ft^2-°F) for a horizontal surface, and 50 Btu/(hr-ft^2-°F) for a vertical surface. Experience has shown that as long as an oxidizing atmosphere is maintained in the bed, tube erosion is not a problem (Refs. 7-9 and 7-10).

Although a high rate of heat transfer to the surface immersed in the fluidized bed is an advantage that enables the total furnace size to be reduced, the heat transfer coefficient itself is relatively constant with respect to the superficial velocity in the normal bubbling regime. This makes turn-down (load reduction), a very different kind of problem than in a non-fluidized bed furnace. In order to turn-down a fluidized bed furnace to a lower operating load, other means than reducing air and fuel flow must be provided to reduce the heat transfer in the bed (Ref. 7-11). In a conventional furnace, convective heat transfer is a function of the flue gas flow rate. In the fluidized bed furnace, if the bed temperature stays the same and the bed is normally fluidized, the heat transfer in the bed will tend to stay constant unless some other mechanism of change is available. Two possible mechanisms are allowing the bed temperature to change, and changing the effective area for heat transfer.

The bed temperature could drop to perhaps 1400° F without serious degradation in the sulfur dioxide sorbent effectiveness. This is useful for trimming the bed temperature or making small heat transfer adjustments. The bed temperature change is also a slow process since a large amount of thermal energy is stored in the bed and its thermal relaxation time can be 10 minutes or more (Ref. 7-11).

The effective heat transfer area can be changed by changing the bed level, or by "slumping" a portion of the bed. At constant bed mass, the bed level varies with superficial velocity. At a higher velocity, the void fraction increases and there are a larger number of "bubbles". The heat transfer surface can be located near the bed surface at maximum load conditions. As the firing rate and the superficial velocity are reduced, the surface will emerge and the effective heat transfer area will be reduced (Ref. 7-12). The bed level can also be varied by changing the bed mass. Bed material can be dumped to lower the bed level, or fresh bed material can be injected to raise the level. Previously dumped bed material can also be reinjected.

Slumping refers to loss of fluidization in a portion of the bed as illustrated in Figure 7-5. This is accomplished by shutting off fluidizing air to a section of the distributor plate. The immersed tubes in the slumped region are surrounded by the hot bed material. However, the bed is static, heat transfer is low, and the thermal relaxation time is correspondingly much greater than when it is fluidized. There is little danger to the immersed tubes, as long as water or steam continues to circulate.

By using the mechanisms of bed level change, bed temperature change, and changes in the bed mass (singly or in combination), it is possible to operate a fluidized bed steam generator over a 2 to 1 turn-down range. Additional turn-down can be achieved by using the slumping mechanism. In a fluidized bed boiler adapted to a railroad locomotive, two stages of slumping (to 50% of bed area, and to 25% of bed area) coupled with the normal 2 to 1 turn-down will result in an 8 to 1 operating range.

G. START-UP AND SHUT-DOWN

From a cold start condition, the bed must be preheated to a temperature of 1000° to 1100° F before self-sustaining coal ignition and combustion is achieved. Preheating can be accomplished by an oil-fired burner. Other possible techniques include the use of a burner to preheat fluidizing air, or the combustion of a gas/air mixture introduced into the bed through the distributor plate. It is not necessary to preheat the entire bed. Once a major segment of the bed is heated and is sustaining coal combustion, the adjacent segments of the bed can be fluidized sequentially and combustion will be transferred by the normal turbulent flow. Care must be taken during start-up to properly manage coal injection into the bed. A large inventory of unburned coal can lead to excessive gasification or thermal runaway and clinkering of the bed.

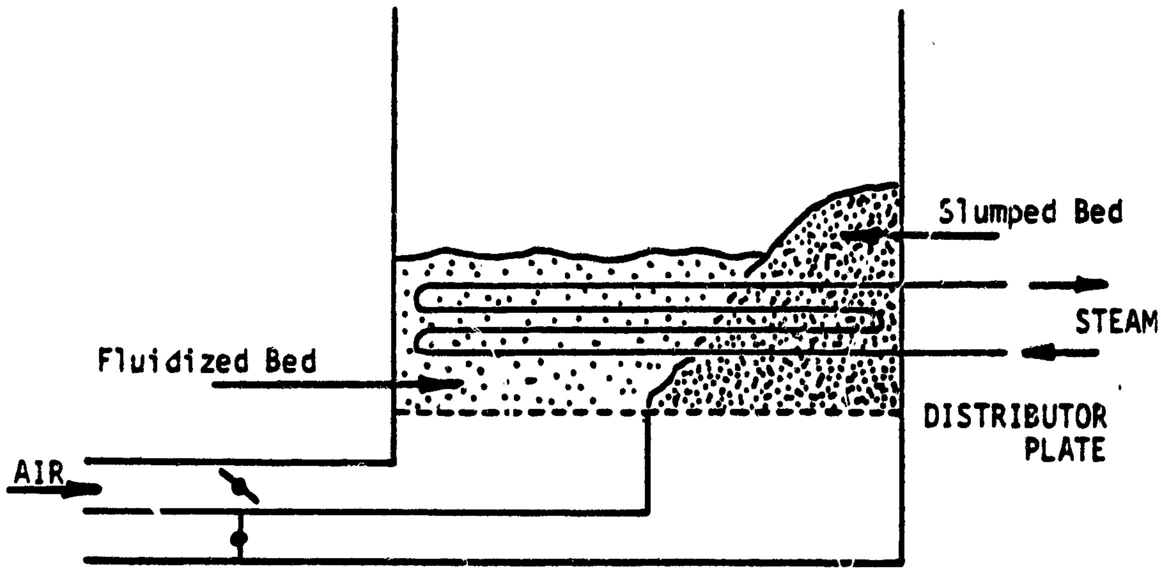


Figure 7-5. Slumping (From Ref. 7-5)

When shutting down a bed by slumping, it is advisable to reduce the coal injection first, to allow the remaining coal to burn-out for approximately a minute, and then to reduce the fluidizing air flow. This reduces the possibility of gasification and the release of combustible gases into the furnace enclosure. When the bed is slumped, heat transfer is essentially cut-off even with the continual flow of water or steam through the immersed tubes. The bed material is an insulator so that heat transfer through it is very poor. The thermal relaxation time of the slumped bed can be several hours, especially if inlet air dampers are tightly closed and no residual air can leak up through the static bed. During this interval, the bed can be quickly restarted by the addition of fluidizing air and subsequent coal injection. Additional preheating with a duct burner or by some other means is necessary if the bed has cooled much below 1100° F (Ref. 7-').

It has been assumed that in accordance with current utility and industrial fluidized bed combustor applications (Ref. 7-8), the fluidized bed combustor locomotive will employ an oxidizing fluidized bed that operates in the bubbling regime with the heat transfer surface immersed in the bed for both raising steam and for superheating. The fluidized bed is also an effective means for coal gasification, in which case the fluidizing air only provides for partial combustion. The gas emerging from the bed is rich in carbon monoxide and unburned hydrocarbons. Combustion is completed in the freeboard region above the bed by the injection of additional overfire air. This approach has the advantage of higher flue gas temperatures and a heat release profile closer to that of a conventional locomotive stoker furnace. It is similar in some respects to the thick fuel bed furnace described earlier. A more extensive evaluation of coal ombustion techniques as they apply to locomotive design must explore this alternative with respect to the locomotive size and configuration, and the overall coal pile to roadbed efficiency.

Incomplete utilization of the coal's heating value because of incomplete combustion, or because of a low cycle efficiency for the process, has a direct impact on the amount of coal that must be carried on-board. This loss also impacts the required coal storage facilities and their locations along the right-of-way. The loss of carbon in the form of fly ash has a negative environmental effect. The efficient burn-up of all of the carbon in the coal, plus the high conversion efficiency of chemical energy into tractive effort are the most desirable locomotive characteristics in terms of overall environmental and systems considerations.

Several atmospheric fluidized bed combustor steam generators have been operated successfully for long intervals and several boiler manufacturers are now offering and installing fluidized bed combustor units for industrial applications. These units bracket the size and steam conditions required for locomotive application. Several are designed specifically to be rail transportable, and their size and shape approaches what would be needed in a locomotive. For example, the boiler for the Industrial Fluidized Bed Combustion Demonstration Plant discussed earlier, would be suitable for a railroad locomotive, and it is almost two times as large as the boiler which would be required for 3000 hp system (Refs. 7-13 and 7-14). None of these boilers, however, have been designed specifically for a locomotive and, therefore, development effort is required to optimize the design for this purpose.

H. INDUSTRIAL FLUIDIZED BED COMBUSTION DEMONSTRATION PLANT

This plant has an A-type drum boiler which would be suitable for locomotive applications. The approximate width is 10 ft, and the boiler was designed so that similar models of different capacities would differ only in their length. The boiler width would be traded off with length, since it is the total heat transfer area that is important. Thus the boiler in Figure 7-6, which is approximately twice the capacity required for a 3000 hp locomotive, would readily scale into a package suitable for such an application.

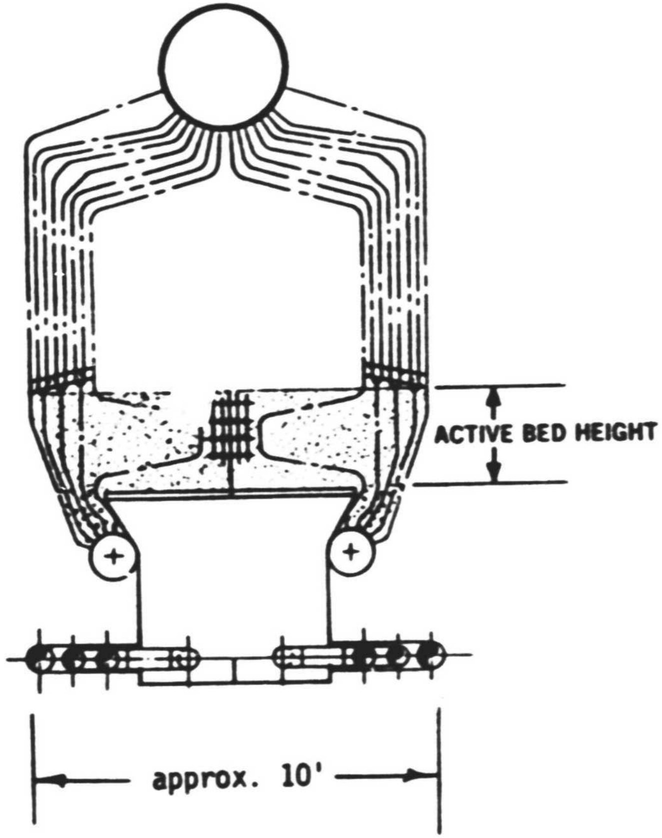
Characteristics of the unit are:

Steam flow	50,000 lb/hr, 560° F/380 psia (superheated)
Bed dimensions	17 ft x 8 ft
Bed depth, expanded	3 ft
Bed temperature	1600° F
Fluidizing velocity	7 ft/sec
Coal	Illinois No. 3; 12,725 Btu/lb; 10% ash; 3.5% sulfur
Calcium/sulfur ratio	4 to 1
Coal feed rate	4900 lb/hr
Limestone feed rate	1600 lb/hr
Reinjection rate	3900 lb/hr
Heat input	62 million Btu/hr

The A-type shop-assembled, natural recirculation, drum boiler was transported by rail from Combustion Engineering in Chattanooga to its operating site at the Great Lakes Training Center in Illinois. The installation was 50% complete in April 1980, and its startup is scheduled for January 1981. The coal and limestone will be mixed before injection.

An A-type drum boiler with grate firing was successfully incorporated in the "John Henry" locomotive (4500 hp) which operated successfully on the Norfolk and Western railroad for three years (Ref. 7-15). The boiler was fabricated by Babcock and Wilcox. This design may be a major influence on the design approaches for adapting a fluidized bed boiler to a locomotive.

In order to estimate the size of the required fluidized bed boiler, the configuration shown in Figure 7-2 was selected. Steam at 615 psia and 900° F is generated in the fluidized bed combustion boiler and expanded in a turbine. Extraction steam is used for deaerator feedwater heating and for a low pressure feedwater heater (with condensate return to water storage), in order to improve cycle efficiency. Three alternatives are considered for handling the exhaust steam. They are non-condensing with steam released to the atmosphere, 50% condensation, and 100% condensation.



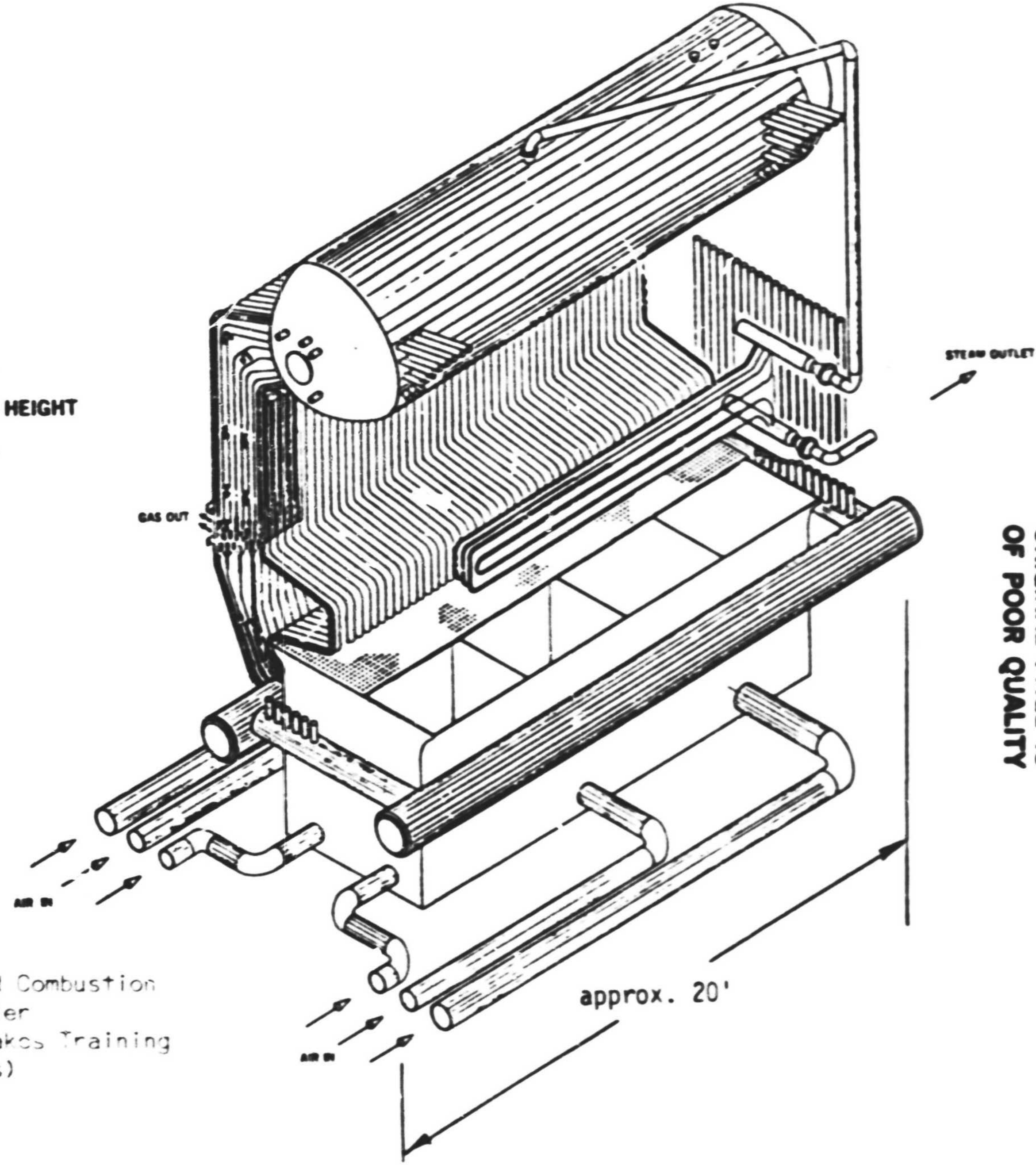


Figure 7-6. Industrial Fluidized Bed Combustion Demonstration Plant Boiler (located at the Great Lakes Training Center Station, Illinois) (From Ref. 7-5)

In the system without a steam condenser, most of the turbine exhaust steam vents directly to the atmosphere. A small portion flows to the low pressure feedwater heater where the steam condensate is drained off to water storage. Another portion is extracted at higher pressure (44 psia) for direct contact feedwater heating in the deaerator. Approximately 10% of the primary steam flow is recycled in this case.

The remaining two alternatives consider partial and total steam condensation in an on-board air-cooled condenser. The turbine back pressure is assumed to be 4 psia. A condensate pump is necessary for delivering flow from the condenser, through the feedwater heater and on to the deaerator. With partial condensation, it is assumed that half of the steam exhausts to the atmosphere at 16 psia from the feedwater heater extraction port, and the other half leaves the turbine exhaust port at 4 psia to go to the condenser. For total condensation, the flow to the atmosphere is reduced to zero, and all spent steam exhausting at 4 psia is condensed.

Auxiliary power is required for the condensate pump, the feedwater pump, the forced draft fan, and the condenser cooling fan. The system is sized to give 3000 hp (net) at the generator output. Additional auxiliary power required for coal and bed material transport, auxiliary steam and air, locomotive control, lights, and train heat has not been considered.

The idealized Rankine cycle diagram is shown in Figure 7-7. This cycle was analyzed for three condensing conditions (100%, 50%, and no condensing) and three output levels (3000, 1500, and 750 rp), resulting in nine separate runs. The assumptions used in the analysis are listed on Table 7-2. They are valid at every output level, except as noted. The cycle analysis results for each condition are summarized in Table 7-3 (Ref. 7-5).

No attempt was made to optimize the condenser nor in any other way to reduce condenser fan power. It is a conventional condenser design (Ref. 7-16), and the fan requires almost 10% of the gross output at 3000 hp. Nonetheless, the condensing cycle has a thermal efficiency of 22.22%. When the locomotive is underway, natural draft can be used to reduce the condenser fan power.

The condenser can be quite large and it may be necessary to have a separate car for it in the train. At the very least, it would occupy a large portion of the locomotive or coal car. Having 100% on-board condensing, however, is advantageous from the railroad system operating point of view because enroute water supplies and extensive water treatment facilities are not necessary. Furthermore, the non-condensing design consumes 13% more coal for the same operating distance.

The following components and subsystems are required for installation in a locomotive:

- Atmospheric fluidized bed combustor and drum boiler
- Economizer
- Deaerator
- Feedwater heater
- Boiler feedpump
- Forced draft fan

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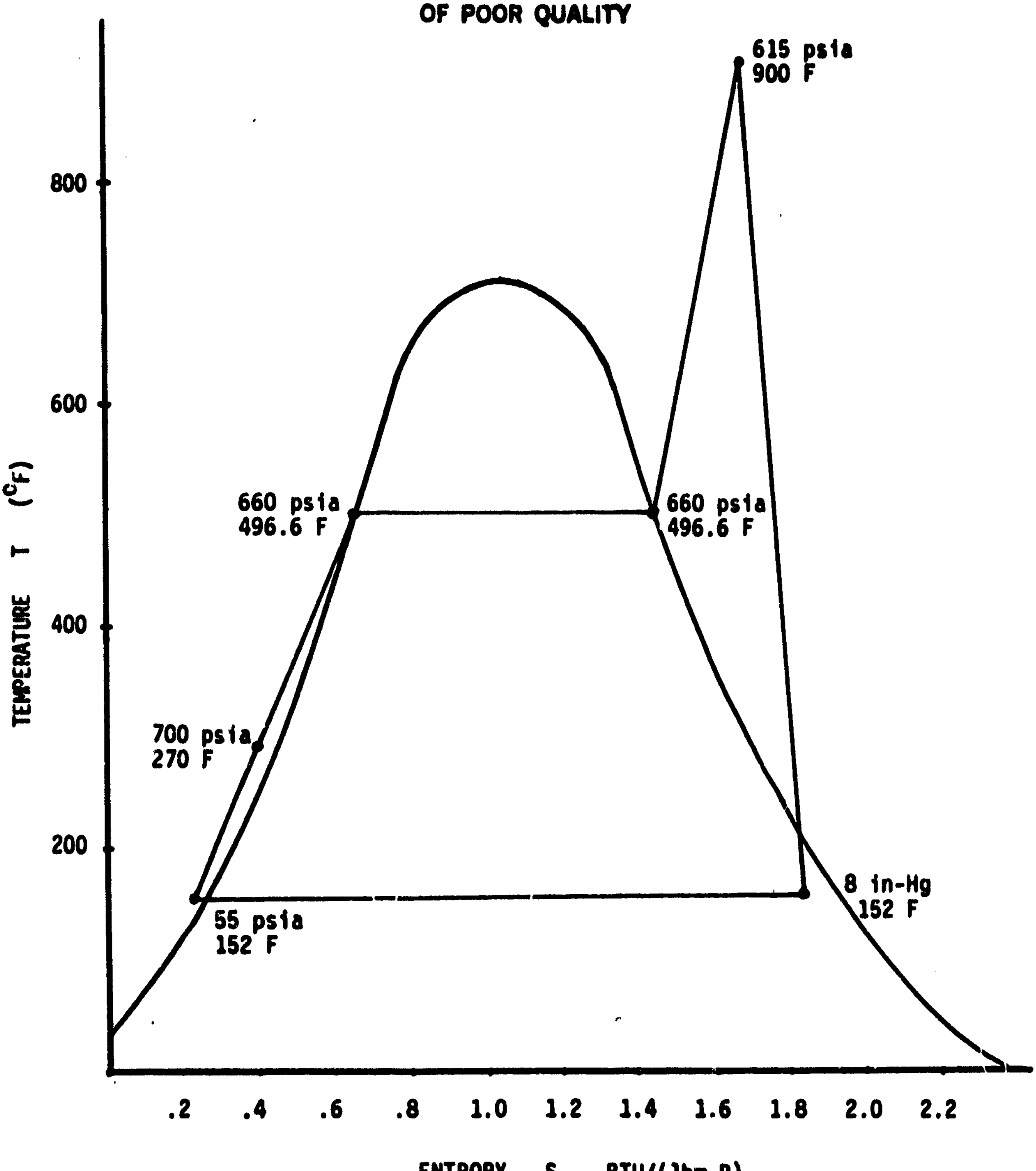


Figure 7-7. Rankine Cycle Diagram
(From Ref. 7-5)

Table 7-2. Rankine Cycle Analysis

Steam Turbine	
Steam inlet conditions	615 psia, 900°F, 1462.5 Btu/lbm
Turbine efficiency	80% at 3000 hp 75% at 1500 hp 70% at 750 hp
Deaerator extraction pressure	44 psia
Feedwater heater extraction pressure	16 psia
Turbine exhaust pressure (condenser)	4 psia
Generator	
Efficiency	96%
Condensate Pump	
Efficiency	80%
Outlet pressure	55 psia
Deaerator	
Pressure (2 psia less than extraction pressure, independent of output level)	42 psia
Boiler Feedpump	
Outlet pressure	700 psia
Efficiency	80%
Fluidized Bed Furnace	
Coal, higher heating value	12,000 Btu/lbm
Furnace efficiency	88%
Forced Draft Fan	
Pressure rise across fan	50 in. of water
Efficiency	80%
Inlet air at 80° F; density	0.073546 lbm/ft^3
Air-Cooled Condenser	
Pressure drop (corresponding to 22,000 lbm/hr of steam flow)	3.27 in. of water
Air inlet temperature	90° F
Temperature rise	40° F
Air outlet temperature	130° F

Table 7-3. Summary of Locomotive Performance at Net Power Output Levels of 3000 hp, 1500 hp, and 750 hp, with 100%, 50%, and No Condensing of Exhaust Steam

Gross Power (hp)	Steam Flow (lbs/hr) Turbine Inlet	Exhausted[a]	To Condenser	Air Flow to Condenser (10^6 lb/hr)	Coal Feed Rate (lb/hr)	Furnace Heat Input (10^6 Btu)	Efficiency (%)
3364	24,770	0	21,982	2.49	2,863	34.4	22.22
3235	26,329	11,683	11,683	1.33	3,044	36.5	20.90
3089	28,098	24,936	0	0	3,248	39.0	19.59
1696	13,138	0	11,680	1.35	1,519	18.2	20.95
1626	13,944	6,198	6,198	.72	1,612	19.3	19.74
1547	14,856	13,207	0	0	1,717	20.6	18.53
856	6,997	0	6,226	.73	809	9.7	19.67
816	7,217	3,211	3,211	.38	834	10.0	19.07
774	7,451	6,630	0	0	861	10.3	18.47

[a] Exhausted to the atmosphere

Air heater
Multicyclones
Fly ash recycle feeder
Coal storage
Limestone storage
Spent bed storage
Coal feeder and transport
Limestone feeder and transport
Spent bed feeder and transport
Water storage
Condenser
Condensate pump
Condenser cooling fan
Steam turbine-generator set

An auxiliary fuel supply is required for cold start-up of the locomotive. It is assumed that cold starts are made at the engine house and that the locomotive does not need an on-board fuel supply for that purpose.

I. PROPOSED CONFIGURATION

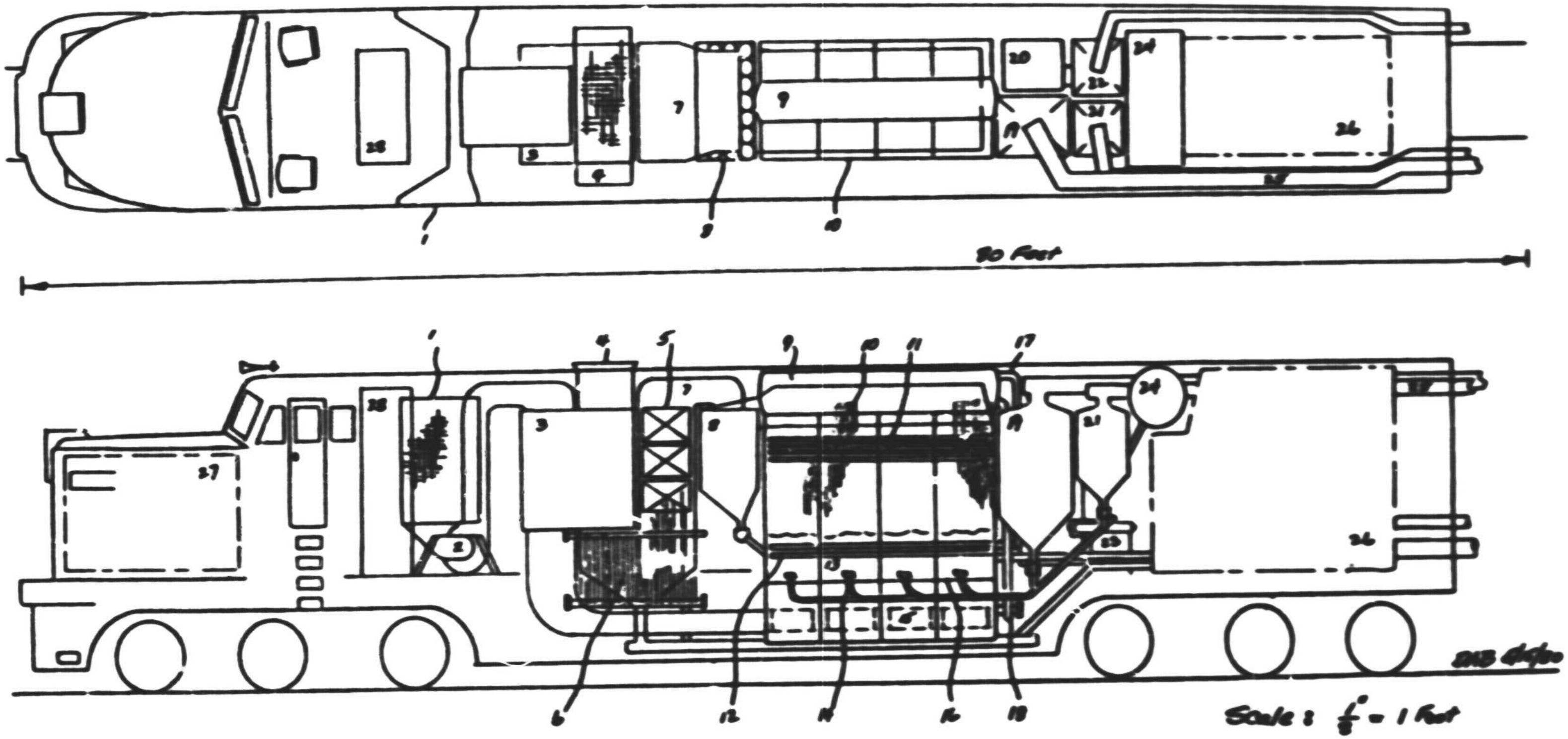
An A-type boiler with the nominal dimensions of 6 ft wide, 12 ft long, long, and 13.5 ft overall height is proposed. It is divided by three internal partial divider walls into four 3 by 6 ft cells. All cell walls are waterwalls. The divider walls are open between adjacent tubes in both the bed and the freeboard (convection) regions in order to permit the circulation of bed material between adjacent cells, the mixing of flue gas in the convection region, and the passage of longitudinal heat transfer assemblies through all four cells. The cells will be operated in any combination of 1, 2, 3, or all 4 cells, which correspond to operation in notch position 2, 4, 6, or 8. Operation in notch position 1, 3, 5, or 7, or proportional operation between notch positions will be accomplished by the turn-down procedures discussed earlier. The idle cells are slumped. If the locomotive operation requires many hours of duty with only one or two cells active, the active cells will be rotated to insure that the fluidized bed temperature in the slumped cells is always greater than 1100° F in order to facilitate rapid restart.

The location of the boiler in the center of the locomotive balances the weight distribution between the forward and rear trucks as shown in Figure 7-8. It is narrow enough for passageway access on each side within the locomotive shroud.

The boiler height is 10 ft between the drum centerline and the air distributor plate. The expanded bed height is 2.5 ft, leaving a 4 ft freeboard and a 3 ft convection region. Exhaust gas is carried through flues adjacent and parallel to the drum, leading to common ash recovery multicyclones and to a convective back-pass containing the economizer and the tubular air preheater.

Preliminary calculations indicate the following arrangement of heat transfer surfaces; the fluidized bed for superheat and boiling, the waterwalls and the convection region for boiling, and the convection back-pass for economizing. The economizer outlet flow will pass through the waterwalls of the convection section back-pass into the drum. With higher deaerator pressure, the amount of economizing surface in the convection region can be decreased and a small increase in the cycle efficiency will result. At 3000 hp with no condensing (maximum heat input case), the approximate allocation of heat transfer is: economizing, 6.8 million Btu/hr; boiling, 20 million Btu/hr; and superheat, 7.3 million Btu/hr.

The locomotive consists of two units. The A-unit, as shown in Figure 7-8, comprises the basic power plant and the major balance-of plant equipment. The B-unit is the tender. It is envisioned to be enclosed in a similar shroud, although somewhat shorter, and will have a complete control cab for fully bi-directional operation. The tender will house the water storage tank and the air-cooled condenser if one is used. The tender will also store the coal, the limestone, and the spent bed material that is withdrawn during normal operation. The major interconnections between the A and B units include: the coal, limestone, and spent bed feeders, the turbine exhaust steam duct, the feedwater heater drains, the feedwater supply pipe, and the control cables. Additional effort is required to optimize the locomotive layout and to determine if a single unit could accommodate all of the locomotive functions including the condensation and storage of solid materials.



1. Air intake
2. Forced draft fan
3. Air preheater
4. Flue gas exhaust
5. Economizer
6. Convection back pass waterwalls
7. Convection back pass duct
8. Multicyclones
9. Boiler drum
10. Boiler waterwalls
11. Convective heat transfer assembly
12. Immersed heat transfer assembly
13. Fluidized bed
14. Coal and limestone feed into bed
15. Air duct inlet to plenum
16. Air distributor plate
17. Steam line to superheater
18. Downcomer
19. Intermediate coal storage
20. Feedwater heater
21. Intermediate limestone storage
22. Intermediate spent bed storage
23. Feedwater pump
24. Deaerator
25. Coal feed line from tender
26. Space for turbine-generator set
27. Crew support, instrumentation, work area
28. Control cabinet

Figure 7-8. Proposed Configuration for 3000 hp Coal-Fired AFBC Steam-Turbine-Electric Locomotive, Unit A (From Ref. 7-5)

J. SUMMARY

If the steam engine is considered only from a thermal efficiency standpoint, it is hard to justify its use. The best thermal efficiency predicted for the advanced steam engines in Table 7-3 is only 22%, about half that of present Diesel engines. The very highest efficiency that might be attained in locomotive service is less than 30%. This justification is based on the cost of coal or low grade oils which can be used in steam engines instead of the much higher priced Diesel fuels. The cost analysis of these and other engines is presented in Section XIV. The cost of coal energy in dollars per million Btu can be one-tenth that of Diesel No. 2 energy in the western mountain areas. Even in New England, where the difference is minimized, coal energy is still one-third of Diesel No. 2 energy. The cost of fuel is only part of the total cost picture but the differences are large enough to justify a close look at coal burning locomotives.

Of the three coal-fired engines (the steam, the Stirling cycle and the closed cycle gas turbine), the steam engine requires the least development. A first generation new steam engine can be commercially available in five years. A second generation engine of the type analyzed could be on the market in about 1990.

If a fluidized bed combustion system uses a metallic heat exchanger the maximum temperature of the working fluid is about 1550° F. At temperatures lower than this, the steam engine system is more efficient than either the closed cycle gas turbine or the Stirling cycle engine. For the increased durability of the heat exchanger, temperatures of 1200 °F to 1300° F would be prefered and, in this range, the Rankine cycle engines are the best. When fluidized beds with ceramic heat exchangers are developed and the outlet temperature of the exchanger approaches 1800°F, the closed cycle gas turbine with its higher efficiency will probably take over. Until then, the steam locomotive offers the most efficient approach to the direct use of coal.

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SECTION VIII
GAS TURBINE ENGINES

A. INTRODUCTION

The gas turbine is not new to the railroads, its use goes back fifty years. The initial attempts to use gas turbines in locomotives were in the early 1930s in Germany. About twenty different systems were tested and discarded after limited service. In order for these gas turbines to compete with the steam and electric locomotives already in use, the gas turbine had to offer significant advantages over the conventional systems. Unfortunately, these early attempts did not result in a competitive gas turbine engine. In all of these applications, industrial gas turbines were used as the main power source for traction (Ref. 8-1).

Following the introduction of aircraft gas turbine engines in the 1940s, there was renewed interest in gas turbines for railroads. This led to developments in two directions. One was the adaptation of aircraft gas turbines to rail service (Ref. 8-2). The other was to design gas turbines specifically for locomotives using aircraft technology. The General Electric gas turbine locomotives used by Union Pacific in the 1950s used simple cycle gas turbines. Figure 8-1 shows one of these locomotives and the engine that was used in it. Coal was used in a few locomotives including the one shown. For most engines, residual oil was used as the fuel and cost about one-fourth as much as Diesel No. 2. These engines performed well in the type of freight service encountered on Union Pacific routes. Their main drawback was their high idle fuel consumption, about 30% of the full load usage. Because of this, the engines were shut down whenever possible and an auxiliary Diesel engine was used to provide heat and electrical power as necessary. The engines, however, were economically justified as long as residual oil was cheap.

Fifty-five locomotives were delivered to the Union Pacific Railroad between 1952 and 1958. The initial order of ten was delivered in 1952 and they were rated at 4500 hp. In 1954, fifteen more of this same model were delivered. In 1958, thirty 8,500 hp locomotives were delivered and 6 years later, these units were uprated to 10,000 hp. However, the end was in sight. Between 1963 and 1965, the twenty five original 4500 hp gas turbine locomotives were traded in on new Diesel units. The turbine locomotives were only 10 years old. The upgraded 10,000 hp units were then traded in on new Diesel units and by the end of 1969, the gas turbine freight locomotive had vanished from the tracks. Improvements in oil refining methods made it possible to convert much of the residual oil into middle distillates and gasoline. The price of residual oil rose and the gas turbine was no longer economically practical.

The gas turbine is used for some passenger trains. In 1966, United Aircraft Corporation was awarded a 2-year contract by the United States Department of Transportation for two three-car passenger trains to be used for a two-phase program:

(1) To demonstrate a vehicle capable of operating at speeds up to 160 mph on standard construction track.

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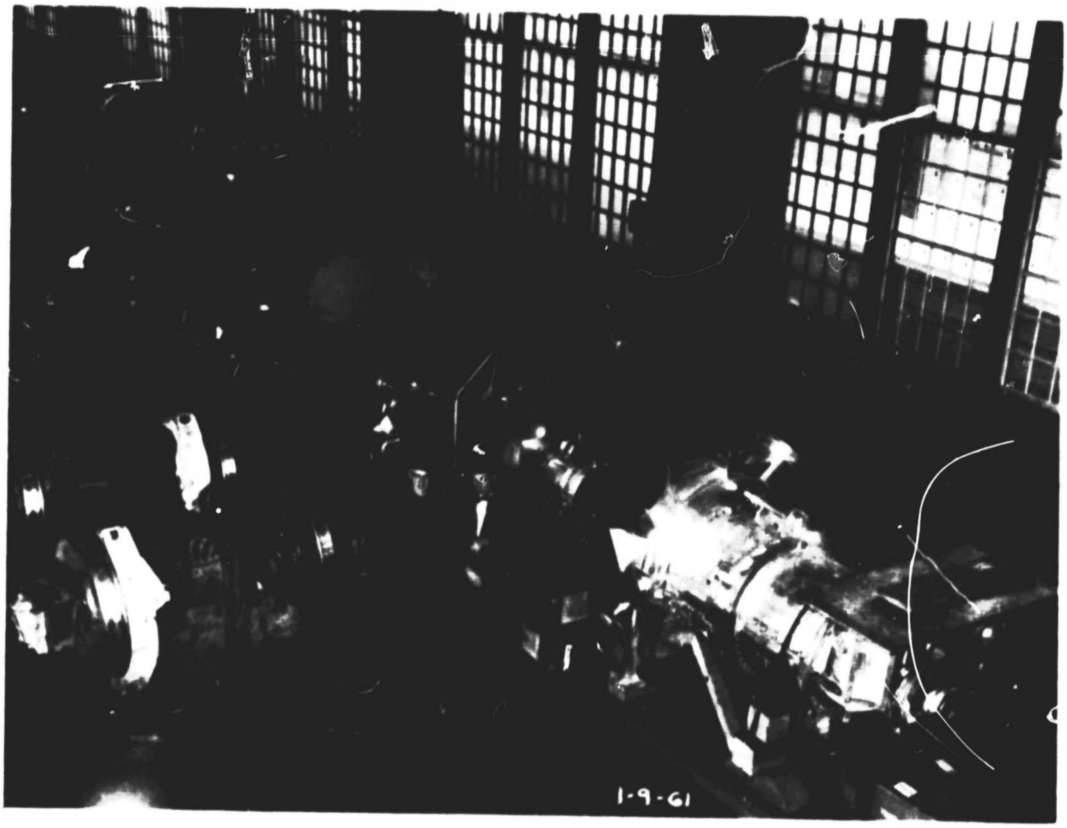

Figure 8-1. Union Pa.ific Gas Turbine Train and Gas Turbine Engine (From Ref. 8-3)

(2) To evaluate service of the vehicles at speeds up to 125 mph.

For the high-speed trials, each train was powered by seven ST6B-65 gas turbines. Each engine was set at an intermittent rating of 455 shaft hp. Six units were used for propulsion and the seventh for driving the generator to provide electrical power for the train. The engines were grouped at each end of the train in power drive cars.

Five seven-car trains of the same type were fabricated by Montreal Locomotive works for use by Canadian National Railways. These were intended for service primarily between Montreal and Toronto. Each train was originally powered by five ST6B-65 gas turbines, four provided traction power and the fifth for electrical power for the train. These trains accumulated over 10,000-service hr on individual engines and over 600,000-service mi on individual trains. Their performance indicated that aircraft derived ST6 gas turbines were reliable in rail service (Ref. 8-4).

Other applications of aircraft derived gas turbines are discussed in Ref. 8-2. There have been a number of developments for passenger trains in Europe, particularly France and Germany. It appears there has been no significant application to freight service in Europe.

General Motors Electro-Motive Division (GM-EMD) began a program to develop a gas turbine engine specifically for locomotives in the mid-1950s. They approached the problem from two directions. One approach was the use of a free piston gas turbine. The other used a more conventional axial compressor and combustor with exhaust heat recovery via a rotating drum regenerator.

The first engine to be developed was the free-piston gas turbine shown schematically on a test stand in Figure 8-2. In this unit, the GM 2-14 siamese gasifier supplied hot gas to a four-stage axial flow reaction turbine. A double reduction split load gearbox was interposed between the turbine and the dc traction generator. It also provided a drive for the gasifier and turbine accessories. The power of the dc generator was absorbed by four traction motors geared to the driving wheels. The free piston gasifier section was planned to burn residual fuel oils of high viscosity, as well as Diesel fuel.

This engine was designed to produce 2000 hp with a gearbox output speed of 835 rpm. The design power rating was met in actual testing of the engine. The best brake specific fuel consumption (BSFC) was obtained at an output of 1900 hp and was 0.445 lb/hp-hr measured at the output of the gearbox. The thermal efficiency was 30.7% and the idle fuel flow rate was 167 lb/hr. The high idle fuel flow rate, mechanical problems with the gasifier, and an efficiency comparable to the Diesel engines of the day eventually lead to the abandonment of the engine. The free piston gas turbine was not markedly better than the Diesel engine it was to replace.

Early in the 1960s, work began on the regenerative two shaft gas turbine at GM-EMD. This engine, the T-45, was first tested late in 1965 and testing continued on into 1971. Figure 8-3 shows a side view of the engine. The large circular area is the cover of the drum regenerator.

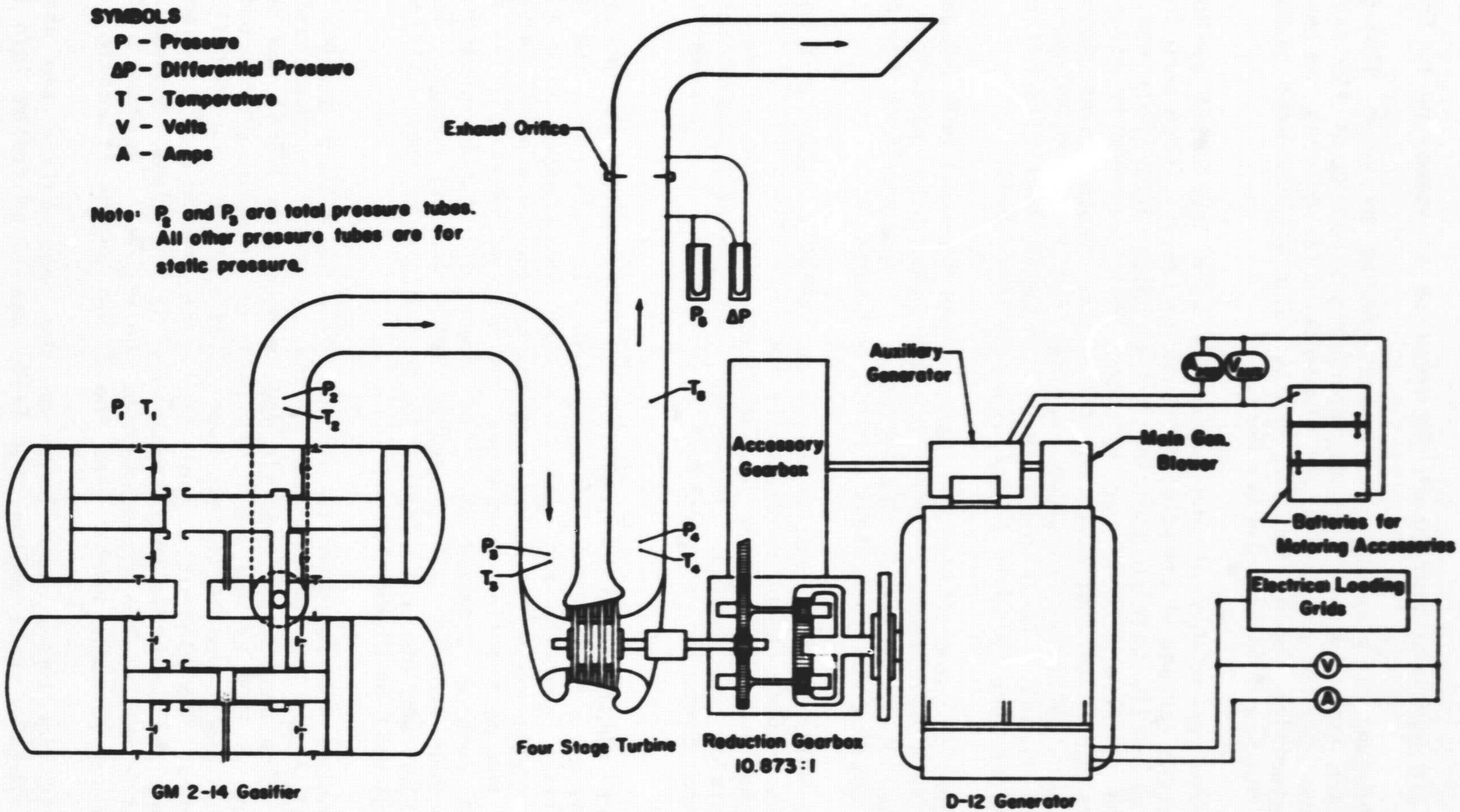

Figure 8-2. Schematic Diagram of Gas Flow and Instrumentation for a Free Piston Gas Turbine Locomotive Engine (From Ref. 8-5)

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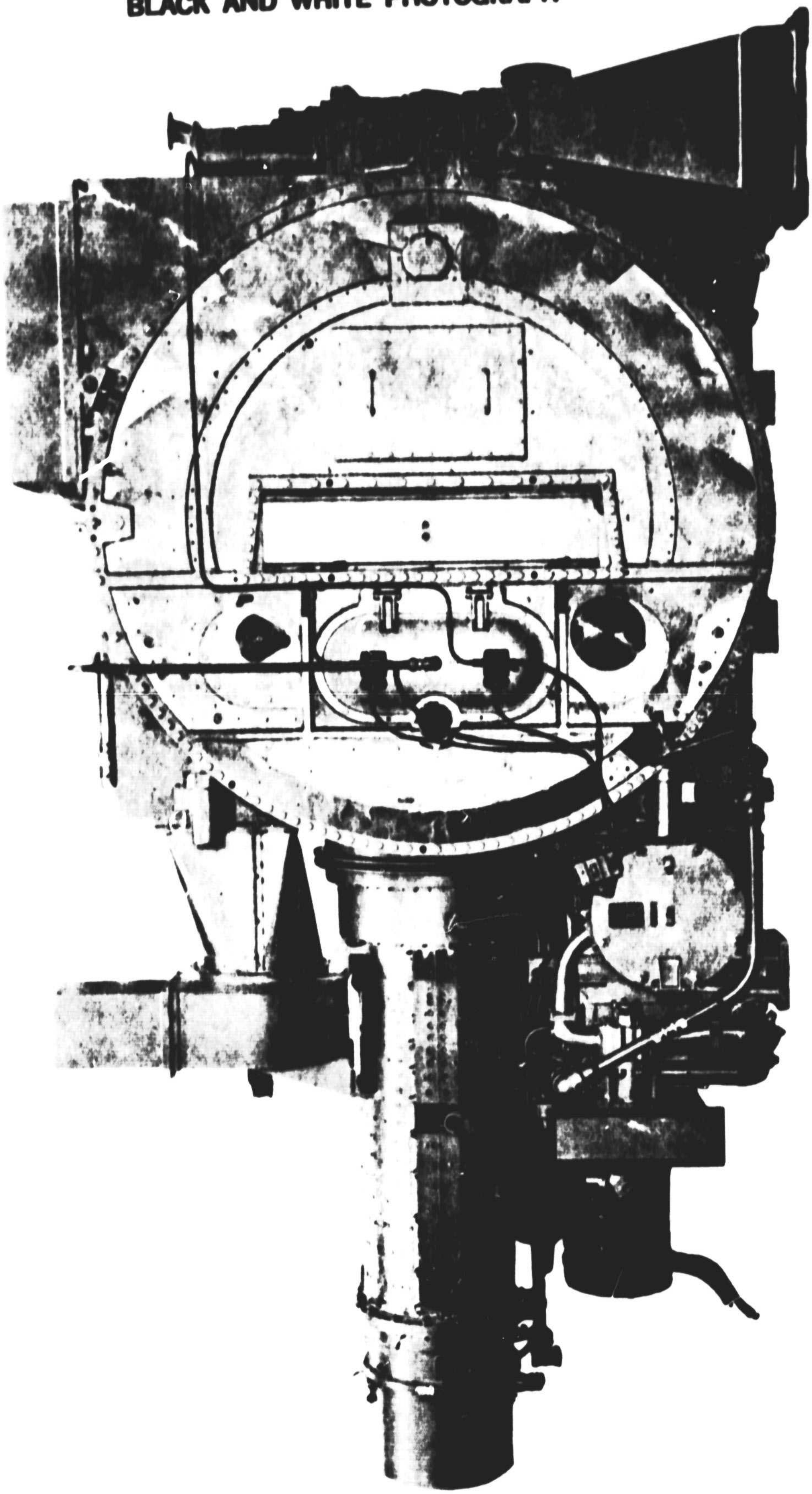

Figure 8-3. T-45 Engine - Left Side View
(From Ref. 8-5)

The oblong within the circle is the combustor cover with the fuel nozzle. The cylinder to the left houses the axial compressor with the air intake at the extreme left. The accessories are mounted below the compressor. The power output shaft is at the right end of the unit.

The design specifications called for a 4500 shaft hp engine with a 0.400 BSFC on a 90° F day. It was specifically designed for a mechanical drive locomotive and a 20,000 hour life. Provisions were made for a higher rating (6300 shaft hp) for shorter life applications. The design, modifications, and testing continued for nearly 6 years. In 1971, the engine met the design objective of 4500 BHP at 1850°F turbine inlet temperature and the gasifier rated speed of 6500 rpm. The power turbine rated speed was 5400 rpm. The BSFC design goal of 0.400 lb/hp-hr was not met. Low turbine efficiencies and a high regenerator pressure drop resulted in a BSFC of 0.436 lb/hp-hr, 9% above the goal. Higher turbine speeds and a 1910° F turbine inlet temperature produced 5671 BHP. This was short of the original goal for the shorter life version of the engine.

The Diesel engine was also being further developed during this period of time. The increasing use of turbochargers on the big Diesel engines was improving their BSFC ratings as well as their power-to-weight ratios. Although it appeared certain that the BSFC of the gas turbine could be reduced to the 0.400 lb/hp-hr level, the Diesel engine was already below that level. The high idle fuel flow rate (177 lb/hour) and the durability problems with the drum regenerators, together with the poorer BSFC of the gas turbine compared to the Diesel, caused the T-45 engine program to be terminated.

The gas turbine has proven to be successful in passenger service although the high cost of fuel is a serious threat to its future. The gas turbine in freight service has not been successful for a number of reasons but primarily because of its high idle fuel consumption and its relatively poor BSFC.

There have been a number of improvements in gas turbines in the last decade and it is time to evaluate the effect of these changes on potential railroad gas turbine engines. The gas turbine engines have a number of advantages. One of them is the relatively high power to weight ratio. Increasing the train speed could improve the competitive position of the railroads in short and medium distance service relative to other transport means. Increasing the speed, demands higher acceleration rates and peak speeds which in turn requires more power at the rails. At the same time, axle loads must be reduced to keep the truck loading and impact forces on the rail within allowable limits. The high power to weight ratio of the gas turbine is its essential advantage in this type of service.

The small size of gas turbine engines compared to Diesels of equal power output makes it possible to modularize the power plant. The exchange of engines when one needs an overhaul or repair would reduce the maintenance time and the time the locomotive is out of service. Utility and aircraft gas turbines require fewer overhauls than piston engines. Railroad Diesel engines generally need new piston rings every 2 to 3 years and a major overhaul once every 7 or 8 years. The Diesel requires new rings and other minor overhaul work every 7,000 to 10,000 hr and a major overhaul at 20,000 to 30,000 hr. Utility gas turbines normally go 30,000 hr between overhauls. If railroad gas turbines had the same time between overhauls as utilities, the time that locomotives are out of service would be significantly reduced. Since the gas turbine engine does not need water cooling, the radiators, pumps, water lines, and the radiator fans are not needed. This simplifies the locomotive and improves its overall efficiency as well. The radiator fans, for instance, use about 100 hp on a 3,000 hp locomotive.

The use of ceramics in turbines and improved exhaust heat recovery systems should improve the brake specific fuel consumption. Only open cycle gas turbines have been tried on the railroads. Closed cycle gas turbines are a new class of engines to be studied for this application. One disadvantage encountered in regenerated and closed cycle gas turbines is the complexity introduced by the heat exchangers. High temperature heat exchangers have been a source of mechanical problems on gas turbines.

B. BRAYTON CYCLE

Gas turbine engines are based on the Brayton thermodynamic cycle. This cycle is shown on a pressure-volume diagram in Figure 8-4. In this cycle, the working fluid (air or some other gas) is first compressed. The compressed gas is then heated at constant pressure either by injecting fuel into it and burning it or by heating the gas in some form of heat exchanger. The hot gas is expanded, usually in a turbine, to produce mechanical power via a shaft or a high velocity jet as in a pure jet aircraft engine.

There are a number of variations to the basic cycle. First of all, there are open and closed cycles. An open cycle uses air from the atmosphere as a working fluid and as an oxidizer in the combustion process. The exhaust goes to the atmosphere. The working fluid in the closed cycle may be air, helium, hydrogen, or any other gas which does not condense in the cycle. After the working fluid goes through the turbine, it goes to a heat exchanger to reject the unavailable energy before going to the compressor again. Heat is added to the compressed gas through a heat exchanger before the turbine stage. The open cycle gas turbine is the simplest and it is the one most commonly used.

Another variation of the basic cycle is a cycle with regeneration. The non-regenerative cycle is commonly known as a "simple" cycle. In a regenerative cycle, some of the heat in the exhaust gas is transferred to the compressed gas between the compressor and the combustion chamber. This arrangement reduces the amount of fuel needed for combustion. Regenerative cycles are used primarily for engines that spend most of their time at part load. An automobile gas turbine is one example of an engine using this cycle. At full load, a regenerative cycle is usually no more efficient than a simple cycle because of reduced regenerator effectiveness and increased turbine temperature drop.

The thermal efficiency of Diesel and industrial gas turbine engines are shown in Figure 8-5 as a function of their load rating. The black dots are the rating of specific gas turbine engines. All, with the exception of the AGT 1500, are simple cycle engines. The AGT 1500 is a regenerative

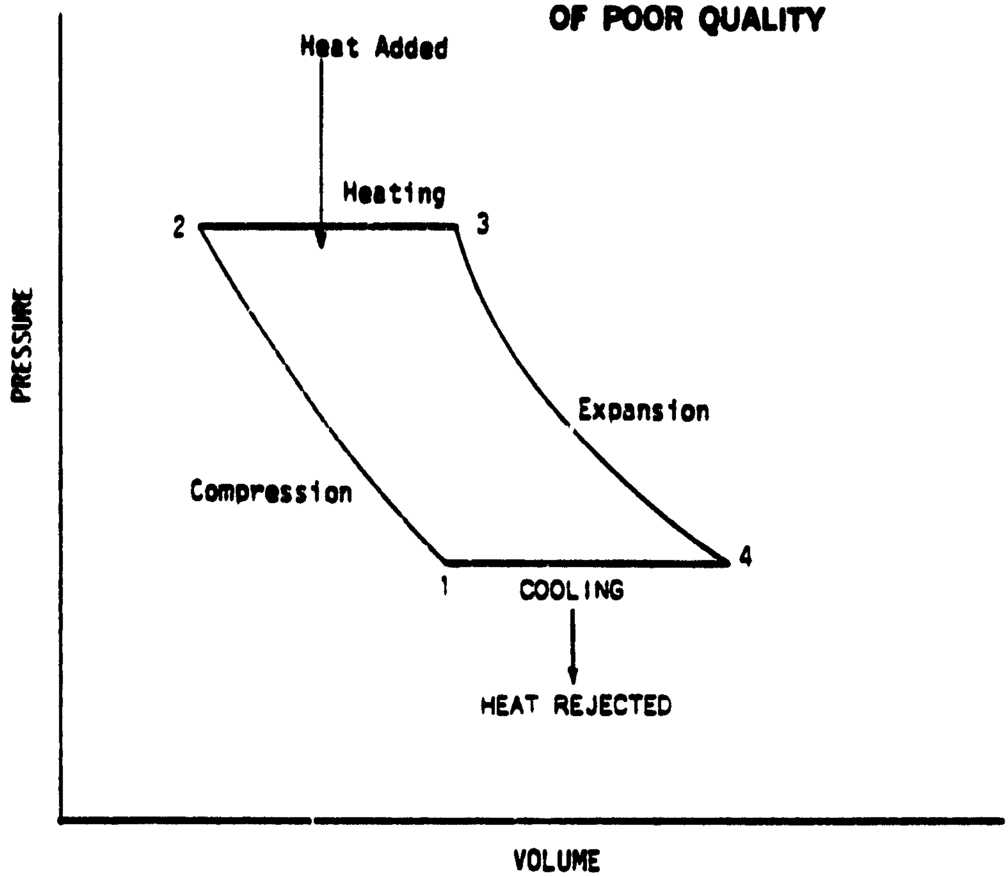

Figure 8-4. Brayton Cycle on Pressure-Volume Diagram

cycle engine. The dashed lines mark the present upper limit of gas turbine technology. The combined cycle technology refers to gas turbines with Rankine bottoming cycles to recover and convert exhaust heat. Medium speed Diesel engines, as used in locomotives, fall just below the upper bound of the Diesel band. A thermal efficiency of 40% at 3000 hp is typical of new railroad Diesel engines.

In all gas turbines, power from the turbine is used to drive the compressor. In those gas turbines where the output power is in the form of shaft power, there are two variations of the system. A single-shaft engine has the compressor, the turbine, and the output load all on one shaft. The other variation is known as a dual-shaft, two-shaft, or free-turbine gas turbine engine. This variation has two separate turbines, one to drive the compressor and the other to provide the output power. Most utility and automotive gas turbine engines are of this type. It is a more flexible design than the single-shaft gas turbine. There are a few gas turbines that are three-shaft engines.

The relative location of the fuel combustor is another design variable. Aircraft and automotive type gas turbines are considered to be internal combustion engines. The fuel is burned within the geometric confines of

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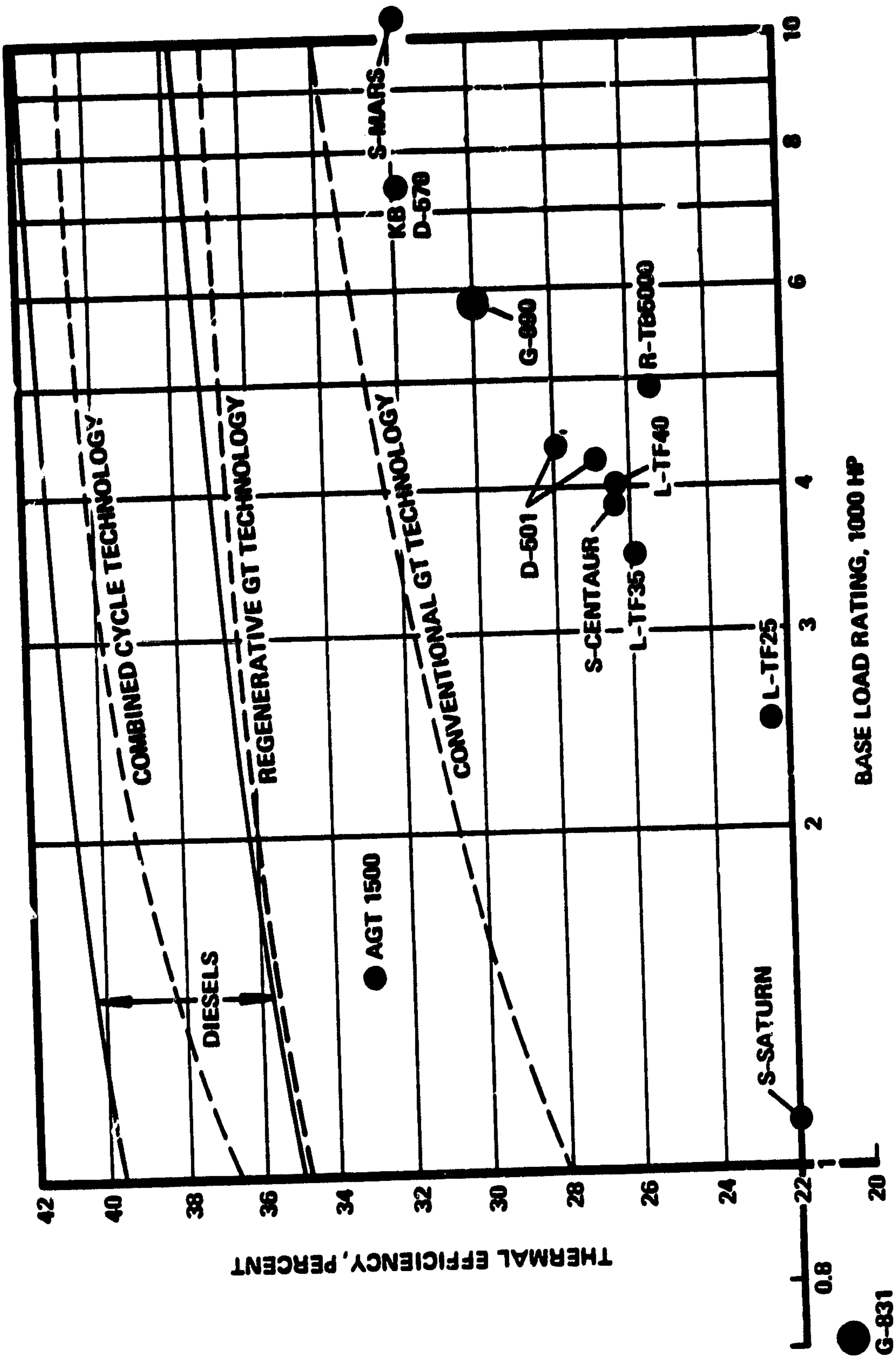

Figure 8-5. Competitive Industrial Engines
(From Ref. 8-6)

the engine. External combustion refers to the systems where this fuel is burned in a combustor separate from the turbomachinery.

It is evident that there are a wide variety of gas turbine engines to be considered for use in locomotives. However, the range of choices can be narrowed down rapidly by considering some of the characteristics of the various engines. The fuel consumption of simple cycle engines at part load is too high to be seriously considered for future locomotives. The Union Pacific-GE gas turbines were of this type. The idle fuel flow rate for a simple open cycle gas turbine is about 30% of its full power fuel flow rate. The idle fuel flow for a Diesel engine is about 3% of its full power fuel flow rate. Even at full load, the simple cycle gas turbines use more fuel than a Diesel with the same rated power.

Therefore, a gas turbine engine for a locomotive should be a regenerative cycle engine. It still could be a one or two-shaft engine as well as using either an open or closed cycle. The choice of either the one or the two-shaft configuration depends more on electrical considerations than on the gas turbine itself. A single-shaft gas turbine acts much like a Diesel engine. The speed of the engine in both cases is directly related to the output power. Since the output voltage of the alternator is speed related, the voltage varies with the power as well. In the two-shaft gas turbine, the speeds of the two shafts are independent of each other, although there is an optimum relationship between them. The voltage from the alternator can be tailored to the needs of the locomotive by controlling the second, or power turbine speed. Variable geometry power turbines are widely used on automotive gas turbines.

Using a gas turbine of either configuration permits the locomotive to use a higher speed alternator. If the alternator was directly coupled to the turbine, it would rotate at about 10,000 rpm, ten times that of the one used with a Diesel engine. The linear dimensions would shrink by a factor of 3 and the weight would be reduced by a factor of 30. The bearings on the alternator would be the biggest problem but they would be less of a problem than those in the gas turbine itself. The use of a slower speed alternator would require a gear box between the gas turbine and the alternator which adds extra complexity and maintenance. The direct coupled high speed alternator is preferable.

The choice of a combustion system depends on whether it is an open or a closed cycle engine. If it is to be a closed cycle, then it must have external combustion. If it is an open cycle engine, it can use either external or internal combustion. Figure 8-6 shows schematically the three types of gas turbine engines which will be considered for locomotive use. Before going into the details of these three types of engines, an overview of the components common to all of them is appropriate.

C. BASIC COMPONENTS

The basic components used in the three types of engines shown in Figure 8-6 are the compressor, the combustor, the turbine, and the heat exchanger. The gas turbine engine uses a very large volume of air compared to most other heat engines. In addition, the pressure ratio (maximum pressure divided by inlet pressure) requirements are quite low for regener-

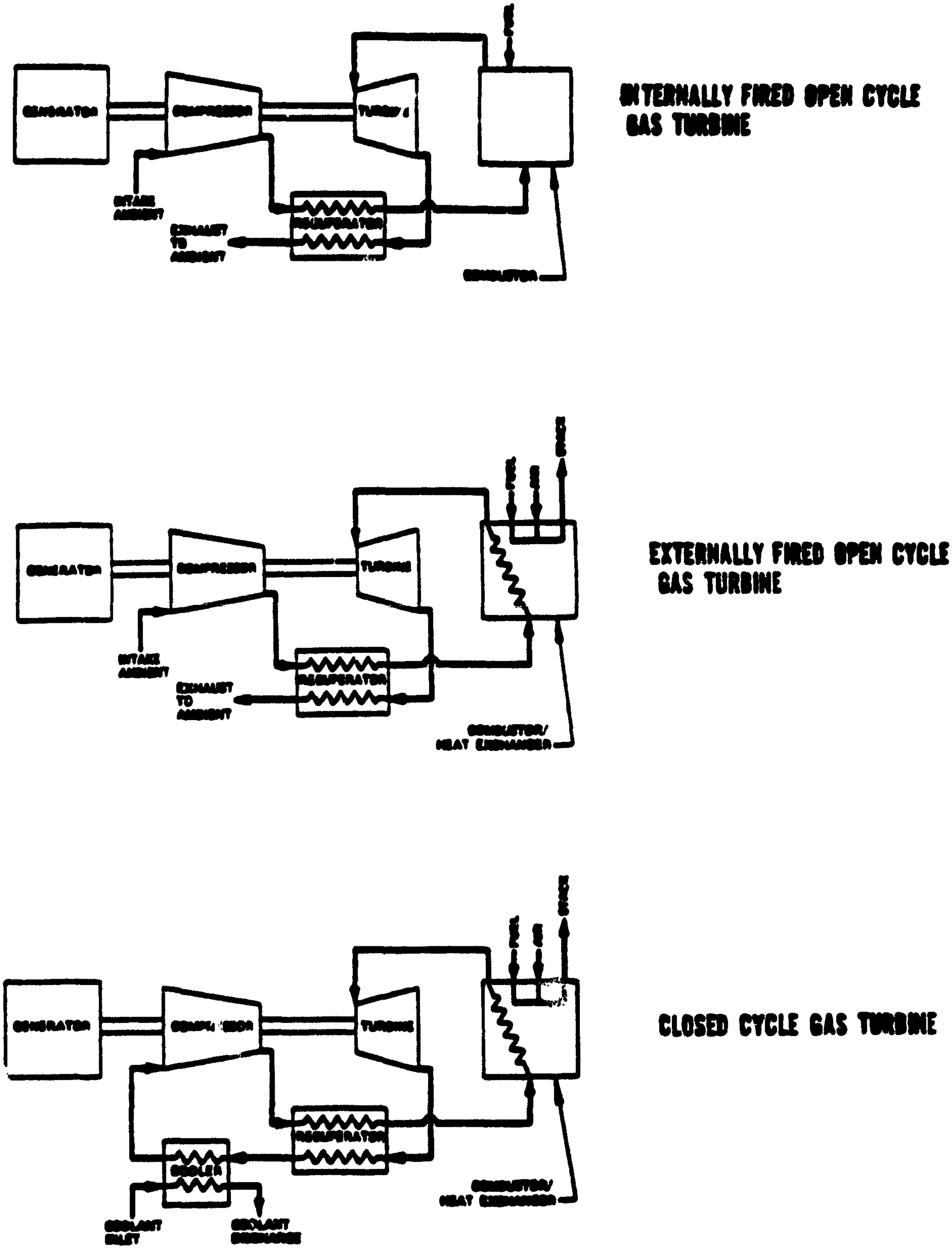

Figure 8-6. Single Shaft, Regenerative Gas Turbine Engines

ative engines. Typically, the optimum pressure ratio is between 3 and 9. Two types of compressors are commonly used in gas turbines; the centrifugal compressor and the axial compressor as shown in Figure 8-7.

The centrifugal compressor accelerates the air to a near sonic speed at the rotor tip. The air from the rotor flows into a series of passages in the diffuser. Within these passages, the kinetic energy of the air is converted to an increase in static pressure. Pressure ratios of 4 to 5 are typical in automotive regenerative gas turbines using a single compressor stage. Higher pressure ratios can be obtained by staging, where two or more compressors are arranged in series. The efficiency of multistage compressors is increased if the air is cooled between the stages. The efficiency of a single stage centrifugal compressor for a 3000 hp gas turbine engine would typically be 80 to 82%.

The axial multistage compressor is comprised of a number of axial flow rotors in series with stators between each rotor to redirect the air. The pressure ratio across the individual stage is small but the overall ratio can be quite large. Many simple cycle gas turbinr have pressure ratios over twenty. On the other hand, the GM-EMD T-45 regenerative gas turbine had 17 stages for an overall pressure ratio of 4.6 and an efficiency of 84.6%

The choice between the two types of compressors depends on the size constraints and on economic considerations. The early aircraft jet engines used centrifugal compressors. Developments in axial compressors which resulted in higher efficiencies and higher pressure ratios when combined with a smaller frontal area made these the preferred units for aircraft. For locomotives, the large diameter of the centrifugal compressor and the length of the axial compressor are not serious constraints but must be considered. The cost factors are even more important. The axial compressor is more expensive to fabricate with its numerous rotors and stators, but it is usually 3 to 5% more efficient in the sizes being considered here. This difference in compressor efficiency results in a 1% to 3% difference in fuel economy between engines using the two different compressors. A life-cycle cost analysis of the two compressors is as important as the technical details in making a choice between them.

There is another aspect of compressor selection which should be mentioned. If the design pressure ratio is 6 or more, then there is a choice between an intercooled compressor and a non-intercooled one. Intercooling has two main effects. First, it lowers the temperature of the compressor discharge air so that the temperature difference in the regenerator of a regenerative engine is greater as shown in Figure 8-8. This effect results in either a smaller recuperator for a given fuel savings or greater fuel savings for a given size recuperator. In addition, it reduces the power required to drive the compressor, thus increasing the cycle efficiency. However, at pressure ratios much below 6, the gains may not be great enough to justify the increased complexity and cost of intercooling. It is a design trade-off that should be explored in the early stages of an engine project.

Just as with the compressors, there are two basic types of turbines. The radial inflow turbine is the analog of the centrifugal compressor.

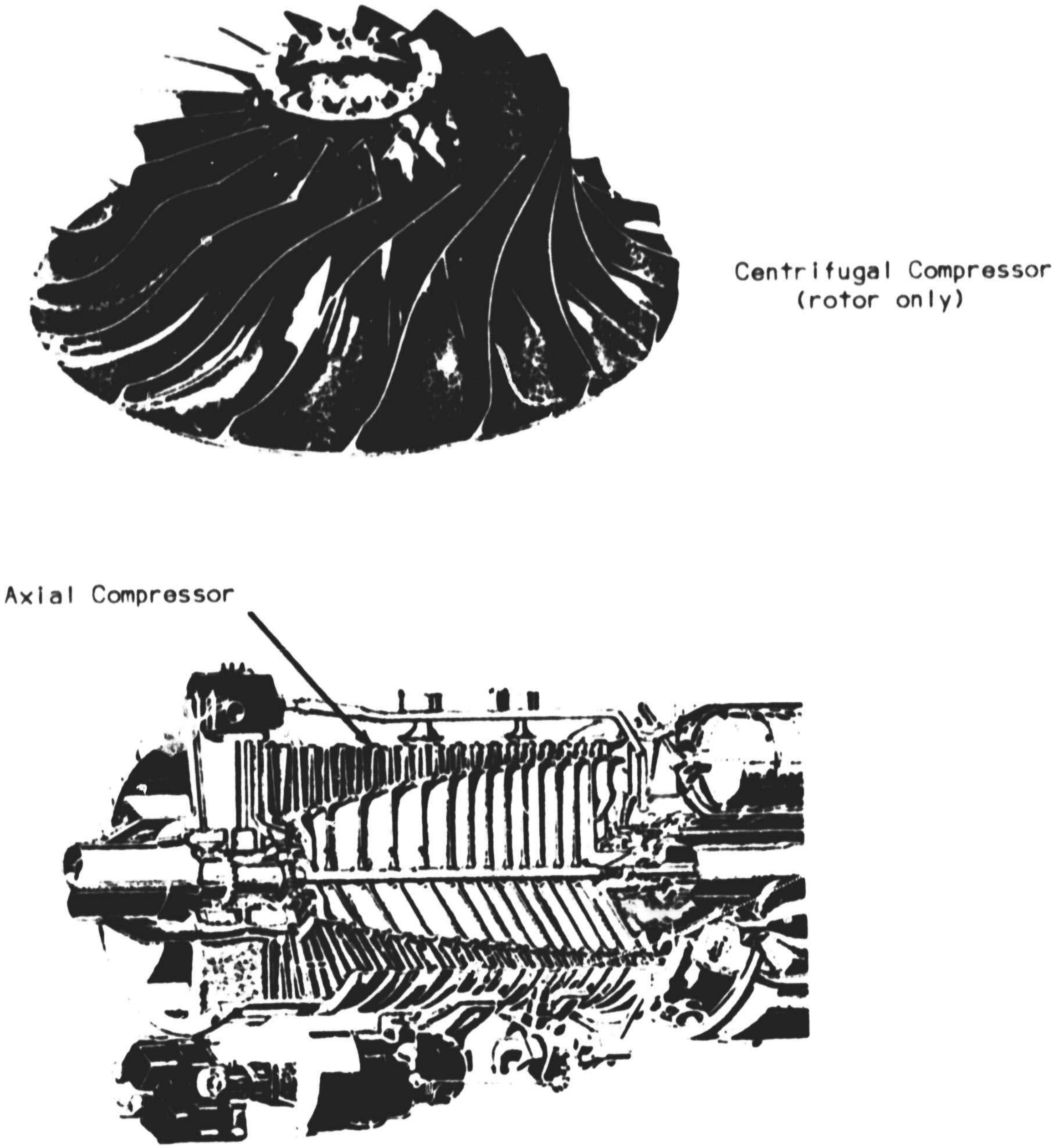

Figure 8-7. Centrifugal and Axial Compressors

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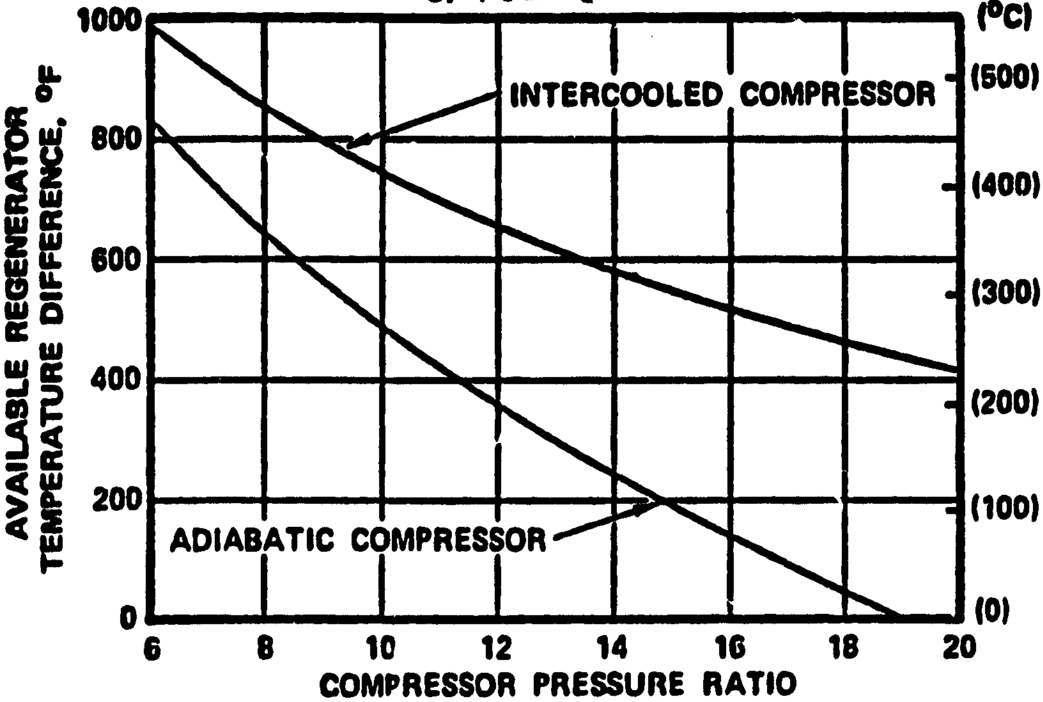
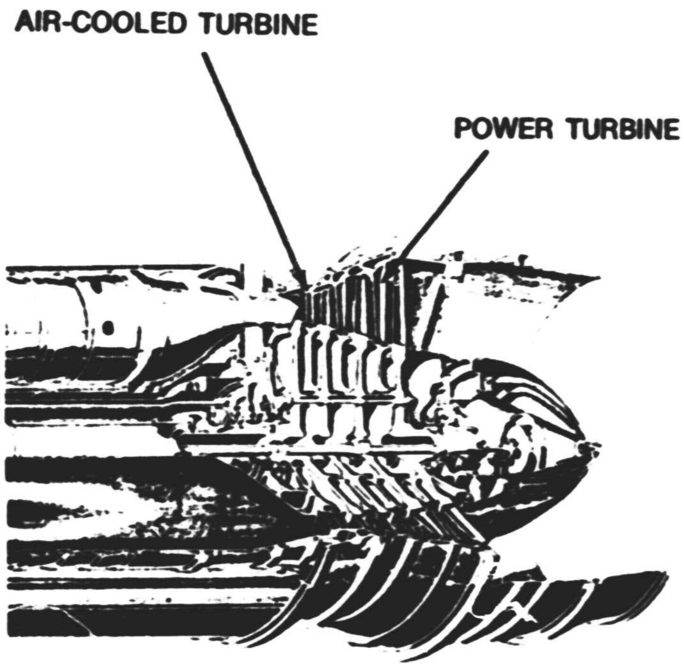


Figure 8-8. Compression-Cooled Regenerative Concept Regenerator Temperature Differences (From Ref. 8-6)

The hot, high pressure gas enters the turbine at its perimeter and flows inward. It is turned 90° and flows out of the turbine in an axial direction. The axial flow turbine is the analogy of the axial compressor. Figure 8-9 shows the two turbines. Most of the comments about the two types of compressors also apply to the turbines. The biggest difference is in the number of stages required for an axial turbine. Because of stall, the pressure ratio across a compressor stage is limited to 1.2 to 1. A common design value is 1.1 to 1. The pressure ratio across a turbine stage can be about 2.5 to 1 without incurring excessive losses. The axial turbine on the T-45 engine had four stages with a pressure ratio of 1.5 to 1 per stage. The efficiency of both types of turbines is in the 80 to 86% range. The axial turbine is normally slightly more efficient than the radial inflow turbine. The choice between the two types of turbines depends on design features, dimensional constraints, and economic factors just as in the selection of the compressor. The choice also depends on whether the engine is a single or two-shaft engine. The radial inflow turbine and the axial flow turbine are both suitable for single-shaft engines. The axial flow turbine is definitely preferred for two-shaft engines. The staging of radial inflow turbines is not recommended because of the complicated ducting requirements.

The heat exchanger used to recover exhaust gas energy, also, comes in two forms, the regenerator and the recuperator. The recuperator is a fixed boundary heat exchanger where heat is transferred through a wall from the exhaust gas stream to the compressor outlet stream. The automotive



Axial Turbine

Radial In-Flow Turbine

Figure 8-9. Turbines

radiator or a tube and shell condenser are examples of fixed recuperators. The regenerator is a periodic flow heat exchanger and normally rotates through the gas stream. A section of the regenerator is immersed in the exhaust gas stream where it absorbs heat. This section subsequently moves to the compressor outlet stream where it releases the heat to the air stream. Diagrams of the two basic forms are shown in Figure 8-10. The typical rotary regenerators for gas turbines are either cylinders (drums) or disks as shown. In both types of regenerators, there is a network of seals which are designed to prevent the high pressure, relatively cold compressor air from leaking into the low pressure, hot exhaust gas stream. The seals must slide across the regenerator surface, stand high temperatures, and be flexible enough to follow the distortions of the regenerator during thermal transients. Although regenerator seals have been in development for 20 years, a completely satisfactory seal has yet to be developed. The regenerator matrix, the heat storage part, was originally made of thin stainless steel strips. Later, ceramic matrices were developed for automotive use. The T-45 engine used a stainless steel matrix in its drum regenerator.

The effectiveness of the heat exchanger is defined as the ratio of the actual heat transferred to the maximum heat available. Regenerator effectiveness at design point may be 85 to 90% and higher at part load. Effectiveness levels of up to 98% are possible with ceramic matrices and up to 94% with stainless steel matrices. The difference is due to conduction losses. Recuperator effectiveness is less than that of regenerators and it is usually in the 75 to 85% range. The size of the recuperator is frequently the limiting factor on effectiveness. An effectiveness of 90 to 95% is possible if there is sufficient room for recuperators in the installation.

The regenerator, although having problems with leakage, has high effectiveness. The recuperator, on the other hand, suffers from thermal fatigue and low effectiveness but uses static seals which do not present leakage problems. Provided that the thermal fatigue problems can be solved, and it appears that they can, then the recuperator would be preferable for the type of service encountered by the railroads. The recuperator should be more reliable and durable, but larger than the regenerator. Regardless of which type is used, the heat exchanger is a major problem area and requires further development.

D. OPEN CYCLE, INTERNAL COMBUSTION GAS TURBINE

This is the type of gas turbine engine that GM-EMD built during the T-45 program. It is certainly the most common and best developed of the regenerative cycle engines. The automotive gas turbines developed by General Motors, Ford, and Chrysler are all of this type.

The major components in this engine as shown in Figure 8-11 are the compressor, one or more turbines, the combustor, and the heat exchanger. In this engine, ambient air goes through an air cleaner into the compressor. Depending on the engine design, the compressor pressure ratio is probably between 3 and 9. Usually a pressure ratio of 4 to 6 is optimum for regenerated engines. From the compressor the air flows through the heat exchanger where it is heated by the turbine exhaust gases. The air

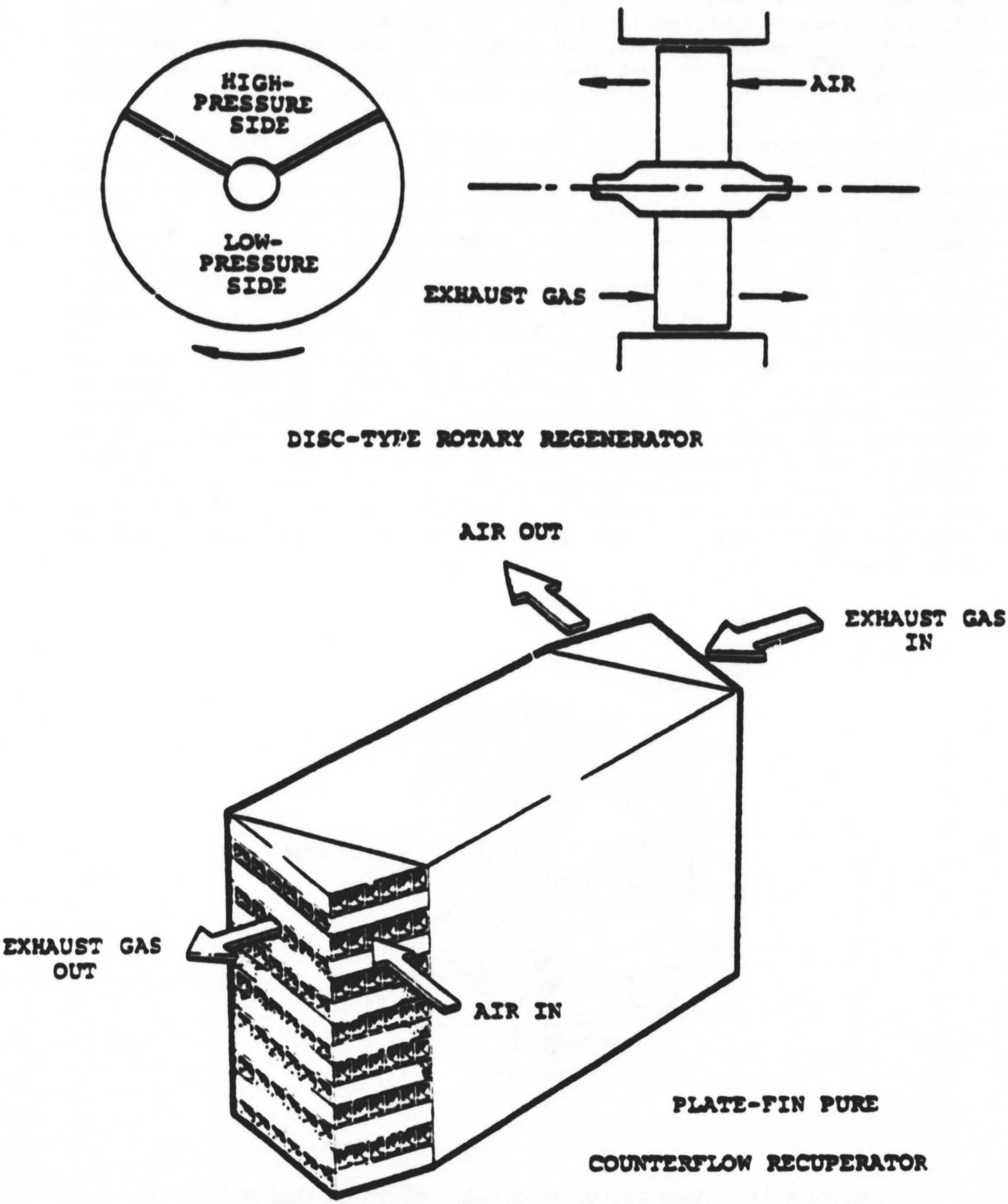


Figure 8-10. Regenerators and Recuperators

then goes to the combustor where fuel is injected and iginited. The hot, high pressure gas resulting from the combustion of the fuel enters the turbine. From the turbine, the exhaust gas goes to the heat exchanger to provide the energy to heat the compressor discharge air. The exhaust gas is discharged to the atmosphere after leaving the heat exchanger.

The only major component in this type of engine not discussed earlier is the combustor. The combustor serves two purposes. One is to provide suitable conditions for the ignition and stable combustion of the fuel and the other is to combine the combustion products and the excess air into a homogeneous mixture before it enters the turbine. Because of material limitations, the turbine inlet temperature must be well below the temperature of a combusted stoichiometric fuel-air mixture. The turbine inlet temperature in industrial service is about 1800° F to 2000° F. When ceramic turbines are developed, the temperature may be raised to 2500° F. The temperature is controlled by operating the engine at extremely lean conditions. The air-fuel ratio is so high (up to 150 to 1) that it could not sustain combustion if the fuel-air mixture was homogeneous. Gas turbine combustors vary greatly in design but most are based on dividing the air stream so that only part of it is used for combustion and the rest is mixed into the combustion products before reaching the turbine. A variety of combustor types have been developed or proposed over the years for internal combustion gas turbines. These are diffusion flame, dual stage, evaporative, and catalytic.

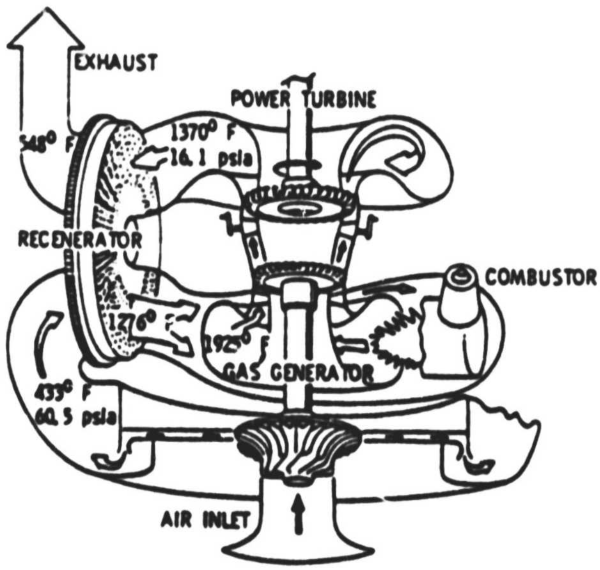


Figure 8-11. Open Cycle, Internal Combustion Engine System
(From Ref. 8-6)

These four combustors illustrate the main types of fuel handling, preparation, and combustion reaction. Although gas turbines can burn a wide range of liquid and gaseous fuels, the most common fuels are liquids. The first two types of combustors are basically designed for liquid fuels.

The third and fourth could be used for either gaseous or liquid fuels. The diffusion flame combustion is the most common type and is usually fueled by liquid hydrocarbons although methanol has been used successfully. The conventional diffusion flame combustor shown in Figure 8-12 utilizes a 90 degree spray atomizing nozzle injecting fuel int a compact primary zone in which about 30% of the compressor air flow is inducted through a number of holes or slots in the combustor liner. The fuel-air mixture is ignited with a spark plug and burns at a near stoichiometric temperature of 3500° F. The remainder of the compressor air is introduced as dilution air through ports downstream of the primary zone. The liner is cooled by the dilution air.

The fuel need not be injected through one nozzle or through one type of nozzle. Figure 8-13 shows a two stage combustor where two different nozzles are used to introduce fuel into the reaction zone. The primary one is at the right where part of the fuel is injected, mixed with air, and ignited. On the left side, is the second nozzle injecting the remaining fuel. This fuel mixed with air is introduced into the already burning mixture from the primary zone. The remaining compressor air is mixed in below the main reaction zone.

In both of the combustors discussed so far, the fuel is atomized as it is injected. The burning takes place around the individual droplet and not in a flame such as a Bunsen burner produces. The rate of combustion is controlled by the rate of heat being absorbed by the droplet and the

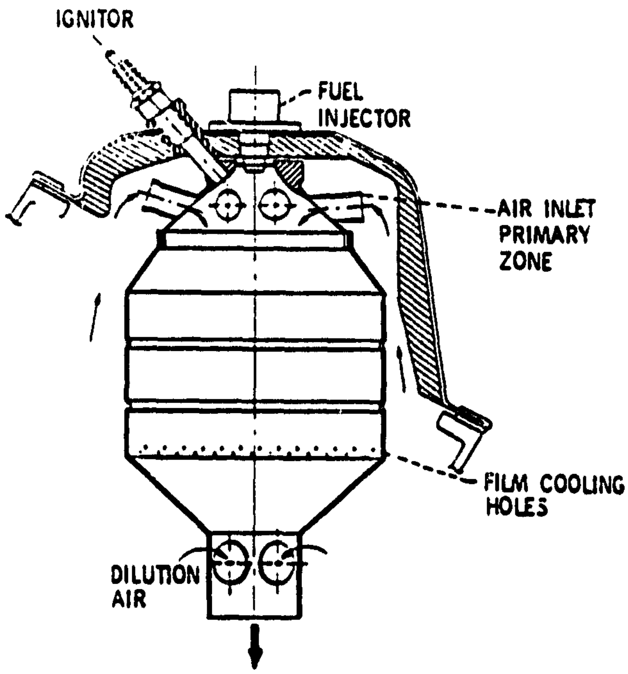


Figure 8-12. Diffusion Flame Combustor

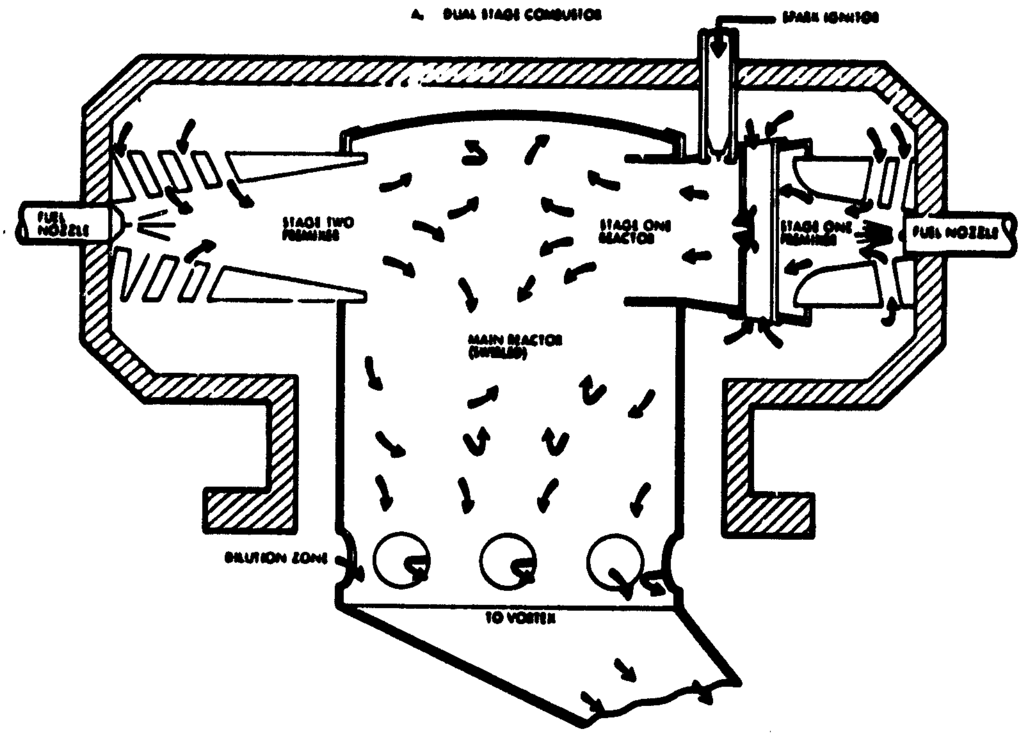


Figure 8-13. Schematic Diagram of a Dual Stage Combustor

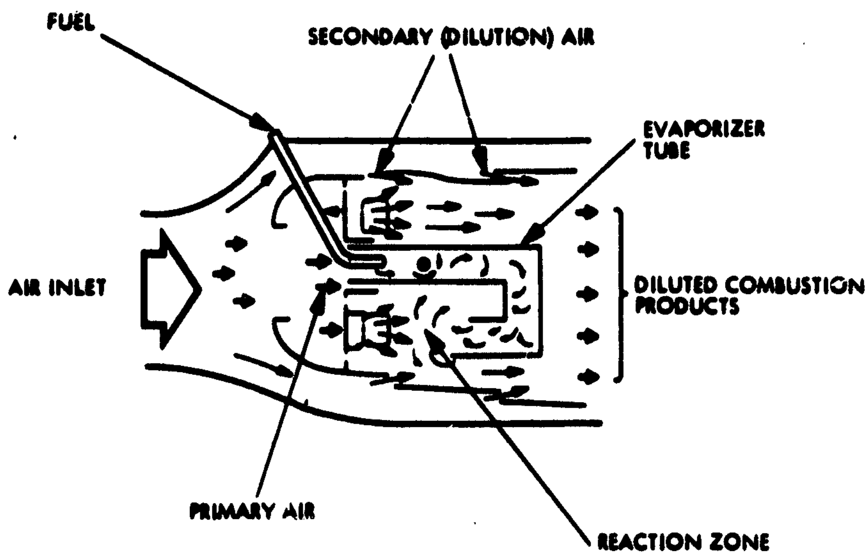


Figure 8-14. Schematic of an Evaporative Combustion Chamber

rate at which the fuel is vaporized at the surface of the droplet. The combustion takes place where the vaporized fuel and the air mix in near stoichiometric proportions.

The third type of combustor eliminates the droplet evaporation mechanism in the combustion process. The fuel is vaporized prior to its introduction into the combustion zone. In some cases, the vaporized fuel is mixed with air prior to being injected into the combustion zone. In this type of combustor, the air-fuel mixture is homogeneous and a conventional flame is present. One version of an evaporative combustion chamber is shown in Figure 8-14. The advantage of this type of combustor is that combustion is usually very clean with little or no soot or smoke being produced. The biggest problem comes from the evaporation process. Many fuels, including most hydrocarbon fuels, may decompose in the evaporator causing deposits to build up that restrict further heat transfer. Incomplete vaporization and evaporator burnout are the common result. This type of combustor is best suited to light hydrocarbons, methanol, liquid hydrogen, and ammonia.

The catalytic combustor is relatively new in the gas turbine field. The fuel reacts with the air on the surface of a suitable catalyst. As a result, very lean mixtures that are well outside the flammability limits will readily react. The temperatures involved are comparable to the required turbine inlet temperature. The high thermal capacity of the bed tends to stabilize the combustion. It also makes the gas turbine harder to start since the bed must be warmed up before fuel is introduced. The reaction rate of a catalyst bed is relatively slow compared to the rate in a flame. To improve both the cold startability of the catalyst bed and to increase the reaction rate, a hybrid catalytic combustor has been proposed. In this design, fuel and air are mixed and partially burned before they reach the catalyst bed. In the bed itself, the reaction is completed and the hot gas leaving the bed is uniform in temperature with very little smoke or soot.

One advantage claimed for the gas turbine engines is their multi-fuel capability. This claim must be qualified to limit it to gaseous and liquid fuels for open cycle, internal combustion gas turbines. Solid fuel, notably coal, was tried in the General Electric turbines used by Union Pacific. Rapid turbine blade erosion resulted and the program was ended. There have been numerous other attempts to use solid fuels but none, for one reason or another, have been successful. Blade erosion and nozzle plugging by partially burned fuel particles were the principal causes of failure.

It is not clear at the present time whether or not a combustor system could be developed to operate on a number of different fuels interchangeably. Generally, injectors have to be sized on a volumetric basis. Fuels such as the alcohols will require larger nozzles than will kerosene. Any gaseous fuel will require even larger nozzles. Changes may be required to the primary zone air holes if a fuel with a substantially different stoichiometric air-fuel ratio is substituted for the regular fuel. The reaction times for some fuels like heavy hydrocarbons may require a longer combustor to insure complete burning than is needed for propane or butane. The catalytic combustor may be especially useful for multi-fuel operation but this has yet to be demonstrated.

The thermal efficiency of a gas turbine varies to some degree with fuel for an engine designed for one specific fuel. Table 8-1 shows the variation in efficiency for an engine designed along the lines of the GM-EMD T-45 engine and for use with JP-5 fuel. The variation in efficiency is from a minus 1.1 percentage points to a plus 2.3 percentage points.

The thermal efficiency of a gas turbine is highly dependent on the turbine inlet temperature. Over the years, the turbine inlet temperatures have been raised by the development of water and air cooled blades, by the development of new super alloys, and the use of ceramics. Figure 8-15 shows the increase in the turbine inlet temperature over the last 30 years. The thermal efficiency of the engine is also influenced by the individual component efficiencies, leakage, pressure losses in the flow stream, the pressure ratio, and compressor-turbine matching.

A sensitivity study was made for an open cycle, regenerative gas turbine engine. Table 8-2 presents the baseline conditions for this engine. Figures 8-16 through 8-20 show the effects of turbine inlet temperature, pressure ratio, compressor efficiency, heat exchanger effectiveness, and turbine efficiency on the engine cycle thermal efficiency. It is obvious from these curves that the real gain in thermal efficiency comes from higher turbine inlet temperatures. The baseline gas turbine engine used in this study had an experimental thermal efficiency of 31.5%. Rematching the engine, together with modest gains in component efficiencies, might have brought it up to 34% efficiency.

The results of the sensitivity study indicates that current technology would make a 37 to 38% thermal efficiency possible. Garrett AiResearch (Ref. 8-6) has indicated that thermal efficiencies in the 42% to 45% range may be attained with components of current production technology using their "CCR" (compression-cooled regenerative) concept. They also indicate that an advanced version (10-20 year time frame) using a 92% effective regenerator, a turbine inlet temperature of 2400° F, and a pressure ratio of 12 to 1 would have a thermal efficiency of 53%. The projected size of this advanced CCR gas turbine, rated at 4000 hp, will be 8.5 ft by 3 ft by 7 ft and it would weigh 8000 lb.

Past experience would indicate that given the constraints on a locomotive for size, reliability, and durability, the maximum gas turbine thermal efficiency which could be realized in the next 20 years will be about 42% for an open cycle, internal combustion engine. The higher efficiencies quoted by Garrett may be attained in utility application where there are less stringent restraints. Garrett's conclusion is that this type of engine is better suited to high horsepower to weight ratio freight service.

E. OPEN CYCLE, EXTERNAL COMBUSTION GAS TURBINE

This general type of engine is very similar to the internal combustion gas turbine except for the combustion system. There are several variations of this type of engine and two are shown in Figures 8-21 and 8-22. The first one is the same as the internal combustion gas turbine except for the combustor which is shown here as an atmospheric fluidized bed (AFB). Only the heat is transferred to the working fluid (air) in this engine.

Table 8-1. Effect of Alternative Fuels on Thermal Efficiency of Gas Turbines

Fuel	Thermal Efficiency (%)	
	Open Cycle Regenerative Gas Turbine	Closed Cycle Recuperative Gas Turbine
JP-5 (Baseline)	31.5	30.4
Bunker Fuel	30.4	29.3
Shale Liquids	31.5	30.4
Coal Liquids	31.5	30.4
Ethanol	32.6	31.5
Methanol	33.8	32.6
Methane	31.5	30.4
Acetylene	31.5	30.4
Hydrogen	33.8	30.4
Carbon Monoxide	30.4	30.4
Ammonia	31.5	30.4
Vegetable Oil	31.5	30.4
Coal	-	29.3
Coal/Oil Slurry	-	29.3
Coal/Methanol Slurry	-	29.3

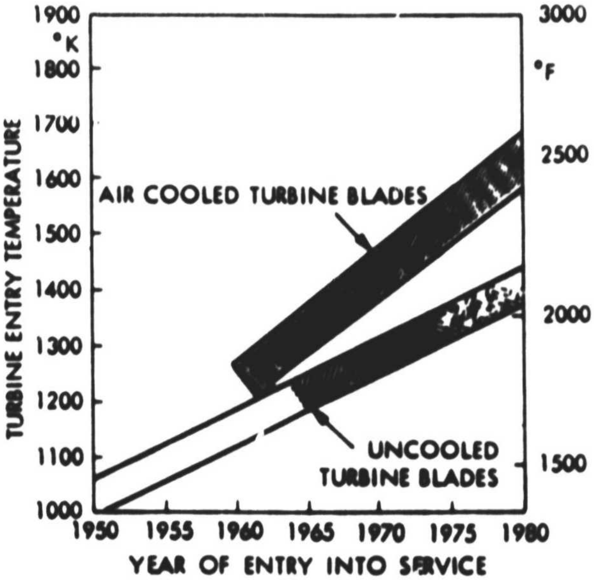


Figure 8-15. Turbine Inlet Temperatures
(From Ref. 8-7)

Table 8-2. Baseline Gas Turbine Locomotive Conditions

Barometric Pressure, in. Hg. abs.	29.30
Compressor Inlet Pressure, in. Hg. abs.	28.86
Compressor Inlet Temperature, °F	90
Gasifier Turbine Inlet Temperature, °F	1850
Accessory Power, hp	103
Compressor Efficiency (total to static), %	85
Burner Efficiency, %	99
Regenerator Effectiveness, %	91.5
Gasifier Turbine Efficiency, %	88
Power Turbine Effi iency, %	84
Gasifier Section Mechanical Efficiency, %	99.35
Power Section Mechanical Efficiency, % (incl. reduction gear box)	98.5
Gasifier Turbine Speed, rpm	6500
Power Turbine Speed, rpm	5400
Compressor Pressure Ratio (total to static)	4.1
Compressor Air Flow, lb./sec. (W_a)	51.15
Gasifier Turbine Cooling Air, % W_a	4.1
Regenerator Leakage, % W_a	4.0
Miscellaneous Air Leakage, % W_a	1.0
Inlet Pressure Loss, % ($\Delta P/P$)	1.5
Exhaust Pressure Loss, % ($\Delta P/P$)	.5
Burner Pressure Loss, %	3.0
Regenerator Pressure Loss, % ($\Delta P/P$) air	.56
Regenerator Pressure Loss, % ($\Delta P/P$) gas	4.44
Shaft Power, shp	4500
Specific Fuel Consumption, lb./SHP-hr	.400
Thermal Efficiency, %	34.2

The engine system shown in Figure 8-22 is a hybrid of the internal combustion gas turbine and the external combustion system of Figure 8-21. The air from the compressor is split, part being used for the fluidized bed combustion and part merely being heated in the bed. The air used for combustion is cleaned up in a series of cyclones and is then mixed with the heated air. The turbines receive a mixture of heated air and combustion products. The fluidized bed in this case operates at a compressor discharge pressure of about 7 atm. This type of pressurized fluidized bed (PFB) combustor like the AFB combustor is especially useful for burning high sulfur coal or residual oil. The bed is made up of granular limestone pebbles that react with sulfur to form calcium sulphate which remains in the bed. Part of the bed is drawn off to remove the spent limestone and to replace it with fresh limestone.

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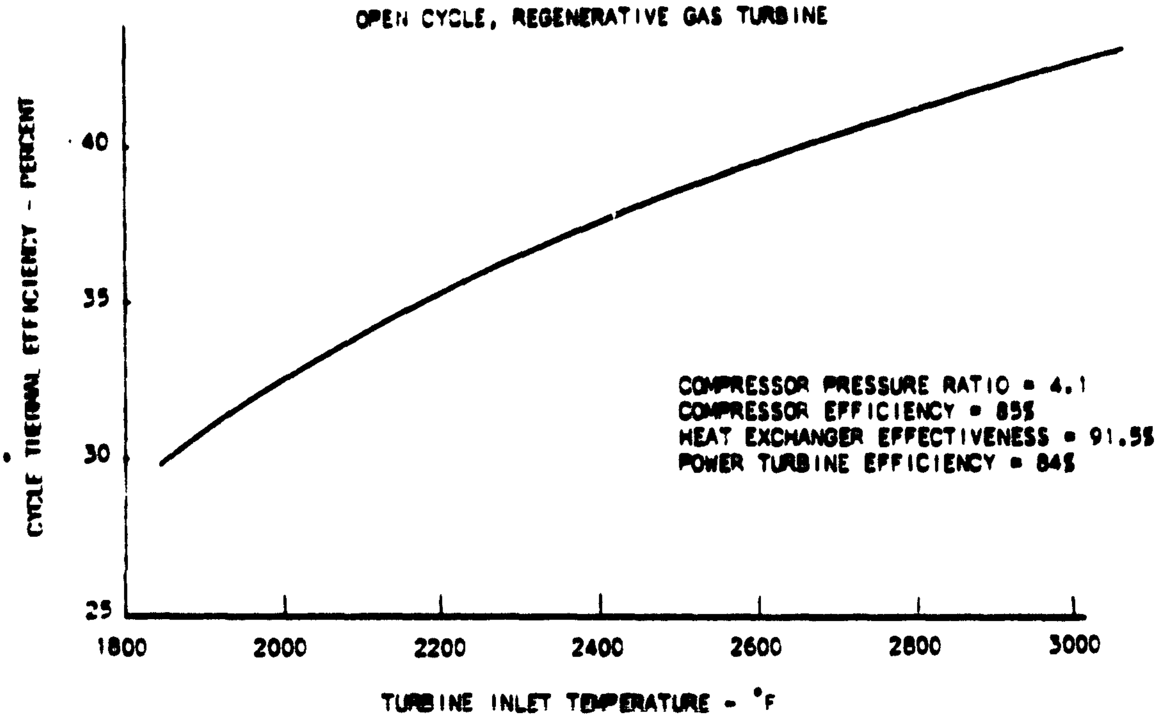

Figure 8-16. Turbine Inlet Temperature vs. Cycle Thermal Efficiency

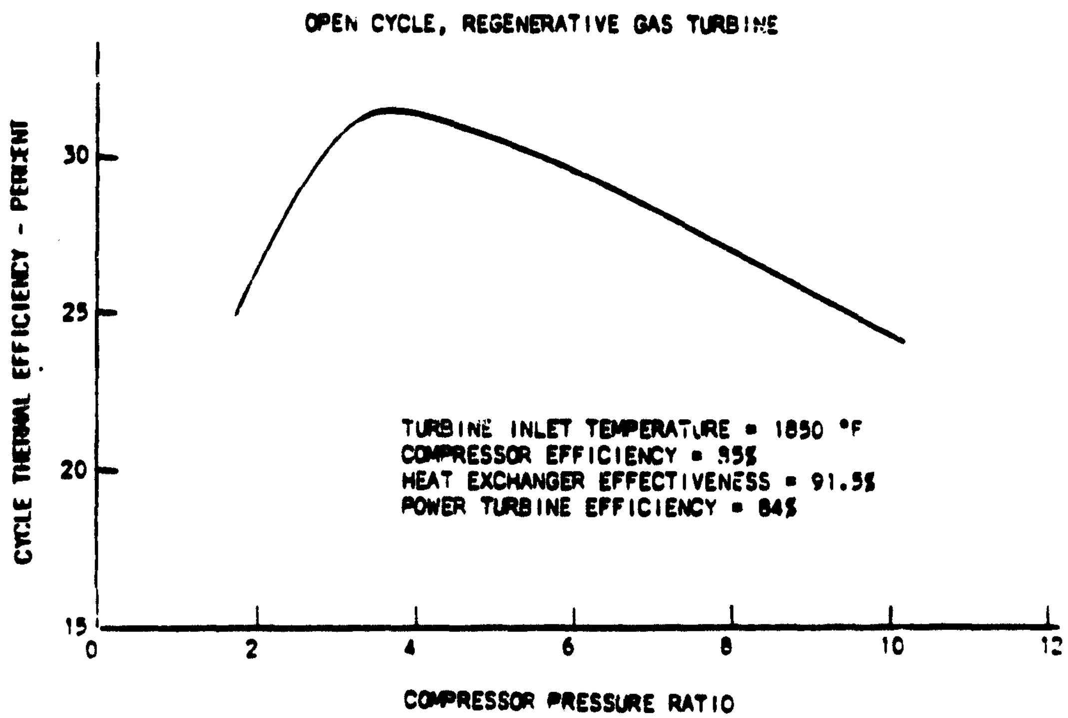

Figure 8-17. Compressor Pressure Ratio vs. Cycle Thermal Efficiency

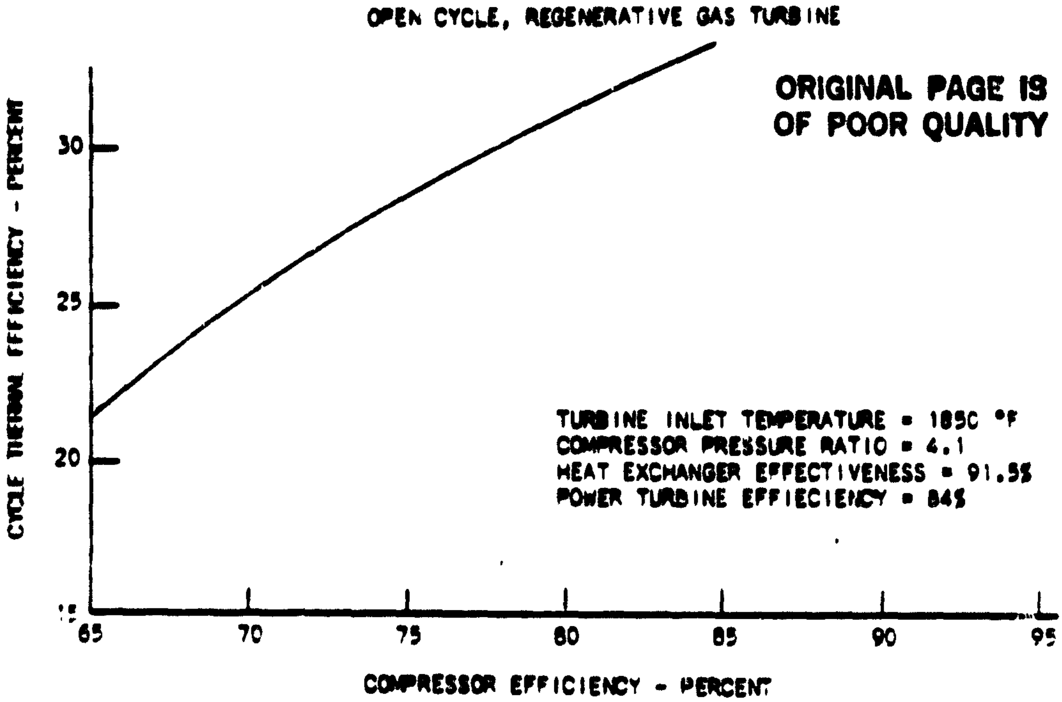

Figure 8-18. Compressor Efficiency vs. Cycle Thermal Efficiency

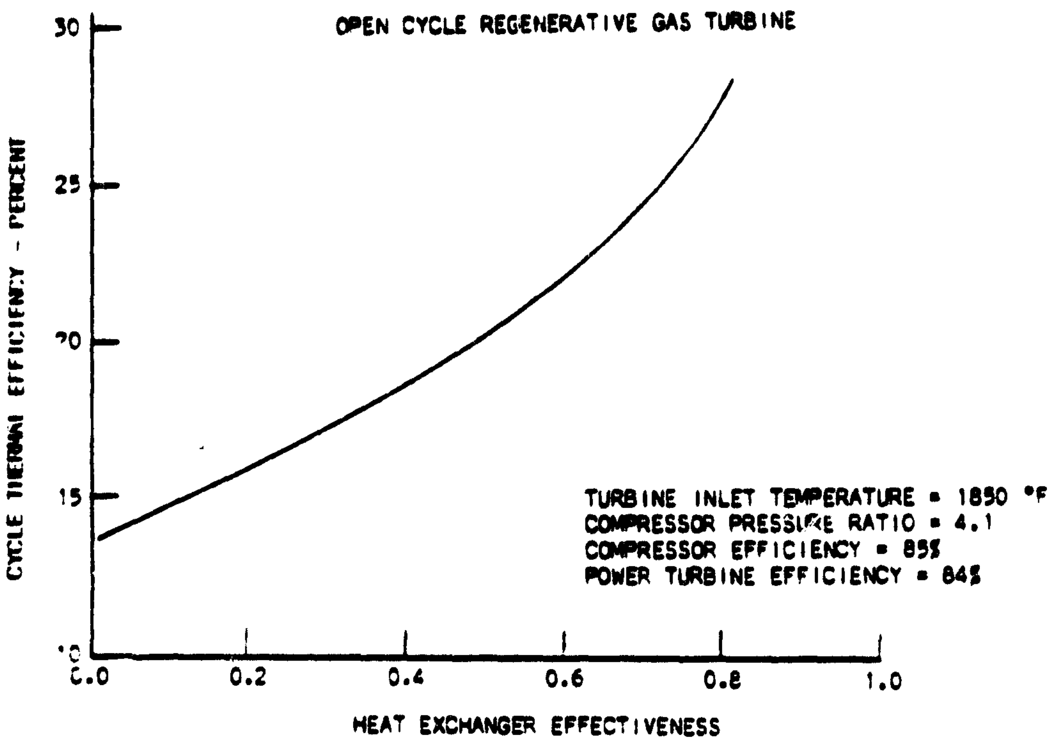

Figure 8-19. Heat Exchanger Effectiveness vs. Cycle Thermal Efficiency

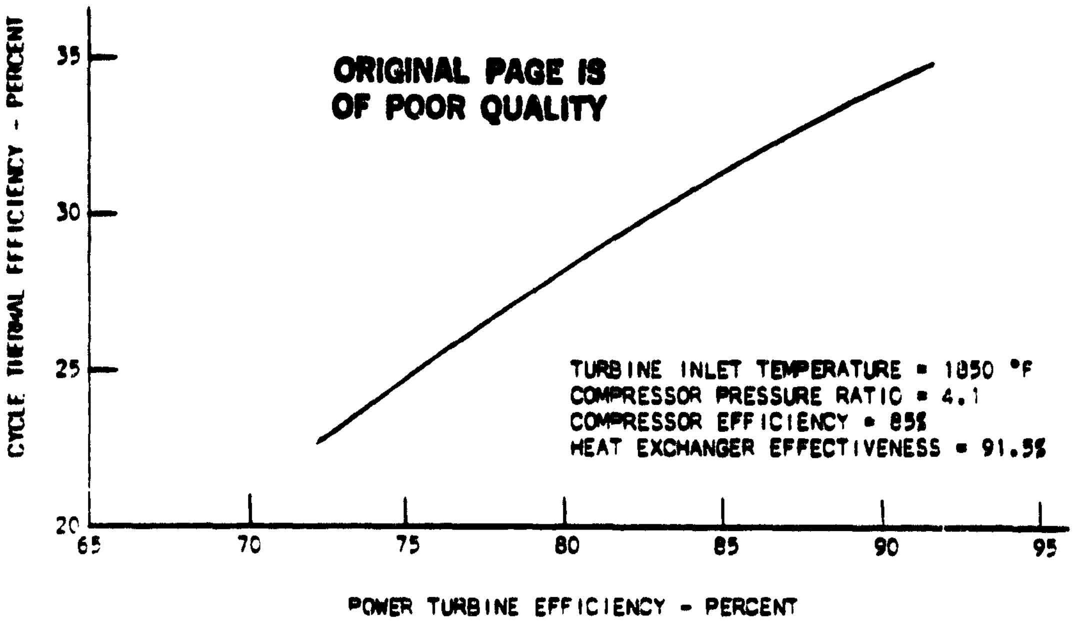

Figure 8-20. Power Turbine Efficiency vs. Cycle Thermal Efficiency

The components of a fluidized bed, either atmospheric or pressurized, are shown in Figure 7-3 in Section VII. The bed is made up of granular particles with a major dimension typically in the range of .020 to .120 in. It could be an inert solid, such as sand, or as mentioned before, it could be a sorbent for sulfur dioxide such as limestone or dolomite.

The bed is fluidized by an air stream flowing up through the bed at a rate sufficient to support its weight. At very low flow rates, the air merely seeps through the bed without disturbing the particles. At the minimum fluidizing velocity, typically 1 ft/sec, the static granular bed begins to move, the voids between particles increase, and the bed assumes the properties of a turbulent fluid. For further increases in air velocity, the pressure drop across the bed remains relatively constant while the voids increase slowly. At some point, the velocity is high enough for larger voids to form which "float" to the surface like bubbles. The bed is now in the so-called bubbling or boiling regime. At even greater velocities, approximately 10 to 20 ft/sec, increasing amounts of bed material are entrained in the air stream and the surface of the bed loses definition.

Some entrainment of very fine bed particles occurs at all velocities. The entrainment velocity of a particle is proportional to its diameter and depending on the bed particle size distribution, some amount of the bed mass is carried away starting with the smallest particles. This process of entrainment is called elutriation. When the air velocity reaches the terminal velocity, the bed is literally blown out through the exhaust. The air velocity is normally chosen to be above the minimum fluidizing velocity in the bubbling regime. Any fine particles such as ash which are blown out of the bed are collected in cyclones for recycling or disposal.

Because of the turbulent nature of a fluidized bed, heat transfer in the bed is very high. The heat transfer coefficients are typically 40 Btu/(hr-ft^2-°F) for horizontal surfaces and 50 Btu (hr-ft^2-°F) for vertical surfaces. The gentle scrubbing action of bed particles against metal tube surfaces, combined with radiant and convective heat transfer, is responsible for the high in-bed heat transfer coefficients. Experience has shown that as long as an oxidizing atmosphere is maintained in the bed, tube erosion is not a problem.

If the bed is being used for sulfur control, the bed temperature is optimized for the sulfation reaction in which SO_2 generated during combustion combines with oxygen and calcined limestone (CaO) to form calcium sulfate ($CaSO_4$). To maintain the bed in the 1500° to 1600° F range for sulfation, it is necessary to continually remove heat energy from the bed. Heat transfer surfaces are immersed within the bed. The high heat transfer coefficients make it possible to have a smaller heat exchanger in the bed

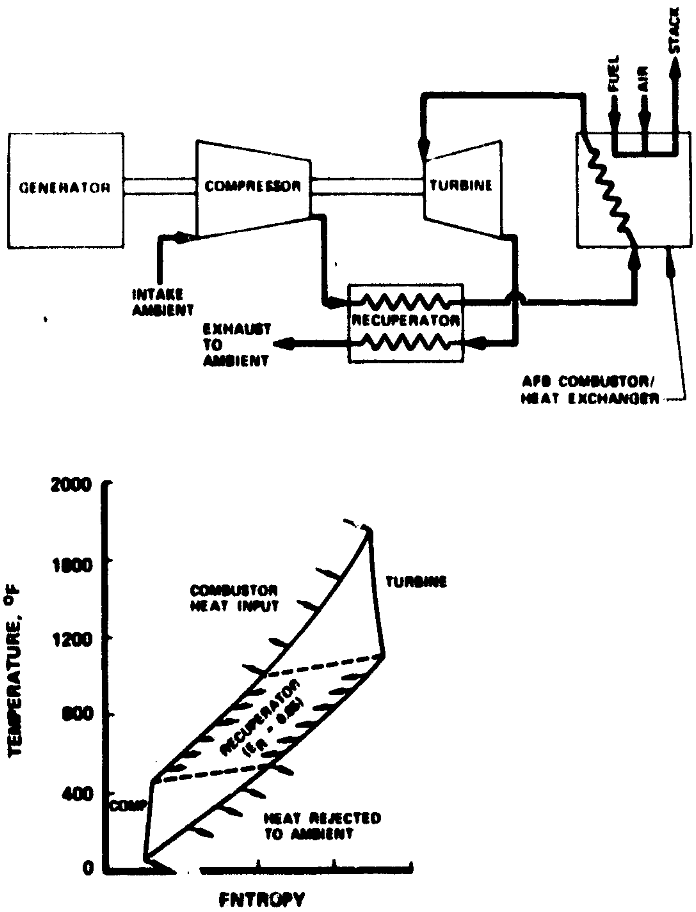

Figure 8-21. Externally Fired Open Cycle Gas Turbine Power System (From Ref. 8-6)

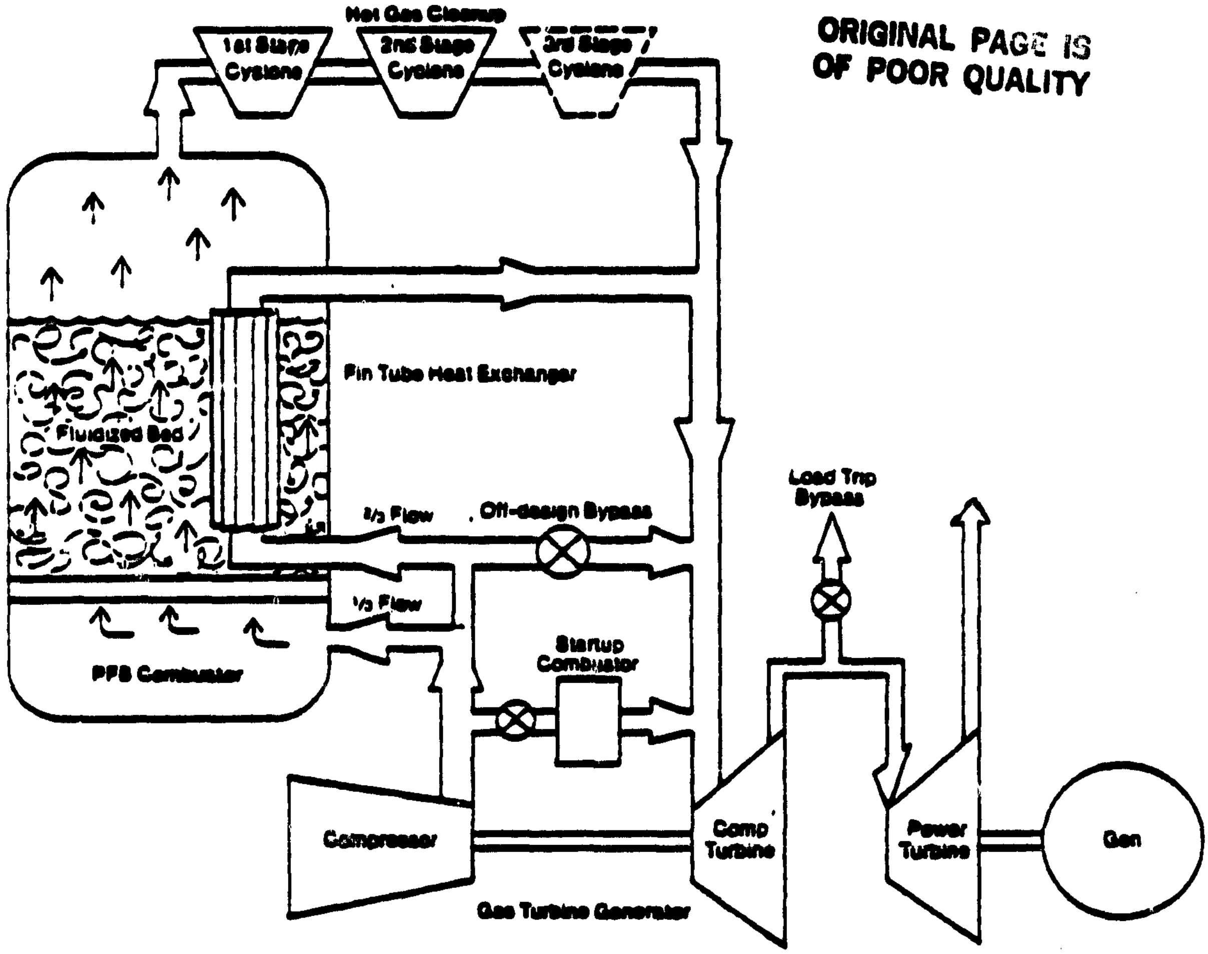

Figure 8-22. Gas Turbine Generator System
(From Ref. 8-8)

than would be possible if it were located in the heated gas above the bed for the same amount of heat transferred. In addition to the in-bed heat exchanger, there may be others above the bed in the exhaust system. Figure 8-23 shows a cut-away of a fluidized bed combustor designed for utility use. The stack heat exchangers are shown. Additional preheaters in the stack can be used. This figure also shows the location of most of the other major features of this type of combustion system.

The fluidized bed combustor has many attractive features, not the least of which is its multi-fuel capability. Once the bed is up to its operating temperature, virtually any combustible material that can be introduced into the bed will burn. The method of introducing the fuel is quite flexible. For example, coal can be mechanically spread on top of the bed or injected into the bed pneumatically from the side or up through the air distributor plate at the base of the bed. Limestone or any other bed material can be introduced separately or can be added to the coal stream. A good candidate for use in a fluidized bed is high sulfur residual or even crude oil.

The fluidized bed is not the only combustion system that could be used for externally fired gas turbines. Since the lighter oils, alcohols, and gases are more suited to the internal combustion gas turbine, the only fuels that must be considered are the very heavy oils and the solid fuels of which coal is the most likely candidate. Unless sulfur cleanup is a problem, heavy oils can be preheated and sprayed into an open chamber combustor. Some solid fuels can be pulverized and pneumatically sprayed into an open chamber.

Another technique which is attractive for low sulfur coal and, possibly for wood, is thick bed gasification. This is not a new technique since it has been used for years in industry and on the Rio Turbio Railway in Argentina (Ref. 8-10).

Thick bed gasification resembles conventional stoker operations except that the bed of coal is very thick, about one foot, and the air coming up through the grate is less than is necessary for complete combustion. If the air flow is adjusted to maintain the bed at about 2000° F, then the coal is gasified and the ash tends to adhere to other ash particles and therefore sink down toward the grate. Combustion is completed by injecting air over the bed into the hot gases. Since only a portion of the combustion air goes through the bed, the air velocity in the bed is low. This low velocity means that the fly ash and carbon particle content of the exhaust will be

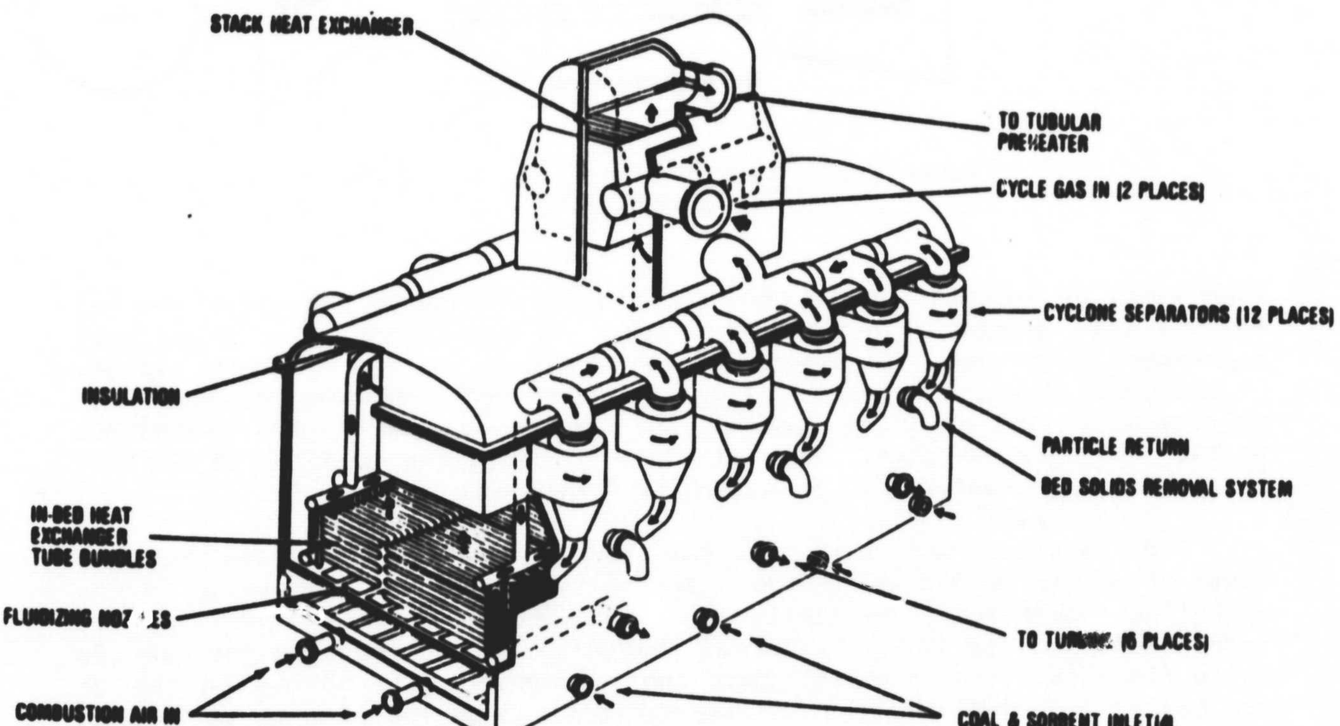

Figure 8-23. Fluidized Bed Combustor Heater
(From Ref. 8-3)

lower than it would be for a thin bed stoker system where all of the air goes through the bed. The combustion efficiency is higher and the smoke is reduced. This type of combustion system is far simpler than the fluidized bed in design and operation. Its initial cost and maintenance would be less and there is no bed material (limestone, etc.) to be replenished. However, this system is more restrictive as to the type of fuel, i.e. solid with a low sulfur content, that can be used.

The thermal efficiency of this gas turbine, as with all gas turbines, is strongly dependent on the turbine inlet temperature. When metal heat exchangers are used, the turbine inlet temperature is limited to 1550° F. The thermal efficiency of the gas turbine alone would be about 25% at this temperature and the overall efficiency including the combustion system would be about 20%. Ceramic heat exchangers (Ref. 8-6) currently under development will permit turbine inlet temperatures to be raised to 2250° F. Advanced ceramics may raise the temperature into the 2500° to 3000° F range. The cycle efficiency of open cycle, external combustion gas turbines at these elevated temperatures is 39% at 2250° F and 45 to 48% at 2500° to 3000° F. If the external combustion heater losses are included, the overall efficiencies are 34% at 2250° F and 39 to 42% at 2500° to 3000° F. At these conditions, a coal-fired locomotive would be as energy efficient as present day Diesel locomotives.

F. CLOSED CYCLE, EXTERNAL COMBUSTION GAS TURBINES

The closed cycle gas turbine is similar to the open cycle in that both have the same thermodynamic characteristics. The primary difference is that in the closed cycle the turbine exhaust is ducted back to the compressor inlet via a cooler where the cycle waste heat is rejected. The closed cycle system is shown schematically in Figure 8-24. This cycle is shown on a temperature-entropy diagram in the same figure. The effect of the recuperator on both heat input and rejection is clearly illustrated.

The working fluid in this engine can be air, helium, argon, krypton or a number of other gases. With a closed cycle, regardless of the type of working fluid, the gas is clean with no products of combustion, dust, sulfur or salts remaining. The turbomachinery and heat exchangers are not subject to fouling, erosion, or corrosion thus insuring a long low-maintenance life.

Another attractive feature of the closed cycle gas turbine is loop pressurization. Loop pressurization refers to the concept of operating the entire cycle well above atmospheric pressure so that high mass flows can be achieved with relatively small flow passages. The result is a very high power output from small, compact turbomachinery. With loop pressurization, high pressure drops through the heat exchangers can be achieved with low percentage pressure losses. This results in significant reductions in heat exchanger size and weight. The high pressures also increase the thermal conductivity of the working fluid to further reduce the heat exchanger size. These effects of reducing turbomachinery and heat exchanger size, permit a significant increase in the effectiveness of the recuperator without incurring a severe weight penalty for the complete system. The higher recuperator effectiveness results in an increased thermodynamic cycle efficiency.

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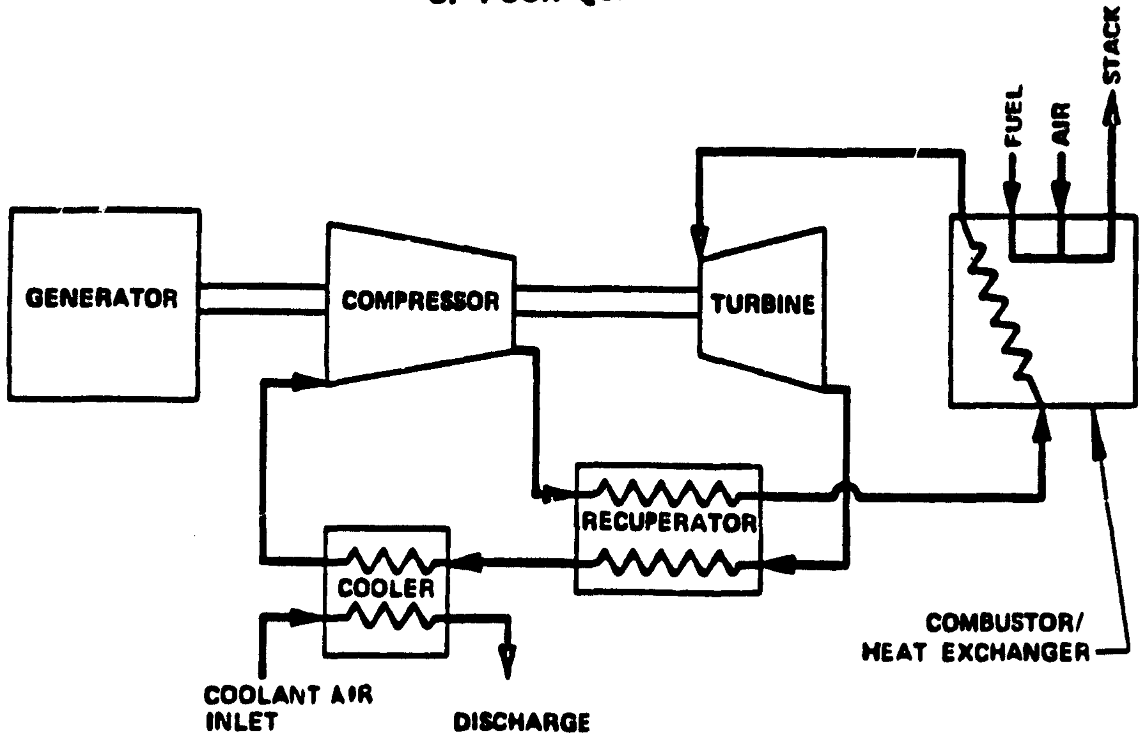

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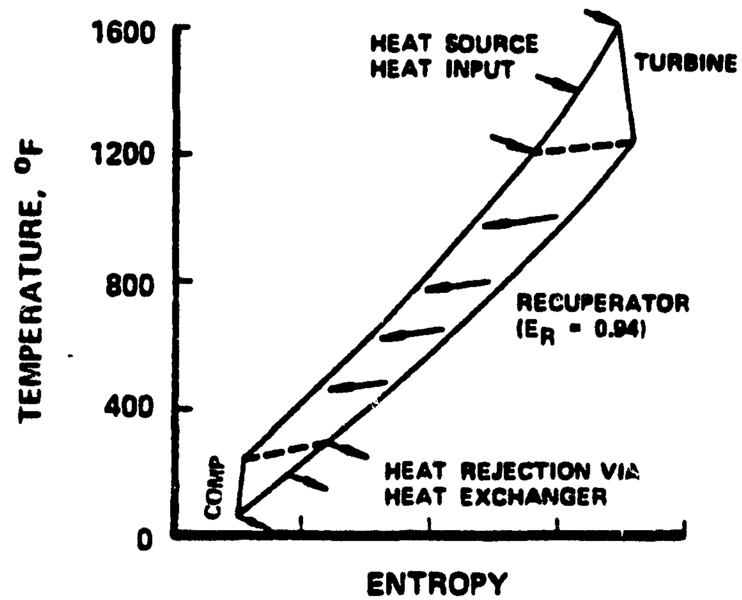

Figure 8-24. Closed Cycle Gas Turbine Power System

By changing the base pressure of the loop, the engine can be made to be highly efficient over a wide power range. Changing the base pressure varies the mass flow rate of the working fluid in the engine and, hence, its output power. This technique permits the turbine inlet temperature to remain constant at its design value. The turbomachinery efficiencies are high because the speed remains constant. The entrance and exit velocities of the turbomachinery blading are unchanged as the load varies. Losses due to separation on the stators and rotors are minimized. Actually, the part load efficiency can rise as the output power is decreased since the heat exchangers which are sized for full load conditions become effectively oversized during part load operations (Ref. 8-6). Figure 8-25 shows this effect for a range of turbine inlet temperatures from 1450 to 1750° F. The efficiency at 20% load is higher than at full load by about two percentage points.

The combustion system for the closed cycle gas turbine would be the same as for the open cycle, external combustion gas turbine previously discussed. The heat exchangers used in the combustion system may be smaller for the closed cycle than the open cycle because of the loop pressurization.

The cooler used for heat rejection could be either an air-to-air heat exchanger as a dual system where the working fluid is cooled in a gas-to-liquid heat exchanger and the liquid is then cooled in a liquid-to-air heat exchanger. The latter system is more complex but permits the heat exchanger, which is rejecting heat to the atmosphere, to have more flexibility in shape and location on the locomotive.

The efficiency of closed cycle engines was touched on earlier and shown in Figure 8-25. A cycle thermal efficiency of about 36 to 40% is possible with metallic heat exchangers in the combustion system and with large recuperators in electrical utility service. In locomotives where space is more restricted, a 30 to 35% cycle thermal efficiency is more realistic. The development of ceramic heat exchangers will permit the turbine inlet temperature to be raised to the 2200 to 2300° F range with a cycle thermal efficiency of about 46% and the overall thermal efficiency including combustion system losses of 40% (Ref. 8-6). Projected technology advances may raise this overall efficiency to 50% by the year 2000 but a more realistic figure for rail service would be in the 44 to 47% range. None of these figures include accessory loads or transmission efficiencies. The overall thermal efficiencies are the ratios of the shaft output of the gas turbines divided by the lower heating value of the coal, nominally 12,000 Btu/lb.

Garrett (Ref. 8-6) has looked at the problem of fitting a closed cycle gas turbine in a six-axle locomotive. Their representation is shown in Figure 8-26. The largest items in size are the recuperator and the fluidized bed heat source. The turbomachinery is quite small by comparison. This installation, in a SD-40 locomotive, is for a 4700 hp gas turbine to replace the 3000 hp Diesel engine. The tractive effort is shown in Figure 8-27 for both the Diesel powered SD-40 and the 4700 hp gas turbine locomotive (GTL). The axle loadings are the same for both locomotives.

The closed cycle, external combustion gas turbine engine coupled with a fluidized bed, coal-burning combustor is a very attractive alternative engine to the present day Diesel engine. The economic considerations of the engines will be considered in a later section.

G. SUMMARY

The gas turbine once had a place on the American Railroads. It may reappear on the rails again but not in the form of the earlier engines. The simple cycle of the General Electric turbines used on the Union Pacific Railroad is too inefficient for use in the future. Even the regenerative open cycle with internal combustion is not efficient enough. These engines need a liquid or gaseous fuel which would be more efficiently used in a Diesel engine.

The external combustion gas turbine engine using coal as a fuel is the better choice. The thermal efficiency of this engine is less than that of a Diesel engine but the fuel is far less expensive. The fluidized bed combustor is the most attractive of the possible different ways to burn the coal. It also permits the use of heavy, high sulfur oils as a fuel for locomotives.

The choice of an open cycle or a closed cycle gas turbine depends on the selection of a working fluid and the relative advantages of loop pressurization. In a locomotive, air is the most probable working fluid for the closed cycle on the basis of cost. The open cycle has one less

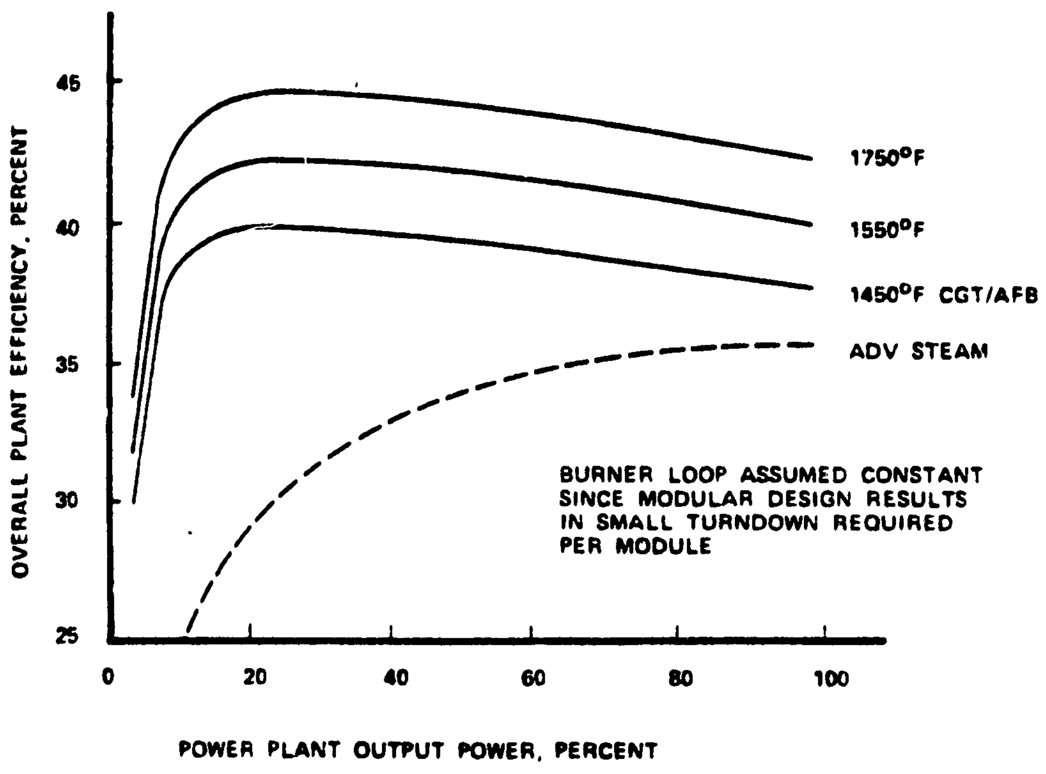

Figure 8-25. Part-Load Power Plant Performance Characteristics (From Ref. 8-3)

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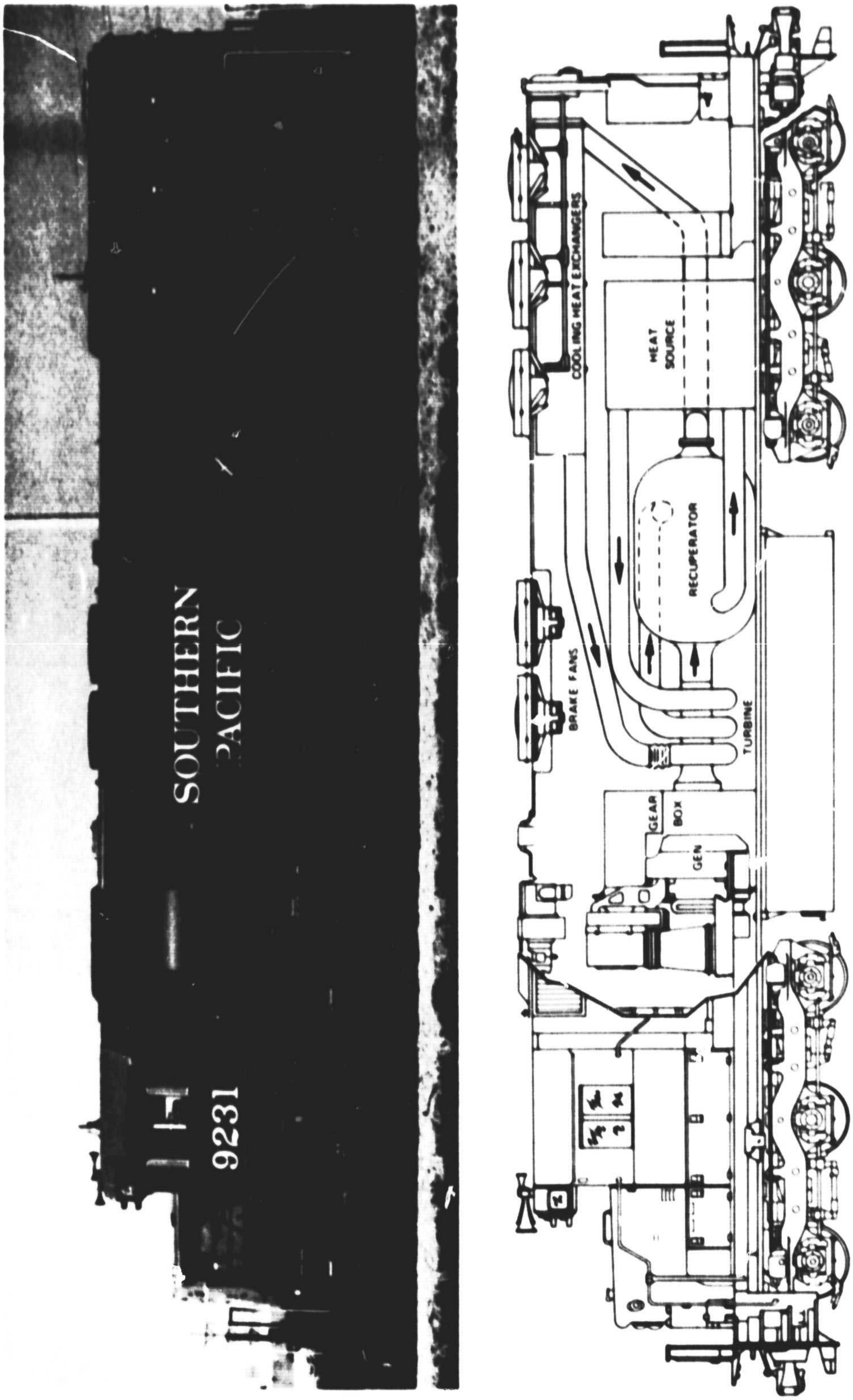

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Figure 8-26. Closed Cycle Gas Turbine Freight Locomotive (From Ref. 8-6)

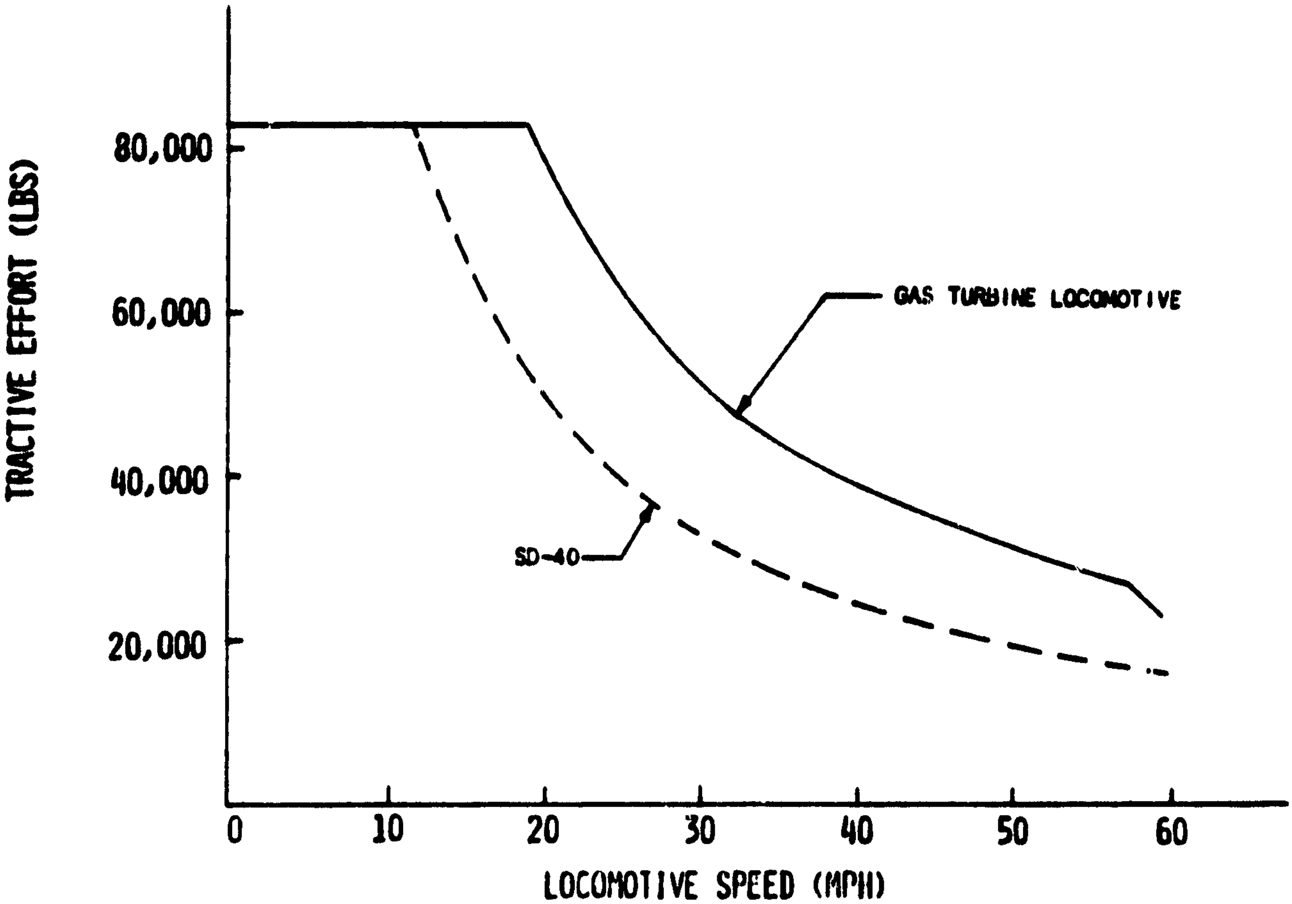

Figure 8-27. Gas Turbine Locomotive Tractive Effort vs. Speed (From Ref. 8-6)

heat exchanger (the gas cooler), than the closed cycle. However, the closed cycle offers loop pressurization with its ability to reduce the size of all of its heat exchangers and its higher efficiency at part load.

The closed cycle, external combustion gas turbine engine using coal as a fuel in an atmospheric fluidized bed is the most attractive of the various gas turbines studied in this project. The open cycle, external combustion engine is a fairly close second choice. The key to both of these engines is the fluidized bed combustor.

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SECTION IX
STIRLING ENGINES FOR RAILROAD LOCOMOTIVES

A. INTRODUCTION

The closed cycle regenerative engine was first constructed around 1816 by Robert Stirling. These early machines were generally referred to as hot-air engines due to the use of air as the working gas. Motivation for the invention is attributed, in great part, to the frequent occurrence of steam engine and boiler explosions during that period of history. The term, hot-air engines, was changed when the Research Laboratories of Philips, in the Netherlands, substituted helium or hydrogen gas as the working fluid. The original designation became inappropriate (Ref. 9-1) and devices of this type are now identified as Stirling cycle engines. The original hot-air engine, a single cylinder displacer type, is illustrated in Figure 9-1.

A Stirling cycle machine is a device which operates on a closed regenerative thermodynamic cycle, with alternate compression and expansion of the working gas occuring at different temperature levels. The temperature of the gas is changed during the cycle by a positive displacement of the gas from the hot side of the engine, through a regenerative heat exchanger, to the cold side. A net conversion of heat to work results. Heat is added externally to the engine continuously. The Stirling differs from the Rankine cycle since the working fluid remains as a single phase, i.e., a gas.

Returning to the historical developments during modern times, in 1937 the N. V. Philips Laboratories revived the engine concept during their search for a silent power plant for a portable electric generator. The first successful development of modern Stirling cycle machines, however, was for cooling applications. This commercial venture was started by Philips in the late 1940s and culminated in 1953 with their introduction of a mass-produced air liquification machine.

The first known licensee of Philips in the United States was the General Motors Corporation. The contractual agreements, signed in 1958, provided for a ten-year information exchange period and had provisions for mutual licensing of patents related to Stirling engines (Ref. 9-2). General Motors original interest was for space power systems, marine propulsion, portable electrical power generation, and eventually propulsion systems for ground transportation. Their large Stirling engine testing activity was moved in 1962 from their Cleveland Diesel Engine Division to the Electro-Motive Division (EMD). These engineering developments represent the first known efforts to investigate the use of Stirling engines specifically for railroad locomotives (Ref. 9-2).

Different versions of this four-cylinder engine were tested using a variable phase angle control system. This permitted the engine to operate in either direction of rotation. Details of this early work have not been published in the open literature, however, reference to the effort is made by Percival (Ref. 9-2). The engine was designed as a split-crankshaft V-8 configuration but only a four-cylinder version was actually assembled and tested. Percival reports that some bearing and vibration problems were encountered during these early development tests.

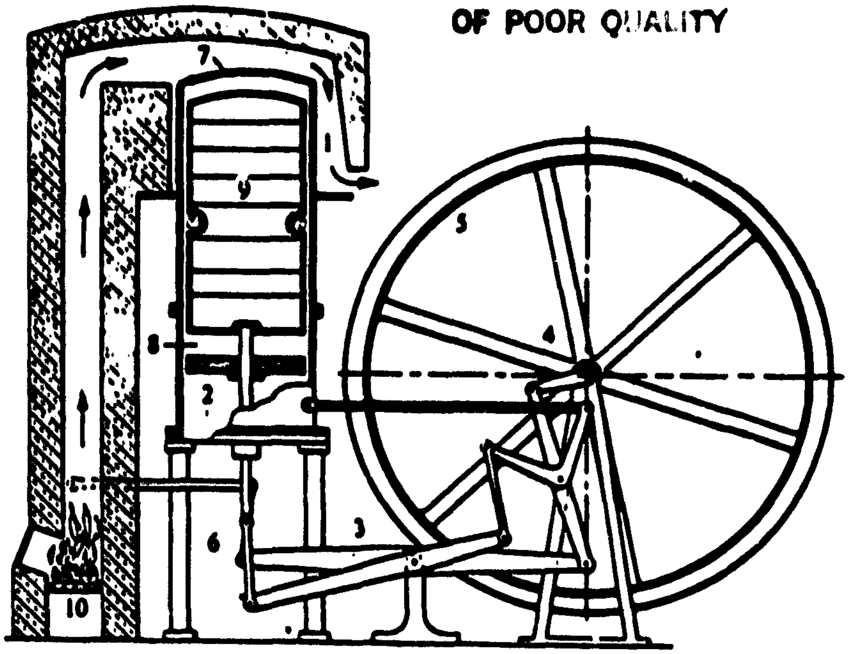


Figure 9-1. Original Stirling Engine Used for Water Pumping, Based upon 1816 Patent Specifications

A management decision was made by General Motors in 1970 to terminate the licensing agreement. Numerous and significant contributions to Stirling engine technology were made by General Motors during this period (Ref. 9-2). Most of these activities were financed by corporation funds, one exception being an Army supported project to develop a small electric power generator set identified as the Ground Power Unit (GPU). One of these early units, designated as GPU 3-2, has recently been rejuvenated and tested by the NASA Lewis Research Center (Ref. 9-3).

In 1968, United Stirling of Sweden was formed and soon thereafter became a licensee of Philips. Since that time significant engineering accomplishments have been made, including the building of large Stirling engines. The first engine tested by United Stirling was designated as the 4-615 (four, single-acting cylinders with a total swept volume of 615 cm^3). The engine used a rhombic drive system and ran at relatively low speeds. This engine was originally planned for installation into an urban bus in 1973 with eventual limited production to follow three years later.

More recent developments at United Stirling have concentrated on smaller engines having four double-acting cylinders for automotive applications. The first direct automotive demonstration has been described by Carlquist (Ref. 9-4) using an experimental V4X engine in both a 1972 Ford Pinto and 1974 Ford Taunas car.

Studies undertaken by the U.S. Department of Energy (DOE) for the purpose of assessing the development status, and potential of large (500-3000 hp) Stirling engines for stationary power purposes have been reported by Ziph (Ref. 9-5), Hoagland (Ref. 9-6), Marciniak (Ref. 9-7) and others. Three contractor/teams have conducted conceptual design studies for large Stirling stationary engine applications. The study effort was supported by the General Electric Company, Advanced Mechanical Technology Inc. (AMTech), and Foster-Miller Associates. Program management is being provided by the Argonne National Laboratory.

The conceptual engine designs evolving from this study are modular in approach and are based upon units of about 500 hp each. The primary heat source being considered for these large engines is coal combusted in a fluidized bed. This probably necessitates the need for some type of a thermal transport system to transfer the usable heat from the combustion source to the engine heater head. A liquid sodium heat pipe or a pressurized closed loop helium source is being proposed for this purpose. Following completion of the conceptual design phase, one or more prototype engine builds are planned to demonstrate the operating characteristics of large coal burning Stirling engines. The technical aspects of this work should be directly applicable to any future locomotive use of Stirling engines.

B. BASIC CHARACTERISTICS OF THE CYCLE AND ENGINE

Like all heat engines, the Stirling produces power by compressing a working gas at relatively low temperatures, adding heat, and then expanding it at higher temperatures. During each cycle of operation, energy is supplied at high temperatures and rejected at relatively low temperatures. The ratio of these two temperatures determines the cycle efficiency of the engine and is a factor in the power output.

There are two types of engines, the single-acting and the double-acting, depending on the arrangement of the basic components. There are five basic components common to Stirling engines. The engine heater provides a means of keeping the working gas hot, the regenerator stores part of the thermal energy, a piston compresses and/or displaces the gas, another piston expands the gas, and a cooler rejects the excess heat to an external source. The arrangement of these basic components are shown for a particular two-piston single-acting engine in Figure 9-2. This engine has an electric heater head and, therefore, the normal combustion related components are not required. A combustion air blower, fuel pump, combustion chamber, and an air preheater which recovers heat from the engine exhaust are the usual engine related accessories.

Nearly all pr ctical Stirling engines have one or more pairs of reciprocating pistons and/or displacers which operate within cylinders. Displacer pistons shuttle the working gas back and forth while the power pistons expand the heated gas to produce power. The two types of pistons move with definite relationship to each other in order to maintain the desired internal volume variation characteristics of the engine. The flow of gas within the engine is achieved solely through these volume changes and without the use of intermittently actuated valves or ports. Various mechanical devices can be employed to maintain the proper phase relationship between the displacer and expansion pistons.

Most modern Stirling engines work on the double-acting principle which means each piston serves both functions. They shuttle the gas back and forth between the hot and cold spaces and also expand the gas to produce net work. This configuration will be described in some detail in a later section of this chapter.

The Stirling engine operates on a closed cycle, heat addition takes place externally, and the process is continuous. This is contrasted to the conventional internal combustion engine which is an open cycle machine with intermittent heat addition.

The working fluid in a Stirling engine is usually high pressure helium or hydrogen gas. Due to the continuous heat addition process and the thermal capacity of the heater walls, it is not possible to rapidly heat and cool the working gas in a single step. However, the gas temperature is varied cyclically by using a displacer or compression piston to transfer the working gas back and forth between the two spaces, one at a constant high temperature and the other at a fixed low temperature.

The ideal Stirling cycle (Figure 9-3) is composed of four discrete thermodynamic processes:

(1) Isothermal compression process: (1-2)

Compression of the working gas occurs while maintaining the gas temperature constant. In order to do this, the heat of compression must be transferred from the gas to the external coolant.

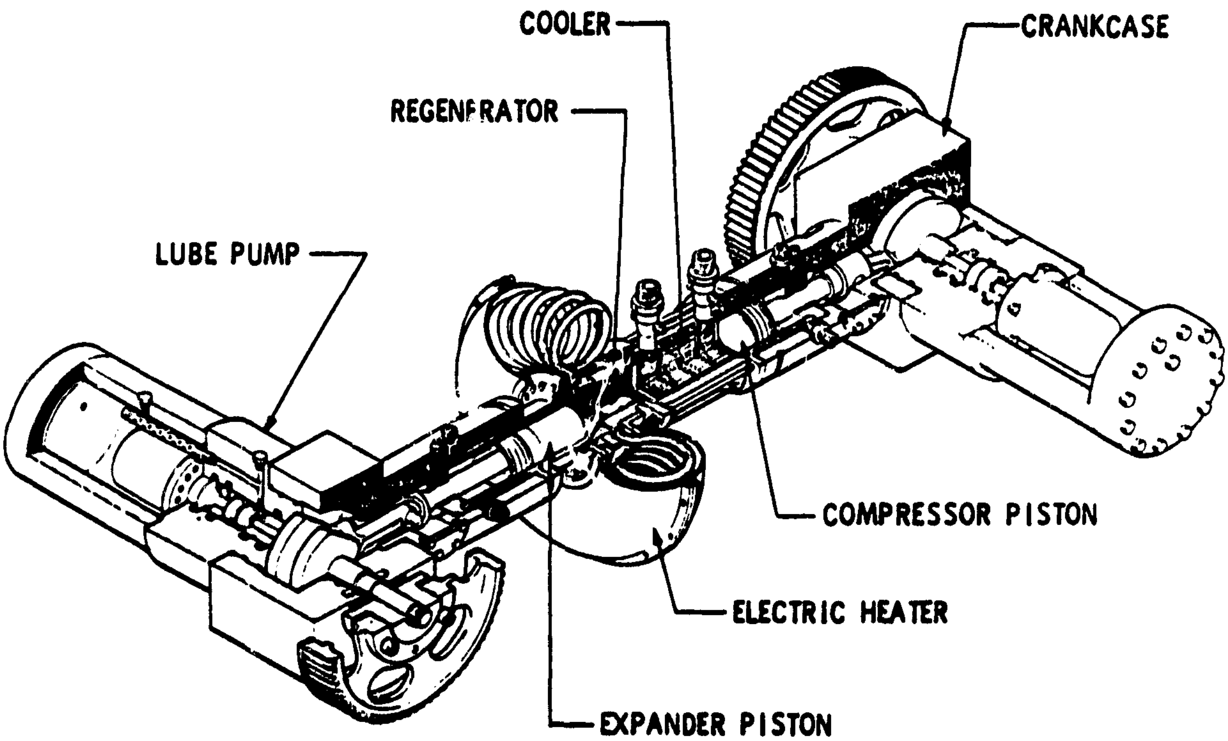


Figure 9-2. A Single-Acting Stirling Engine Configuration Showing Basic Components

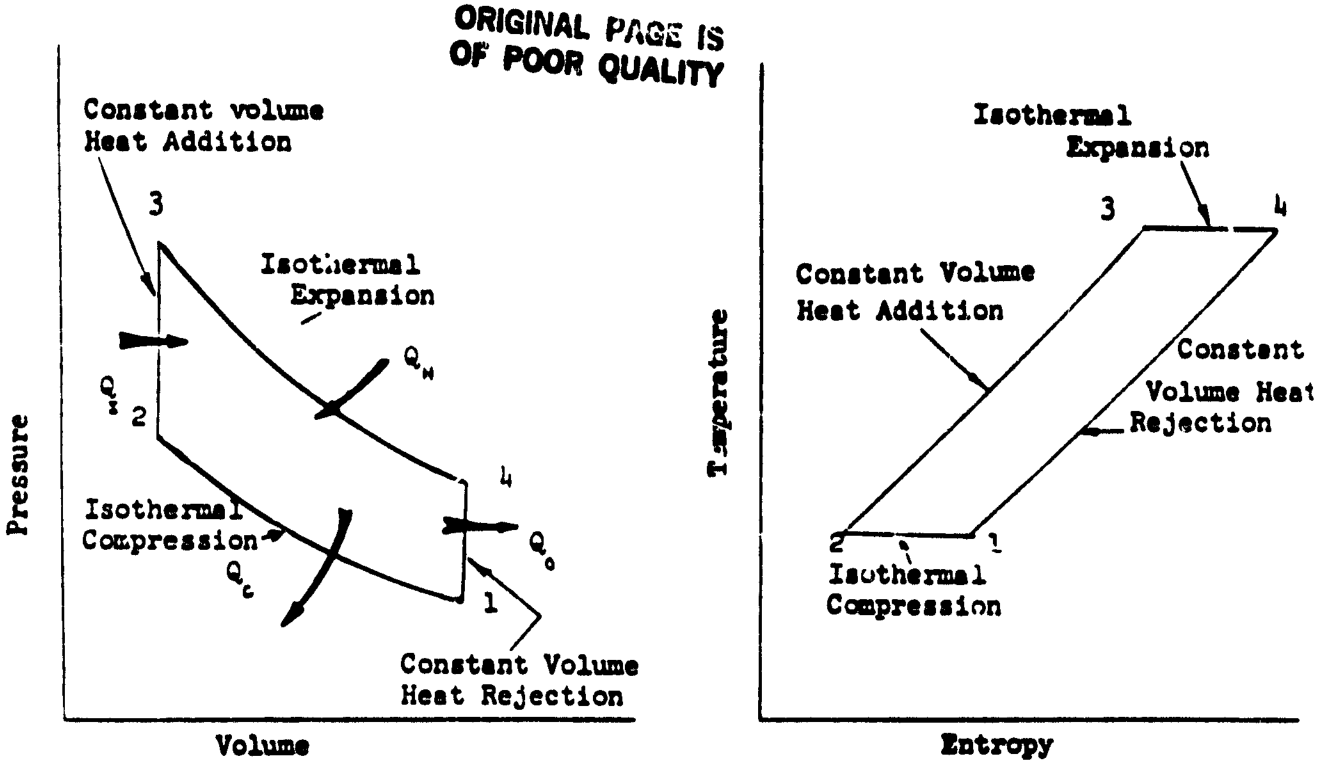


Figure 9-3. Pressure-Volume and Temperature-Entropy Diagrams for the Stirling Cycle

(2) Constant volume regenerator process: (2-3)

The working gas is displaced from the compression space to the expansion space while maintaining the gas volume constant. During this step the gas passes through the regenerator, is heated, and because the total volume is fixed, the gas pressure increases further.

(3) Isothermal expansion process: (3-4)

Expansion of the working gas occurs while maintaining the gas temperature constant. In order to do this, heat must be transferred to the working gas from an external source. It is during this expansion process that useful work is done by the engine.

(4) Constant volume regenerative process: (4-1)

Movement of the working gas from the expansion space back to the compression space while maintaining the gas volume constant. The gas passes through the regenerator, is cooled, and because the volume is fixed the gas pressure decreases.

Ideally, the regenerative heat transferred in process 2-3 has the same magnitude as in process 4-1; the only heat transfer externally between the engine and its surroundings is the heat supplied at T_{hot} and the heat rejected at T_{cold}. This heat supply and heat rejection at constant temperature satisfies the requirement of the second Law of Thermodynamics for maximum thermal efficiency. In addition, for the ideal cycle, all heat transfer and gas flows are reversible, thus the efficiency of the ideal Stirling cycle is the same as the Carnot cycle. The principal advantage of the Stirling cycle over the Carnot cycle lies in the replacement of two isentropic processes by two constant volume processes, which greatly increases the area of the P-V indicator diagram. Therefore, to obtain useful work from the Stirling cycle, it is not necessary to operate at very high pressures and swept volumes, as in the Carnot cycle.

C. ACTUAL STIRLING ENGINE CYCLE

An approximation of the ideal cycle can be achieved by employing two single acting, opposed pistons which move synchronously with simple harmonic motion (Figure 9-2). Typically, Stirling engines operate with the expander pistons leading the compressor pistons by approximately 90° crankshaft rotation. With harmonic motion, the piston speed varies from zero velocity at both the top and bottom-dead positions to a maximum value at the intermediate crank angle positions. This fact, coupled with the above mentioned phase angle relationship, approximates the internal volume, pressure, and temperature conditions necessary for the ideal Stirling cycle just described.

Instead of the various thermodynamic operations occurring as four discrete steps, the entire cycle now occurs during one continuous process. Consequently, the gas displacement process overlaps with the expansion and compression processes, thereby rounding off the corners of the ideal pressure-volume relationship. The real engine operates at a sufficiently high speed so that the relatively slow heat transfer process results in imperfect regeneration. In addition, the available heat transfer surface area in the hot and cold cylinders causes the expansion and compression steps to be intermediate between the isothermal and isentropic processes.

These more realistic processes combine to produce the actual pressure-volume curves shown in Figure 9-4. Well designed engines can achieve an indicated efficiency of about 65-70% of the Carnot efficiency, but when mechanical and combustion system losses, finite heat transfer, and auxiliary power requirements are considered, the net engine efficiency is typically only about 50% (or less) of the Carnot value. Even though the efficiency of real Stirling engines is less than the ideal or theoretical values, they still offer the potential of significant improvements over conventional internal combustion engines.

D. TYPES OF ENGINE DRIVE SYSTEMS

Many different mechanical arrangements have been suggested for Stirling engines over the years; few of them have materialized into actual engine concepts. Walker (Ref. 9-1) classifies single-acting Stirling engines into two types; either piston-displacer or dual-piston machines. An example of each type is shown schematically in Figure 9-5. The piston-displacer Stirling engine uses one power and one displacer piston for each cylinder; these pistons operate approximately 90° out of phase. The power piston is

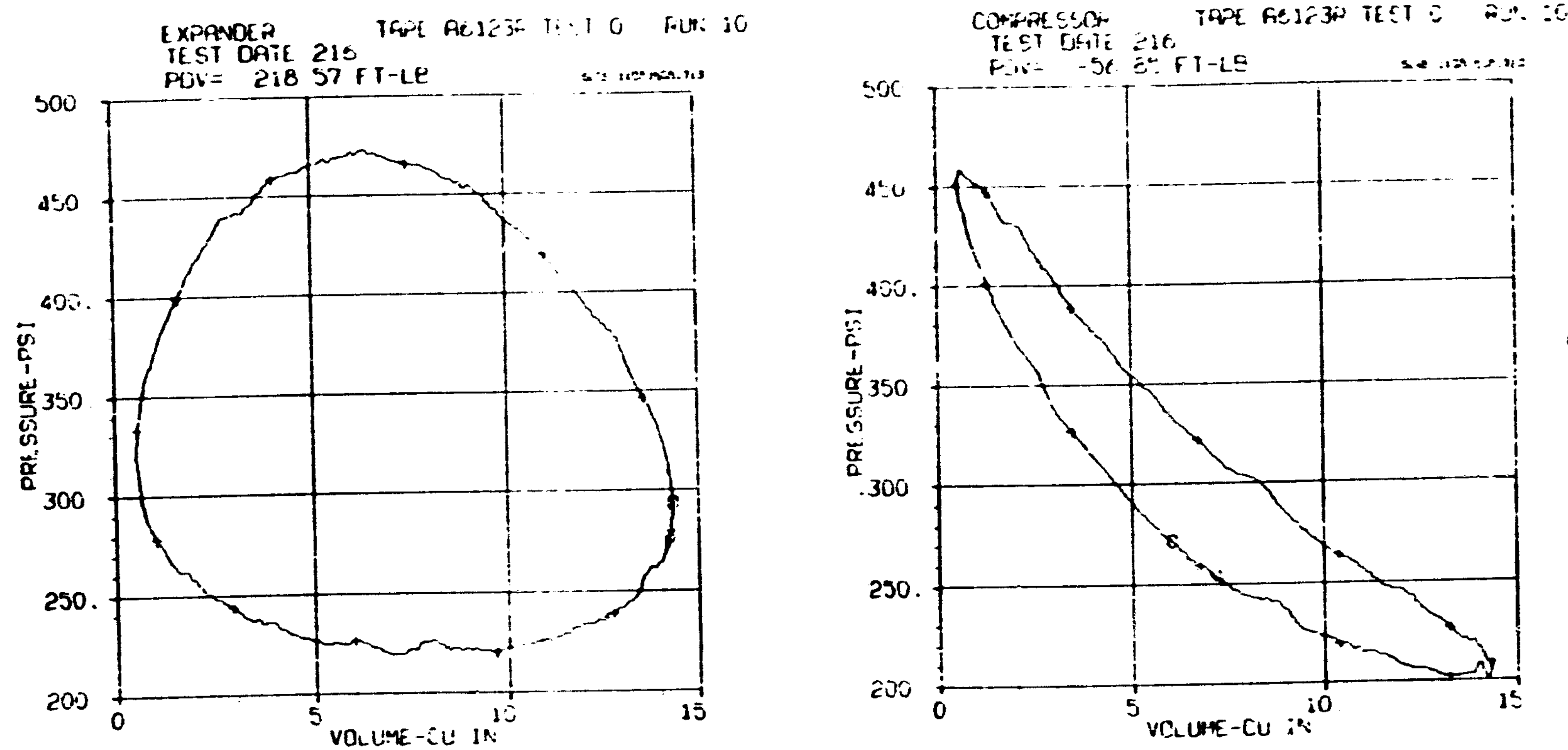


Figure 9-4. Experimental Pressure-Volume Diagrams for Expander and Compression Spaces

usually fitted with Teflon-type sealing rings and operates with significant pressure differences across it, and extracts net positive work during each cycle. The displacer piston, moves the working gas back and forth from the hot expansion space through the heater, regenerator, and cooler to the cold compression space. The only pressure difference across the displacer piston therefore, is due to the small flow resistance through the the heat exchanger components.

An example of an actual dual-piston engine configuration was shown in Figure 9-2. This particular laboratory type test engine has two single-acting opposed pistons in separate cylinders, one for the hot space (expander) and one for the cold space (compressor). The two working spaces are separated by the heater, the regenerator and the cooler. The opposed piston configuration can be adapted to small engines having pressurized crankcases and rotary shaft seals, but is less attractive for large engine applications.

Another general type or classification for the multi-cylinder Stirling engine is based on the use of double-acting pistons; each piston provides both the power and displacer functions. As shown in Figure 9-6, each of the multiple double-acting cylinders has a hot space on the head end of the engine and a cold space on the rod end. Thus, the hot space of one cylinder is connected through a heater, a regenerator, and a cooler to the cold space of an adjacent cylinder to form a single unit, as illustrated. The pistons are phased 90° for a 4-cylinder engine. The double acting principle can be applied to engines having from 3 to 7 cylinders; however, maximum efficiency is achieved for engines having 4 to 6 cylinders (Ref. 9-5).

Rinia (Ref. 9-5) invented the multi-cylinder, double-acting Stirling configuration early in the Philips engine development program. It was abandoned soon afterward due to the sealing and lubrication problems. This

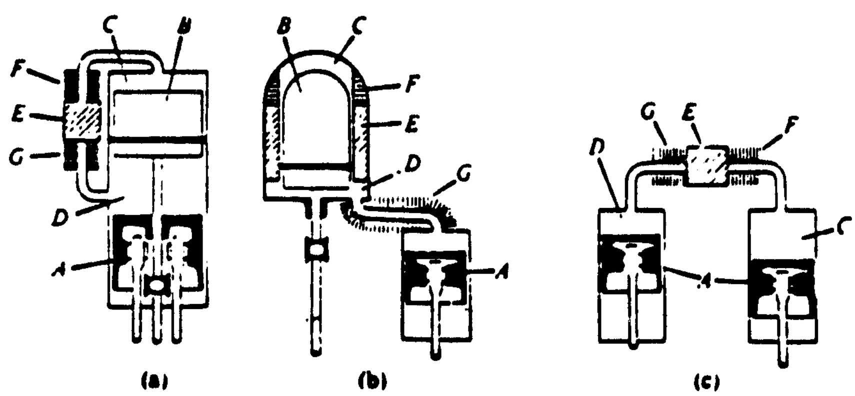


Figure 9-5. Types of Single-Acting Engine Arrangements (From Ref. 9-1)

(a) Piston-displacer, in the same cylinder
(b) Piston-displacer, in separate cylinders
(c) Two-piston machine

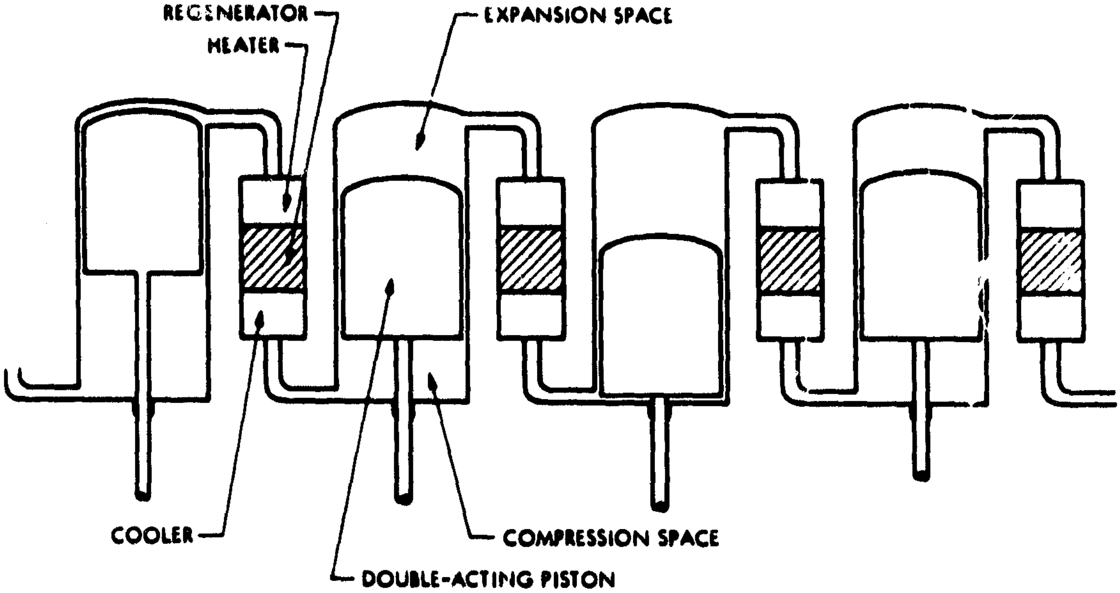


Figure 9-6. Schematic Representation of a Double-Acting Four-Cylinder Stirling Engine (From Ref. 9-5)

configuration has recently been revived because it is the most compact, lightest weight, and lowest cost arrangement.

Double-acting Stirling engines have been configured with the cylinders placed in-line, in a Vee arrangement and in a co-axial or parallel arrangement. Drive mechanisms have included a single crankshaft for the in-line and Vee, and double-parallel crankshafts for the co-axial cylinders in a barrel arrangement. These configurations are illustrated in Figure 9-7.

Engines with parallel cylinders provide a compact heater head installation which simplifies the interface with the heat source. Separate drive shafts are needed for each row of cylinders or the use of a swashplate is required. The parallel cylinder engine has increased dead volume, but reduces thermal expansion problems and simplifies the heat source integration.

Various methods of converting reciprocating motion to rotary motion have been successfully implemented for Stirling engines. The easiest method employs the use of the conventional crankshaft and connecting rod (Figure 9-8). This approach was taken in the first engine design developed by Philips. A disadvantage is that piston side forces are generated which produces excessive ring wear. This can be minimized, however, by the use of a sliding cross head member to take the lateral loads. Another design frequently employed since it provides a dynamically balanced system, is the rhombic drive mechanism.

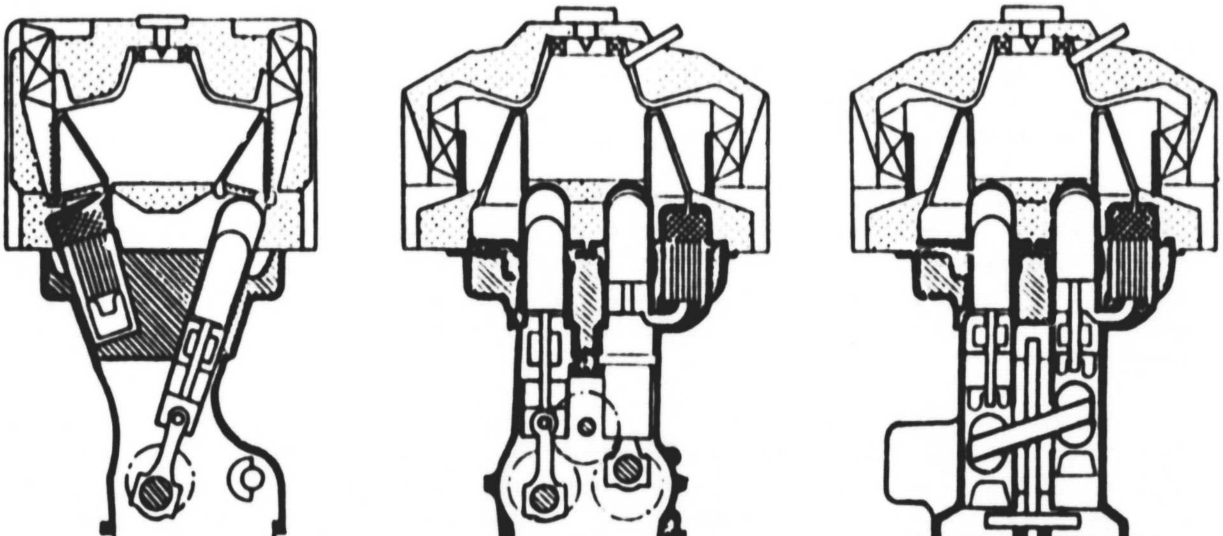
The rhombic drive, which is shown in Figure 9-8, contains two counter rotating crankshafts linked to a centrally located piston rod yoke by means of matched connecting rods. Synchronization is achieved by two helical timing gears, mated to an idler gear. The advantages of the rhombic drive relate to balanced inertia forces, low vibration, improved seal life, and variable phase options.

The disadvantages of this mechanism are that it requires a large number of mechanical components. In addition, closer manufacturing tolerances and quality control measures are necessary. All of these factors translate into higher manufacturing costs.

Alternative crankshaft drives on the other hand, are compatible with both double and single-acting machines. Single and dual crankshaft drives are illustrated in Figure 9-7. The crankshaft drives are similar to those used in reciprocating internal combustion engines and are, therefore, more compatible with present mass production methods. This fact, in combination with fewer components, results in lower manufacturing costs.

A third type of drive that is presently being employed in Stirling engines is the swashplate (Figure 9-9). It has also been widely used in hydraulic pumps and refrigeration compressors. Advantages of this type of drive include dynamic balancing, compact size and favorable torque characteristics.

In terms of cost, the swashplate should be comparable to conventional, crank drive systems. The wobble plate drive is similar in principle to the swashplate; however, it employs conventional roller bearings instead of hydrodynamically lubricated sliders. The wobble plate drive demonstrated by Philips offers potential in a variable stroke engine.



(a) (b) (c)

Figure 9-7. Double-Acting Stirling Engine Cylinder Arrangements (From Ref. 9-6)

(a) Vee design - single crank
(b) Parallel cylinders - twin cranks
(c) Swashplate configurations

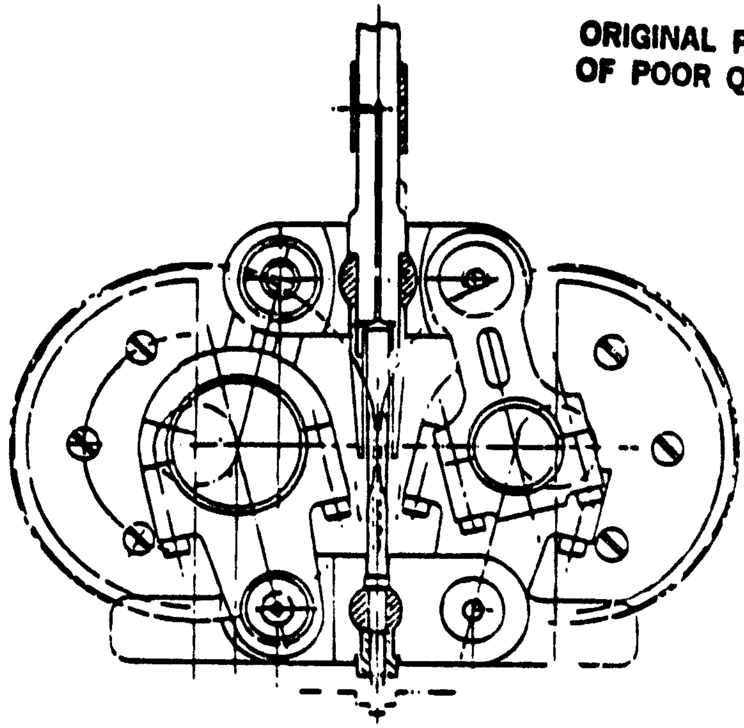


Figure 9-8. Rhombic Drive Design
(From Ref. 9-8)

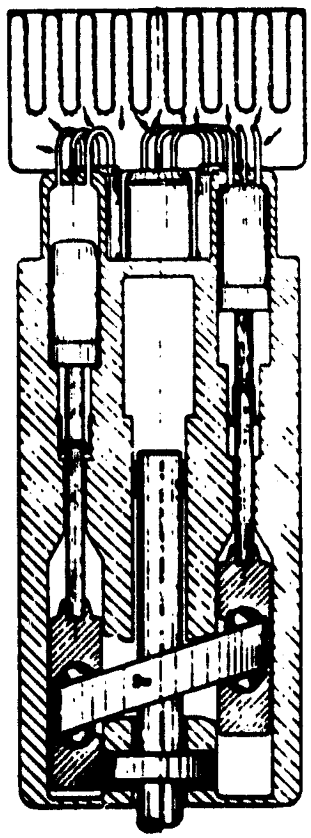


Figure 9-9. Swashplate Drive Design
(From Ref. 9-8)

E. ENGINE SELECTION

Engine size and weight are important considerations for automobiles because they influence vehicle size, cost, and performance. High efficiency is more important at part throttle than at full power since most of the vehicle operation occurs at road load conditions. Automotive engine designs usually strive for compactness and low specific weight considerations which lead to the selection of high-speed, light-weight designs.

Locomotive engines, on the other hand, would not be limited to the same extent by weight and size considerations. As a result, considerations such as high efficiency, durability, and reliability would influence the selection of larger, heavier, slower speed engines that are more conservatively rated for operation near their best efficiency point.

The automotive application of Stirling engines has received considerable attention in recent years. However, much of the technology developed for this activity will undoubtedly apply to other areas. Some of the important similarities and differences between automotive, locomotive, and stationary engine applications have been noted by Hogland (Ref. 9-6) and are presented in Table 9-1.

F. ENGINE PERFORMANCE MAP

In order to perform a typical locomotive duty cycle analysis, it is necessary to know the performance characteristics of Stirling engines at full, part load, and idle conditions. Correction factors are needed to determine the effect of variations in heater head and coolant temperatures on engine efficiency and power output.

The Stirling engine map shown in Figure 9-10 was scaled to 500 "design" horsepower from a larger existing United Stirling engine and is optimized using a liquid metal thermal transport system (Ref. 9-6). It was constructed using appropriate scaling factors and by making allowances for various engine auxiliaries such as the combustion blower, oil and water pumps, fuel pump, and atomizing air pump. The performance map does not include, however, any allowance for auxiliary equipment associated with the locomotive. If the locomotive installation employs air cooling, for example, then an appropriate allowance would be needed for the fan.

It can be seen from Figure 9-10 that the predicted maximum thermal efficiency of about 40% for this engine occurs near half speed and maximum pressure. The operating point for this particular engine was selected at 850 rpm (about 70% speed) and 2000 psi cycle pressure to provide approved efficiency and durability. The actual selected operating point efficiency is 39.6%. The estimated idle fuel consumption for a 500 hp Stirling engine is about 22 lb/hr at 300 psi mean pressure (Ref. 9-9).

The engine map shown in Figure 9-10 is based upon a hot gas temperature of 1500°F and an average cooling water temperature of 86°F. As shown by Figure 9-11, both the engine power and efficiency are influenced by these two temperatures. For example, as pointed out by Hogland (Ref. 9-6), increasing the heater temperature from 1350°F to 1500°F, increases the power 12% and the thermal efficiency 10%. Similarly, at the cold end

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Table 9-1. Comparison of Application Requirements

	Automotive[a]	Locomotive	Stationary[a]
Power Rating	50 - 200 HP	1000 - 3000 HP	500 - 2000 HP
Compactness	Very Important	Less Important	Not Important
Gross Weight	Very Important	Less Important	Not Important
Efficiency	Part Load Efficiency Is Important	Part Load Efficiency Is Important	High Load Efficiency Very Important
Heat Rejection load	Important	Less Important	Secondary Importance
Required Rotational Speed	up to 5000 RPM	300 - 1100 RPM	500 - 1000 RPM Usually Constant
Required Response Time	Fractional Second	Fractional to Few Seconds	Fractional to Few Seconds
Noise	Important	Secondary Importance	Secondary Importance
Low Emissions	Important	Not as Critical, but Highly Desirable	Not as Critical, but Highly Desirable
Fuels	Distillate	Residual or Solid Fuels Preferred	Residual or Solid Fuels Preferred
Durability	About 4000 Hours	Over 15,000 Hours	Over 15,000 Hours
Cost	Initial Cost Very Important	Life Cycle Cost Very Important	Life Cycle Cost Very Important

[a] From Ref. 9-6

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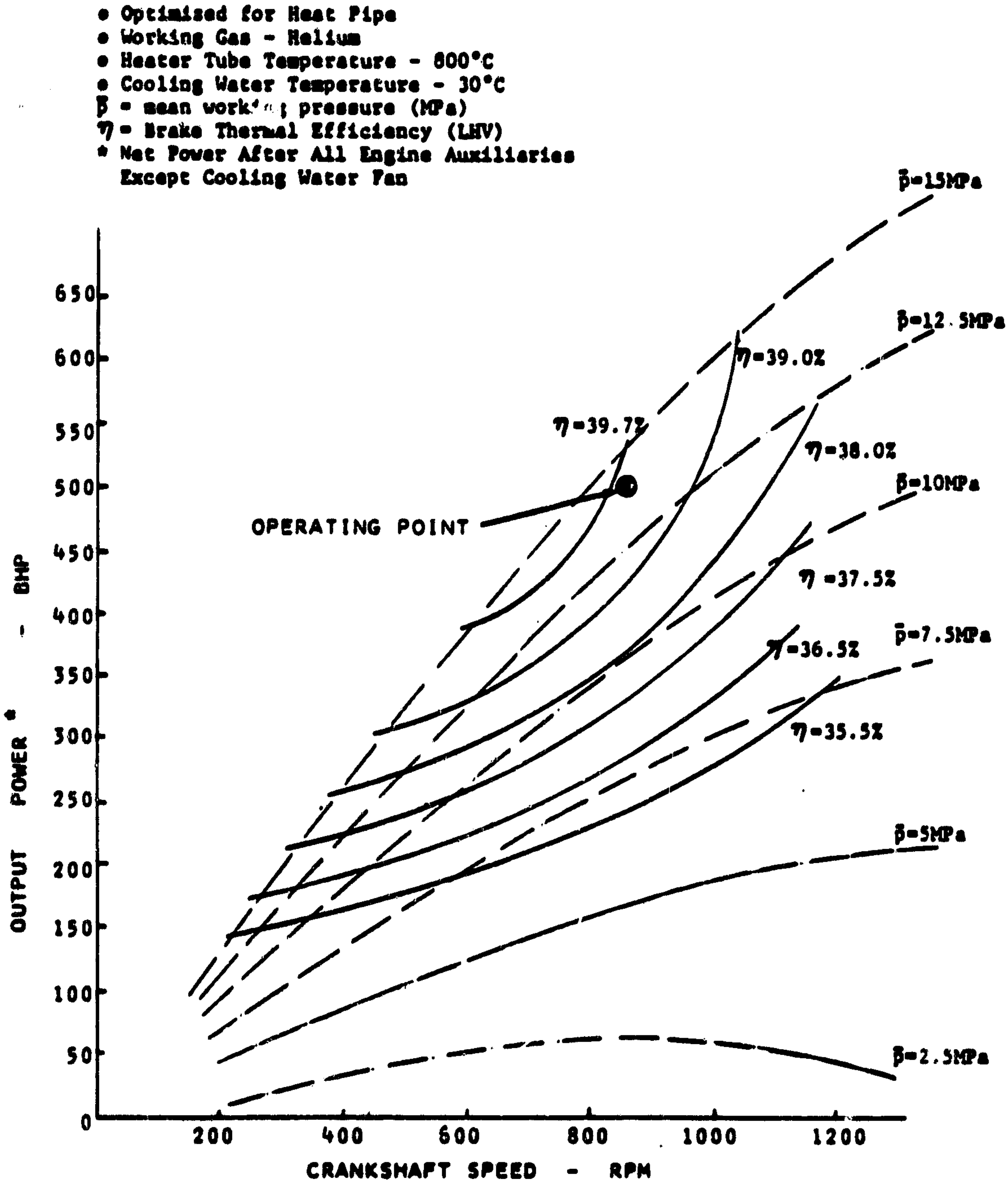

Figure 9-10. Estimated Performance Map for Representative Stationary Stirling Engine - Basic Four-Cylinder Power Unit (From Ref. 9-6)

(Figure 9-11), decreasing the coolant temperature from 160°F to 86°F increases power 18% and efficiency 13%. Correlations of this type are needed to adapt Stirling engines to specific applications.

One problem, in locomotive service, is that the coolant temperature depends on the ambient air temperature, air flow rate, and the radiator size. Seasonal and geographic changes will have profound effects on engine output, power, and efficiency. Using Figure 9-11 as a guide, winter in Minnesota could lower the coolant temperature to the 60 to 90°F range which would result in 15 to 20% increase in power and efficiency. On the other hand, summer in the desert would cause the coolant temperature to rise to the 200°F level resulting in a 10 to 15% decrease in power and efficiency as compared to the baseline engine of Figure 9-11. This range of winter to summer temperatures could cause a 35% variation in power and a 25% variation in efficiency. Diesel and gas turbine engines show a similar behavior pattern but the effects are not nearly as great.

G. HEAT BALANCE

Of the total energy supplied to a Stirling engine, the useful energy (shaft work) is typically between 30 to 40%. About 50% of the supplied energy is rejected to the cooling water. Figure 9-12 shows the typical part-load performance characteristics of the Stirling engine. This data was obtained from the results of a study by Phillips (Ref. 9-7). Comparative data is presented in Table 9-2 for some alternative engine types. It shows that the Stirling engine is one of the most efficient alternatives.

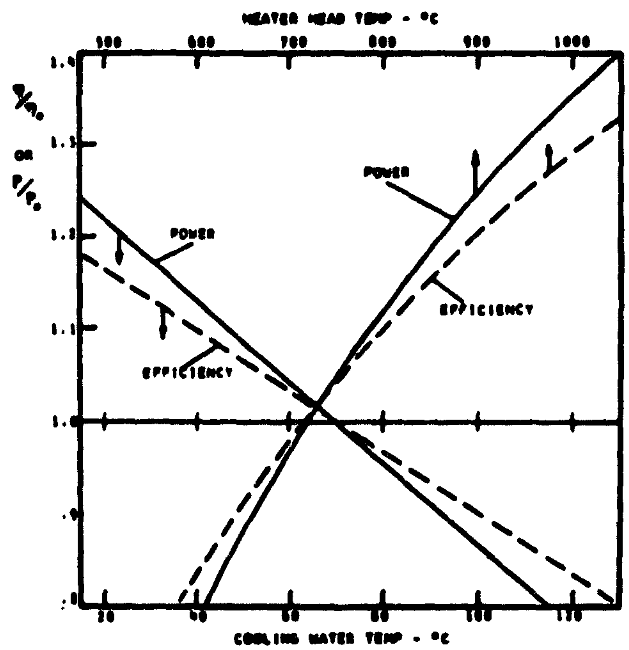

Figure 9-11. Effect of Operating Temperature on Stirling Engine Power and Efficiency (From Ref. 9-6)

H. ENGINE SIZE AND WEIGHT

Using the modular approach, previous studies indicate that four cylinder modules of about 500 hp provide the maximum cost benefit. A 2000 hp engine could be constructed from four modules, each cylinder having a nominal output of 125 bhp. Generally, as in Diesel engine practice, there is some small gain in thermal efficiency as the number of cylinders increase because of improved mechanical efficiency and more efficient accessories.

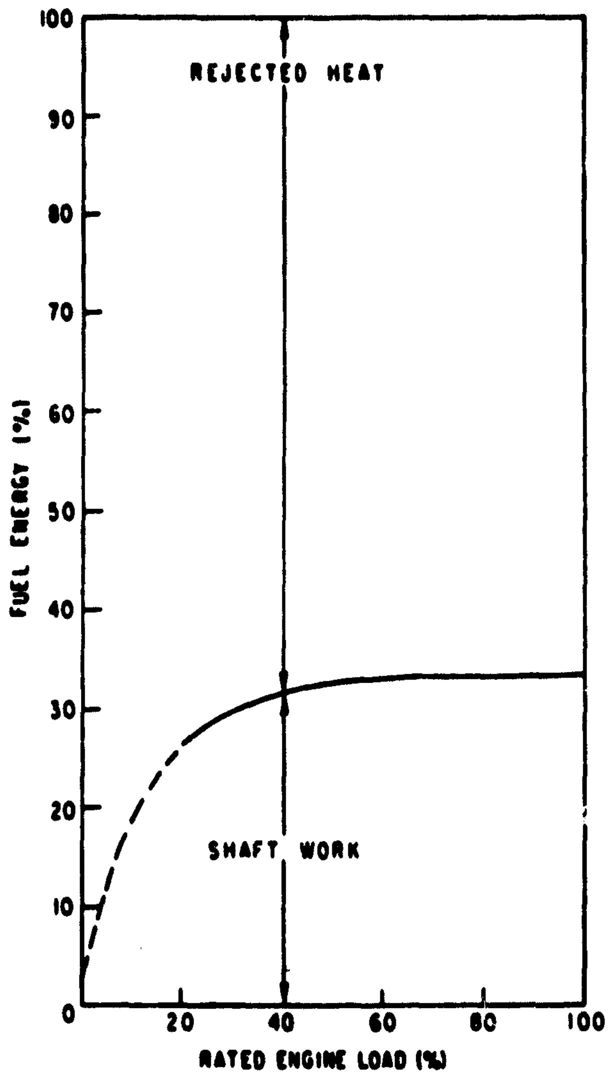

Figure 9-12. Stirling Engine Heat Balance at Part Load (From Ref. 9-7)

Table 9-2. Typical Heat Balance for Several Engine Types at Full Load

Engine Type	Net Efficiency (Ref. 9-7)	Net Efficiency (JPL)
Diesel	36	40
Adiabatic Turbo-Compound Diesel	47	47
Adiabatic Turbo-Compound Diesel with Bottoming Cycle	—	54
Gas Turbine (Simple)	25	25
Gas Turbine (Regenerative)	38	36
Stirling (Current)	34	32
Stirling (Advanced)	46	38

Studies of the size effects for displacer Stirling engines were made by Phillips and others during the 1960s. The results became known as "scaling rules". They considered how scaling geometrically similar rhombic drive engine designs would affect power, efficiency, stresses, heat transfer, and sealing. The engines covered by these rules range from about 10 bhp to over 3000 bhp. The scaling rules can also be applied to other Stirling engine designs, such as the double acting type.

The piston diameter for a 125 bhp per cylinder engine should be about 7 in., and have a piston weight of approximately 35 lb. (Ref. 9-6). The piston stroke would be 4 in., and the engine speed at maximum power would be approximately 1200 rpm. Engines of 4-cylinder, 6-cylinder, 8-cylinder, 12-cylinder and 16-cylinder (all employing the same basic cylinder size and internal components) would have outputs of 500, 750, 1000, 1500, and 2000 hp respectively. Thus, the 4, 8, 12, and 16-cylinder engines would be built up from multiples of 4-cylinder, double-acting modular power units.

Table 9-3 presents the specifications for a typical 2000 hp, 16-cylinder engine designed for stationary power applications (Ref. 9-6). The overall estimated dimensions are 143 in. long, 50 in. wide and 68 in. high. The engine weight is approximately 32,000 lb, which represents a specific weight of about 16 lb/hp. This is almost identical to the specific size and weight of an equivalent Diesel engine used for similar service.

In comparing the physical size of a crankshaft drive Stirling engine to the Diesel, the former would have greater overall height for two reasons. First, the crankshaft drive of the Stirling requires the addition of a cross head slider mechanism and second, the power piston is fitted with a thermal heat barrier (dome) which necessitates the use of a longer cylinder. Both of these factors combine to increase the height of the engine as measured from the oil pan to the crown of the piston to be about 25 times the crank throw (Ref. 9-1). In contrast, conventional Diesel practice calls for a height of about 10 times the crank radius. The estimated width however, of both the Stirling and Diesel engine has been found to be about the same.

As part of the conceptual design study by General Electric (Ref. 9-10), artists conceptions of two possible stationary power configurations were developed. These are illustrated in Figures 9-13 and 9-14. The first shows a 1500 hp double acting Vee/axial inline configuration having a single crankshaft drive mechanism. The second represents a 1000 hp rhombic drive engine. Both configurations employ a high pressure gas heat transport system. The figures are of interest since they convey the overall size of these particular engine types.

I. HEAT EXCHANGER SIZE

Heat exchangers represent the most critical aspect of the overall Stirling engine design. The three engine heat exchanger units are the heater head, the regenerator, and the cooler. In addition, there is an air preheater in the combustion system. The location and function of each component is identified in Figure 9-15 (Ref. 9-1). The internal (dead) volume of the three engine heat exchangers represents a compromise in design between the need to maintain sufficient surface area for heat transfer and its adverse effect upon engine performance because of reduced pressure ratios.

The heater head transfers heat from the external source (i.e., combustion gas, solar, nuclear, etc.) to the engine working gas. Similarily, the cooler transfers heat from the working gas to the external coolant at the sink temperature. The regenerator serves a dual purpose by both storing and releasing heat within the cycle. It also acts as a thermal barrier and isolates the hot and cold working spaces of the engine. The preheater transfers heat from the exhaust gas to the inlet combustion air and thereby increases the overall engine system efficiency.

Table 9-3. Modular Engine Design Specifications (From Ref. 9-6)

Parameter	Value
Power	125 hp
Speed	1200 rpm
Stroke	4.00 in.
Bore	7.00 in.
Hot Side Temperature	1400 °F
Cold Side Temperature	68 °F TO 250 °F
Displacement	153.94 $in.^3$ per cylinder 1.23 $in.^3$/hp
Working Fluid	Helium
Pressure, Mean	1500 psi
Maximum	2000 psi
Engine Type	RINIA
Phase Angle	90 degrees
Module Size	500 hp

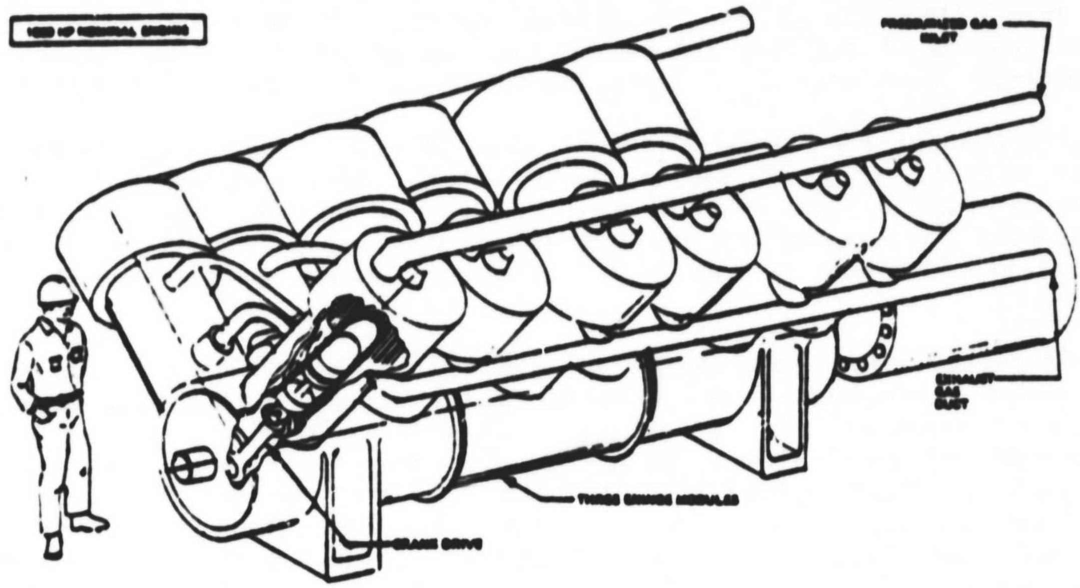

Figure 9-13. Conceptual Layout for a Double-Acting, Vee Stirling Engine Configuration (From Ref. 9-8)

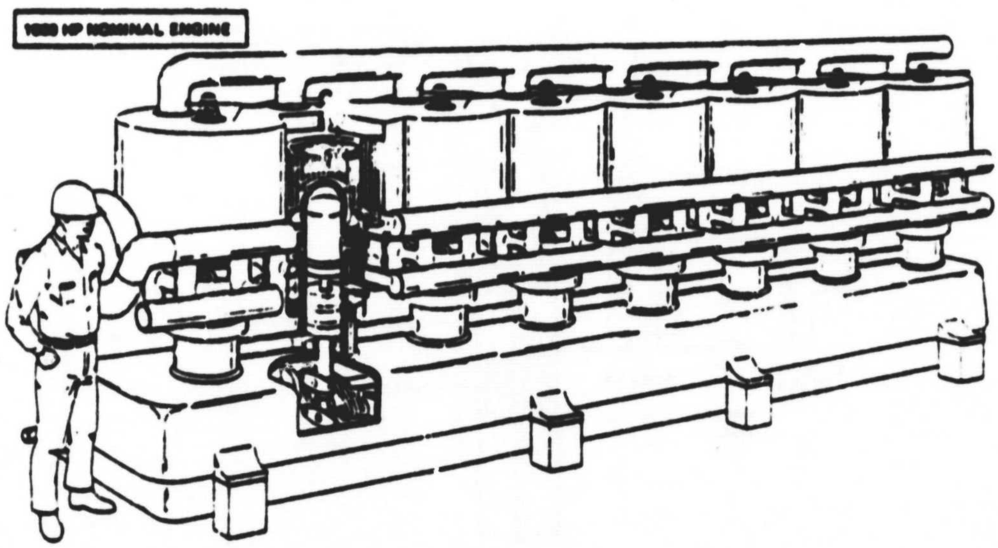

Figure 9-14. Conceptual Layout for a Single In-Line Stirling Engine Configuration (From Ref. 9-8)

Stirling engines have a more demanding cooling requirement (see Table 9-2) than either Diesel or gas turbine engines. As mentioned previously, the efficiency of the Stirling engine improves as the heat sink temperature is reduced; therefore, it is very important to minimize this temperature.

The overall efficiency of a fuel-fired Stirling engine depends heavily on the effectiveness of the air pre-heater. Requirements become more severe as heater input temperature levels increase to maximize engine efficiency. For locomotive applications, ceramic pre-heaters may have to be developed to achieve the high efficiency potential of the Stirling engine.

Stirling engine heater heads are usually fabricated from finned tubes as a means of increasing the external surface area. These heat transfer surfaces are susceptible to fouling if clean burning fuels are not used. For locomotive applications, depending upon the fuel burned, it may be necessary to utilize a heat transport system which transfers the combustion heat from the source to the heater tubes. The combustion of most liquid fuels would not require such a heat transport system but if coal or another solid fuel is used, it would be necessary. Effective indirect heating can be achieved, for example, using a liquid metal heat pipe system or a pressurized helium or hydrogen convective heat transport loop.

From an engine performance standpoint, the heat pipe is probably the preferred heat transport system. It provides a uniform heater head temperature distribution, minimizes the temperature difference between the

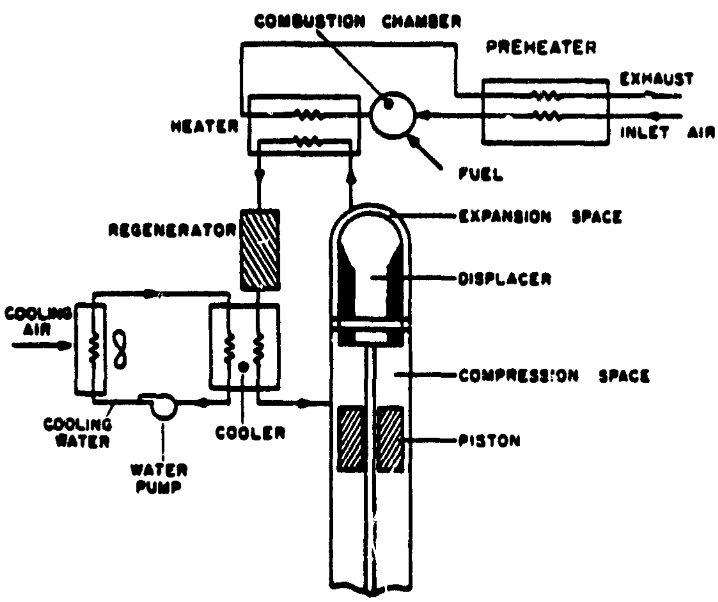

Figure 9-15. Location of Stirling Engine Heat Exchangers (From Ref. 9-1)

combustor and heater head heat exchangers, and requires no auxiliary power. An alternative heat transport system uses forced circulation of a high pressure, low molecular weight gas such as helium or hydrogen. Both helium and hydrogen possess thermal properties which enable a high heat transport rate to be achieved with reasonable pumping losses. Although hydrogen possesses better thermal transport properties, helium is considered to be more attractive because of its lower safety hazard.

J. FUEL TYPES

The Stirling engine can operate from any heat source of sufficiently high temperature. Normally, heat from the combustion of a distillate fuel is utilized. But other liquid, gaseous or solid fuels, as well as solar heat, stored heat, or nuclear reactor heat sources can be successfully used.

For locomotive applications, most gaseous liquid and solid fuels could be considered as possible energy sources. The main consideration for the selection of a particular fuel type relates to its availability, storage capability, and the type of efficient combustion system which can be devised. The candidate combustion systems presented in Table 9-4 have been identified by General Electric (Ref. 9-10) for use with stationary power systems with various fuel types. Of these systems, the most attractive candidate for locomotive use is the atmospheric fluidized bed (AFB). It is especially attractive when high sulfur coal is the fuel. Although Table 9-4 is primarily concerned with coal and residual oil, the Stirling engine is capable of utilizing any form of heat.

The exhaust emissions requirements for automotive engines are currently very stringent and are expected to become even more limiting in the near future. The interest in the Stirling engine for automotive use is based primarily on its superior low emission characteristics, its multi-fuel capability, and its fuel economy potential. In stationary power and locomotive engine applications, however, the exhaust emissions requirements are presently less severe. Distillate fueled Stirling engines can meet the requirements, as do current Diesel and gas turbine engines. Stirling engines burning solid fuels and residual fuels are also expected to meet the current and expected statutory emissions regulations, although with some fuels, sulfur oxides, and particulates may become a problem. Although noise is not as critical for locomotive application, the low noise characteristic of the Stirling engine represents an added advantage. The fuel flexibility of the Stirling engine combined with its high efficiency makes it a very attractive candidate for locomotive service.

K. OPERATION AND CONTROLS

Four different types of control methods have been used with Stirling engines; mean pressure, dead volume, variable phase angle, and variable engine stroke. The first two of these are the most frequently employed. With mean pressure control, a governor opens a supply valve and additional gas flows from a high pressure reservoir tank into the engine, increasing the mean pressure level. This process continues until the desired speed is attained. When engine power is to be decreased, the governor opens the dump valve, decreasing the engine operating pressure until the desired speed is attained. Both of these processes are very rapid. By-pass control is

Table 9-4. Combustion System Selection Summary (From Ref. 9-10)

System	Advantages	Disadvantages	Comments
Stoker	No preparation of coal. Simple design. Available in small sizes.	Low efficiency. Low heat transfer rate. Slow transient. No air preheat permissible. Large excess air required.	Requires exhaust gas sulfur recovery. Not suited for heat pipe application.
Pulverized Coal Burner	Fast transients. Good turn down.	High cost of pulverizer. High auxiliary power req'd. Low heat transfer rate. Corrosion of air preheater. Large excess air req'd.	Present capacity > 100 x 10^6 BT/HR. Development of special mills req'd. Requires exhaust gas sulfur recovery. Not suited for heat pipe.
Cyclone	No preparation of coal. High flexibilty for low-grade coal.	High maintenance cost. Low heat transfer rate. Corrosion of air preheater. Large excess air req'd.	Liquid slag discharge requires fusion. Temp. < 2400°F Present capacity 100 x 10^6 BTU/HR. Requires exhaust gas sulfur recovery. Not suited for heat pipe.
Fluidized Bed	Fuel flexibility - Coal - Wood - Oil, oil shale - All combustibles Increased heat transfer rate	Low temperature 1750°F. Slow transient. Large quantities of flyash.	Good turndown requires special attention. Good match for heatpipe.
Residual Burner	Low cost. Simple/fast control. Easy fuel-handling.	Low heat transfer rate. Vanadium corrosion. Preheater corrosion.	Requires exhaust gas sulfur recovery. Diesel may be more economic.

presently used with mean pressure control to prevent overspeeding upon loss of engine load. The by-pass valve between the buffer space and compression space is also operated by the engine governor. In the event of an overspeed, the governor opens the by-pass valve which rapidly changes the phase and and amplitude of the working fluid pressure. In this manner overspeed control of the engine is achieved.

Dead volume control decreases power output by adding additional volume to the cycle. As a result, the pressure fluctuation decreases, thereby decreasing power. The advantage of this system is that the cycle mean pressure remains constant and no working fluid has to be supplied or pumped off. The part load efficiency is very good, being slightly better than the mean pressure control system.

The earlier large Stirling engine work of General Motors (Ref. 9-2) utilized the variable phase angle method for power control. This method ultimately controlled the direction of engine rotation. Phase control is achieved by varying the phase angle between the power and compression pistons of single-acting machines employing dual crank shafts. Some type of mechanical/hydraulic mechanism is required to change the relative angular position of the two crankshafts.

Engine stroke variations and, therefore, power control can also be accomplished by varying the angle of the swash-plate. The change in stroke and the resulting change in engine displacement and power have been shown to be very fast. A new design of this type has been recently reported by Meijer (Ref. 9-11).

The most likely control system for locomotive use is the mean pressure control system. Using this method, engine torque is proportional to the amount of working gas stored in the system, which directly effects the mean pressure level. The mean pressure control system developed by United Stirling (Ref. 9-12) is shown schematically in Figure 9-16.

Stirling engine heater heads are maintained at a constant temperature independent of power output. The control sensor, usually a thermocouple, is located within the combustion space and connects to an electronic controller. As shown in Figure 9-17, the sensor output controls the position of an air throttle valve used to maintain the proper air-fuel ratio. This particular system is used by United Stirling in combustors burning gasoline and/or distillate fuels. Similar control schemes could be developed to maintain constant heater head temperatures when burning other types of fuel.

L. REGENERATIVE BRAKING

Perhaps the single most unique characteristic of Stirling engines that sets them apart from other alternative power plants is the concept of regenerative braking with direct conversion of energy to thermal storage. This inherent feature makes the Stirling engine particularly attractive as a propulsion source where high payloads are involved and changes in the grade occur. The principle of the combined regenerative braking system with energy storage has been described by Walker (Ref. 9-13).

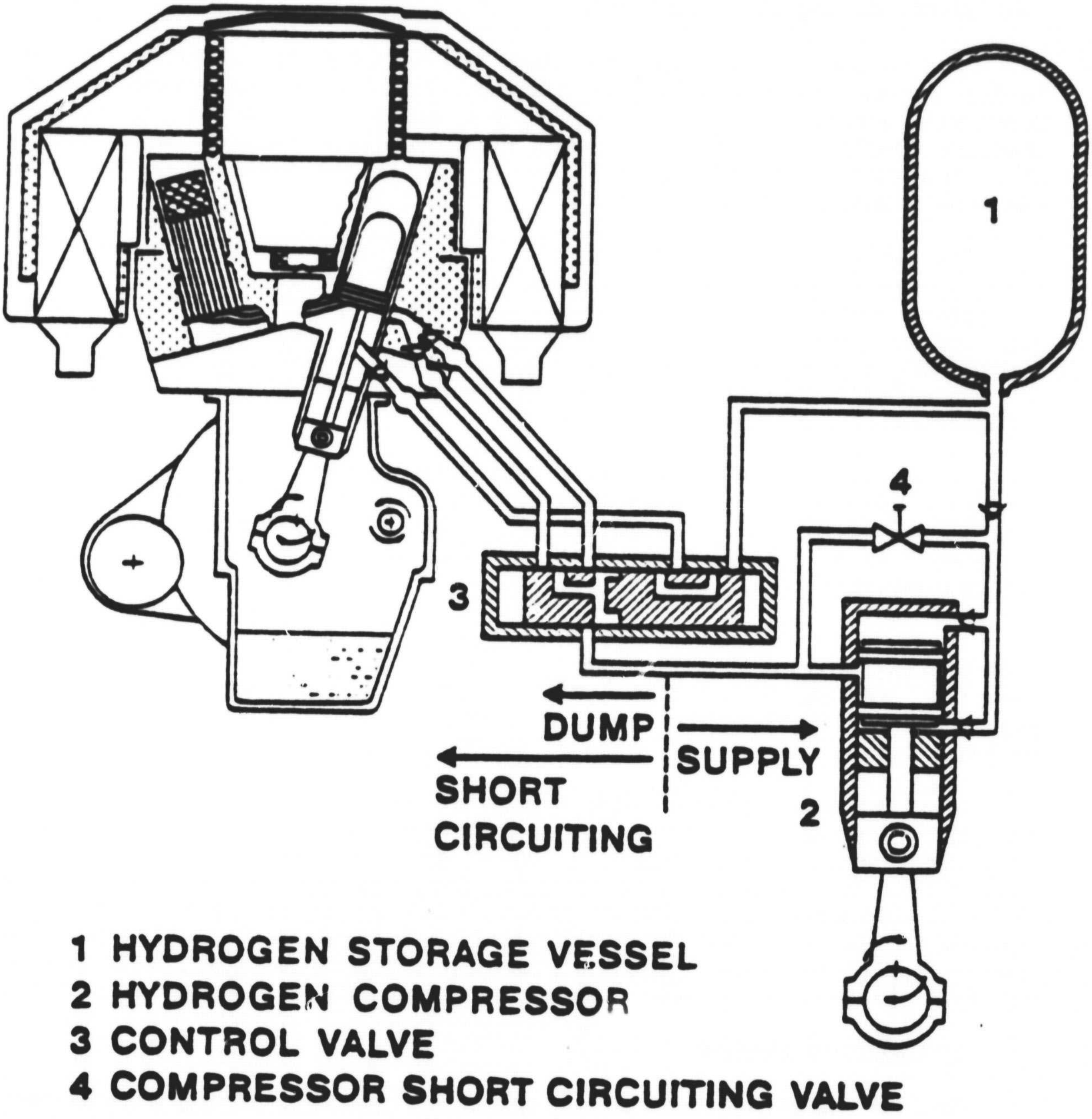

Figure 9-16. Schematic Diagram of a Stirling Engine Power Control System (From Ref. 9-13)

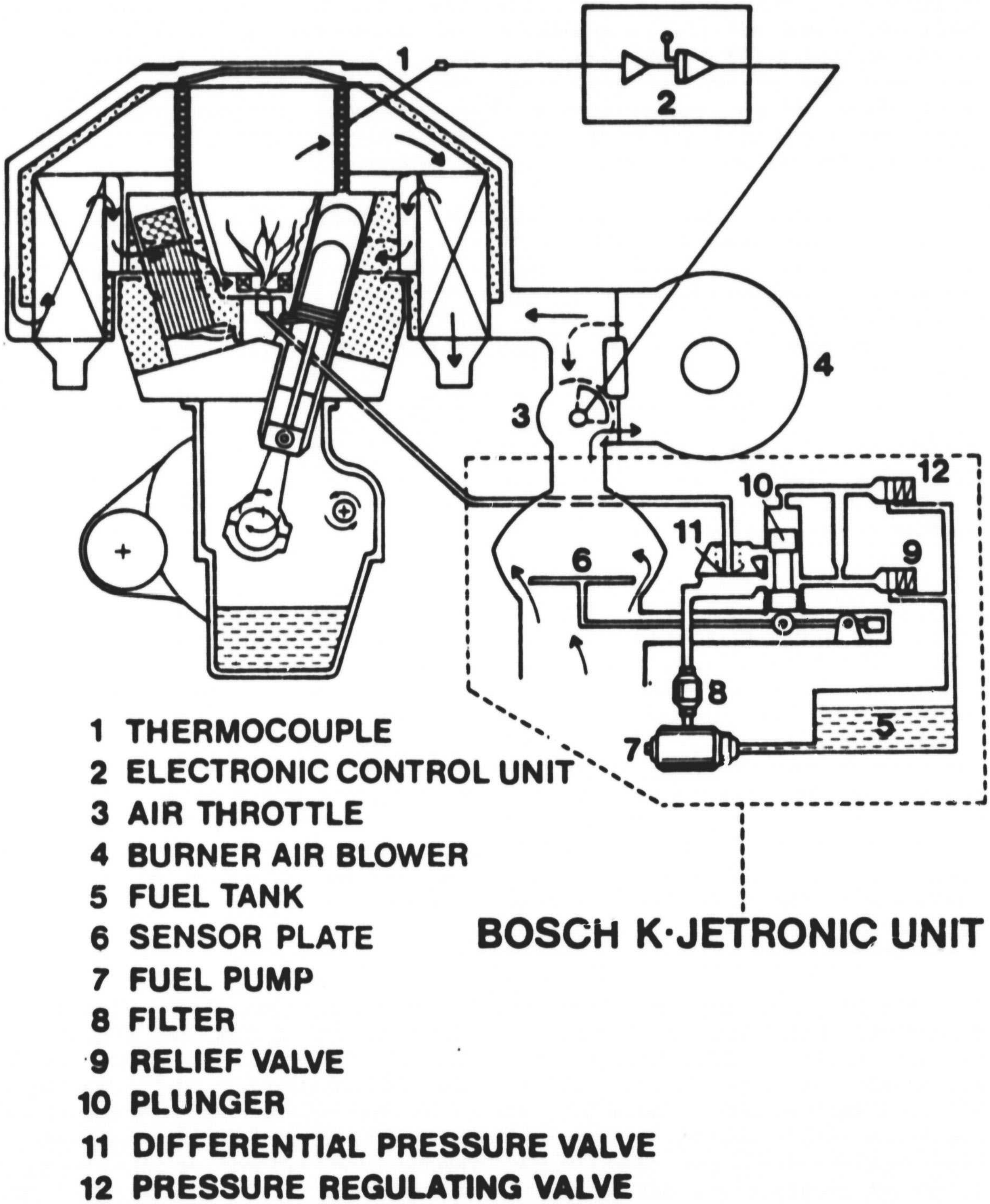

Figure 9-'7. Schematic of the Temperature and Air-Fuel Ratio Controls (From Ref. 9-13)

Using the double acting engine configuration previously described (Figure 9.6), the normal direction of heat flow is into the engine at high temperature and out of the engine at low temperature while producing work at the output shaft. The expansion space of one cylinder is connected to the lower end of the succeeding cylinder through the heater, regenerator and cooler. If the connections between cylinders are reversed, i.e., the upper end of one cylinder to the lower end of the preceeding (not succeeding) cylinder, the engine will try to run in reverse rotation.

If sufficient power is applied to the engine shaft to drive the engine in the original forward direction it will operate, not as a prime mover, but as a heat pump. According to Walker (Ref. 9-13), by simply switching connections between adjacent cylinders, the engine can be converted from a prime mover using high temperature heat and producing power, to a heat pump which absorbs power and generates high temperature heat. This high temperature heat can be stored in a molten salt such as lithium fluoride which has a high latent heat capacity in the temperature range of interest. The heat can be returned to the engine via a liquid sodium heat pipe over a temperature range of 1000°F - 1600°F. Preliminary studies conducted by Walker for mining trucks have shown that for loaded downhaul operations, regenerative braking in this manner can result in a 25 to 50% fuel savings. Further advantages are the reduced wear on mechanical braking systems.

A similar system could be applied to locomotives equipped with large Stirling engines and some type of thermal energy storage. The system would absorb power when operating in the retard (down grade) or normal braking position with a loaded system and then use the stored thermal energy to help propel the train on level ground or on grades. While this is a very intriguing concept, the vast amount of energy involved in train operation and the lower energy density of most thermal storage systems do not make it feasible for locomotive applications. The amount of thermal storage medium necessary for a 70 car train would add another 3 to 4 cars for the energy storage. The savings in fuel from reusing the regenerative energy is offset by the increased use of energy necessary to move the additional cars over the entire route. Except for some special cases, the savings are negated by the increased train weight.

M. SUMMARY

The Stirling engine offers high efficiency and fuel flexibility advantages that make it a possible candidate for locomotive propulsion. Its use would be advantageous primarily in terms of fuel conservation and multi-fuel utilization. In addition to the above noted advantages, Stirling engines also provide good part-load characteristics, low emissions, low noise and vibration. The development of large Stirling engines is significantly different from those intended for automotive use since a different strategy and set of goals are required. It is assumed that this development would take place as part of the large Stirling stationary power systems program. It is questionable if the needed large engine development activities could be totally supported on the basis of potential locomotive usage.

The engine configuration which appears to hold the greatest promise for the power range under consideration is the double-acting type previously under development for passenger cars by the Ford-Phillips team and currently by United Stirling of Sweden. Some of its advantages over the displacer type engines include one piston per cylinder instead of two, fewer burners and controls for a multi-cylinder design, simpler balancing methods, lower weight and volume, and subsequently lower cost. The development of large locomotive Stirling engines will maximize its fuel flexibility and high thermal efficiency.

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SECTION X
FUEL CELLS FOR LOCOMOTIVES

A. INTRODUCTION

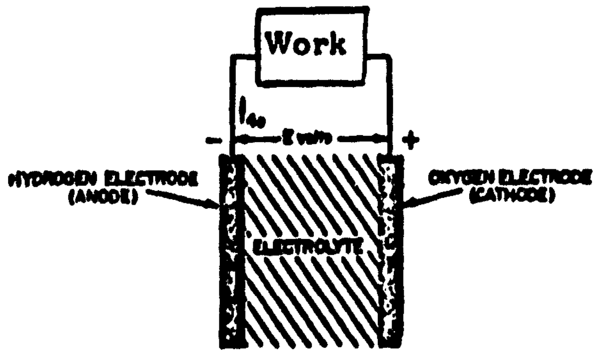
Nearly all of the prime movers used in the transportation field are heat engines using either internal or external combustion. A group of prime movers that do not use heat in the power conversion process are the fuel cells. They resemble batteries but are energy converters rather than energy storage devices. They can also be described as consumable electrode batteries or externally rechargeable batteries. In a fuel cell, a fuel and an oxidant are reacted in the presence of an electrolyte. In trying to understand the principle of operation of a fuel cell, recall the classic electrolysis demonstration in which water was dissociated into proportional volumes of oxygen and hydrogen in an inverted H-shaped burette via application of dc electrical power.

Since water can be dissociated into its components by applying electricity, these same components should react so that they yield water and electricity. This is precisely how a fuel cell works. Such a reaction was first performed by W. R. Grove in 1839, using oxygen and hydrogen as the fuels, with inert electrodes and sulfuric acid as the electrolyte. In fuel cells, oxidizers and fuels are supplied continuously and the electrodes remain largely unaltered.

Fuel cells offer an excellent source of electrical energy to drive locomotives. This is primarily because (1) fuel cells convert chemical energy directly to electrical energy and (2) fuel cells are not Carnot-cycle limited. Each reaction takes place at the electrodes to produce electricity and heat. The reactions typically associated with a fuel cell are shown in Figure 10-1.

Usually, the oxygen from air is utilized as the oxidant at the cathode. Hydrogen, derived from numerous hydrogen containing fuels such as coal, petroleum-derived fuels, pipeline gas, and methanol, is the best fuel. It is the most reactive and therefore, has the highest efficiency of all fuels which could be used at the anode. Because the electrochemical reaction utilizes the available free energy of the fuel at a constant (isothermal) fuel cell temperature, its efficiency is not confined to the same restrictions that apply to classical heat engines.

Fuel cells are commonly classified into one of four types; (1) acid (2) alkaline, (3) molten salt, or (4) solid oxides. The first two have advanced the furthest technologically since they operate at temperatures below approximately 400° F. The last two cell types operate above 1100° F and have material problems even though they do have higher efficiencies than the others. Between the acid and alkaline cells, the latter is more efficient but suffers from a severe limitation; no (or at most an extremely low concentration of) carbon dioxide can be present in either the fuel or oxidant feedstream. This is because the carbon dioxide reacts with the electrolyte to form carbonates which reduce the cell performance. Therefore most, if not all, of the carbon dioxide in the air must be scrubbed. Of even greater importance is the fact that the hydrogen fuel must also be devoid of carbon dioxide. The sources, supplies, and storage of hydrogen



Schematic Representation of a Cell

Anode:	$H_2 + 2OH \longrightarrow 2H_2O + 2e^-$
Cathode:	$1/2\ O_2 + H_2O + 2e^- \longrightarrow 2OH^-$
Overall:	$2H_2 + O_2 \longrightarrow 2H_2O$

Reactions

Figure 10-1. The Hydrogen Fuel Cell (From Ref. 10-1)

make this a tremendous problem for the alkaline fuel cell. This problem does not exist for an acid cell because the carbon dioxide does not react with the electrolyte or interfere with the level of performance.

Of the acid electrolyte fuel cells, phosphoric acid is the most advanced. The combined advantages of low temperature operation (<400° F), the use of present-day materials, reasonable efficiency (50%), and carbon dioxide resistance are factors responsible for this progress. Progress has been made on both acid and the alkaline fuel cells since 1967 and these improvements are illustrated in Table 10-1. The power density of the alkaline fuel cell doubled in 10 years and the catalyst loading has dropped by a factor of five. The last point requires further explanation to fully describe its importance.

The commercial use of fuel cells require that they use conventional fuels. Hydrogen is not a conventional fuel and is not expected to be one for many years. Fuel cells are the most efficient and have the longest life when fueled by hydrogen, therefore, some means of producing hydrogen from conventional fuels is necessary. The available methods for producing hydrogen from conventional hydrocarbon fuels all result in substantial amounts of carbon dioxide. Thus, the tolerance of the phosphoric acid fuel cell to carbon dioxide is a definite advantage for this system.

Typically in a fuel cell system that produces dc power, two components are integrated; (1) a fuel processor and (2) a fuel cell "stack" (i.e., multiple fuel cell units). This permits a variety of fuels to be converted to a hydrogen-rich gas for use at the anode of the fuel cell stack. While this element reduces the overall efficiency of the system compared to the fuel cell stack by itself, it provides some advantages such as: (1) a range of fuels can be used, (2) a continuous supply of hydrogen can be maintained, and (3) the highest efficiency and longest life can be achieved.

The most efficient fuel processor system that can be used with a phosphoric acid fuel cell stack is a steam reformer. The chemical reactions associated with steam reforming and the subsequent water-gas shift reaction in the shift converter are as follows:

Steam Reforming Reaction	$H_2O + (-CH_2-) \longrightarrow CO + 2H_2$
Water-gas Shift Reaction	$H_2O + CO \longrightarrow CO_2 + H_2$
Total Fuel Processor Reaction	$2H_2O + (-CH_2-) \longrightarrow CO_2 + 3H_2$

The thermal efficiency of the integrated fuel processor and the fuel cell unit is 40% as compared to 50% for the fuel cell unit alone. A diagram illustrating this type of system is shown in Figure 10-2. A comparison of the efficiencies of various power systems at different load levels is shown in Figure 10-3. The efficiencies of four different power systems as a function of size are shown in Figure 10-4.

Table 10-1. Improvements in H_2/Air Fuel Cells (From Ref. 10-2)

	Alkaline		Acid		
	1967	1977	1967	1977	1977
Current density (mA/in.^{2})	650	1300	2000	2000	2600
Cell Voltage (V)	0.75	0.7	0.55	0.65	0.60
Power density (mW/in.^{2})	500	900	1060	1260	1550
Operating temperature (°F)	150-170	180	320	375	80
Thermal efficiency (%)	51	47	37	44	41
Catalyst loading (mg/in.^{2})	6	<1.0	130	5	25
Start-up time	3.5 hr	5 min (1969)			15 min
Life (hr)	5,000		>12,000		600

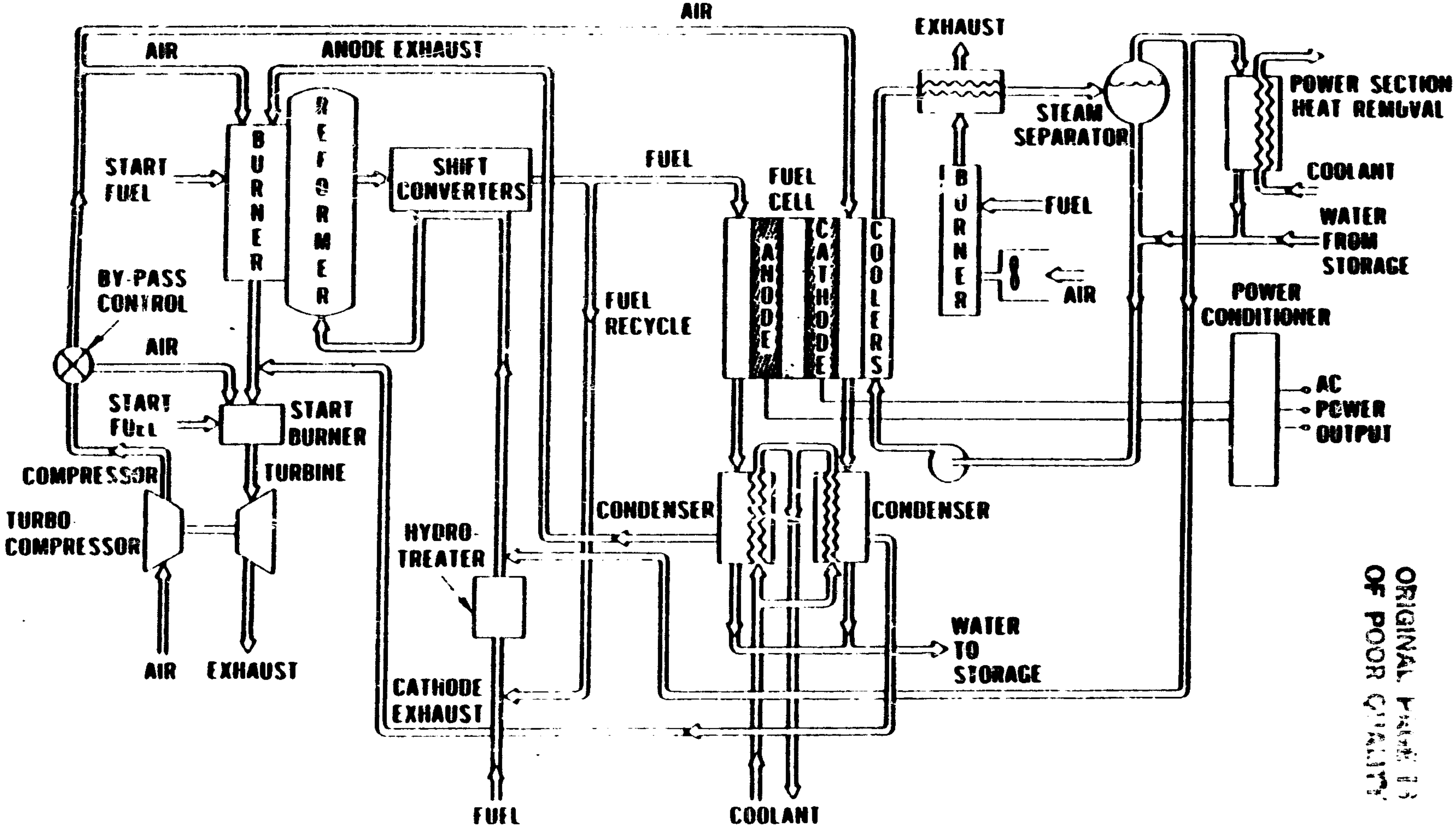


Figure 10-2. Schematic of United Technology Corporation Phosphoric Acid Fuel Cell System

The development of an integrated fuel processor and the phosphoric acid fuel cell system has reached the point where a 4.8 MW unit is to be demonstrated in the Consolidated Edison grid in New York in 1981. A smaller (1.5 kW) unit has been built for the U.S. Army (Figure 10-5) and it is a fully integrated system. This unit is part of a program directed at developing a family of silent, lightweight power sources with ratings of 0.4 to 5.0 kW. Methanol is the fuel and it goes through a low temperature steam reformer before being used at the anode of the fuel cell. Air is used as the cathode and phosphoric acid is the electrolyte.

Fuel cells rated at 40 kW and 60 kW have been built and tested. Table 10-2 shows the fuel consumption and the efficiency as a function of load for the methanol and the propane fueled cells. The estimated performance for the 60 kW reformed methanol-air fuel cell system is shown in Figure 10-6.

The fuels to be delivered to the Consolidated Edison unit will be pipeline gas and clean naphtha. The expected thermal efficiency to ac power, based on the lower heating value of the fuel, is approximately 38 and 37% respectively for the two fuels. If methanol were to be used as a fuel, it has been projected to result in a thermal efficiency of 39%. A recent publication for the DOE suggests that megawatt-size units can be expected to be 37.9 and 44.2% efficient when ac power is generated, and 39.5 and 46.0% when dc power is generated using naphtha and methanol, respectively. Detail of the 4.8 MW fuel cell is given in Table 10-3.

B. PHOSPHORIC ACID FUEL CELL SYSTEM DESCRIPTION

The phosphoric acid fuel cell is made up of several repeating components sequentially placed together. A conceptual diagram showing the relationship of the fuel and oxidant feed, electrodes, electrolyte, and current collectors is presented in Figure 10-7. Intercell cooling channels are placed throughout the stacked cells to remove heat and maintain a uniform and relatively constant temperature across the plates. A plot showing the progress in improving cell performance is shown in Figure 10-8. Parameters which affect cell performance are gas composition, temperature, pressure, impurity levels, and internal resistance.

The efficiency of the conversion process from the chemical energy in the fuel feed gas to the dc electrical output is a function of the electrochemical efficiency and the heating value ratio. The electrochemical efficiency is the product of the thermodynamic, and the electrical efficiencies of the cell. The heating value ratio is the ratio of the hydrogen heat energy to the total combustible gas heat energy in the fuel feedstream. This term links the conversion efficiency of the fuel processor to the fuel cell thermal efficiency. The hydrogen concentration in the fuel feedstream is a direct function of the fuel processor operation.

The basic materials needed to produce fuel cells are carbon for the electrodes, teflon for seals, platinum as the electrode catalyst, and phosphoric acid. Phosphoric acid fuel cells have been built and tested continuously over the past 10 years with fuel processors supplying the necessary hydrogen. Although the demonstration phase of the large (100 Kw to 5 MW) units is slated to begin within the next year, there are still some difficult problems to be faced. The cost of platinum and the amount needed in the cells is a serious concern. The loss of electrolyte in operating cells is another problem. The limited electrochemical activity at the air cathode is a barrier to the further improvement of cell performance. These problems and the need for the standardization of phosphoric acid fuel cell components constitute the obstacles which must be overcome in the next decade.

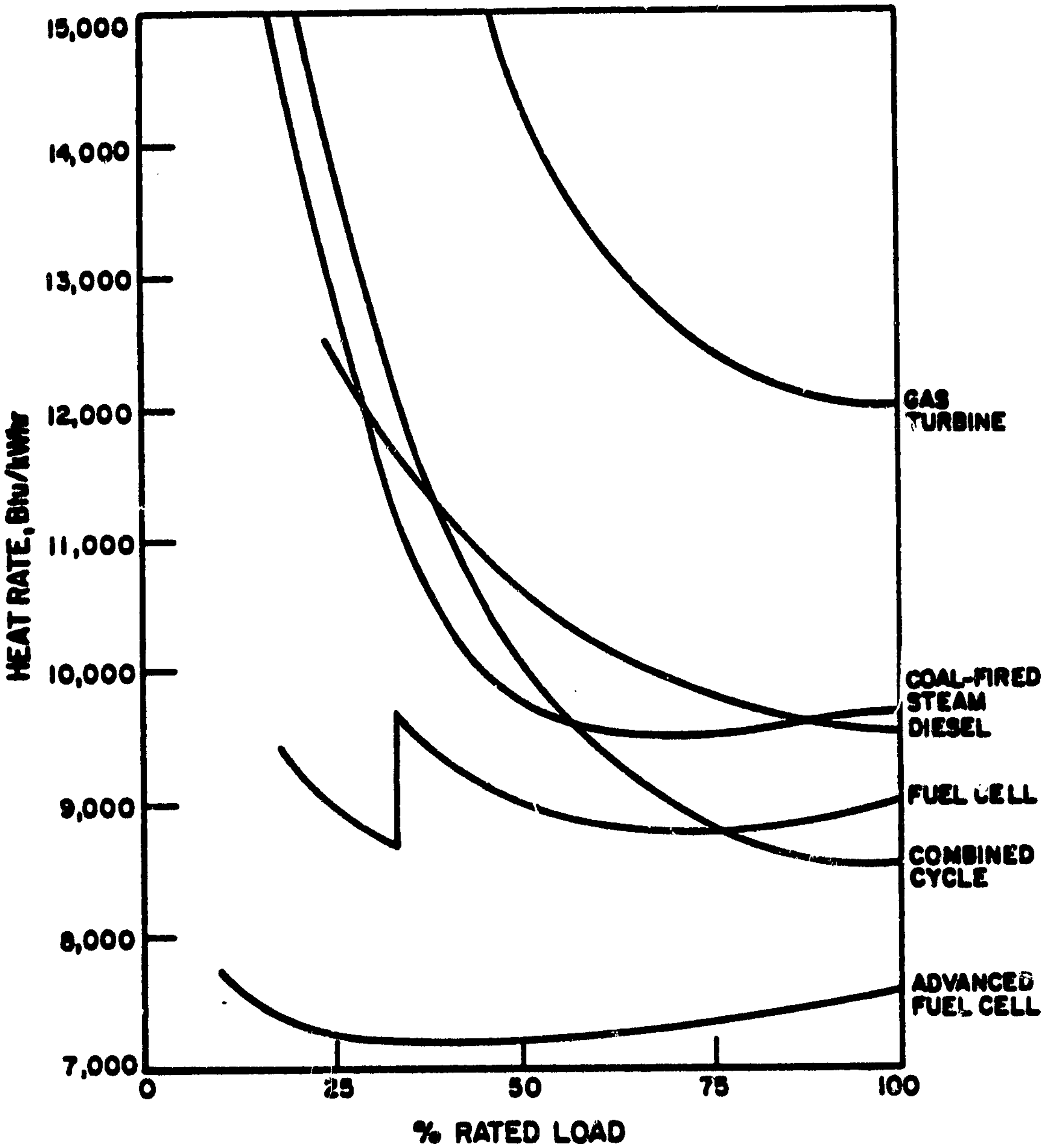


Figure 10-3. Comparison of ac Power System Heat Rates
(From Ref. 10-4)

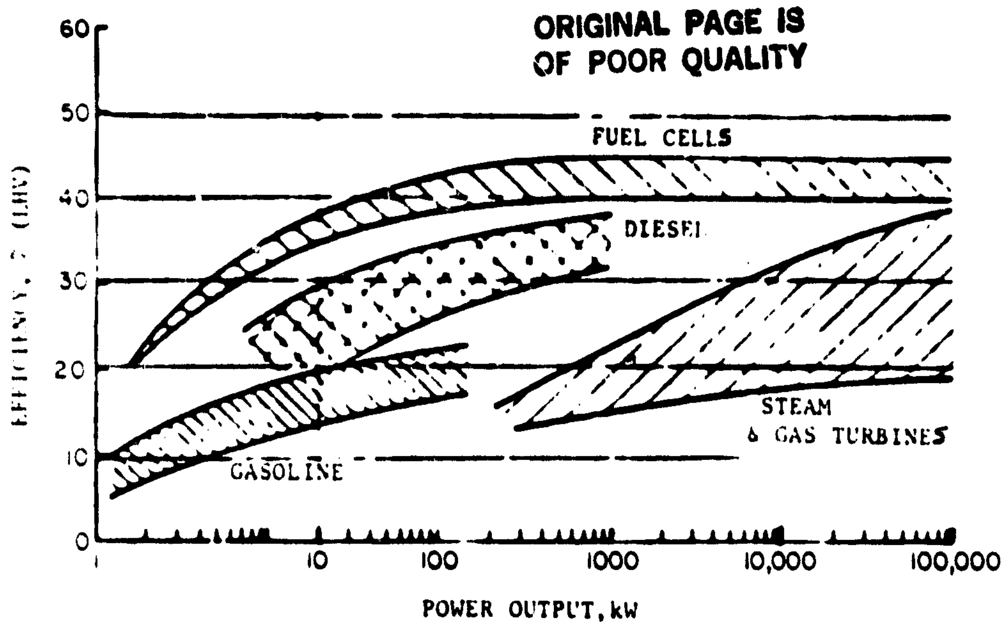


Figure 10-4. Power System Efficiency Comparison as a Function of Size

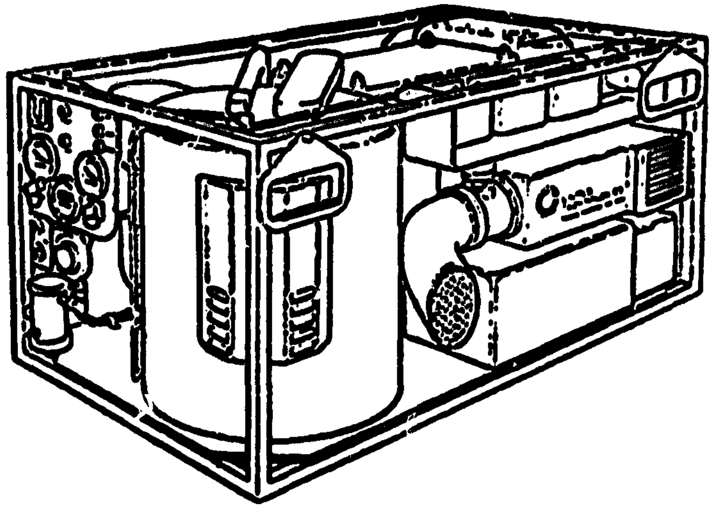
C. FUEL PROCESSOR

For locomotive applications, the key element in providing an efficient, compact fuel cell unit is a processor that is capable of converting the fuel into a hydrogen-rich gas. The primary components of the fuel processor in a phosphoric acid fuel cell system are two catalytic reactors. One is the steam reformer and the other is the shift converter.

Methanol and naphtha are the only two fuels likely to be used in a locomotive fuel cell. The methanol steam reformer operates at temperatures below 610° F while the naphtha reformer requires a 1725° F temperature. Because of the efficiency of the steam reformer process, very little shift conversion is needed for methanol. On the other hand, naphtha requires extensive shift conversion in order to reduce the carbon monoxide concentration in the fuel cell feedstream to the required one percent maximum level.

The energy necessary for the steam used in the reformer is important to the overall efficiency of the processor. A measure of the steam requirements is the steam-to-carbon ratio which is about 1.5 to 1 for methanol and 3.5 to 1 for naphtha. These are the minimum amounts of steam necessary to convert the carbon in the fuel to carbon dioxide without depositing free carbon in the catalyst bed. Carbon deposition in the fuel processor results in catalyst failure which necessitates its replacement. The higher steam rate for naphtha is reflected in its larger steam converter unit. The extensive shift conversion required for naphtha makes the conversion unit larger as well.

The fuel processor for converting methanol to a hydrogen-rich gas is more efficient, smaller, and operates at a lower temperature than the



Characteristic	Units	Value
Specific power	W/lb	8.6
Power per unit volume	kW/ft^3	.22
Lifetime	hr	6,000
Rated output	kW	1.5
Weight	lb	175
Volume	ft^3	7
Fuel consumption	lb/kWh	1.22
Start time	min	15
Mean time between failures	hr	1500
Temperature range	°F	-65 to +125
Number of starts		2000

Figure 10-5. 1.5 kW U.S. Army Methanol Fuel Cell Power Source (From Ref. 10-1)

Table 10-2. Fuel Consumption and Power Plant Efficiency As a Function of Load for 60 kW System

	Fuel Consumption lb Mole/hr		% Efficiency Base on HHV[a]	
% Load	Methanol	Propane	Methanol	Propane
100	1.5765	.5225	41.6	41.3
75	1.1631	.3810	42.3	42.5
50	0.7685	.2548	42.7	42.4
25	0.3946	.1343	41.5	40.2
Hot Idle	0.0623	.0208		

[a]HHV - higher heating value

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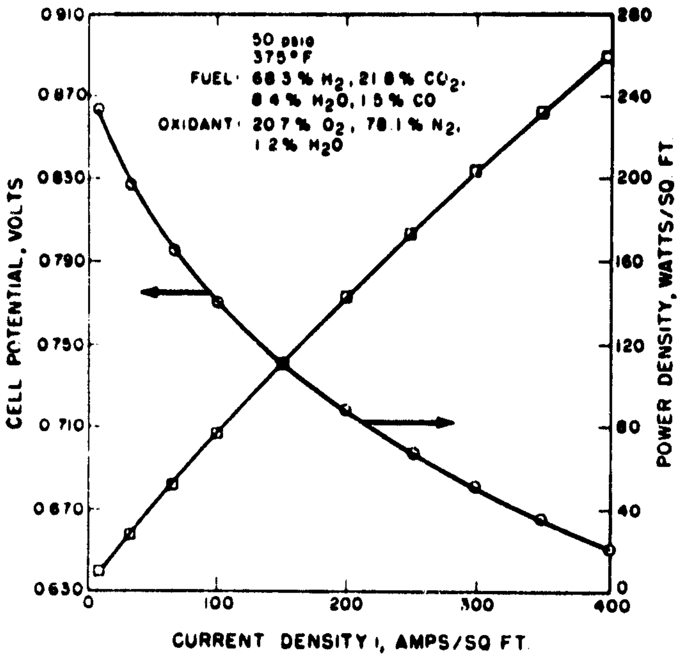

Figure 10-6. Estimated Performance Curve for 60 kw Reformed Methanol-Air Fuel Cell System (From Ref. 10-2)

processor for naphtha. The space saved by the methanol fuel processor as compared to the naphtha processor is offset by the increased volume of fuel required on-board the locomotive. The energy density of methanol is about half that of naphtha so that the volume required, therefore, is twice that of naphtha. The difference in the amount of steam, hence, water required for steam reforming tends to balance out the space requirements.

In addition, the methanol could be stored on-board the locomotive as a premixed solution with water, thereby requiring the metering of only one fluid. The alternative to this approach is to have an extensive water reclamation and purification system. A methanol fuel processor / phosphoric acid fuel cell system is depicted in Figure 10-9.

D. ADAPTATION OF A METHANOL-PHOSPHORIC ACID FUEL CELL SYSTEM TO A LOCOMOTIVE

If an existing 3000 hp Diesel-electric locomotive were to be retrofitted with a fuel cell system, the fuel processor and fuel cell would be required to have the capacity of 2.5 MW and would replace both the Diesel engine and the alternator. To fit in a locomotive, the fuel cell system must be contained in a space 24 ft long, 8 ft wide, and 6 to 10 ft high and can weigh no more than 27 tons.

Using the dimensions and specifications of the 4.8 MW ac fuel cell, an estimate can be made for a 2.5 MW dc fuel cell. The 4.8 MW unit is fueled by pipeline gas or light naphtha and, therefore, the fuel processor is larger than is required for a methanol processor. Figure 10-10 shows the plant layout of the 4.8 MW ac module. Approximately one-half of one fuel processor pallet and one complete power section (fuel cell stack) pallet as shown should provide the 2.5 MW output. The steam reformer and high temperature shift conversion units on the 4.8 MW number 2 fuel processor pallet are depicted in Figure 10-11. The high temperature shift unit is not required for methanol and the steam reformer unit would be approximately one-half the size shown. The major components on the other fuel processor pallet are not necessary for methanol processing.

Table 10-3. Characteristics of 4.8 MW Utility Fuel Cell Power Source

Heat Rate	9300 Btu/kWh at rated power 9000 Btu/kWh at 30% of rated power
Power	
Power rating	4.5 MW (net ac) at sea level, 660° F at unity power factor
Minimum power	25% of rated power
Standby	Zero output power
Life	
Design life (with scheduled overhaul maintenance)	20 years
Fuels	
Liquids	Naptha or selected kerosene
Gaseous	Natural gas
Load-Response Time	
Minimum power to rated output	15 seconds
35% power to rated output	0.5 second
Electrical Output	
Power form	3 phase; 60 $\pm$ 0.1 Hz
Voltage	13.8 kV ac
Operation	
Control	Automatic, from remote or on-site controller
Modes	Load, spinning reserve, standby off, and no-load power-factor correction
Start	Four hours from 70° F
Miscellaneous	
Ambient temperature	-30° F to + 110° F
Acoustic noise	55 dB(A) 100 feet from power plant perimeter
Heat-rejection method	Air-cooled (dry cooling tower)

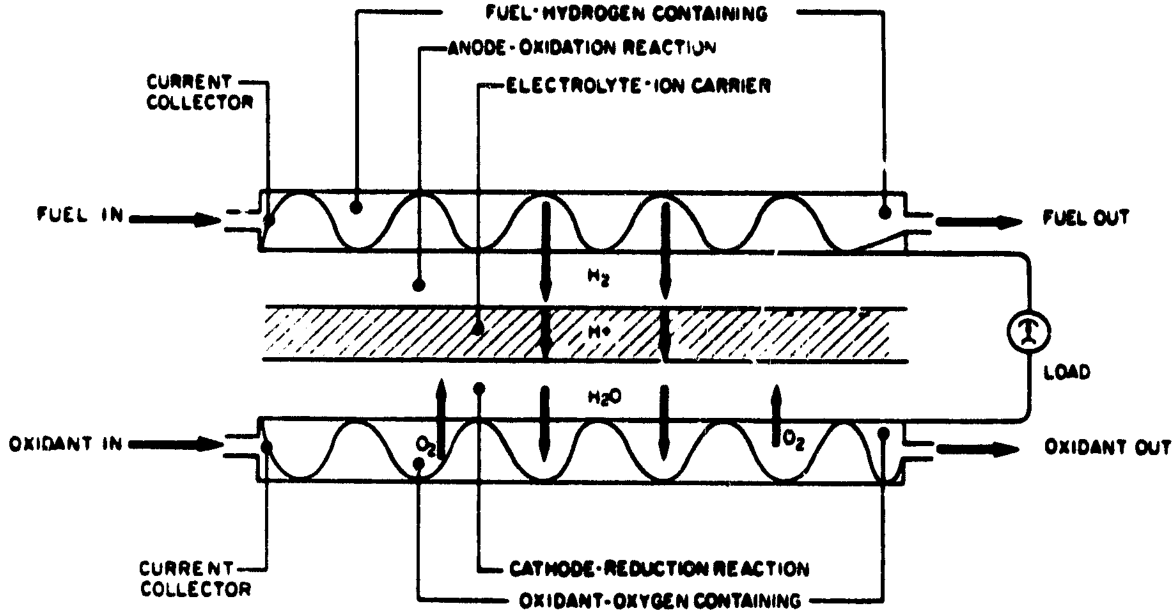

Figure 10-7. Conceptual Diagram of a Phosphoric Acid Fuel Cell

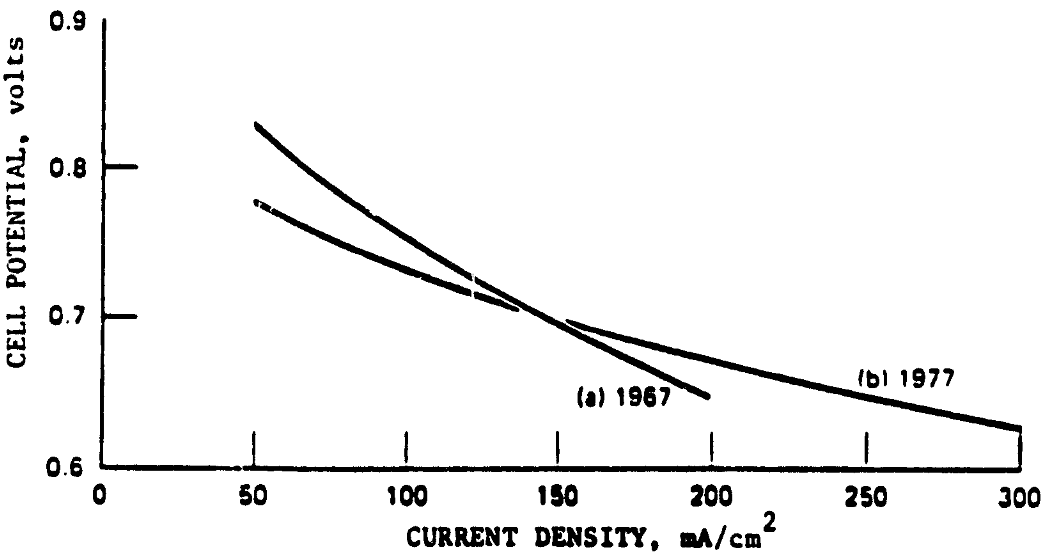

Figure 10-8. Phosphoric Acid Fuel Cell Polarization Curves

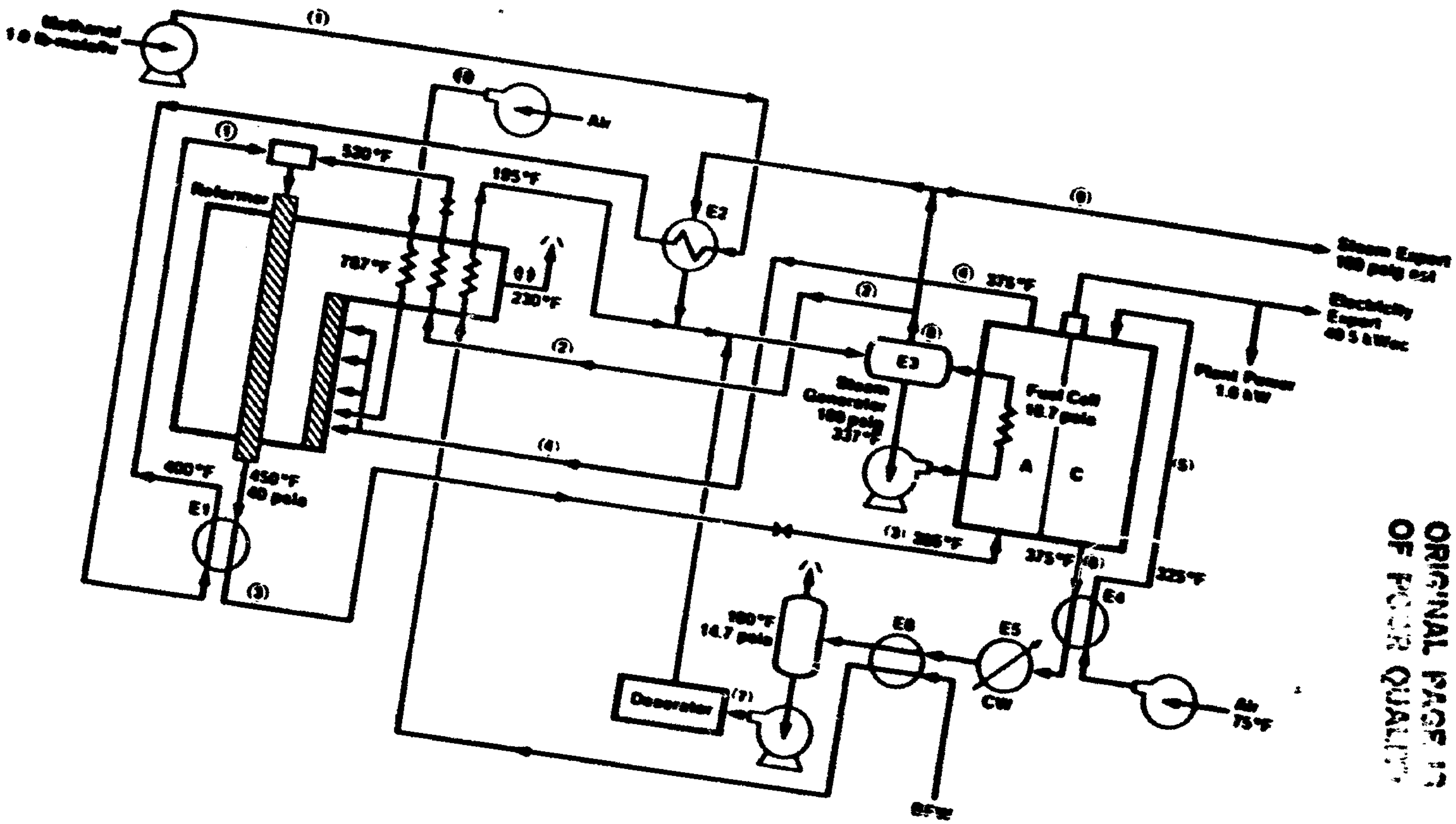

Figure 10-9. Methanol Reformer/Phosphoric Acid Fuel Cell System

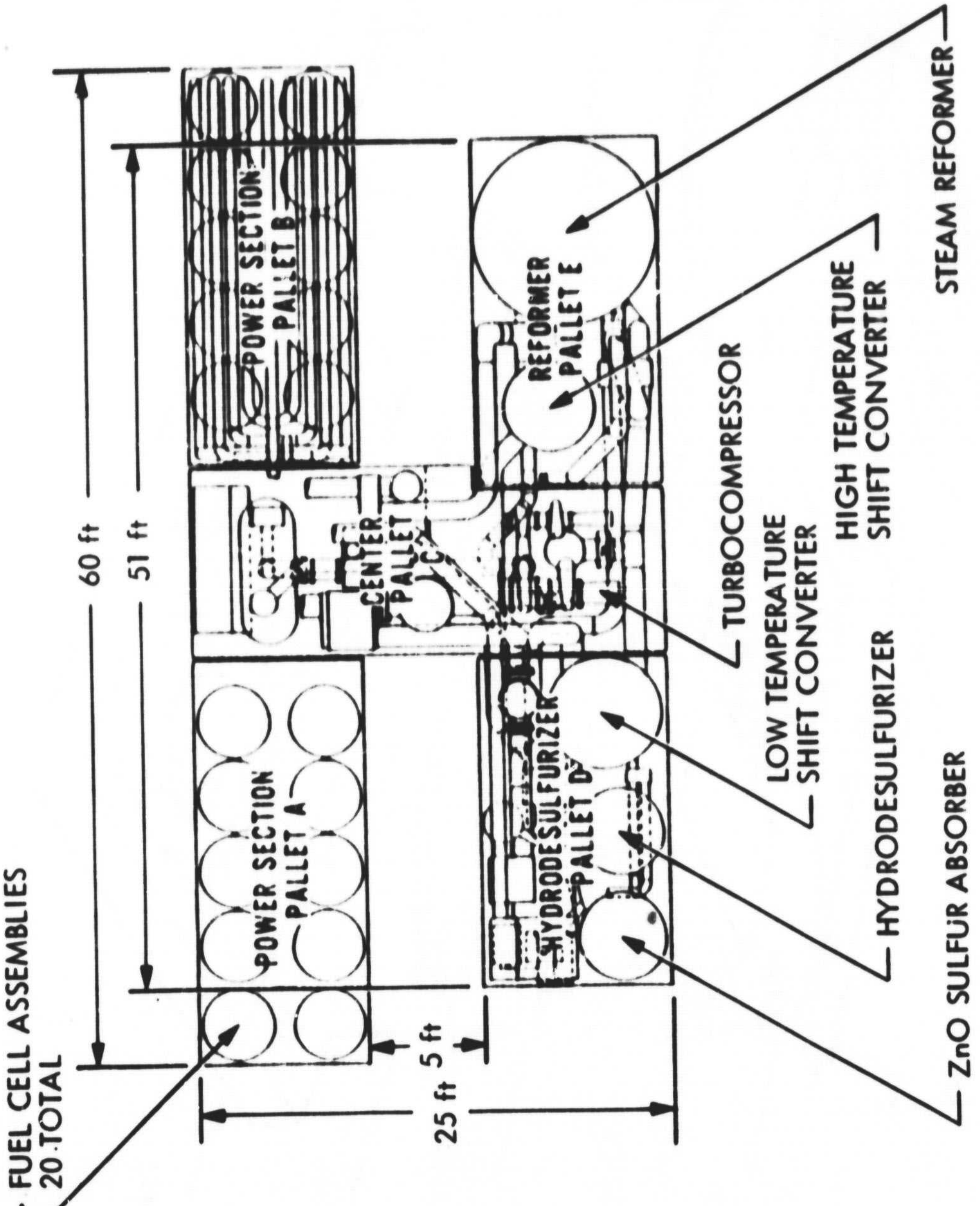

Figure 10-10. Module Demonstrator Pallet Definition of dc Module

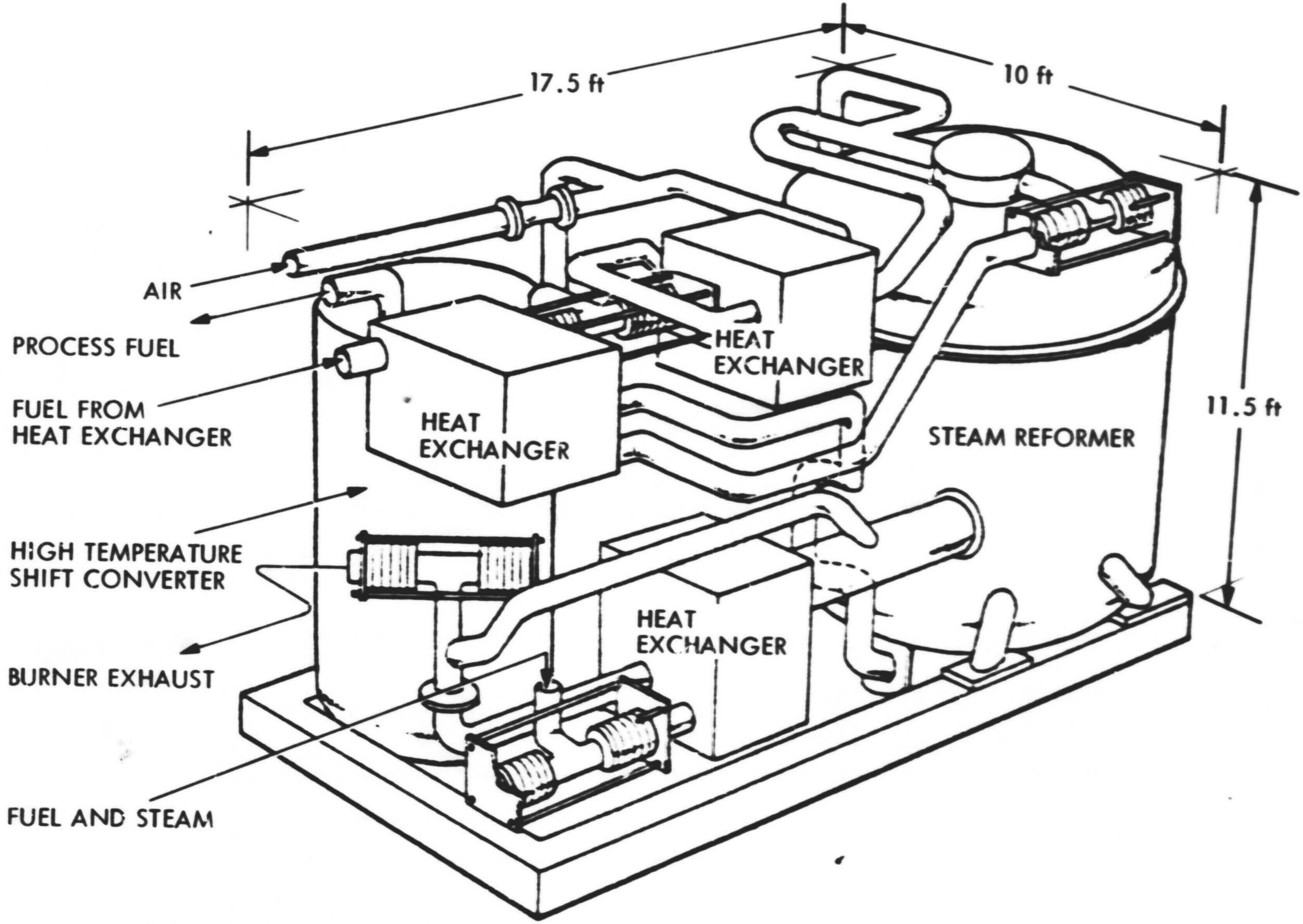

Figure 10-11. Fuel Processor Pallet E
Pallet Weight - 70,000 lb

The projected unit would be 14 ft long, 8 ft wide, and 10 ft high and would house the fuel processor and controls. It could possibly weigh 14 to 15 tons. One fuel cell section would occupy a space approximately 18 ft long, 9 ft wide, and 10 ft high with a weight of approximately 28 tons. Based on these projections it appears that with some packaging modifications, an advanced methanol fueled phosphoric acid fuel cell comparable in rated power to the Diesel-engine could be mounted on a locomotive chassis. However, the fuel capacity, now normally 2000 to 4000 gal of diesel fuel would be insufficient for the same range by nearly one-half because of the difference in the energy content. In addition, if a methanol/water mixture is used as previously described, a total capacity increase of approximately four times would be necessary and a tender is required for the train. However, one tender could serve two locomotives.

Figure 10-12 shows the size and packaging of a 40kW fuel cell being developed for the Gas Research Institute and DOE to meet the energy needs of buildings and for industrial use. The major contractor for this project is United Technologies, Inc. Details of this fuel cell are given in Table 10-4.

The controls necessary to operate the dc traction motors using the power from a phosphoric acid fuel cell are expected to follow the approach used in vehicular applications by the U.S. Army. A schematic of this system is shown in Figure 10-13. The elements involved in the control system are (1) a silicon control rectifier (SCR) which interfaces between the operator and the power system and (2) a feedback loop connecting the fuel cell power output and the fuel flow. While the SCR must be specifically designed to match the fuel cell voltage-current output to the dc motor requirements for a given operational setting, this type of control is not unusual. Basically, this type of control is currently in use and is commonly referred to as a thyrister. The feedback control loop within the fuel cell system is a load-following trim system which maintains the balance between the electricity demand and the fuel flow. The fuel cell control system is designed to follow the load on a utility peaking duty cycle and it is therefore, expected to meet the operational demands of a locomotive.

E. SUMMARY

Many factors must be taken into account in assessing whether fuel cells should or could be utilized in locomotives. Consideration must be given to advantages, availability, adaptability, cost, fuel, ease of operation and future locomotive needs. Table 10-5 shows some of the characteristics of both the Diesel-electric and the fuel cell locomotives.

If fuel cells were currently as available as the Diesel-electric systems, the efficiency comparison of the two would favor the fuel cell, both at full load and at part load as shown in Figure 10-3. This fuel cell system would operate on coal-derived methanol. The availability of light distillate fuels from coal is a future possibility but coal-derived methanol is possible in the near future using current technology. Although the fuel cell can be operated on fuels other than methanol, the size constraints of the locomotive would require considerably more development in the fuel processing section of the system. The operational controls

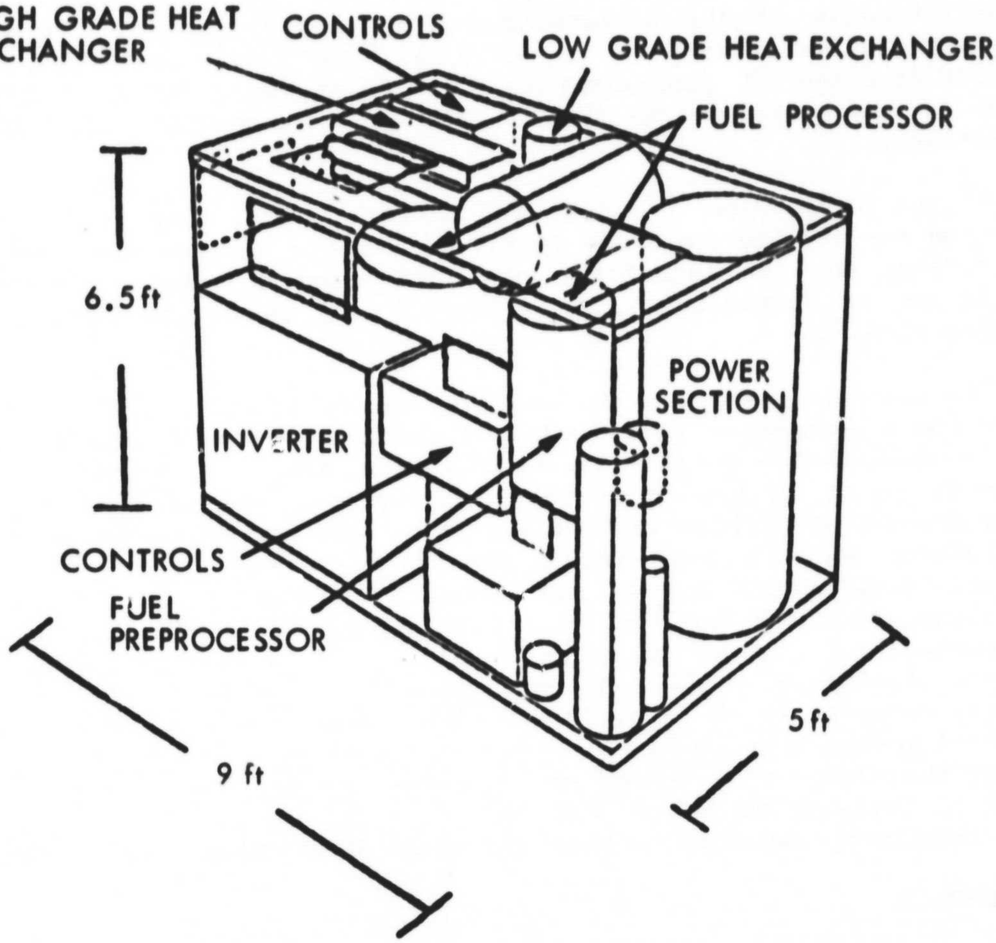

Figure 10-12. 40 kW Fuel Cell Power Source

Table 10-4. United Technology Fuel-Cell Data

1.	Fuel-cell	Reformed gas/air
2.	Cell	Phosphoric Acid
3.	Size (kW)	40
4.	Status	Demonstrated hardware
5.	Volume (ft^3)	
	a. Cell stack	50
	b. Fuel processing	30
	c. Thermal management	11
	d. Water recovery	14
	e. Heat recovery	3
	f. Reactant supply	4
	g. Inverter	11
	h. Plumbing	40
6.	Weight (lb)	
	a. Cell stack	1800
	b. Fuel Processing	820
	c. Thermal management	450
	d. Water recovery	250
	e. Heat recovery	85
	f. Reactant supply	140
	g. Inverter	520
	h. Plumbing	800
7.	Operating conditions	
	a. Temperature	375°F
	b. Pressure	4 in. - 5 in. water
8.	Operating characteristics	
	a. Voltage	163 V
	b. Design	287 A (overall efficiency based on H.H.V. of fuel)
	c. Start-Up time	Reformer zero hours, Stack 3 hours
	d. Life continuous operation	8000 hours

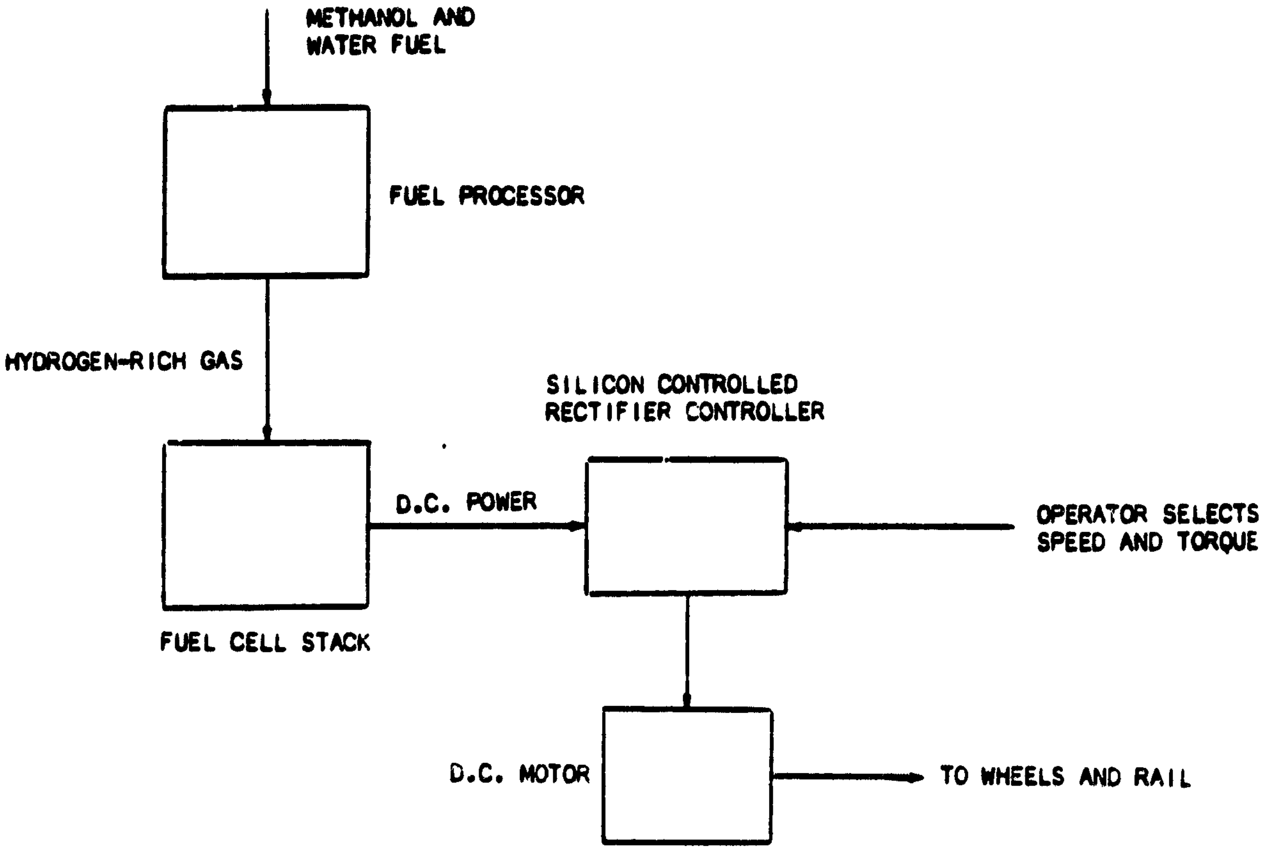

Figure 10-13. Fuel Cell/Drive Train Control Interface

required for a fuel cell require minor modifications which can be met with current technology. Fuel cells have demonstrated their ability to follow the load for simulated utility demands for both the small 40 Kw units and the 1 MW units in test facilities. This indicates that the response time is adequate for locomotive use.

A commercial fuel cell system can be available within the next 10 years. There is little doubt that the fuel cell could be adapted to meet the locomotive performance requirements using methanol as a fuel. It is possible that utility sized fuel cells could be commercialized in the next 7 years. These systems would have to be modified for locomotives because their size requirements are less than half of the smallest utility unit of 5 MW. Downsizing and repackaging will be necessary to meet the particular size and weight constraints of the locomotive. If this work was undertaken in parallel with the utility fuel cell program, a system could be completed for testing in about 6 to 8 years. A less ambitious approach would probably require 10 years before a test unit could be available.

Over and above all these factors are the future needs of the railroad. If the use of coal-derived fuels and the improved efficiency are sufficiently important, the expense of adapting fuel cells may be justified. As fuel cell technology advances toward commercialization for utilities, they will become less expensive. Time is the determining element.

Table 10-5. Design Characteristics of a Fuel Cell Power Source for a Locomotive

Characteristic	Units	Diesel Requirement	Fuel Cell Configuration
Peak Power	kW	2000	2000
Voltage	V	600	500
Max. Dimensions (LxWxH)	ft x ft x ft	24 x 9 x 8	24 x 9 x 8
Max. Volume	ft^3	1700	1700
Max. Weight	tons	60	60
Peak specific power	W/lb	17	22
Peak power / unit volume	kw/ft^3	1.20	1.05
Rated efficiency	%	40	40
Rated heat rate	Btu/bhp-hr	6400	6400
Start time (from cold)	min	30	120
Time (no load to full load)	sec	10	3
Operating life	hr	50,000	150,000
Time between major overhauls	hr	15,000	40,000
Pollutants			
HC	g/bhp-hr	1.5[a]	0.0001
CO	g/bhp-hr	125[a]	
NO_x + HC	g/bhp-hr	10[a]	0.07
Opacity	%	15-50[a]	None

[a]1979 Requirements for Heavy Duty Diesel Engines

F. SECTION X REFERENCES AND NOTES

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SECTION XI
OTHER ENGINES AND TRANSMISSIONS

A. INTRODUCTION

The engines already discussed are the most likely candidates for locomotive applications. Four other engines: the Naval Academy heat balance engine, the stratified charge rotary engine, the reacting gas Brayton cycle, and the sodium heat engine, are discussed in this section. Three of them are only at the laboratory stage of development so their use in locomotives is many years away. However, they do indicate the direction that engine research is taking.

Two of the engines are based on the Otto cycle. The Otto cycle engines (spark ignition engines) have been used in rail cars but they have never been seriously considered for main line service due to their inefficiency. Gasoline engine powered rail cars were used on branch lines as early as 1905. General Electric produced gasoline engine powered rail cars from 1909 to 1917 at which time they discontinued production of the gasoline engine in order to concentrate on railroad electrification. In 1922, H. L. Hamilton formed the Electro-Motive Engineering Co. with the intent of designing a gasoline-electric railcar. From 1922 to 1930, this company produced rail cars powered by their own gasoline engines and by engines produced by the Winton Engine Co. In 1930, both companies were purchased by General Motors. The present Electro-Motive Division of GM, the largest producer of Diesel-electric locomotives in the U.S., started out as a producer of gasoline engines for rail service.

The spark-ignited Otto cycle engine may yet be a viable locomotive engine. There are two new variations of Otto cycle engine that look promising. These are the Naval Academy Heat Balance Engine and the stratified charge rotary engine. In addition to the Otto cycle engines are the reacting gas Brayton cycle engine which is a a novel form of a gas turbine, and the Sodium Heat Engine, a true heat engine that bears a remarkable resemblance to a fuel cell.

The alternative transmissions are the hydrokinetic, hydrostatic, and manual transmissions similar to those used in cars and other vehicles. There are some advantages to these transmissions but also some serious drawbacks. These are discussed later in this section.

B. NAVAL ACADEMY HEAT BALANCE ENGINE

The Naval Academy heat balance engine (NAHBE) is based on a non-adiabatic compression process that utilizes retained heat and shock waves to enhance the combustion efficiency. Improvements in thermal efficiency on the order of 43% over the conventional Otto cycle engines have been claimed for some off-peak operating regimes. Figure 11-1 shows the combustion process which entails the following sequence:

(1) Piston approaches top dead center, intake valve opens, and intake stroke begins.

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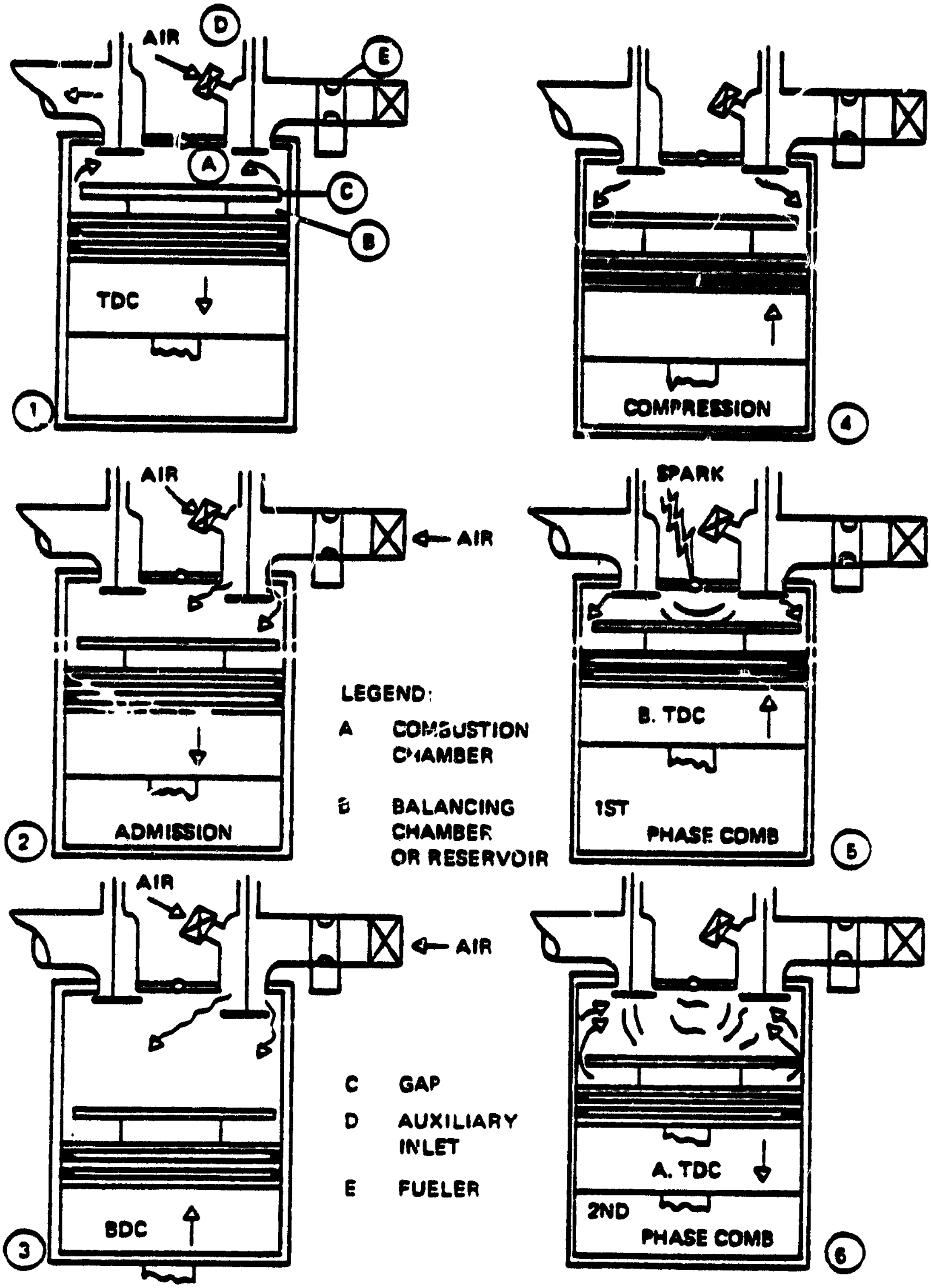


Figure 11-1. Naval Academy Heat Balance Engine Cycle
(From Ref. 11-1)

(2) nitial portion of intake stroke air enters the cylinder through auxiliary inlet (D), followed by a fuel-air charge from venturi (E).

(3) Charge is stratified with a very lean composition just above the piston.

(4) Compression forces air into reservoir (B), also known as balancing chamber.

(5) Ignition causes rapid pressure buildup with large pressure ratio occurring across gap (C). Subsequent shock compression wave propagates under piston cap with expansion wave propagating upward into combustion chamber.

(6) Shock compression under cap builds pressure to higher value than above cap causing air to flow to combustion chamber (A), (Ref. 11-1).

The efficiency gains are due to the two non-ablabatic thermodynamic processes that take place during the cycle. The first process is the transfer of heat to and from the piston cap. The second is the heating of the gas by shock waves generated during the passage of the air into the balancing chamber.

Research into the Naval Academy heat balance engine was funded by The Office of Naval Research (ONR). It was directed at quantifying the contribution of the shock waves and the effect of heat retention by the piston cap. This research effort was completed by the beginning of FY78. Based on the substantiation of the theory, ONR planned a two-year program to:

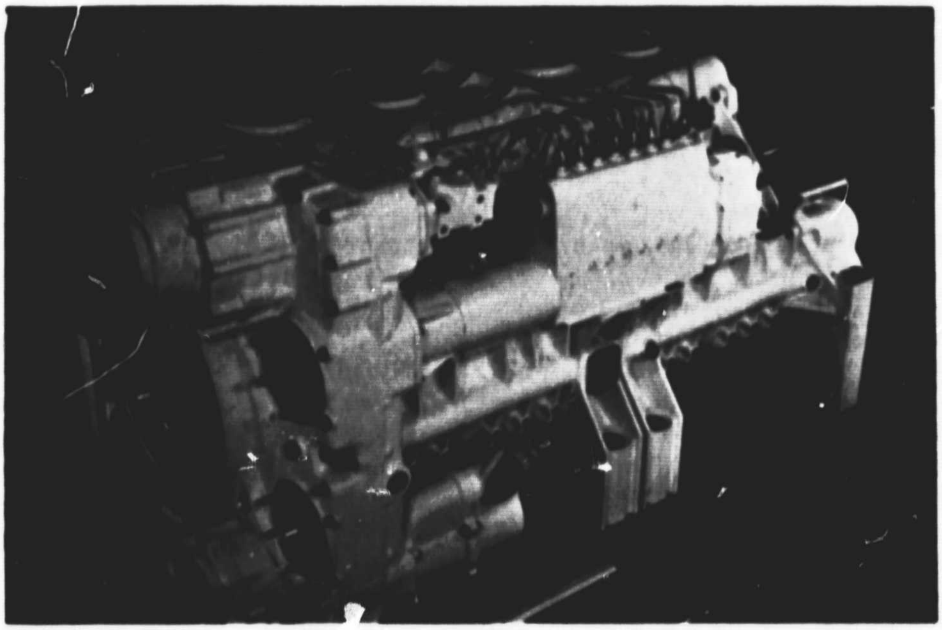
(1) Complete computer modeling of the combustion process and develop a new engine design.

(2) Construct and test an engine based on the NAHBE theory.

The brake specific fuel consumption is expected to be about .34 lb per hp-hr. This value is 25% less than the .45 lb/hp-hr typical of most spark ignition engines and is comparable to that of many current Diesel engines. The improvement of 45% at part load would bring the efficiency of this engine close to that of Diesel engines over a wide range of speeds and loads.

C. STRATIFIED CHARGE ROTARY ENGINE

Curtiss-Wright, under contract to the U.S. Marine Corps, is developing a stratified charge rotary combustion engine for military vehicular applications (Ref. 11-2). The stratified charge combustion process as developed by Curtiss-Wright is designed for operation with a wide variety of fossil fuels, including Diesel fuels No. 1 and No. 2, gasoline, JP-4, and JP-5 while maintaining a fuel consumption level comparable to open chamber Diesel engines.



C-W Stratified Charge RC engine model RC4-350

Figure 11-2. Stratified Charge Rotary Engine
(From Ref. 11-2)

This engine is shown in Figure 11-2. The vital statistics of this engine are:

Number of rotors	4
Displacement	350 $in.^3$ per rotor
Weight (dry)	1860 lb
Length	61 in.
Width	30 in.
Height	29.5 in.
Output power, peak	1500 hp
Output power, locomotive rating in Notch 8	950 hp
Thermal efficiency (estimated)	35%
Brake specific fuel consumption (estimated)	.39

The two outstanding features of this engine are its multifuel capabilities and its high power to weight ratio, about 0.5 hp/lb. By comparison, the General Motors [illegible]A 3300 hp engine weighs 36,300 lb or a little over 10 lb/hp. The General Electric FDL16 engine is even heavier. It weighs 45,000 lb for a 3300 hp output or 13.6 lb/hp. Even if the rotary engine is derated to 900 hp, four of them would produce 3600 hp but would weigh only 7550 lb. It may be possible to have high powered locomotives that do not exceed the axle loading limits while retaining a moderate fuel efficiency.

D. REACTING GAS BRAYTON CYCLE

The third new engine is this reacting-gas turbine. The efficiency of an ideal Brayton cycle engine is not affected by the choice of the working fluid. However, in a real engine, the molecular weight of the working fluid affects both the compressor and turbine efficiencies and, hence, the actual cycle thermal efficiency.

The efficiency of a closed cycle gas turbine engine can be improved, theoretically, if the working fluid changes molecular weight as it goes through the engine. A low molecular weight is preferable in the turbine and a high molecular weight is preferred in the compressor. The working fluid should decompose when it is heated to the turbine inlet temperature and recombine when cooled to the compressor inlet temperature. The change in molecular weight leads to an increase in the net power output and, therefore, in the thermal efficiency relative to a cycle using a constant molecular weight working fluid. The performance of such a closed cycle is heavily dependent on the reaction kinetics of the gas. If the decomposition and recombination reactions are too slow, then the variation in molecular weight between the turbine and the compressor would not be great enough to yield a significant improvement in efficiency (Ref. 11-3).

Nitrogen tetroxide (N_2O_4) is a suitable fluid in such an application. During the decomposition reaction, the nitrogen tetroxide decomposes into two moles of nitrogen dioxide (NO_2). Heat is required during this reaction which will take place over a temperature range of 70° to 340°F. The amount of heat required is 270 Btu/lb of nitrogen tetroxide. The nitrogen dioxide can be further decomposed into nitrogen oxide (NO) and oxygen (O_2). This reaction occurs in the temperature range between 280°F and 1600°F and requires the addition of 530 Btu/lb of NO_2. The decomposition process results in a reduction of the molecular weight of the working fluid from 92 for N_2O_4 to 32 for the mixture of NO and O_2.

A schematic of the engine system is shown at the top of Figure 11-3. Nitrogen tetroxide goes into the compressor. From the compressor, it passes through the recuperator where the first reaction, N_2O_4 to $2NO_2$, takes place. The nitrogen dioxide goes to the main heater where the second reaction, $2NO_2$ to 2NO and O_2, takes place and where the gas mixture is heated to the turbine inlet temperature. From the turbine, it returns to the recuperator where heat is given up and the gas mixture reverts to nitrogen dioxide. The nitrogen dioxide goes into the cooler where the waste heat is rejected and the gas converts back to nitrogen tetroxide at the compressor inlet temperature.

The energy flows for two closed cycle gas turbines using N_2O_4 and helium as the working fluids are shown in Figure 11-3. Each diagram is for the optimum pressure ratio of the working fluid. Both are for a turbine inlet temperature of 1500°F. The big difference between the systems is the amount of energy required to drive the compressor. The N_2O_4 compressor requires only 12.4 units, less than one-fourth of the 57.6 units required for the helium compressor.

Figure 11-4 shows the thermal efficiencies of the two closed cycle gas turbines. The thermal efficiency of the helium engine peaks sharply

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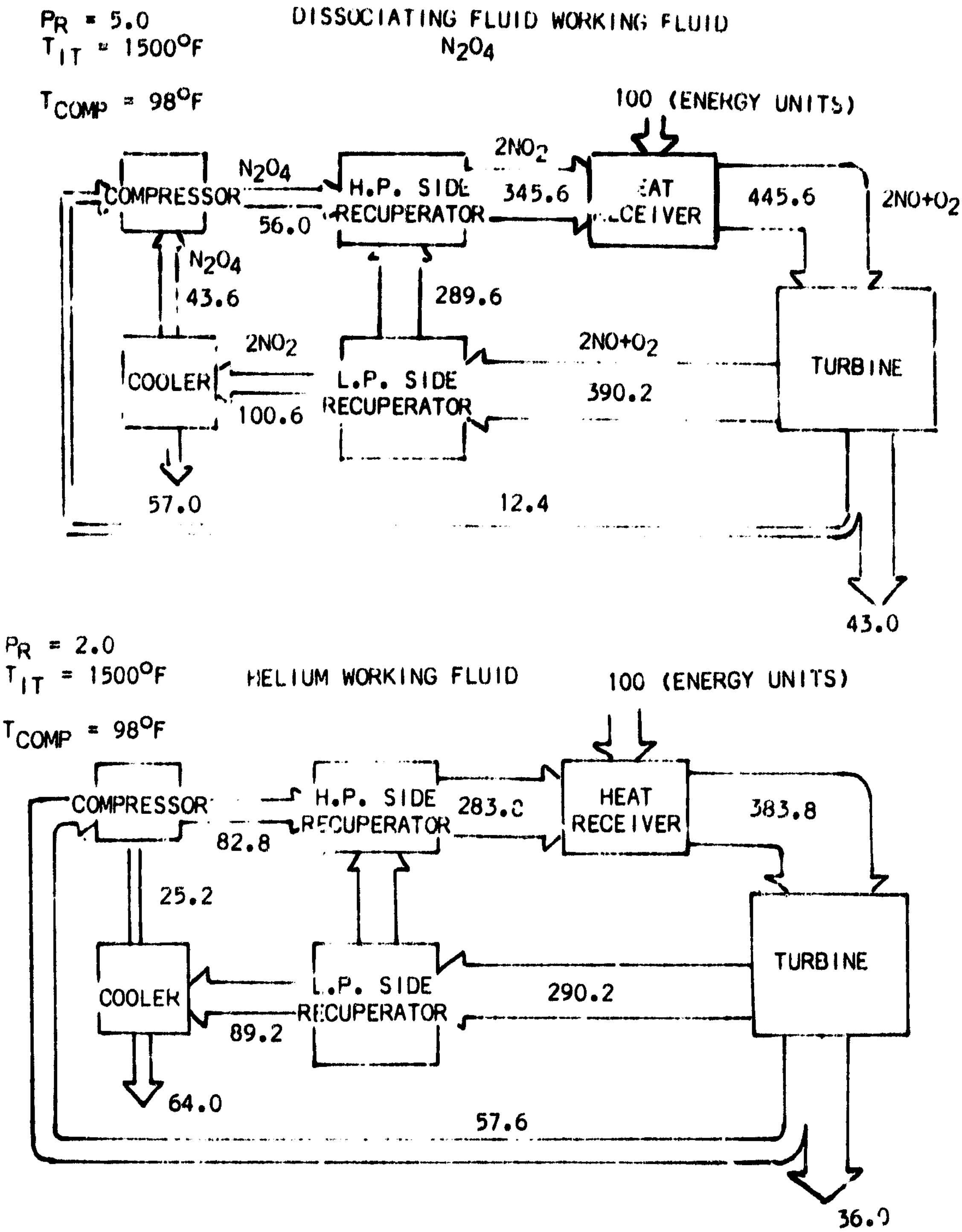

Note: All numbers refer to unspecified units of energy

Figure 11-3. Comparisons of Energy Flow Diagrams
(From Ref. 11-3)

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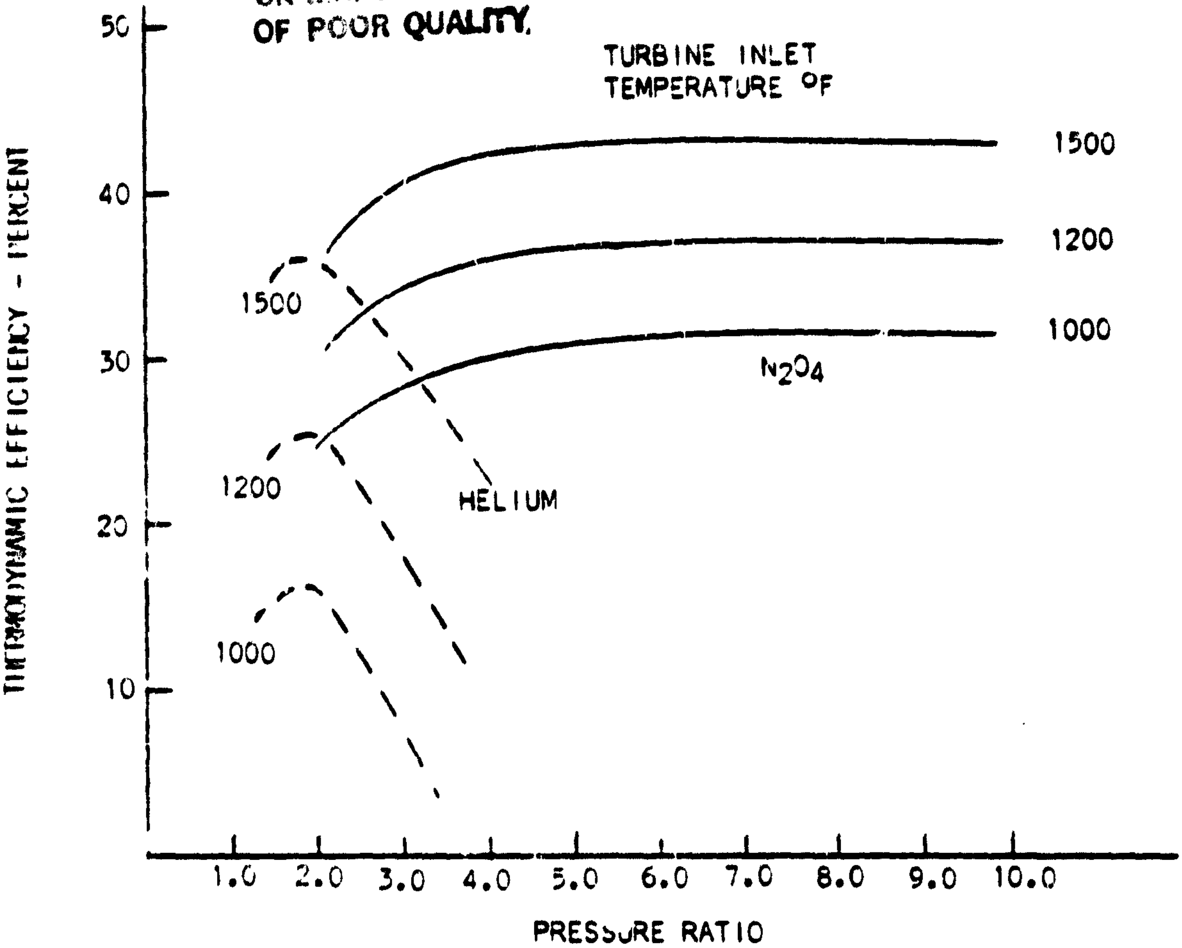


Figure 11-4. Comparison of Thermodynamic Efficiencies for a Closed Brayton Cycle using N_2O_4 and Helium as the Working Fluid (From Ref. 11-3)

at a pressure ratio of 1.9 to 1. The efficiency of the N_2O_4 engine, on the other hand, is relatively independent of pressure ratio at pressure ratios greater than five. At a turbine inlet temperature of 1500°F, the N_2O_4 engine has a peak efficiency of 43% while the helium engine peak efficiency is only 36%. The flat efficiency curve for N_2O_4 indicates that part load efficiency of this engine would probably be much better than that of the helium engine if the design pressure ratio is above seven to one.

There is a large amount of data available on the properties, the handling requirements, and the material compatibility of nitrogen tetroxide. It has been used for many years by the liquid rocket industry as a storable propellant. However, there are two serious problems associated with the use of nitrogen tetroxide. One problem is that it is very toxic and requires special handling. The other is that it is extremely reactive and this material problem is aggravated by the high temperatures encountered in the cycle.

This engine offers a high thermal efficiency at a relatively low turbine inlet temperature. In fact, very good efficiency can be achieved at a temperature low enough that a metal heat exchanger could be used in a fluidized bed combuster and the bed temperature would be at the optimum level for sulfur dioxide control. This engine has a high development and safety risk factor which must be weighed against the efficiency gains that can be achieved.

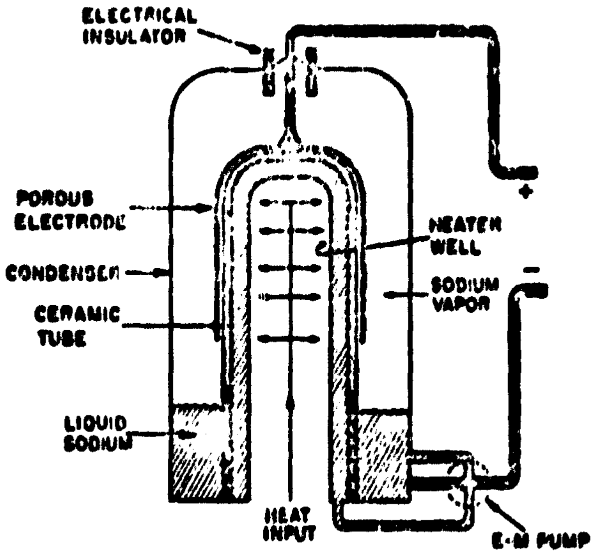
E. THE SODIUM HEAT ENGINE

The sodium heat engine (Ref. 11-4) is a novel device capable of the direct conversion of heat into electrical energy. This engine operates continuously as an ion concentration cell with the required concentration gradient maintained by the input heat source and the output heat sink. Figure 11-5 is a schematic of two versions of the sodium heat engine with the main components identified. It uses an electromagnetic pump which does not have any moving parts. The only moving part in the engine is the sodium itself. In operation, sodium is expanded nearly isothermally through a beta-alumina solid electrolyte with electrical energy being extracted during the process. Theoretical analysis, combined with the measured parameters of operating sodium heat engines, predict an overall thermal-electrical conversion efficiency of 25-30% at 1500° F for single cells. These calculations use component performance levels already achieved and do not assume the use of bottoming cycles to recover exhaust heat. The operating characteristics of the sodium heat engine, especially its high efficiency, make it well suited for the generation of electrical power from chemical fuel in a compact unit.

As shown in Figure 11-5, a closed container is partially filled with the liquid sodium working fluid. The container is physically divided into a high and a low pressure region by the pump and the tubular membrane of the solid electrolyte. There is a porous metal electrode attached to the electrolyte tube. The inner section of the device shown at the top of Figure 11-5 is maintained at a high temperature by the heat source while the outer section is kept at a lower temperature by a heat sink. The temperature difference between the two regions produces a sodium vapor pressure differential across the solid electrolyte membrane.

During operation, sodium travels in a closed loop through the device. Starting in the high temperature region, the heat source raises the incoming liquid sodium to the higher temperature level. Since beta-alumina has high conductivity for sodium ions and negligible electrical conductivity, the same number of electrons must leave the high temperature zone as the number of sodium ions that enters the beta-alumina. The sodium ions migrate through the electrolyte in response to the pressure differential. After passing through the external electrical load, the electrons are recombined with the sodium ions at the porous electrode-electrolyte interface. Neutral sodium then evaporates from the porous electrode, and passes through the vapor space to the condenser. The condensed liquid sodium is returned to the high temperature region by an electromagnetic pump to complete the cycle.

The processes occurring in the solid electrolyte and at its interfaces are nearly equivalent to an isothermal expansion of the working fluid. The thermodynamic analysis of this device can be found in Reference 11-4.



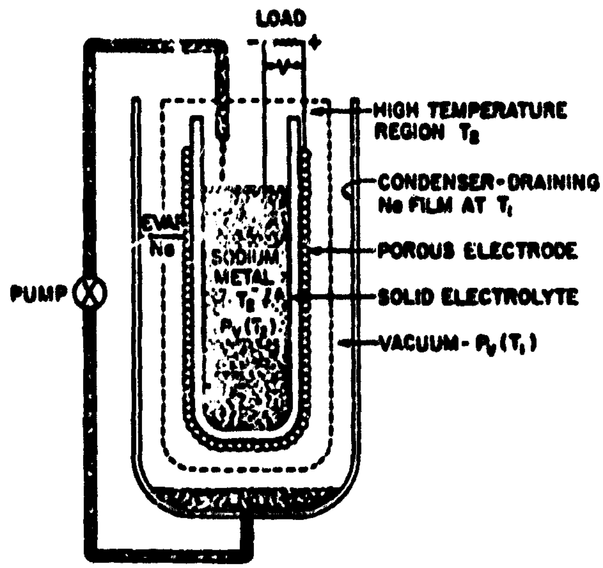
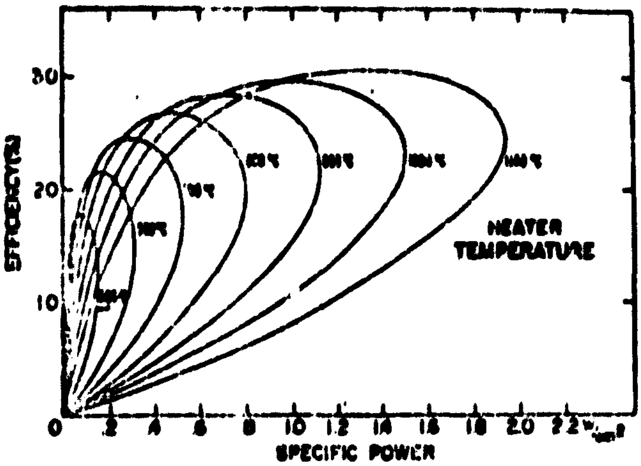


Figure 11-5. Schematics of Two Versions of the Sodium Heat Engine (From Ref. 11-4)



- Calculated power-efficiency performance of SHE vs. T_2. R_0 = 0.18 $\Omega \cdot cm^2$, T_1 = 200°C, Z = 20.

Figure 11-6. Calculated Power-Efficiency Performance (From Ref. 11-4)

It indicates that the sodium heat engine efficiency, neglecting extraneous heat leaks from the hot to cold sides, can be as high as 90% of the Carnot cycle efficiency. Figure 11-6 shows the thermal efficiency of the device taking into account internal resistance, a heat sink temperature of 400° F, electrical lead resistance, radiation, and other losses. This more realistic appraisal indicates that the efficiency is about 60% of the Carnot cycle efficiency at the conditions shown.

The maximum efficiency of the engine is shown in Figure 11-7. The efficiency at maximum power is about 20% lower than the maximum efficiency. Since maximum efficiency is achieved at very low power levels, this device is quite unusual in that the efficiency increases at part load. Nearly all heat engines show either a slight increase or a decrease as the load is reduced.

As expected the efficiency increases as the heater temperature is increased. The thermal efficiency also varies with the heat sink temperature as shown in Figure 11-8. This engine is also unusual in that within a range of temperatures from 200° F to 600° F, the thermal efficiency is nearly independent of the sink temperature. At a sink temperature of 400° F, the thermal efficiency is slightly higher than it is at 200° F. Above 600° F, the efficiency falls off rapidly with an increase in temperature.

The device is far from ready for use in a locomotive. There is much research that is needed but it has considerable potential. The maintenance of such an engine would be small compared to the Diesel engine-alternator combination that it would replace. If the projected efficiency of 35 to 40% for a series connected set of cells is realized, it would be competitive

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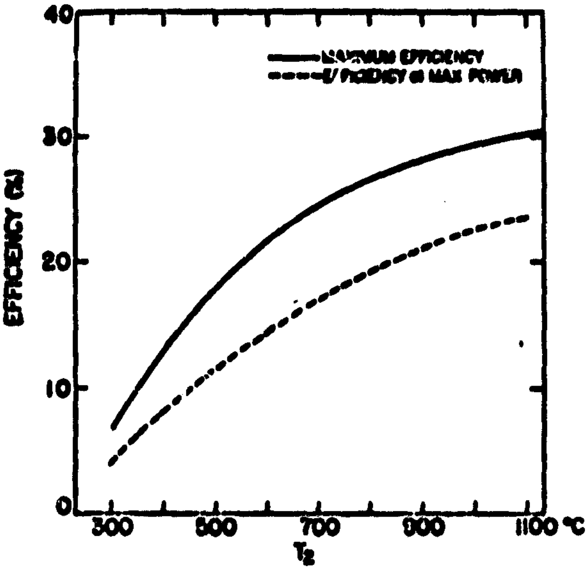
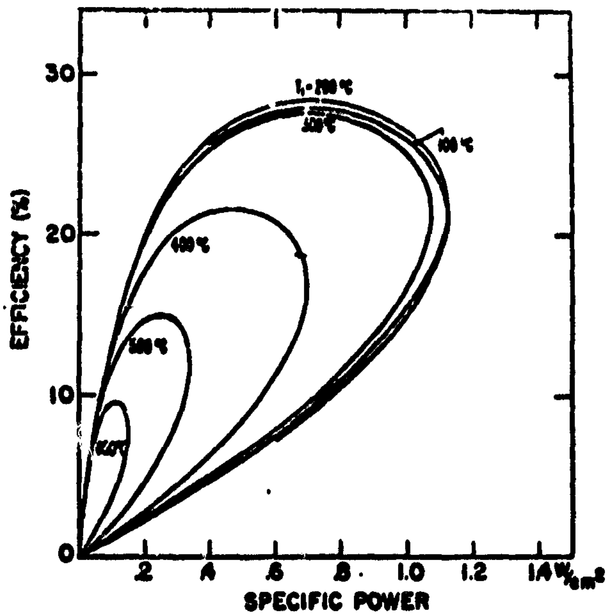


- Efficiency vs. T_2, optimized leads. R_o = 0.18 $\Omega \cdot cm^2$, T_1 = 200°C, Z = 20

Figure 11-7. Efficiency vs. Input Temperature Optimized Leads (From Ref. 11-4)



- Calculated power-efficiency performance of SHE vs. T_1. R_o = 0.18 $\Omega \cdot cm^2$, T_2 = 900°C, Z = 20

Figure 11-8. Calculated Power-Eff ency (From Ref. 11-4) with present Diesel engines and much simpler. If a coal-fired fluidized bed combustion system was used as the heat source, then this engine would be competitive with closed cycle gas turbines operating at much higher temperatures. It is too early to tell if these predicted results can be achieved, but the development of this engine should be followed.

F. ALTERNATIVE TRANSMISSION SYSTEMS

Almost all locomotives in the U.S. have electric transmissions, however, other transmissions could be used. A mechanical transmission similar in nature to the manual transmission found on cars and trucks is one possible type. Hydraulic transmissions are also viable candidates. In fact, hydraulic transmissions have been in limited use for many years on the railroads.

There are two basic types of hydraulic transmissions, hydrokinetic and hydrostatic. The hydrokinetic is characterized by high velocity fluid streams that transmit power. The torque converter used in automobiles is a typical example of a hydrokinetic transmission. Another example, the hydraulic coupling, is the hydraulic analog of a slipping clutch. The hydrostatic transmission uses high pressure and limited fluid flow to transmit power. It consists of a pump and one or more hydraulic motors making it the hydraulic equivalent of the electric transmission. It is used on a wide variety of equipment ranging from garden tractors to large off-highway construction equipment. The hydrostatic transmissions are primarily on low speed equipment while the hydrokinetic transmissions are employed for higher speeds. Both types have been used in locomotives. This chapter examines these alternative transmissions on locomotives as a means of saving fuel. The two main considerations are the efficiency of the transmission itself and the matching of the transmission to the engine. Both can have a strong effect on locomotive fuel efficiency.

G. HYDROKINETIC TRANSMISSIONS

The simplest hydrokinetic transmission is the hydraulic or fluid coupling which consists of two main parts; the impeller and the turbine. The input power goes to the impeller and the output shaft is connected to the turbine. A schematic of a fluid coupling is shown in Figure 11-9. The fluid is accelerated outward by the impeller to point A where it leaves the impeller and enters the turbine. The kinetic energy of the fluid is transferred to the turbine as the fluid slows down and flows inward to point B where it leaves the turbine and re-enters the impeller. Power from the input shaft drives the impeller. The flow through the turbine provides power at the output shaft. A centrifugal force field is set up in both the impeller and the turbine. If they are both running at the same speed, the two fields cancel each other out and there is no fluid flow and no power transmitted. In order to transmit power, there must be a difference in speed between the impeller and the turbine. The "speed ratio" of a coupling is the ratio of the output speed to the input speed and it is normally less than one. The torque across a coupling is constant, independent of the speed ratio. The efficiency of any hydrokinetic transmission is the product of the speed ratio and the torque ratio. The "torque ratio" is the ratio of output torque to input torque. Since the torque ratio is always one, the efficiency of the coupling is the same as the speed ratio. The input power that is absorbed by the impeller is given by this equation:

$$P = C_A N^3 D^5 \tag{11-1}$$

where, P is the power N is the rotational speed of the impeller and D is the diameter of the impeller. The C_A term is a variable which includes the unit conversions and the effect of the coupling design (inlet and exit angles of the impeller and turbine). In some designs, C_A is also a function of the speed ratio. In order to improve the fluid flow between the turbine and impeller at off-design conditions, either one or both may be split into two or more attached parts connected by overrunning clutches.

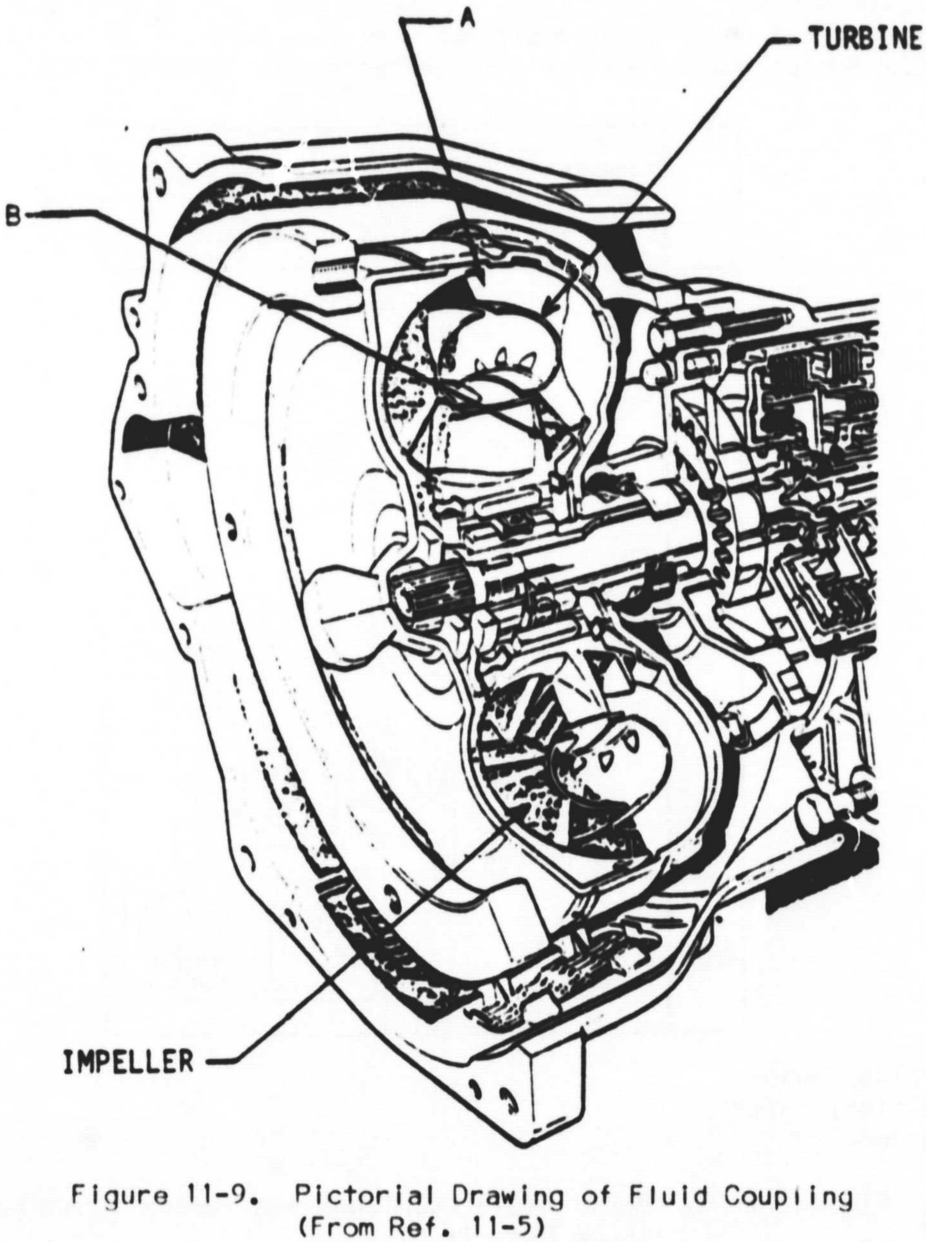
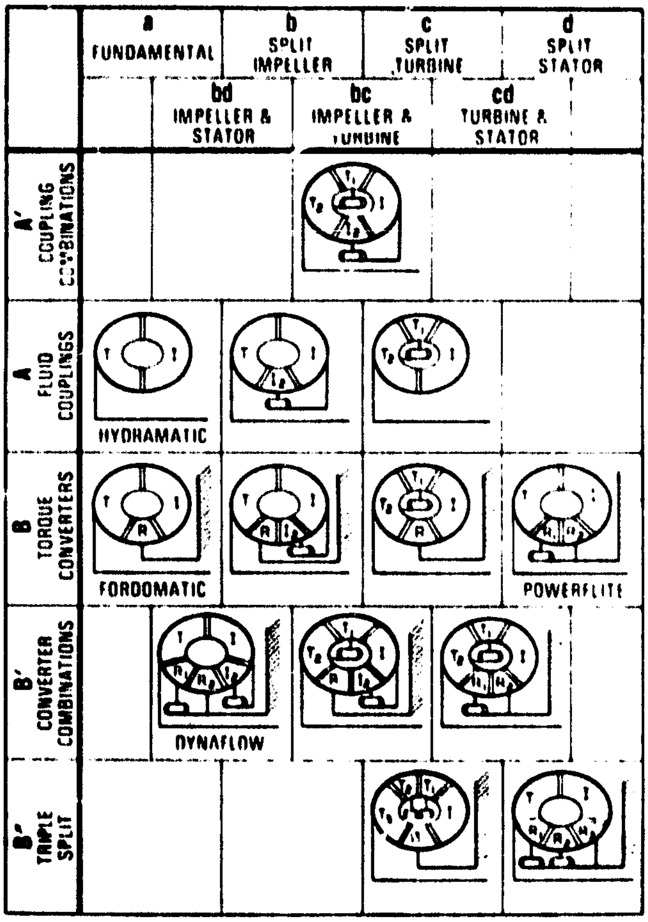


Figure 11-9. Pictorial Drawing of Fluid Coupling
(From Ref. 11-5)

The schematics in rows A and A' of Figure 11-10 show some of the variations in fluid couplings.

A torque converter is similar to a fluid coupling except that a stator or reactor is used in the converter. The torque converter differs in performance from a fluid coupling by having greater output torque than input torque at low speeds. This torque multiplication is largest when the turbine is stalled (output speed equals zero). Depending on the design of the converter, the stall torque ratio can be as high as 5, but in automotive practice, it is normally 2.0 to 2.2. The stall torque ratio is determined by the number of elements (impellers, turbines, and reactors) used and by their relative location in the converter.

The efficiency of a torque converter is the product of its speed ratio and its torque ratio just as in the case of a fluid coupling. Since the torque ratio across the converter is greater than one in the



T - Turbine, output
I - Impeller, input
R - Reactor

Figure 11-10. Basic Fluid Couplings and Torque Converters (From Ref. 11-6)

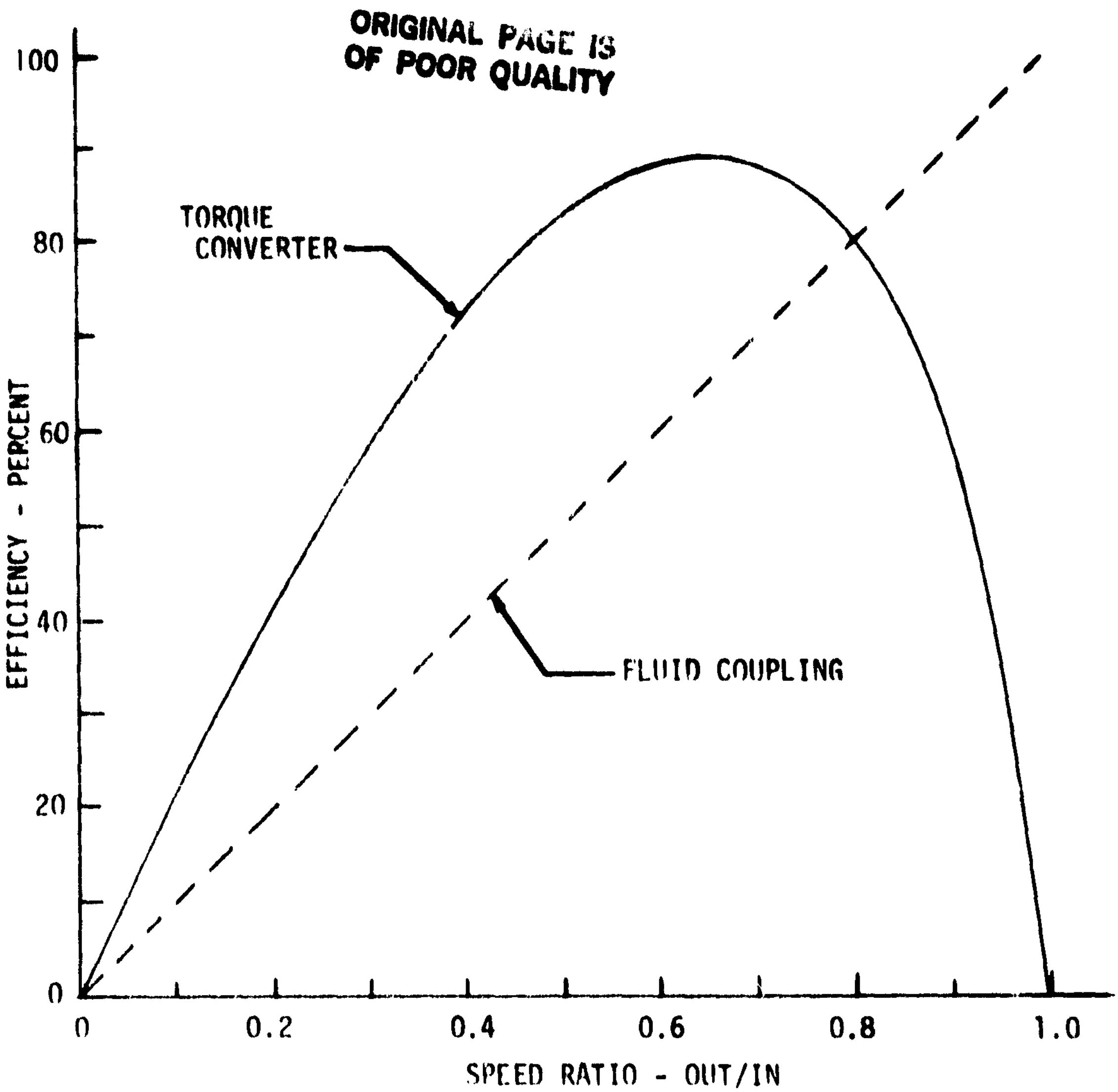


Figure 11-11. Efficiency of Typical Fluid Coupling and Torque Converter

low speed range, its efficiency is higher than that of a fluid coupling. In the high speed range, approaching a speed ratio of one, the torque ratio approaches zero as does the efficiency. Figure 11-11 shows the efficiency of a torque converter and a fluid coupling as a function of the speed ratio. The point where the two curves cross is the coupling speed. At this point, the reactor in a commercial converter is released and allowed to rotate with the turbine, effectively changing the converter into a coupling with a resultant gain in efficiency. In general, the higher the stall torque ratio, the lower the peak efficiency of the converter and the lower the coupling point speed ratio.

The size and design of a torque converter must be carefully matched to the engine and the type of locomotive service if the most efficient system is to result. Equation 11-1 applies to torque converters as well as fluid couplings. The coefficient C_A will differ since it is a function of the design. It is also a function of the speed ratio as shown in Figure 11-12 for a typical automotive torque converter. In the literature on torque converters, a K factor is used which is called the absorption speed factor. It is defined as:

$$K = N/\sqrt{T} \tag{11-2}$$

where, N is the rotation speed of the impeller and T is the input torque. The power capacity factor (C_p) is also used in the literature and is the product of C_A and D^5 in Equation 11-1. The power capacity factor

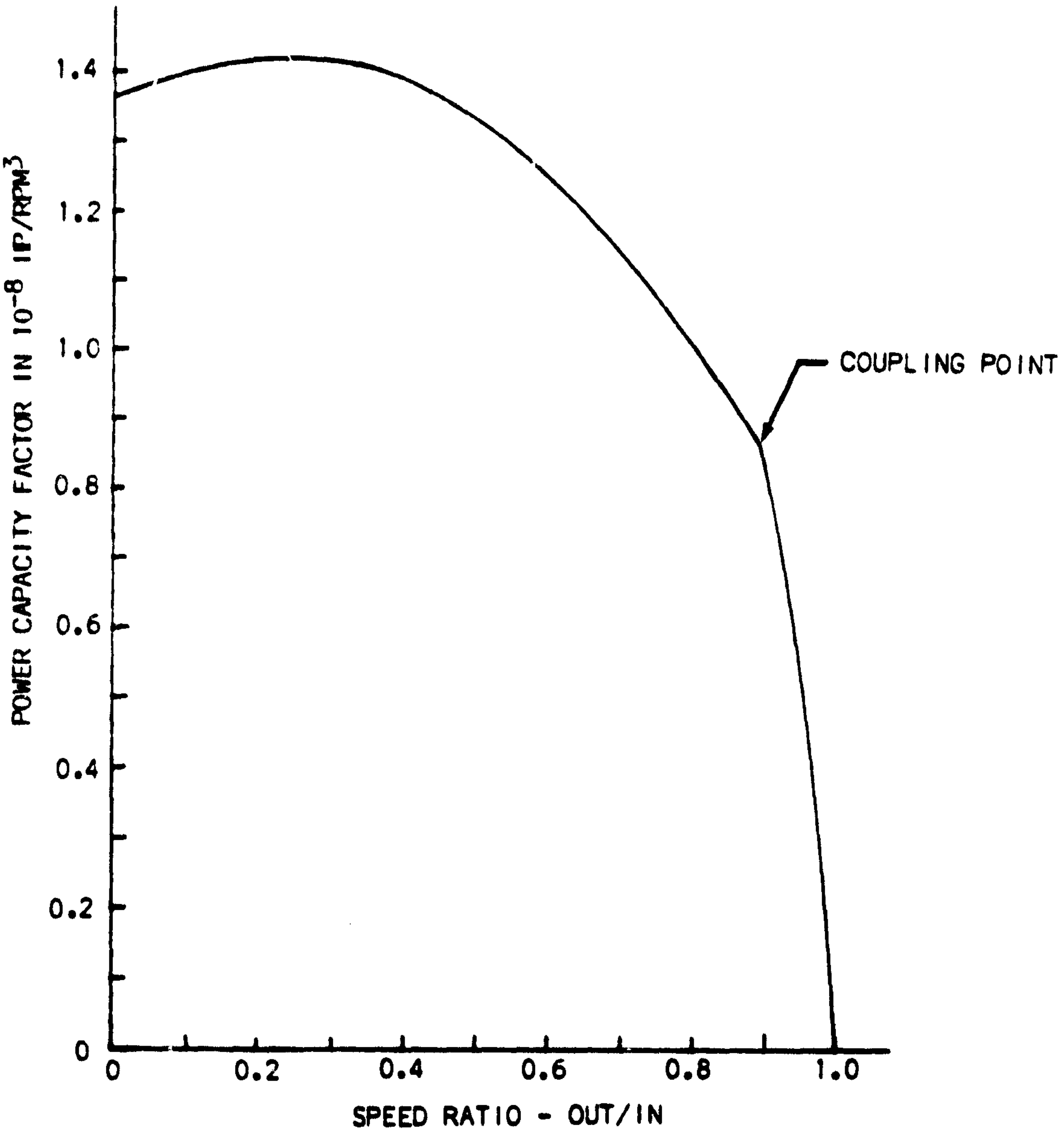


Figure 11-12. Power Capacity Factor

and the absorption speed factor are related through this equation:

$$C_p = m / K^2 \quad (11-3)$$

where m is a constant for unit conversion. C_p is a more convenient unit to use in computation since, at a speed ratio of one, it is zero whereas K is infinite at this value.

There are a wide variety of converter designs, some of which are shown in Figure 11-13. The torque multiplication ratio is rarely sufficient for any practical application. One or more gear sets can be used with the converter to increase the overall torque ratio and to help keep the torque converter operating in an efficient speed range. Figure 11-14 shows a number of combinations of gears and torque converters. The effect of these combinations can be to improve the efficiency and the torque ratio across the entire speed ratio range. Figure 11-15 shows the performance of an output-coupled shift reactor combination. This system acts like several different converters put together. In the low speed ratio range, the efficiency and torque multiplication are high. If this system used only a simple converter, the efficiency would be penalized in the high speed ratio range. The gear-converter combination has high efficiency at both low and high speed ratios.

A torque converter in locomotive service would have to be larger than the automotive converters which are in the 10 to 12 in. range. Depending on the details of the converter, a locomotive converter would need to be 49 in. in diameter to absorb 3160 hp at 1000 rpm and to have an efficiency of 95%. If the efficiency was lowered to 90%, the diameter would be 40 in. for the same power and speed. This range of diameters is compatible with the space available in a locomotive. A gear set will be necessary to match the locomotive speed to the engine speed without operating the torque converter in a low efficiency range. When the locomotive is operating in notch 8, it is desirable to have the converter efficiency around 95%. In this range, the converter is running like a coupling with a speed ratio of 0.95.

There are two conditions which define the limits of operation in notch 8. One is high speed (60 - 70 mph) operation on level track. The other is low speed operation (15 to 20 mph) on ascending grades. Assuming 40 in. wheels, the overall gear ratio from the axle to the converter output is 1.70 for a speed of 70 mph and 5.95 for a speed of 20 mph. If a ratio of 1.70 is used in the differentials, then the gear set would have a range of 1.0 to 3.5. Using an eight speed gear set, the gear ratios would be 1.0, 1.19, 1.43, 1.71, 2.05, 2.44, 2.93, and 3.51. A neutral position would be used for idle.

In Figure 11-16 from Ref. 11-7, a typical torque converter power/speed relationship is shown as a broken line. A typical railway duty curve for a Diesel-electric locomotive is shown as a solid line. Although the two curves are not identical, they are close enough that the performance and fuel consumption would not differ significantly. A torque converter transmission does not provide much dynamic braking. A converter acts like a fluid coupling during the coast and braking operation. As a

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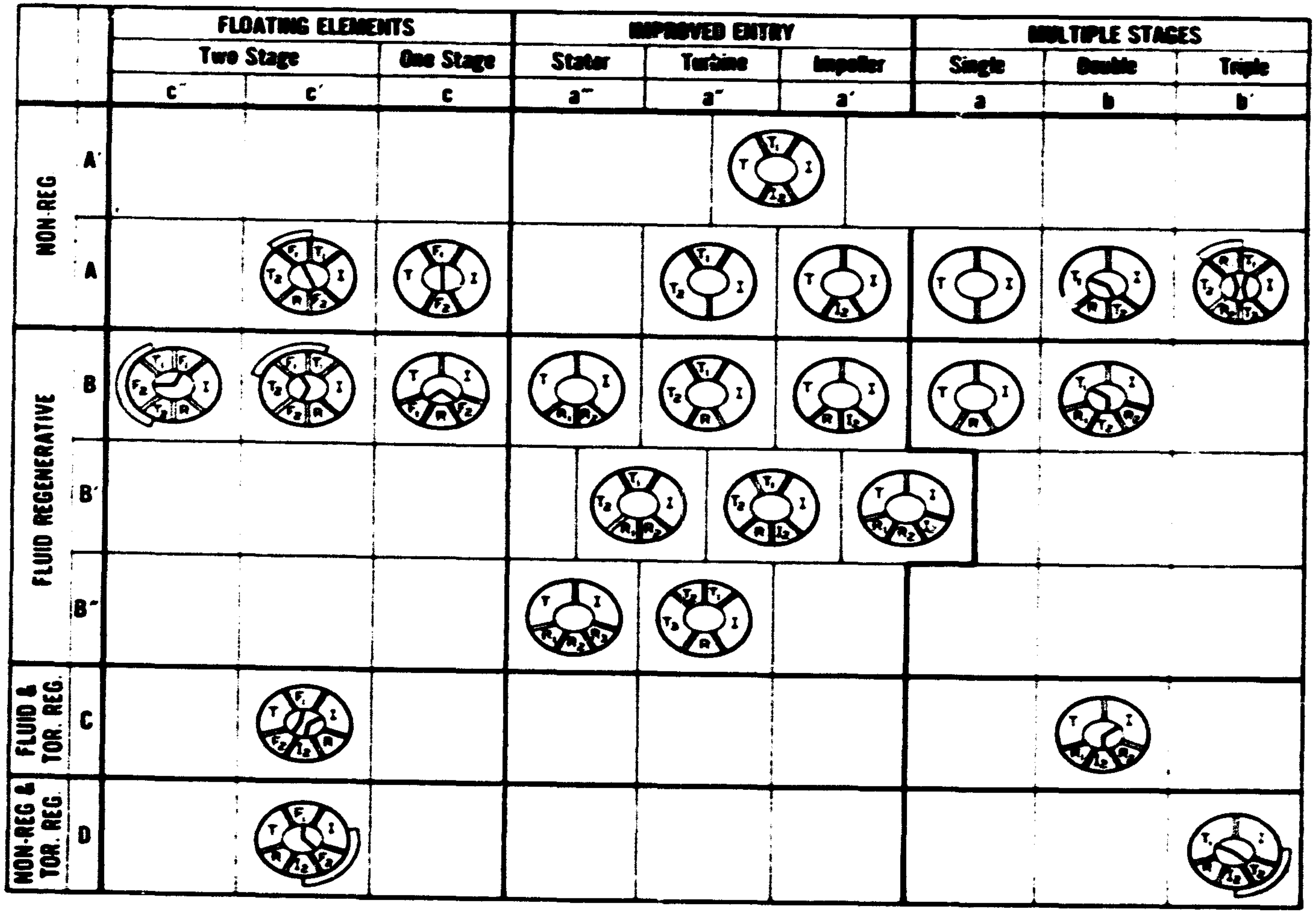

Figure 11-13. Summary of Hydrodynamic Systems
(From Ref. 11-6)

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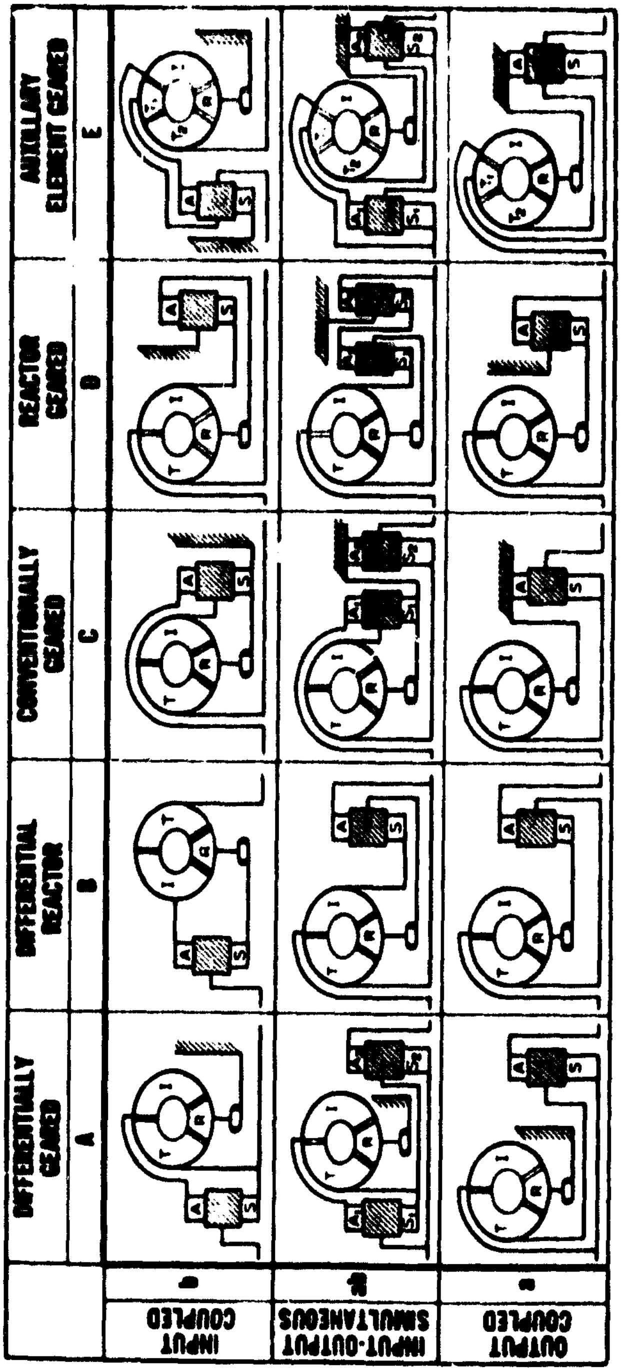

Figure 11-14. Summary of Geared Converter Systems
(From Ref. 11-6)

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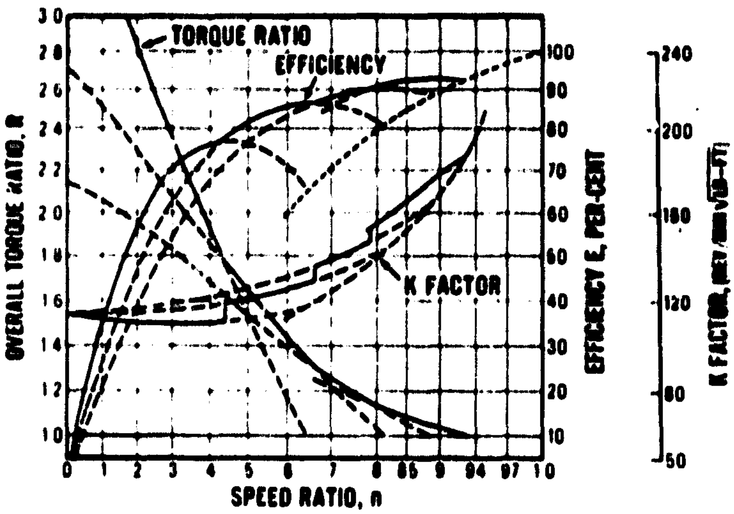


Figure 11-15. Output-Coupled Shifted Reactor
(From Ref. 11-6)

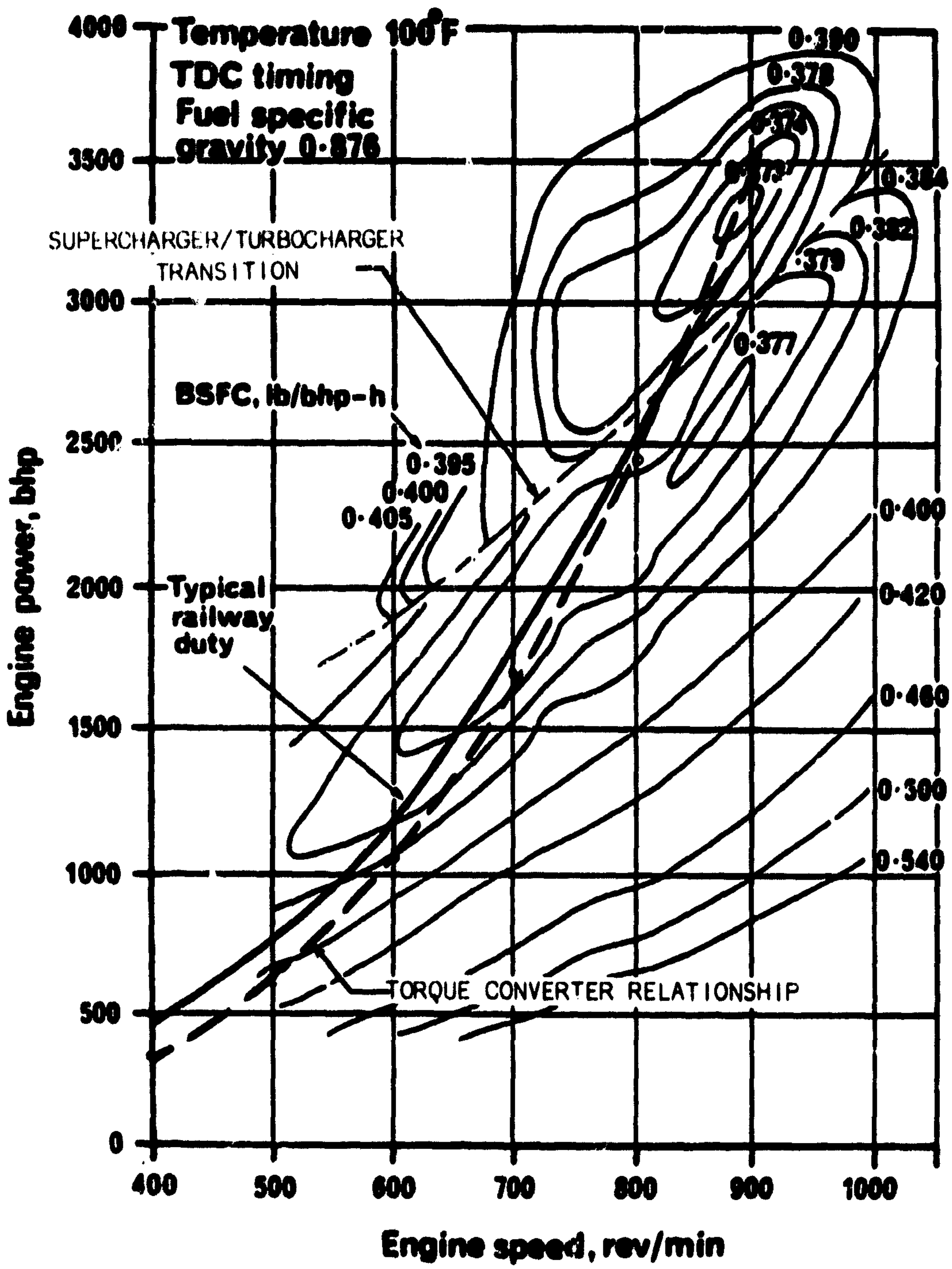
result, there is little power absorbed by the engine. Another means of providing braking power is required with this type of transmission.

It is feasible to design a torque converter that has 95% efficiency during steady speed operation. The additional gearing, drive shafts, and differential would amount to about 5% in loss so that the system efficiency from the torque converter input to the rails is 88 to 91% over a wide range of output loads. This is comparable to the performance of electric transmissions in the same type of service.

The torque converter transmission would need in addition to the converter, an automatic shifting gear set, driveshaft, and differential. The differential is needed to compensate for differences in wheel diameters. If the axles were geared together directly, two axles with different wheel diameters would try to rotate at different speeds. Since they can't, one would be forced to slip on the rail causing rapid wheel and rail wear. A differential allows each axle to rotate independently. The mechanical complexity of the torque converter transmission and its lack of significant braking ability puts it at a disadvantage relative to the electric transmission. The torque converter has little to offer when compared to the electric transmission.

H. HYDROSTATIC TRANSMISSION

The hydrostatic transmission consists of a hydraulic pump, a hydraulic motor, and the connecting high pressure lines. It is, in fact, the hydraulic analog of the electric transmission in use now. The pump, which corresponds to the alternator/rectifier is usually a positive displacement type and may be a vane, lobed, gear or piston pump. For high power level installation, the piston pumps are preferred. The piston pumps are either



Source: Ref. 7.3

Figure 11-16. Diesel Engine Map
(From Ref. 11-7)

a crankshaft type with Inline or Vee cylinder arrangements or a swashplate type with a variable plate angle. In general, the pump is a high pressure, low flow device. Pressures of 3000 psi are common and a few pumps run up to 5000 psi. These high pressures cause serious problems if the pump and motor are any distance apart. High pressure hoses or metal lines are subject to failure especially in the harsh vibrational environment of a locomotive. The motor, like the pump, is usually a positive displacement device. In many cases, both the pump and the motor are the same type of unit. Sometimes, the two units are in the same housing with the lines internally connected.

Theoretically, the hydrostatic transmission can do anything that the electric transmission can do. Unlike the electric transmission, it does not have a minimum continuous speed to limit its use at low speeds. However, it does have hydraulic fluid which has both a viscosity and a vapor pressure. Since the fluid lubricates both the pump and the motor, it must have a reasonably high viscosity. High viscosity, however, tends to reduce efficiency and to increase the heating of the fluid. The vapor pressure limits the speed of the fluid. Excessively high speeds lead to cavitation and damage to the transmission. In general, viscosity and vapor pressure limit the speed of the transmission. For this reason, the use of hydrostatic transmissions is usually limited to low speed applications. Their use in the railroad field has been limited mostly to switch engines which are typically low speed, high traction locomotives. For this type of service, they are well suited. For line haul locomotives, the limitations of the hydostatic transmission prevents it from being a viable competitor to the electric transmission.

I. MECHANICAL TRANSMISSIONS

The manual transmission in an automobile is a typical mechanical transmission. However, the eight, ten, and thirteen speed manual transmissions used on trucks are similar to those that could be used on locomotives. Since this type of transmission is step-wise variable in speed (discrete changes in gear ratio), a slipping element is needed to synchronize the engine and the wheels during shifting and to allow operation at very slow speeds (creeping). In automobiles and trucks, a clutch is used as the slipping element, usually a dry clutch for low power operation and a wet (oil filled) clutch for higher power level.

The main reason for considering a mechanical transmission is its high efficiency (96% to 99%) over the full range of locomotive speeds. However, as discussed in the section on hydrostatic transmissions, driveshafts and differentials are needed to connect the transmission output to the wheels. These components will reduce the overall efficiency by 2 to 5 percentage points. The efficiency, measured from the engine output to the rail, would be about 94% ± 3% over the locomotive speed range. The gearset might have either eight or ten ratios and is likely to have semi-automatic shifting. A manually shifted transmission is not practical in this large size, and the widely varying loads due to different consist weights would make the shift control on a fully automatic transmission extremely complex. In a semi-automatic transmission, the engineer selects the gear ratio desired and the transmission makes the actual shift change. The engineer will have eight notch positions and eight to ten gear ratios to select from in his handling of the locomotive. The selection of a notch position-gear ratio set depends heavily on the experience of the engineer. The energy savings which results from the use of a mechanical transmission is about 3% at high locomotive speeds to about 10% at speeds near 15 mph. This gain could easily be negated by the improper selection of the notch position-gear ratio set.

J. SUMMARY

In general, all of these alternative transmissions are technically feasible. However, none of them display a significant advantage over the electric transmission. Even the 3% to 10% efficiency gain of the mechanical transmission is not enough to make it worthwhile to spend the necessary money to develop it.

K. SECTION XI REFERENCES AND NOTES

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SECTION XII
REGENERATIVE ENERGY STORAGE SYSTEMS

A. INTRODUCTION

In recent years the use of dynamic braking has become standard practice in American rail freight operations. In fact it now is virtually mandatory that the locomotive drive system be able to absorb energy thereby reducing the heat energy input to the wheels both for long descending grades and for decelerating train operations. The electric motor, either dc or polyphase ac, has this regenerative capability, and the fact that the energy is returned in electrical form facilitates its handling and ultimate disposition. Table 12-1 lists the magnitudes of the energy and power involved. The data used in this table are the rolling resistance values of 5.7 lb/ton at 40 mph and 16 lb/ton at 70 mph. These data are for 70 ton cars and are compatible with Davis data at 40 mph. The energy penalty for high speed operation is apparent in Table 12-1. It requires almost three times the energy to go a given distance at 70 mph than it does at 40 mph. Selection of train speeds, however, is a route and traffic specific decision and is not considered further. This energy is irreversibly dissipated on a continuous basis, and is simply the price of running trains or any mode of transportation at the selected speed.

The ratio of the kinetic energy to the dissipated energy per mile is the distance the train could be moved at the given speed using only its kinetic energy. As it turns out, this distance is relatively independent of speed and is 3.6 to 3.9 mi, throughout the speed range of 40 to 70 mph and, more important, it is quite small when compared with the distance a train moves between stops in cross-country or almost any intercity service.

Grades can be a different matter, particularly if they are steep, e.g., 1.5% or greater, and long, e.g. 10 mi or more. Here, a considerable amount of energy is involved. The amount is determined only by the difference in elevation between the top and bottom of the grade and the mass of the train. There are usual train resistances which must be overcome as the train descends such as aerodynamic, flange, bearing, etc., but if the grade is in excess of 0.3% (15.84 ft/mi), there may be, depending on speed, excess energy which must be dissipated. The amount of this excess energy can be estimated from the following equation:

$$E = 0.0517\ T_t\ G_p\ D \quad \text{hp-hr}$$

where E is the total energy in horsepower-hours, T_t is the total weight in tons of the train (locomotive and consist), G_p is the percent grade and D is the distance along the track in miles. As an example, assume that a unit coal train of 68 100-ton cars, drawn by 3 GP40 locomotives is descending a one percent grade that is 7 mi long. The total weight of the train is 9346 tons and the total energy is 3382 hp-hr. If the train is going to maintain a speed of 20 mph, energy must be dissipated at a rate of 9664 hp. At 30 mph, it is 14,496 hp. The requirements of the train to dissipate energy on a down grade may very easily be greater than its ability to generate it in the locomotives on up grades.

Table 12-1. Energy and Power for a Typical Train

	Speed 40 mph	Speed 70 mph	Grade 1%
Resistance lb/ton	5.7	16.0	20.0
Power hp/ton	.61	3.0	---
Dissipated Energy kWh/mi/ton	.011	.032	.040
Kinetic Energy kWh/ton	.040	.125	---
Kinetic/Dissipated Energy mi	3.6	3.9	---

There are two ways for the train to dissipate the excess power on a downgrade. One is the train brakes, operated by air and acting on the wheels of each car. The other is the dynamic braking system in the locomotive. In this system, the dc motors act as generators and absorb power from the wheels to create electricity. The electricity is fed to resistor grids where the power is dissipated as heat. The grid capacity limits the amount of power which can be handled and the maximum speed of the train. In the earlier example, grids which could handle 10,000 hp would be satisfactory at 20 mph but not at 30 mph which would result in a 45% overload.

In notch 8 operations, 3382 hp-hr of energy represents about 1200 gal of fuel. Just as this amount of energy must be dissipated in descending the hill, this amount and more is needed for a similar train to climb the hill. Ideally, the 1200 gal of fuel could be saved if the energy from a descending train could be transferred to an ascending one. The real world conditions make it very unlikely that two trains of equal mass would be on the hill at the same time going in opposite directions. However, the idea of saving the energy from the descending train is very attractive from a fuel or energy conservation standpoint. Obviously, the energy must be stored for reuse later. In storing it, there are three things to be considered: (1) where is the energy going to be stored?, (2) how is it going to be stored?, and (3) how should the stored energy be used?

B. ENERGY STORAGE

The energy can be stored either on or off the train. If it is stored on the train, the energy density is important and limits the "how to store it" question to such devices as batteries, flywheels, hydraulic accumulators and pneumatic systems. If it is stored off the train, these devices could also be used as well as pumped storage and electric utility lines. In this last method, the electricity produced by the descending train is put into the utility grid and power is delivered by the grid to the ascending trains. How the power is to be used is the third question. Using it to power trains climbing the hill has been discussed as one option. Another is to use the power to propel the same train along the track after it has reached the bottom of the hill. As an example, using the train described earlier, the energy derived from its 7 mi descent could drive the train about 20 mi along a level track. This type of operation may be more energy efficient than storing it for use in helping trains ascend the grade. This would be the case if, for example, the number or weight of the trains in one direction was significantly different than in the reverse direction or if the number of trains were few and the internal losses in the energy storage device were high.

Energy storage on the train is not practical from a weight standpoint. Batteries have an energy density of 8 to 15 Whr/lb. The train used earlier as an example would require 168,000 lb of batteries at the 15 Whr/lb rating to absorb all of the energy. This is assuming the batteries were totally discharged at the start. Normally, a 50% discharge level is about as low as is feasible to go and still get a reasonable life out of the battery. This would double the weight to 336,000 lb. Two large cars would be required just to carry the batteries. This is only on an energy basis. Batteries are not very good on a power density basis. Typically, they run 10 to 30 W/lb. At 20 mph, a train develops 9664 hp or 7200 kW. At 10 W/lb, the batteries would weigh 719,968 lb. Ten watts per pound is a reasonably high power density for the charging of batteries. They will deliver power at the 30 W/lb rating. Even if advanced batteries double these ratings, and none appear to have that potential, they would still require several cars just to carry them. The power density of flywheels is ten times as great as for batteries but the energy density is about half to three-quarters as good. Even under the best conditions the size of the flywheel will be in the 300,000 to 400,000 lb class. Hydraulic and pneumatic systems are even less favorable.

There is another means of utilizing the energy derived from dynamic braking. This approach is to use the electrical power to dissociate water in an electrolytic cell and to store the hydrogen in a metal hydride or as a compressed gas. The hydrogen could be used as a fuel in the locomotive in a fuel cell or sold to industry as a feedstock for a variety of uses. Going back to the earlier example of a train descending the hill, the 3382 hp-hr of energy could be converted into 150 lb of hydrogen assuming about 90% electrolytic cell efficiency. One hundred and fifty lb of hydrogen could be stored in 1200 lb of lithium hydride, 2000 lb of magnesium hydride, or 8000 lb of iron-titanium hydride (Ref. 12-2). The weight of the container and the required heat exchanger is not included but can be estimated to weigh at least as much as the hydride itself. This energy storage system consisting of a water supply, electrolytic cell, metal hydride storage bed, and heat exchanger could not be installed in a locomotive but would have to be housed in a separate car. It must be concluded that on-board train energy storage is not feasible today or in the foreseeable future.

The energy storage system must, therefore, be located away from the train and the power transferred from the train to the storage site. This type of system has been studied by Lawson and Cook at Garrett AiResearch (Ref. 12-3) in considerable detail. Some of the alternative energy storage concepts investigated are shown in Table 12-2. The battery costs per kWhr

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Storage Concept	Round Trip Efficiency	Installed $/kwhr	Cycle Life	Service Life
Battery	60	70	1000	2 Months
Pumped hydroelectric	76	1000	10^6	30 Years
Regeneration to utility	92	120*	10^6	30 Years
Compressed air	37	N/A	10^6	30 Years
Flywheel	91.2	270	10^6	30 Years

*Site-dependent

Table 12-2. Alternative Storage Concepts
(From Ref. 12-3)

Total Capacity	7.33 Mwhr
Usable Capacity (2 to 1 speed range)	5.5 Mwhr
Maximum Speed	2037 rpm
Diameter	13.5 ft
Length	17.28 ft
Weight	6[illegible]4.4 tons
Peripheral speed	1440 ft/sec
Vacuum requirements	10 torr
Loss at 100 percent speed	136 hp
Loss at 50 percent speed	48 hp
Moment of inertia	855,800 lb-ft-sec^2
Spin-down time (100 to 50 percent speed)	87 hr

Table 12-3. Characteristics of Optimum Steel Flywheel
(From Ref. 12-3)

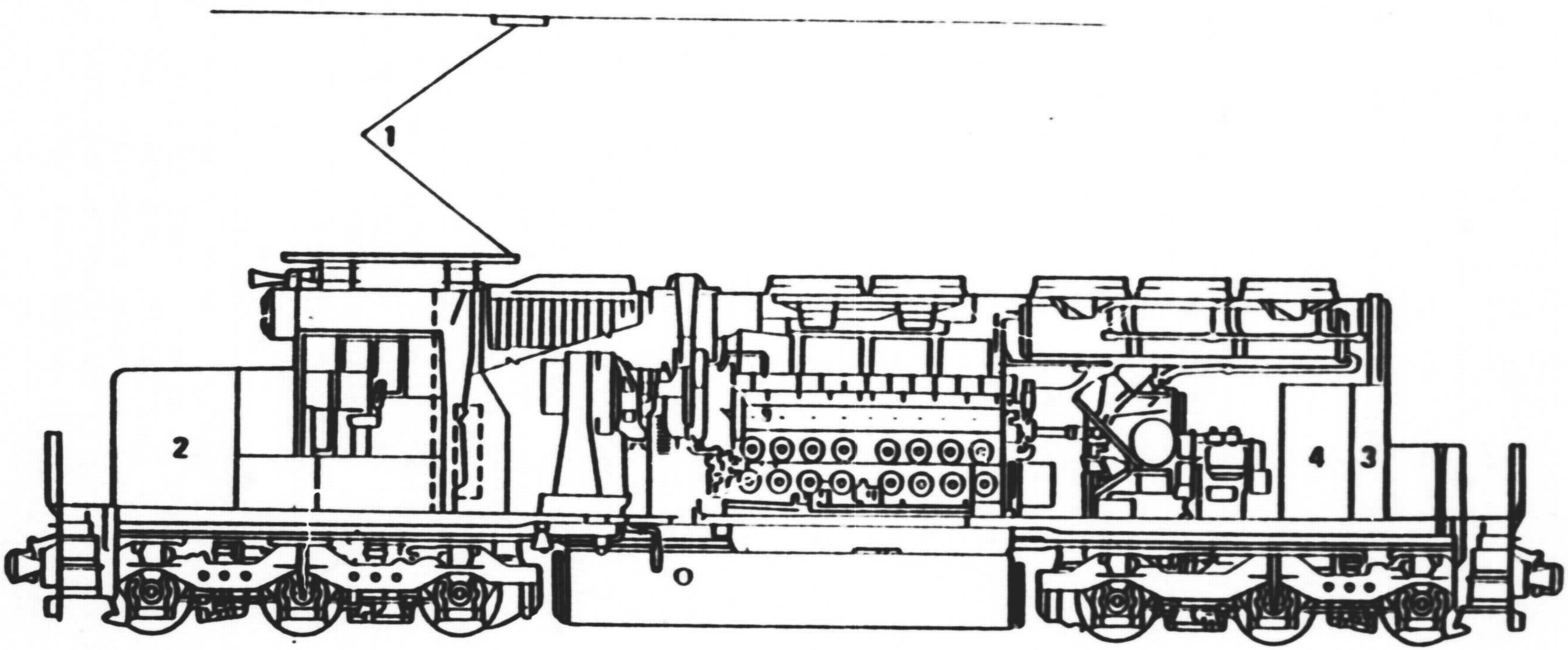
are low but their lives are short. Their life-cycle costs over a 30 year period are excessive. This study concludes that the wayside energy storage system using a flywheel as the storage medium is the most attractive approach. Details of their optimum steel flywheel are shown in Table 12-3. The energy density for this wheel is 6.11 Whr per pound which is about half that of present batteries but better than the other storage systems. They also conclude that supplemental power would be needed when stored energy is not adequate and that a tie-in to an electrical utility is preferable to the on-board Diesel engine or a wayside generating unit. Details of the system can be found in the report.

A wayside energy storage system consisting of a water supply, electrolytic cell, metal hydride storage, and a fuel cell, together with the required electrical controls would be better than the flywheel at some sites. Flywheels suffer from a steady dissipation of energy due to windage, bearings, and seals. In areas where the traffic is irregular either in schedule or train weight, the hydrogen energy storage system can be more efficient.

One of the key features of the wayside energy storage system is the development of the dual-mode locomotive. A locomotive of this type, derived from an SD40 unit, is shown in Figure 12-1. These locomotives are standard Diesel-electric units fitted with a pantograph, transformer, and a thyristor converter which will permit it to operate as an electric locomotive on electrified track or as a Diesel on nonelectrified track. An important consideration is that it allows an evolutionary change-over to electrification rather than an abrupt one.

Diesel locomotives are typically engine power limited in the middle speed range (15 to 60 mph) and the traction motors are underutilized. In this speed range, the motors are capable of producing up to five times as much power as the Diesel engine on a continuous basis. On a steep hill, the speed of a train may be only 15 mph because of the Diesel power limitation. Electrification, either from wayside energy storage or from a utility, would make it possible to use the full capacity of the traction motors and the train could climb the hill at 30 mph. On grades where helper engines are normally used, dual-mode locomotives and electrification can reduce or eliminate the need for the helpers with a resultant saving of both fuel and labor. The elimination of helpers frees locomotives for other uses, reduces transit time and reduces labor costs. The fuel savings amount to the total fuel that would have been consumed by both the line locomotives and the helpers. While it is a small fraction of the total fuel consumed by all the railroads, it may be a substantial amount for the individual railroad.

The dual-mode locomotive has advantages not related to energy storage. This type of locomotive provides a partial solution to the high cost of electrification. It allows the costs to be spread over a longer period of time and to electrify those routes or sections of routes where the benefits are greatest. The flexibility of the dual-mode locomotive is a decided advantage. It can provide continuity of operations during power outages and reduce the dependence of the railroads on petroleum fuels. Its development is encouraged.



1 PANTOGRAPH
2 TRANSFORMER
3 CONVERTER
4 CHOKE

Figure 12-1. Dual-Mode Locomotive

Up to this point, only the effect of the grades has been considered. Since only a very small portion of any cross-country or even inter-city route has a significant grade, it is necessary to put the potential fuel savings into perspective. A specific example is presented as an illustration. Consider a 200 mi operating district with 160 mi of relatively flat grades and 20 mi each of uphill and downhill with an average grade of 1.5%. The 20 mi uphill requires a potential energy input of 1.2 kWh/ton. At 40 mph, which would be a very good uphill speed, the dissipated energy for the 20 mi is .22 kWhr/ton. This dissipated energy is small compared with the potential energy and the potential energy is independent of speed. The downhill energy may be considered to be zero, the excess energy typically is dissipated in resistor grids but could be recovered for other use. For the level 160 mi, the dissipated energy is 1.76 kWhr/ton at 40 mph or 5.12 kWhr/ton at 70 mph. Comparing the recoverable 1.2 kWhr/ton potential energy of the hill with the 1.98 kWhr/ton dissipated energy at 40 mph and the 5.34 kWhr/ton dissipated energy at 70 mph it is readily seen that at 40 mph, the potential energy and the dissipated energy are comparable while at 70 mph the recoverable potential energy is only a fraction of the irreversibly dissipated energy. These energies are measured at the wheel-rail interface and do not include efficiencies associated with the transfer, storage or re-use of the energy.

Undoubtedly, the wayside energy storage system can save a considerable amount of fuel at each grade where it is installed. However, the number of grades which are suitable for its use are very limited. Lawson and Cook identified the thirty-four prime candidate sites shown in Table 12-4 and estimated that another 40 to 50 potential sites exist in the U.S. The fuel savings are projected to be about 56 million gal per year on the three routes analyzed and it may be estimated that about twice this amount may be saved if all of the potential sites are utilized. The total savings is of the order of 110 million gal per year or about 2.5% of the total fuel consumption of the railroads. As a means of reducing the total fuel consumption it is, by itself, not a very effective method. It can represent, however, a first step to system electrification of the railroads and electrification will result in a substantial reduction in petroleum fuel usage.

C. SECTION XII REFERENCES AND NOTES

12-1. Poole, Ernest C., "Costs, A Tool for Railroad Management", Simmons-Boardman Publishing Co., New York, 1962.

12-2. Rohy, D. A. and Nachman, T. F., "Development of Lightweight Hydrides", Solar Turbines International presented at DOE Contractor Coordination Meeting, Detroit, Oct. 1979.

12-3. Lawson, L. J. and Cook, L.M., "Wayside Energy Storage Study", AiResearch Manufacturing Co. of California prepared for Federal Railroad Administration, Department of Transportation, Feb. 1979.

Table 12-4. Wayside Energy Storage Prime Candidate Grades (From Ref. 12-2)

Grade Index No.	Railroad	Identification	Ranking Factor
035	Union Pacific	Baker - Weatherby	32.4
036	Union Pacific	Union Junction - Powder River	69.7
037	Union Pacific	La Grande - Duncan	32.88
056	Union Pacific	Cheyenne - Laramie	112.75
061	Union Pacific	Echo - Wahsatch	57.5
063	Union Pacific	Orr - Milepost 40	43.8
088	Union Pacific	Elgin - Crestline	27.8
089	Union Pacific	Borax - Las Vegas	27.12
090	Union Pacific	Kelso - Nipton	27.46
121	Southern	Braswell Mountain	37.4
145	DM&IR	Duluth	33.8
175	Southern Pacific	Cascades (South)	41.2
176	Southern Pacific	Cascades (North)	39.7
183	Southern Pacific	Sierras (Roseville - Sparks)	41.6
195	Southern Pacific	Colton - Indio	58.0
202	Denver & Rio Grande Western	Helper - Springville	34.1
206	Denver & Rio Grande Western	Denver - Granby	30.6
220	Burlington Northern	Wenatchie - Skykomish	*
222	Burlington Northern	Easton - Auburn	*
226	Burlington Northern	Garrison - Missoula	*
227	Burlington Northern	De Smet - Dixon	*
230	Consolidated Rail Corp.	Harrisburg - Pittsburgh	10.8
240	Atchison Topeka & Santa Fe	San Bernardino - Victorville	69.5
242	Atchison Topeka & Santa Fe	Needles - Goffs	50.7
243	Atchison Topeka & Santa Fe	Flagstaff - Canyon Diablo	50.3
244	Atchison Topeka & Santa Fe	Bellemont - Flagstaff	50.2
246	Atchison Topeka & Santa Fe	Eagle Nest - Williams Junction	55.8
247	Atchison Topeka & Santa Fe	Hackberry - Pica	56.1
248	Atchison Topeka & Santa Fe	Topock - Kingman	56.3
251	Atchison Topeka & Santa Fe	Gallup - Belen	54.1
252	Atchison Topeka & Santa Fe	Belen - Silio	29.1
255	Atchison Topeka & Santa Fe	Vaughn - Fort Sumner	56.9
261(a)	Black Mesa & Lake Powell	Page - Milepost 31	†
(b)	Black Mesa & Lake Powell	Milepost 44 - Kayenta	

*Traffic data not available

†The ranking technique cannot be applied to BM & LP since the railroad is electrified and this distorts the rankings. This is because the simplistic approach adopted is only valid when comparing similar (in this case diesel) railroads.

SECTION XIII
LOCOMOTIVE OPERATIONS

A. INTRODUCTION

Locomotive operations refers to the manner in which locomotives are purchased, dispatched, and operated on the tracks. The areas which affect operations are adhesion, traction motor characteristics, engine power, number of axles, slip control, and more. These items influence affect the fuel consumption of the locomotive fleet in a number of ways. They include the operating speed for best fuel economy, the number of helpers needed, the weight of the locomotives used, and the type of motor (ac or dc) used on the locomotive.

The fuel savings which result from changes in operational procedures are alluded to in this work but no quantitative results are included. The actual fuel savings are site sensitive and are also heavily influenced by company policies, speed limits, and the engineer in the cab.

Three aspects of operations will be discussed here. The first section of this chapter is devoted to a discussion of traction, power, and adhesion. This section is the introduction to the factors influencing the wheel to rail interface and describes how they can affect fuel usage and locomotive operations. The second section is a brief discussion of ac versus dc traction motors and their influence on locomotive dispatching. The final section is an analysis of locomotive operations over a specific section of typical western track. It looks at site specific factors, alternative operational procedures, and some possible new equipment which could be useful in specific cases. It does not claim to solve all concerns but it points out the kinds of problems that the railroads deal with in their day-to-day operations.

B. TRACTION, POWER, AND ADHESION CONSIDERATIONS

When comparing different types of motive power for locomotive application, it can be beneficial to go back to the fundamental relationships between adhesion, weight, and power. The basic constraints, limits, and trade-offs can then be evaluated. It is assumed in this discussion that the locomotive axle loading is 35 tons since this appears to be the current American standard for heavy main line freight operation. The discussion and comments, however, can be applied to other axle loadings as well. The discussion is limited to the use of electric motors for traction power. This would appear to be trivial since all existing road locomotives use an electric transmission. However, there are several alternatives that can be used in the future such as the reciprocating steam drive and the hydraulic transmission. The operational aspects of these transmissions have not been investigated. It is further assumed that dynamic braking is required. For any type of drive system, it is desirable that the braking energy be dissipated somewhere other than at the wheel rim.

Figure 13-1 is a tractive effort - speed curve based on a single axle with a 35 ton load. This figure is designed to facilitate the comparison of different sizes of equipment, changes in gear ratios, different power levels, and other relationships. All parameters are compared to the rail

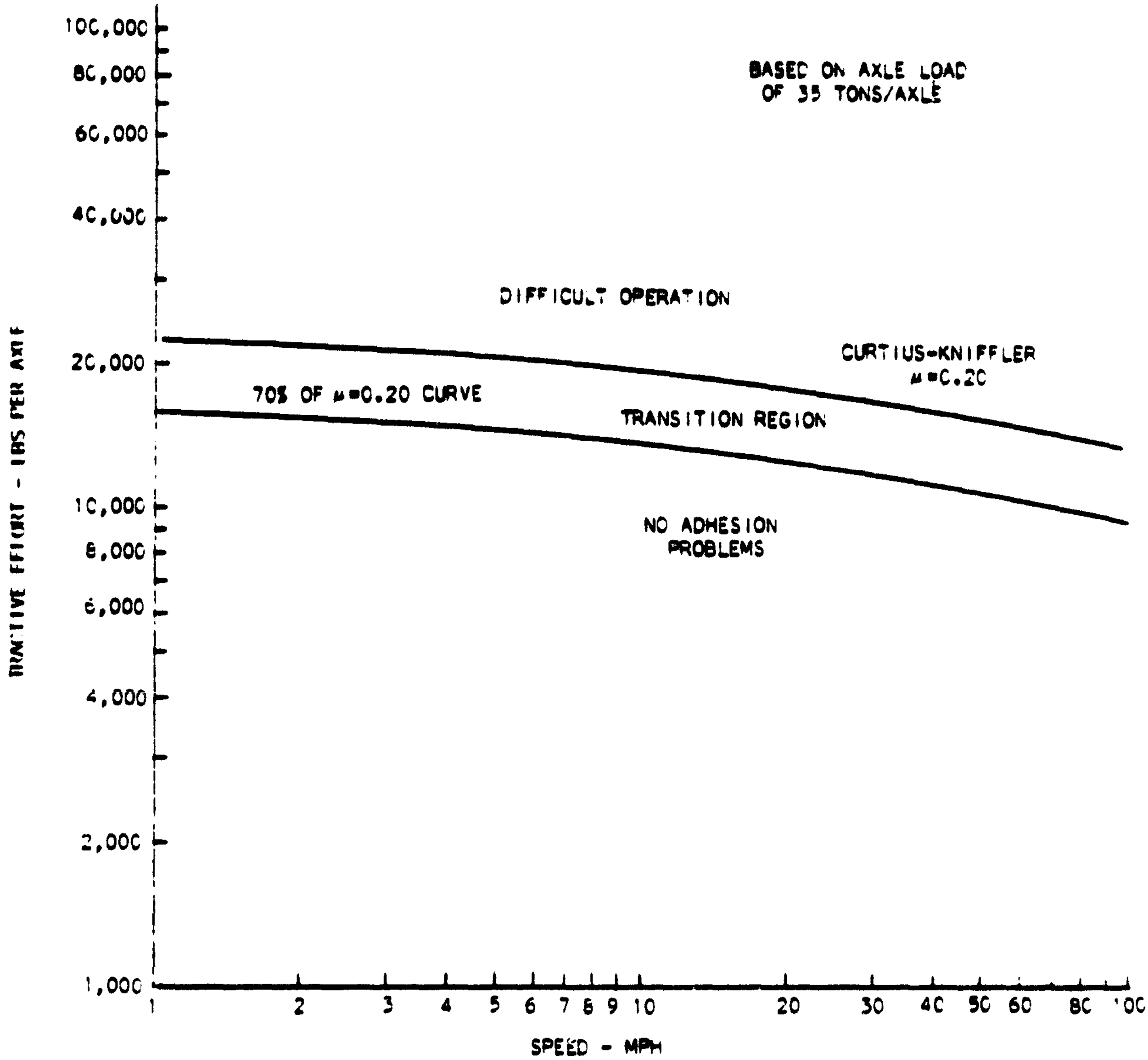


Figure 13-1. Tractive Effort vs. Speed Curve

to wheel interface. Two adhesion lines are shown in Figure 13-1. The top line is the Curtius-Kniffler curve and the bottom line is at 70% of the tractive effort of the top line. The purpose of these lines is to divide the operating region into three areas. Below the bottom line there is no adhesion problem; good rails, good wheels, and good weather are assumed. Operation above the top line cannot be guaranteed and operation significantly above the top line is not possible. However, a locomotive may be designed to operate above the top line at low speeds and for limited time durations. The region between the lines is a transition region. Satisfactory operation can be expected under most conditions. Wheel slip detection is mandatory however, and corrective action must be taken

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to insure continuous operation. The boundaries between these regions are not the sharp lines shown. Adjustments should be made for specific applications.

In Figure 13-2, the continuous tractive effort ratings of the General Motors-EMD SD45, SD40, and GP40 locomotives have been added to Figure 13-1. All locomotives are geared for 70 mph maximum speed, 35 tons per axle nominal weight, and they use the same D-77 motor. The small decrease in tractive effort starting at 11 mph for the SD45 and GP40 is due to the power matching equipment. All of these locomotives can be operated continuously in a multiple unit mode and above the minimum continuous speed

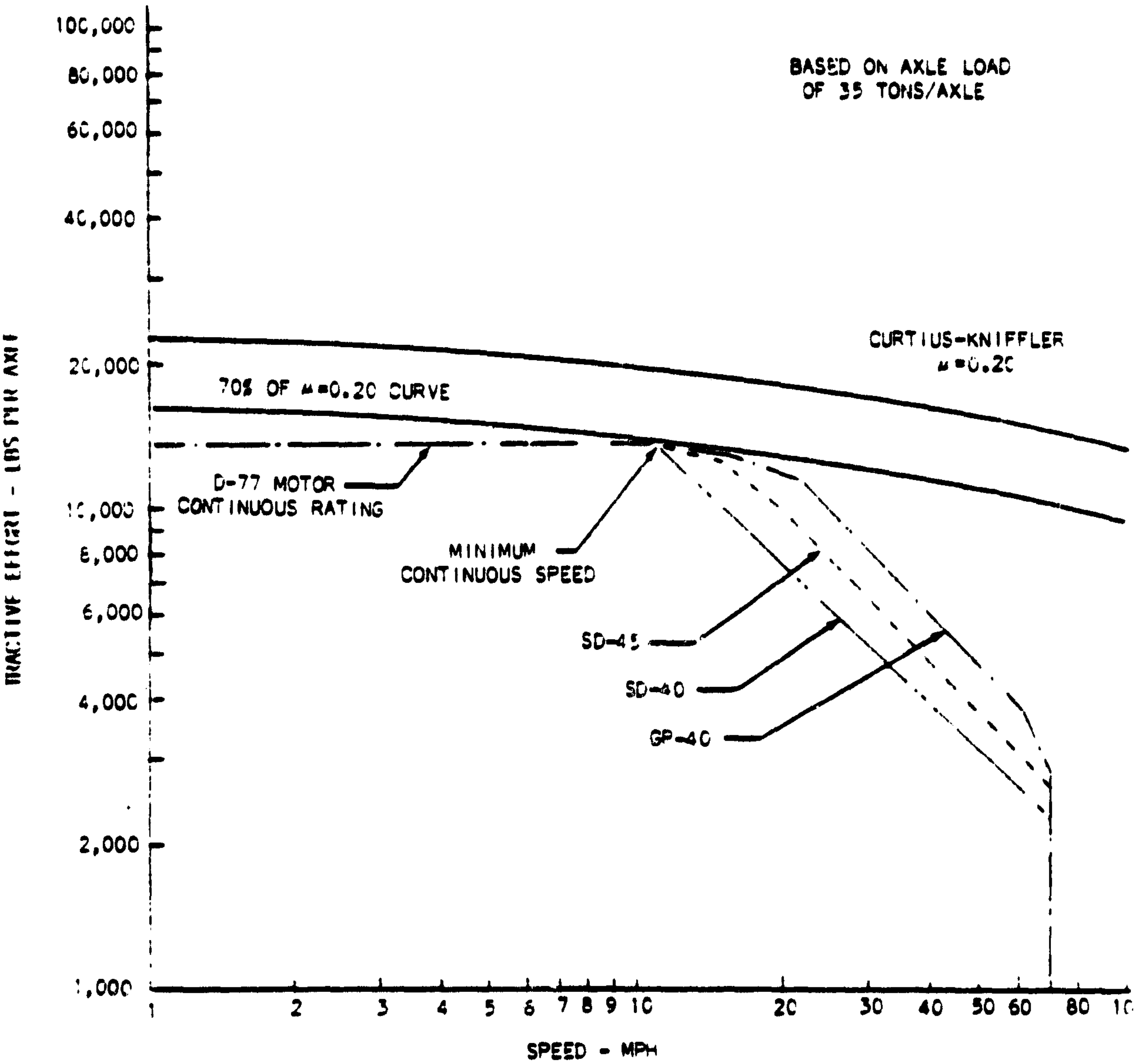

Figure 13-2. Continuous Tractive Ratings

of 11 mph. The decline in tractive effort of the GP40 near 70 mph is apparently a motor limitation. At low speeds, the continuous ratings of these locomotives are very close to the lower adhesion line. The GM-EMD short time rating at zero speed is shown just below the top adhesion line. In the middle to high speed range, the Diesel engine is the factor that limits the tractive effort while at speeds below the minimum continuous speed, the limit is the thermal characteristics of the motors.

Normalized speed-tractive effort curves are presented in Figures 13-3, 13-4, and 13-5 for a dc series motor, a dc separately excited motor, and a three phase inverter-driven ac induction motor. In all cases, the speed has been normalized to the top speed and the tractive effort has been normalized to the continuously rated low speed tractive effort.

Figure 13-3 is the Westinghouse type 1461 motor used on the Metro-liner cars in the electrified Northeast Corridor. It is a series motor using field shunting with a continuous rating of 300 hp at 600 V and a maximum speed of 3850 rpm. The short time current limit is 725 A and the continuous rated current is 415 A for a ratio of 1.75 to 1. These values are from the Westinghouse motoring curve 1461-C. The letter designations and corresponding operating conditions are given in Table 13-1. The segment BC, for example, is for continuous, constant power at rated voltage and rated armature current with increasing field shunting. In practice, it would be a series of saw-tooth steps as the field current is shunted by successive connections of the shunting elements.

Figure 13-4 is a tractive effort - speed curve for the ASEA RC-4 motor. It is a large dc separately excited motor used for heavy duty locomotives. This figure is taken directly from tractive effort versus locomotive speed curves for continuous and maximum currents. The range of values for field current and armature voltage was not given. The segments AB and FG are undoubtedly equivalent to those of the series motor presented in Table 13-1. Beyond point B, the lower curve is comparable to the continuous rating. The upper curve is a short time rating which continues all the way to maximum speed. The rated power of this motor is 1225 hp with a maximum speed of 2100 rpm. When used in light weight high-speed passenger service, the short time tractive effort is well above the Curtius-Kniffler adhesion curve.

Figure 13-5 shows the Brown Boveri type QD 535 N4, three-phase induction motor driven by two 750 kVA inverter modules. The rated power is 1080 hp with a top speed of 3700 rpm. The curve shown is the continuous rating. There is no significantly long or short-term rating possible with this type of motor. In contrast, a dc motor can be operated at twice the continuous thermal rating for a short period of time by simply applying a higher voltage to the armature.

The basic rating of a traction motor is determined by two different requirements. They are continuous low speed tractive effort and maximum speed. The motor rating is some fraction of the product of these two terms. However, these two quantities are completely independent of each other. They do not even occur simultaneously. For the three-phase ac drive, the fraction can be one-third or lower. For the dc motors of the ASEA design, the value is .45. This is probably near the lower limit and values between .45 and .7 are reasonable for dc machines. In Figure 13-6, the dc series motor characteristics with no field shunt are superimposed on the GM-EMD D-77 motors. This is a reasonable comparison and indicates that the D-77 motor at 40 mph is potentially capable of delivering as much as 1500 hp at this speed. At this point it should be noted that although the GM-EMD series of locomotives have been used in the discussion as examples, the same remarks will hold for the General Electric and for other manufacturers' having motors with similar ratings.

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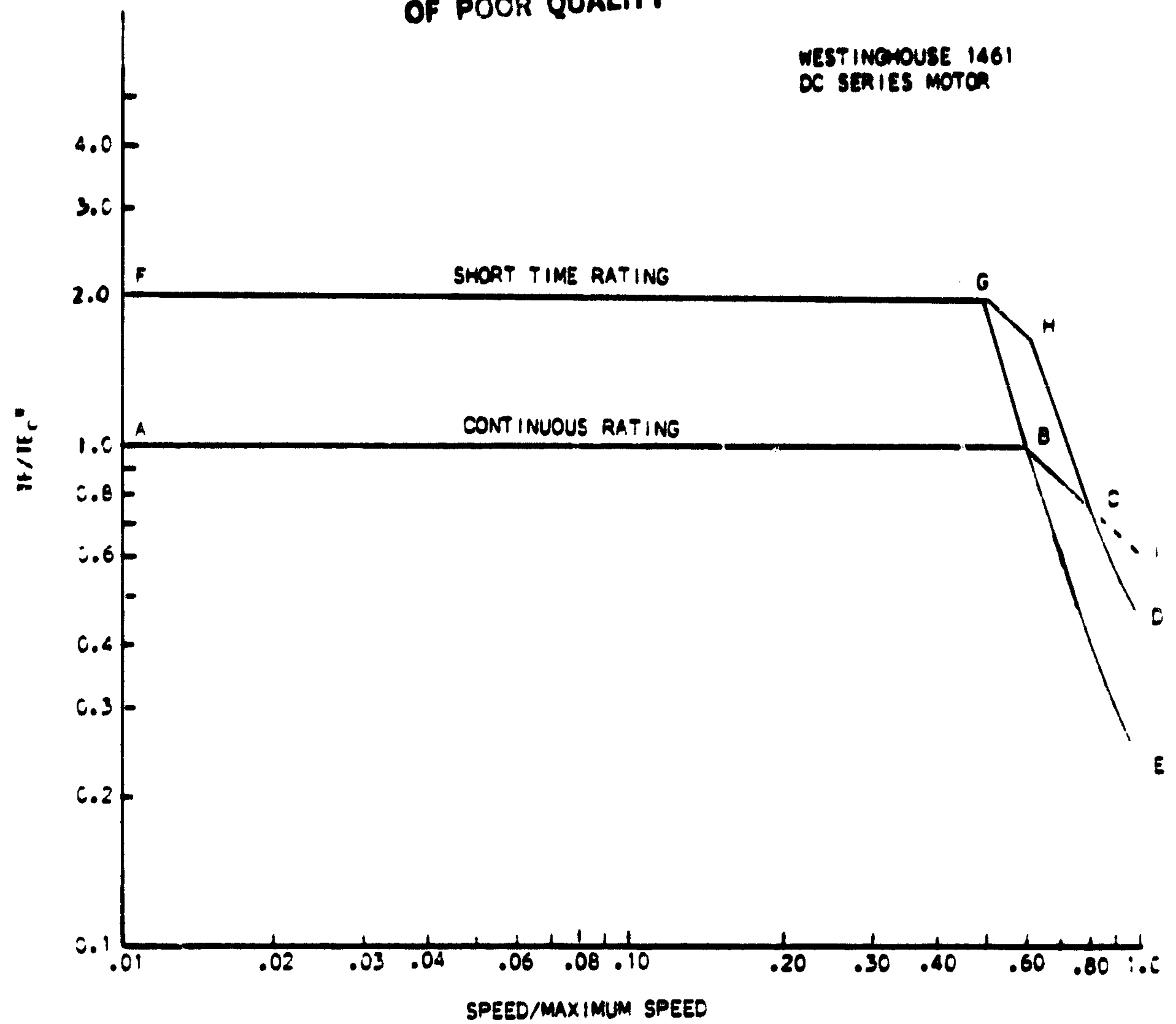

*Tractive effort divided by continuous rated tractive effort

Figure 13-3. Normalized Speed vs. Tractive Effort Curve

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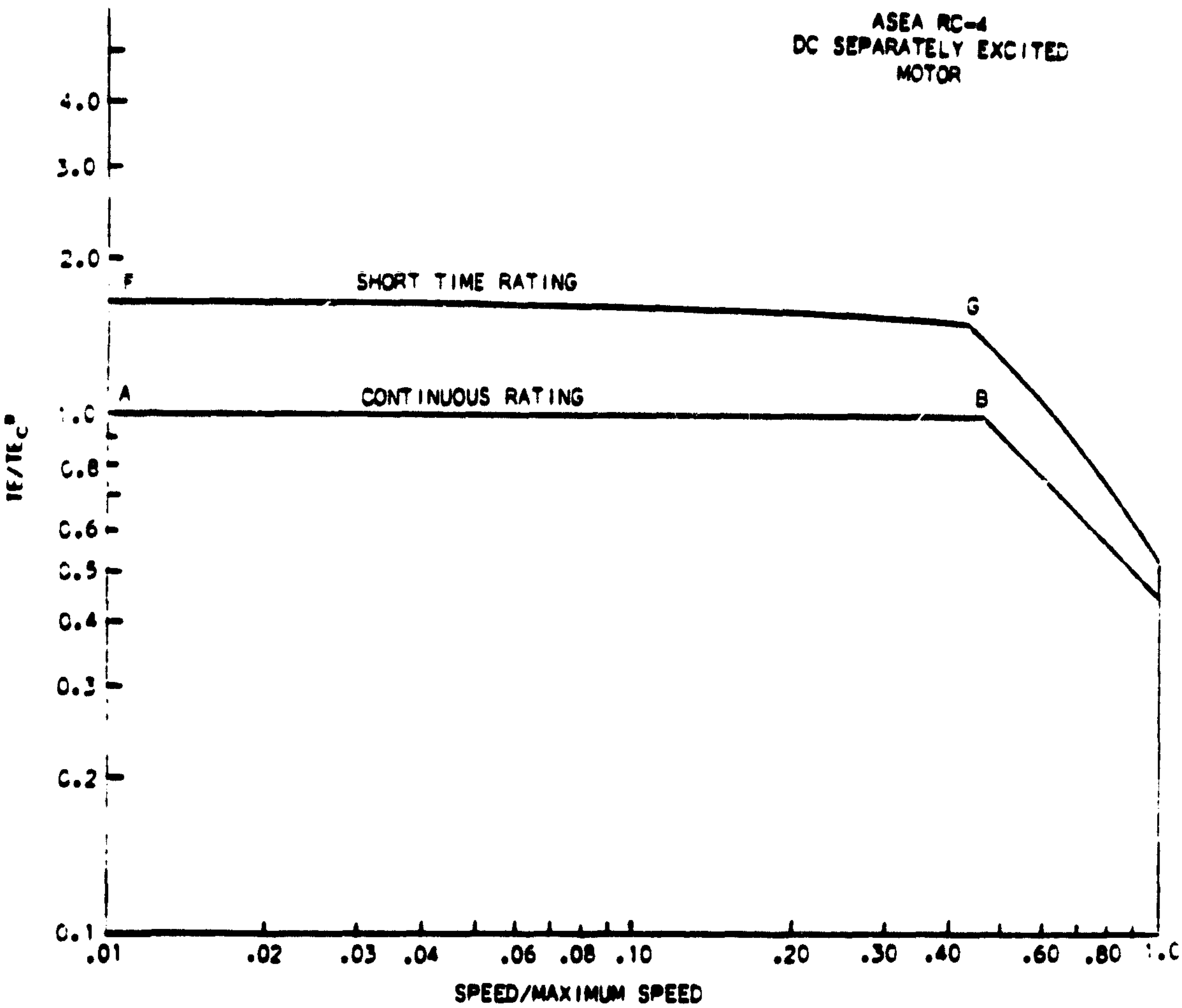
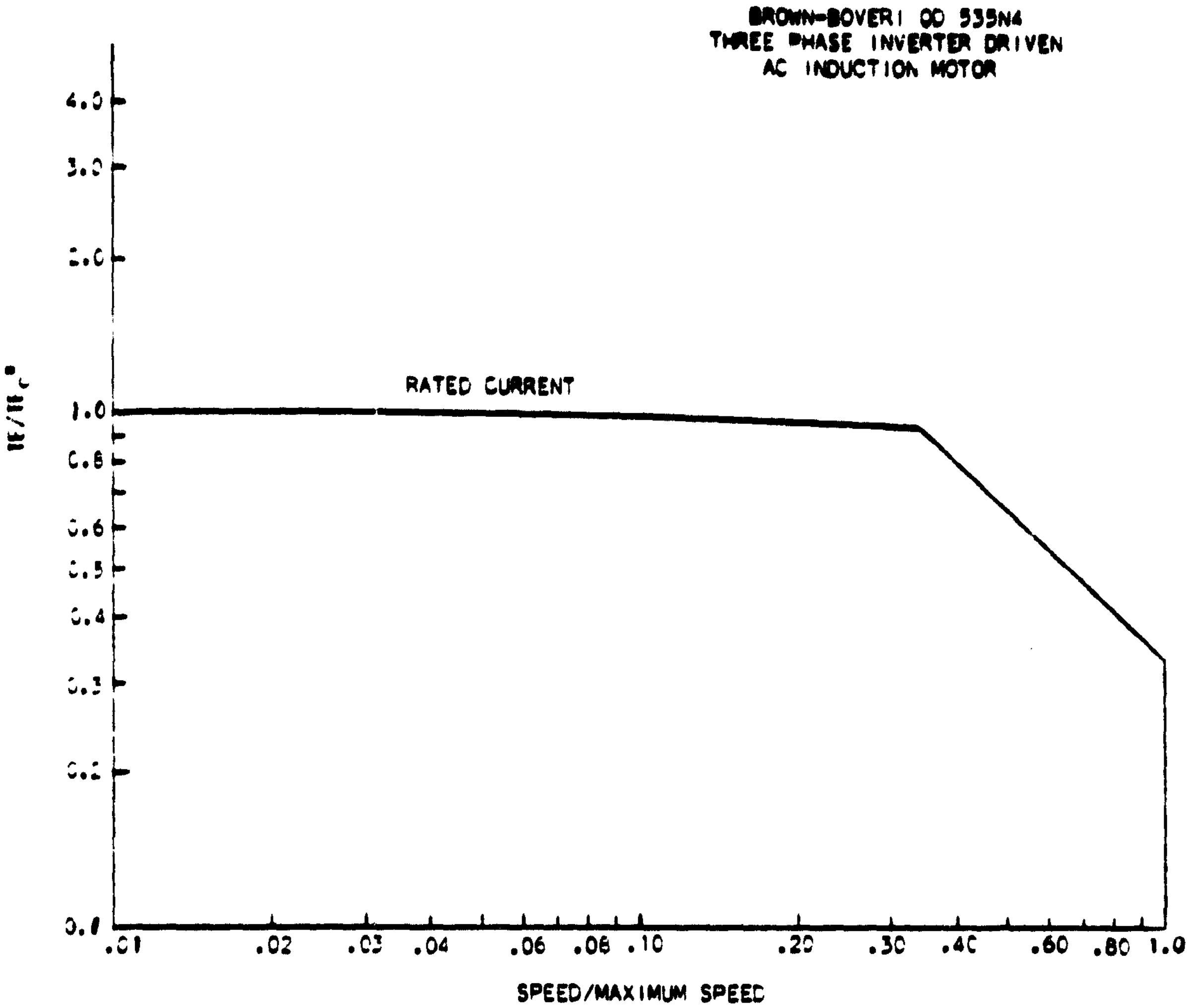
Railroad operational considerations involve three areas; adhesion, power per unit weight of both the train and the locomotives, and also site and company specifics. All these items interact with each other to some degree.

The adhesion limit is a fundamental physical fact and must be dealt with directly. For limited speed operation, a direct approach is to increase the driven axle load. This is not appropriate for the ordinary main line freight operation. However, there are exceptions. In the case of slow operation over a continuous stretch of mountainous territory, special, permanently assigned, heavy weight helpers and extra heavy track



* Tractive effort divided by continuous rated tractive effort

Figure 13-4. Normalized Speed vs. Tractive Effort Curve



* Tractive effort divided by continuous rated tractive effort

Figure 13-5. Normalized Speed vs. Tractive Effort Curve

construction could be used. In the more normal situation, the train speed is high enough that power availability is the real limitation. The adhesion problem involves how to reach the power limited speed, particularly when long periods at high tractive effort are needed. In these cases, wheel slip detection is certainly a requirement. The simplest method is to reduce the tractive effort to a level that assures full adhesion. The other extreme is to use insipient wheel slip detection and operate at the instantaneous wheel slip limit. This approach requires sophisticated equipment and the net benefit must be weighed against cost and maintenance considerations. Another approach is to assign enough locomotives so that satisfactory operation is achieved within the adhesion conditions that exist at the time and in that place.

Table 13-1. DC Series Motor Operating Conditions
(From Refs. 9-3 and 9-4)

Points	Armature Current	Field Shunt	Armature Voltage	Power
AB	Rated	None	Increasing	Increasing
BC	Rated	Increasing	Rated	Rated
CD	Decreasing (below rated)	Maximum	Rated	Decreasing
FG	Short time	None	Increasing	Increasing
GH	Short time	Increasing	Rated	Short time
HC	Decreasing (to rated)	Maximum	Rated	Decreasing
GB	Decreasing (to rated)	None	Rated	Decreasing
BE	Decreasing (below rated)	None	Rated	Decreasing
CI	Decreasing	Maximum	Above rated	Rated

Traffic requirements, route characteristics, and company policy are all factors in the selection of locomotive power per unit train weight values. Satisfactory operation can be achieved with power-to-weight ratios as low as .5 to .75 hp per trailing ton for low to moderate speeds and grades. Speeds of 70 mph for expedited container and trailer trains requires about 16 lb of tractive effort per total ton on level grade or about 3 hp per total ton at the rail. Operation up a 2% grade at 15 and 25 mph requires about 1.8 and 3.0 rail hp per total ton respectively. The wide range of specific train power required in day-to-day operation is well matched by the multiple unit locomotives now in current use.

The maximum drawbar pull limits can further require that locomotive groups be used in more than one location in the train. The use of fewer and larger locomotives in each train is seen as a possibility in the foreseeable future. The individual ratings of these new locomotives will be greater than the present ones but probably limited to a 50% increase in power and almost certainly to less than a 100% increase.

Instead of three to five locomotives running as a multiple unit set, two or three larger locomotives will be used. Alternatively, one large and one or more lower powered units could be used together. These arrangements would decrease the total locomotive weight for those applications where the weight has a significant effect on fuel consumption. For the same fuel consumption it would be possible to increase the train speed.

Maximum speed, or rather the range of speeds, is another topic that should be considered. It is assumed here that in general there is no need to significantly increase the speed for any type of freight beyond existing capabilities. There are exceptions for bad track or any other reason that limits speeds to the 5 to 10 mph range. Another exception is the possibility of moving some heavy items like coal or sugar beets faster than

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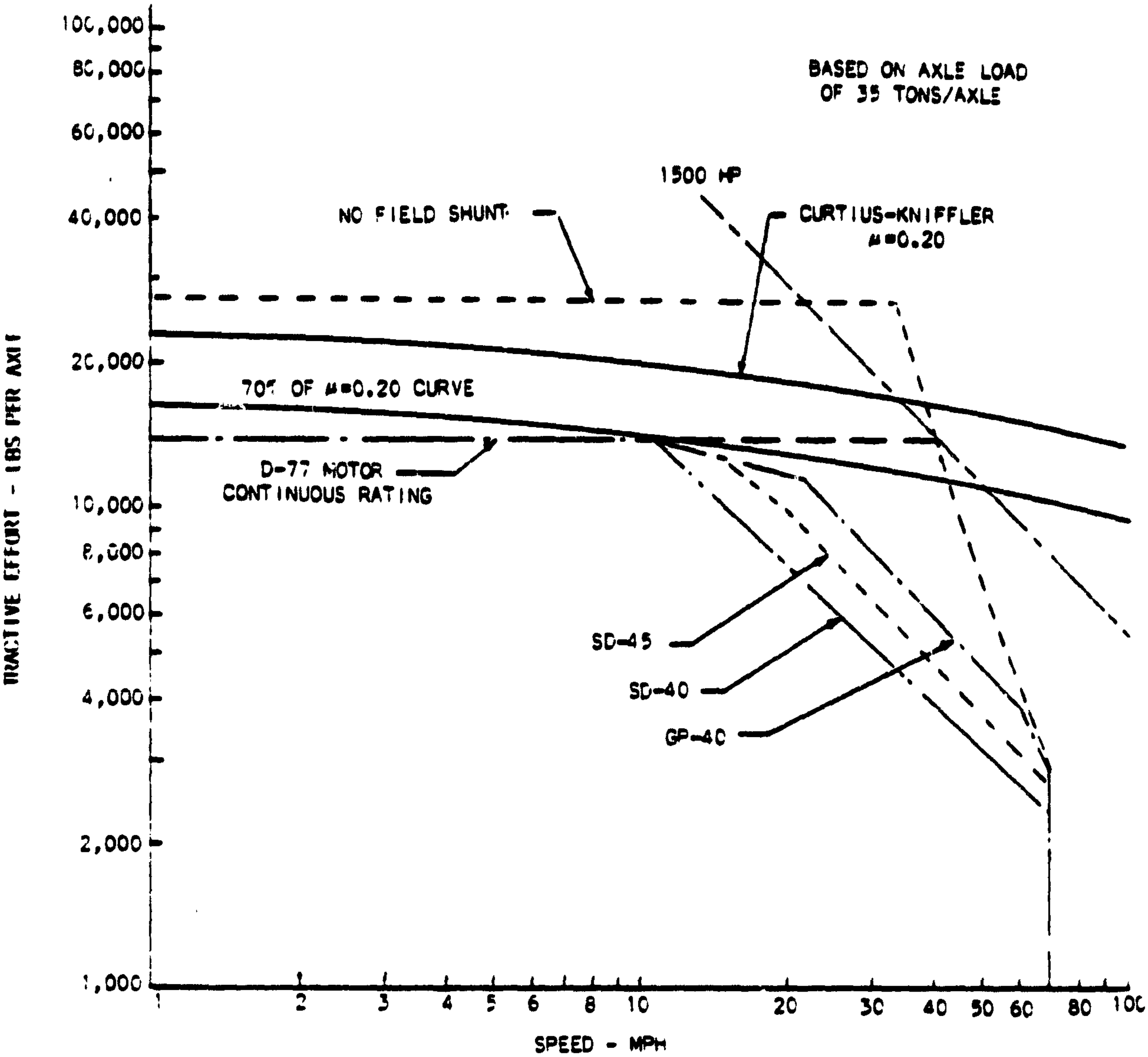

Figure 13-6. DC Series Motor Characteristics

present typical speeds so as to clear the line for movement of additional traffic.

The range of operating speeds is determined by the maximum available horsepower, the weight necessary for adhesion, and the minimum continuous speed. The six-axle, 210 ton locomotive geared for 70 mph operation, carries excessive weight as far as adhesion requirements are concerned at speeds above the minimum continuous speed. The four-axle unit with the same Diesel power plant is better as far as power to weight ratio is concerned but it does not have the low speed tractive effort capability. It is further compromised when power matching is used to enable multiple unit operation with all classes of locomotives. If the maximum speed of

70 mph is reduced, the limiting factor for continuous low speed tractive effort may no longer be the traction motor but will, instead, be adhesion related.

The use of electric motors for the final traction drive provides the simplest method for coupling a narrow speed range prime mover to the broad speed range traction requirements. The poly-phase variable voltage ac traction motor can have advantages over the historic commutation motor. However, the solid state power electronic drives must be developed to full maturity for operation in the extremely difficult railroad environment. From a practical point of view, the fundamental limit at low speed is adhesion. It is suggested that this is really an operational problem. Just how much tractive effort per ton of driven axle weight must be provided in low speed normal operation, and perhaps more important, what abnormal conditions must the train be able to handle on the road without assistance? The specific power rating (hp/ton) of the present locomotive is certainly not seriously deficient since a moderate increase in horsepower could be beneficial, particularly in very heavy high speed mainline operation. As has always been true, electrification becomes attractive as the traffic and power requirements increase. In the area of total horsepower per locomotive group, the single, large, very high power unit that would replace 4 or 5 present Diesels would have the distinct disadvantages of too great a drawbar pull at low speed, lack of redundancy, and lack of flexibility for general use. On on the other hand, extensive flexibility does not come without some compromise. If a road has only one type of locomotive, it may have to be the six-axle, high horsepower class that can climb hills but will carry excess weight for high speed, level track operation.

C. AC AND DC TRACTION MOTOR EFFECTS

When specifying locomotives, the primary consideration is whether to use a four or six-axle configuration. Above 18 mph, there is no difference in locomotive performance because of the limitation imposed by the engine output capability. Therefore, high-speed operation, above 18 mph, requires only four-axle locomotives, whereas operations at slower speeds require the six-axle locomotive in order to operate effectively below 18 mph.

If a four-axle locomotive can be developed which has the same operating characteristics as a six-axle locomotive, by increasing the continuous tractive effort capability of the traction motor and the assumable adhesion level resulting from an improved slip control, then this locomotive will be attractive to those railroads that dispatch at low or "drag" speeds.

The level of adhesion assumed by railroads varies from 16 to 25% dependent on the railroad, its operating policy, and the type of terrain and climates encountered. The most common adhesion level assumed is 18%. This figure has built into it many safeguards which reflect not only the conditions which may be experienced over the route but also the consequence of wheel spin.

In a typical locomotive today, the control is on an "all-axle" basis. If one axle slips, the tractive effort must be reduced on all axles and the entire locomotive becomes unproductive. In an ac motored locomotive, if wheel slip occurs, the torque developed is immediately and automatically reduced since the electrical slip is reduced and this action automatically corrects the wheel slip. Therefore, for the same slip risk, a substantially higher adhesion level can be assumed for an ac motored locomotive than for a dc motored locomotive, and it has been suggested by studies that up to 28% is feasible. In order to achieve the same performance as the dc motored six-axle locomotive, the AC motored locomotive requires an adhesion level of 27%.

As previously stated, in order for a railroad to be able to take full advantage of the four-axle ac motored locomotive, it must dispatch its trains in such a manner that the number of locomotives used is determined by the low speed tractive effort considerations on the ruling grade rather than on the high speed tractive effort considerations associated with high power and fast journey times. The dispatching policies of various railroads are summarized in Table 13-2, from which it can be seen that the majority of railroads dispatch in such a way that the four-axle dc motored 3,000-hp locomotive would be adhesion or traction motor limited and, hence, the popularity of the six-axle locomotive. The adoption of ac motors and the attendant level of control makes it possible to assume a higher level of adhesion than is possible with dc motors for the same slip risk.

The elimination of the commutator results in more rotor space being available for developing torque and also raises the permissible speed of the motor. These two effects combine to give a motor which can develop more torque than a dc motor over a wide speed range and can, therefore, exceed the performance of the dc motor.

In the comparison of ac and dc drives, it has been noted that the ac motor can be lower in rated power than the dc motor since it can deliver its rated power from a lower fraction of its top speed. On the other hand, the dc motor does have a short time overload factor of 175 to 200%. There are other differences such as motor rotational speed which probably requires double reduction gearing for the ac motor, but it is smaller in size and weight than its dc power equivalent.

The energy consumption of the four-axle locomotive compared with the six-axle locomotive will be less because of the reduced dead weight in the train. Locomotive weight can account for 10% of the total train weight in drag operation and, therefore, the reduction in locomotive weight from 200 tons to 140 tons represents a significant fuel saving of up to 4%.

D. ROUTE OPERATIONS

The route selected for this discussion is the section of the Southern Pacific Transportation (SPT) Company railroad between West Colton, CA. and Yuma, AZ. It is an FRA Class 7 line transporting over 40 million gross ton-miles per year. The total length of this segment of line is 197 mi. The west end includes Beaumont Hill. Eastbound the route rises 1650 ft over a distance of 27 mi. The maximum one-mile-average uncompensated grades averaging one mile are 1.8% eastbound and 1.9% westbound. The remaining distance of 135 mi to Yuma is level compared with the hill at the west end. In the westbound direction, the line is single track over the flat section with sidings spaced about 7 mi apart. Traveling up the

Table 13-2. Summary of Railroad Dispatching Policy (From Ref. 13-1)

	Minimum Speed on Ruling Grade (mph)		
Railroad	Drag	Medium Speed	Manifest
AT & SF	12.5	17.5	20
Chessie	10 to 12	15 to 18	20 to 30
CMSP&P	11	>11	>11
Conrail	11	11	20
D&RGW	11	>11	>11
Mopac	11	>11	>11
Seaboard	11	>11	>11
Southern	11	20	25
UP	11 to 14	14 to 17	20 to 25
WP	11	>11	>11

hill westbound there are numerous multiple and overlapping passing tracks. There is a greater route length of multiple track than single track in this section. Going downhill in the westbound direction, the route is double track. These tracks are equipped for operation in both directions. The entire 197 mi route operates under centralized traffic control (CTC).

The route is essentially a through-line with no intermediate yard operation. There are two intermediate branch points where loaded westbound sugar beets and iron ore trains join the route. There is only a small amount of freight service to, from, or between other local points. Except for about a half dozen tracks at Indio near the base of the westbound grade there are no places other than sidings for holding trains. There are no crew changes for through trains (Ref. 13-2).

There are a full range of freight types on this line and hence a full range of desired speeds and service priorities. A large portion of the expedited trains are westbound and consist of single commodity runs, e.g., auto p ts and general freight forwarding traffic. They are medium to higher weight trains and have the highest priority. They are often scheduled to arrive at their final destination at a specified time of the scheduled day. At the other extreme are the heavy westbound beet and ore trains. Because of their local origin they can be scheduled for about the same time each day. Ore trains do not operate every day and beet trains are very seasonal. Cars for both these trains must return empty eastbound to the branch points within the district. These trains form yet another extreme, they are some of the longest and lightest trains operated on this route. The rest of the traffic falls in between.

There are a considerable number of loads moving from California to the population centers of Tucson and Phoenix. Another group originates or terminates in Texas or further east. Some trains have a significant number of empty cars, while others consist almost entirely of loaded cars. The key item here is that the traffic is diversified and that any operational approach must handle a varying mix of traffic types.

F. DIESEL LOCOMOTIVE OPERATIONS

The vast majority of the locomotives are 3000, 3300 and 3600 hp, six-axle, heavy units. There are a few four-axle units. Virtually all of these high horsepower locomotives were acquired from the late 1960s onward. A few lower horsepower units from earlier times are still in service.

There is no identifiable pattern of locomotive use. The locomotives can and do appear to be used interchangeably in all facets of service. Units of the same power, different power, and from different manufacturers operate together, in 2 or 3 unit helper combinations or in 2 to 5 unit road combinations. The 2 and 3 unit combinations tend to stay together on sequential assignments. Because the Yuma-Colton route segment is part of a long transcontinental route, some units appear, traverse the route, and disappear. Some would reappear several days or a week later on return trips. On the other hand, several groups made repeated helper runs over the hill. One three unit combination made 18 such trips over 8 days. Four other combinations made 10 or more trips in the same short time.

Locomotives are assigned to provide a given range of horsepower per trailing ton. Manufacturer's engine horsepower ratings are used rather than the lower power developed at the rail. The locomotive weight can become a significant fraction of the total train weight particularly if the train power is increased.

For slow non-expedited trains, additional helper power is required for Beaumont Hill. For expedited high speed trains, the power assigned allows the trains to reach a maximum speed of 70 mph on the level while maintaining speed on the hill section even when helpers are not assigned.

F. TRAIN SPEEDS AND SCHEDULES

Expedited trains are assigned power levels greater than 3.5 hp per trailing ton for the entire route between West Colton and Yuma. This power level permits operation at the 70 mph limit speed on the level grade. These "identified" freight trains are allowed 40 mph speeds over the hill.

On a 1.5% grade, the approximate total train resistance at 30 mph is 35 lb per ton necessitating 2.8 rail hp per train ton. Thus, trains at 4 hp per trailing ton can maintain a reasonable speed over the hill. Many freight trains can make the 197 mi trip in less than 4 hr 30 min. In comparison, the westbound passenger train is scheduled for 3 hr 40 min. The fastest eastbound trains make the 197 mi trip in 5 to 7 hr which is a little slower than the expedited westbound runs.

The westbound iron ore and sugar beet trains are at the other extreme. The typical ore train is a 100 car, 12,000 ton run that is picked up at the intermediate interchange point with about 0.83 hp per trailing ton. Both swing and rear helpers are added at the base of the grade, nine locomotives total, resulting in about 2.5 hp per trailing ton. They are picked up very early in the morning, outfitted with helpers about 8 a.m. near Indio, and go the 60 mi distance over the hill in 4 to 5 hr although it can take up to 8 hr on occasion. The beet trains originate from a branch line with 3 to 4 road locomotives at 1.5 to 2.5 hp per trailing ton. The beet trains sent to two destinations are treated differently. Those ultimately bound for Bakersfield over the Cajon and Tehachapi grades are given two rear helpers for a total of 5 units and about 3.5 hp per trailing ton. All five locomotives go through to Bakersfield. Trains bound for the Coast route are given the usual 3 unit unit helper which is then removed after the hill. With four original road engines these trains range close to 4 hp per trailing ton on the grade. Like the ore trains, the beet trains arrive at the base of the upgrade at about the same time each day, 8 to 10 A.M., but unlike the ore trains they typically spend extended times there. Once committed to the hill, however, the trip times are very respectable, as short as 3 hr for about the 55 to 60 mi grade.

Between the expedited and the ore or beet trains there are several groups that are distinguishable by the power level used. Some of these are identified freight trains that make the 197 mi westbound trip in 6 to 8 hr using about 3 hp per trailing ton and no helpers over the hill. There is another group that are assigned 1.25 to 2.25 hp per trailing ton on the flat portion and use helpers on the hill. On the hill, the power level is at least 2.5 and often goes as high as 3 to 3.5 hp per trailing ton. These trains take 8 hr or longer for the 197 mi westbound trip. Many of them spend significant times on the sidings, particularly on the single track east end portion of the route while the high priority trains are passing in both directions.

G. OPERATIONAL LIMITS

Descending grade operation is governed by two considerations. They are a requirement for safety that is satisfied by the availability of automatic air brakes and the desire to use electrical brakes as the primary braking system to minimize the use of the air brakes. On the Beaumont hill descending grade, a speed limit of 20 mph for trains exceeding 80 tons per operative brake is in force. Many trains operate at less than 80 tons per operative brake and these "identified" trains are allowed speeds up to 40 mph.

Electrical braking capability is limited in an absolute way by the fraction of the train weight on the driven axles. Ascending grade operation

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is subject to the same limitation. Stated another way, the locomotive must at least be heavy enough to prevent slipping on the non-level track. The maximum physical grade on Beaumont Hill is 1.9% and thus the absolute minimum permitted weight ratio (weight of locomotives divided by the total weight of the train at standstill) is 0.095 to 1.0. It should be recognized that additional locomotive weight may be required for either descending or ascending a grade. From observed data, the minimum ratio on this route is about 0.125 to 1.0. This is the value for the iron-ore trains. The fast trains that require a large value of hp per ton will have ratios of 0.16 to 1.0 to 0.18 to 1.0 simply because many locomotives are needed to supply that level of power.

Figure 13-7 gives the horsepower at the rail per total train ton versus the speed for various grades. The values used for rolling resistance on level grade are given in tabular form in the same figure. These values are a composite of existing rolling resistance data. They are a little larger than the Davis data for speeds up to 40 mph which is the upper limit for the original Davis data. Between 40 and 70 mph it is felt that the extrapolation of the Davis quadratic form gives values that are too low. Similarly the value from the Illinois Institute experimental data for the 40 to 70 mph

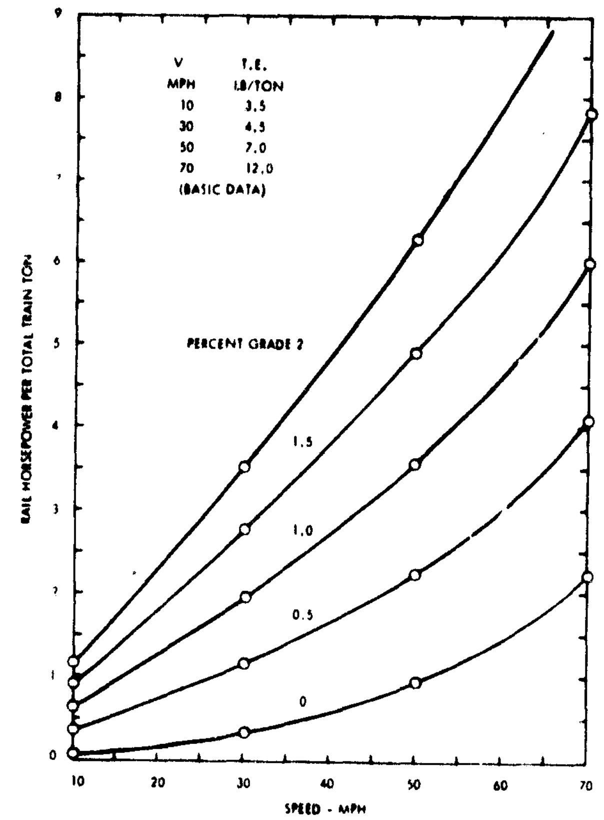

Figure 13-7. Rail Power vs. Speed for Different Grades

range are felt to be too high. The values used here are a compromise between these extremes. In fact, over the entire speed range, they are closer to those used by the German Federal Railways.

H. LEVEL AND ASCENDING GRADE POWER REQUIREMENTS

The energy required to move a train a given distance is proportional to the above rolling resistance values. Thus, at 10 mph, it requires about .0094 hp-hr/ton-mi. At 50 mph, the requirement is double this value or about .0188 hp-hr/ton-mi. At 70 mph, the energy required increases to .0322 hp-hr/ton-mi or nearly 3.5 times the low speed requirement.

On the other hand, going up a 1.9% grade requires .1006 hp-hr/ton-mi just for the grade alone to which the rolling resistance values for the speed must be added. The relative cost of high speed operation is very dependent upon the prevailing conditions.

I. DIESEL HELPER OPERATION

From these power requirements, it can be seen why helper operation is used over the hill for all except the high speed expedited trains. Table 13-3 shows the grade-speed relationship for a range of power per trailing ton. Values of 1.0 to 2.0 hp per trailing ton are commonly used for Diesel power on flat runs for the slower trains and values of 4 and more on expedited trains. A horsepower per trailing ton values of 2.55 hp per trailing ton results in a speed of 15 mph on 1.9% grade and appears to be the nominal minimum used on the hill.

J. TRAIN AND LOCOMOTIVE SIZES AND SPECIFIC POWER RATINGS

Virtually all main line freight trains are operated with more than one Diesel locomotive. Hence the concepts of horsepower per train ton and per locomotive ton are correct measures of train performance independent of train length or size. At low speeds where operation is adhesion limited rather than power limited, locomotives are located in several places in the train in order to not exceed the drawbar force limit. Typically 18 locomotive axles located together will develop the nominal 250,000 lb drawbar limit. In many cases more than 18 axles are used at the head of an expedited train. These trains are power limited at their high operating speeds on level grades. When these trains are operated on hill grades, maximum drawbar pull can be exceeded if the speed is allowed to drop excessively without reducing the throttle setting.

In this study it is assumed that drawbar pull limits are satisfied by proper placement of the locomotives on the train. This is not a trivial operational matter. For analysis purposes the specific power, i.e., horsepower per train ton, can be taken as the single measure of train performance independent of the length, weight, number of cars, or the number of individual locomotives. The concept of specific train power, in horsepower per train ton will be applied to trains for various types of Diesel locomotives currently in use to the all-electric and to other combination locomotives.

There is a relationship between the locomotive power to its weight ratio, the power to trailing weight ratio, and the power to total train weight. Table 13-4 shows this relationship for a number of road locomotives, both Diesel and all-electric. The specific consist power in power per trailing ton is shown for two values of power per total train weight. They are rated at one and three horsepower per ton. This performance description applies only in the power limited region above the base speed. To quantify this, the base speed was calculated and is in Table 13-4. The base speed is the speed at which the assumed maximum tractive effort, utilizes all the available rail horsepower. It is closely related to the minimum continuous speed, particularly for the first locomotive in Table 13-4. Power matching is typically used in the other Diesel units, so this definition is not always true. However, the values of base speed given in

Table 13-3. Representative Traction Capabilities (From Ref. 13-2)

Rail hp (train ton)	Grade (%)	Speed (mph)	Grade (%)	Speed (mph)	Nameplate hp (trailing ton)
0.75	.8	15	0	44	1.0
1.36	1.5	15	0	58	2.0
1.65	1.9	15	0	64	2.55
2.2	1.9	20	0	70	3.59
1.5	1.9	23	0.3	65	4.21

Representative traction capabilities, as derived from Figure 13-2 data; assuming $\frac{\text{hp (Rail)}}{\text{hp (Rated)}} = 0.75$ and a U33 locomotive.

Table 13-4. Representative Locomotive Capabilities (from Ref. B-2)

Locomotive Power/Wt. Ratio (hp/ton)	Specific Consist Power (trailing ton) (3 hp/train ton)	(1 hp/train ton)	Base Speed (mph)
3000/205 = 14.6	3.77	1.073	10.2
3300/208 = 15.9	3.70	1.067	11.1
3600/206 = 17.5	3.62	1.061	12.2
3000/139 = 21.6	3.48	1.048	15.1
3500/140 = 25.0	3.41	1.04	17.5
7000/140 = 50.0	3.19	1.02	35.0

this table assume that full horsepower is available for all locomotives. For convenience of comparison, the same 75% transfer efficiency coefficient, as used for Diesel-electrics, is also used for all-electric. The 7000 hp and the 50 hp per ton ratings thus represent gross ratings of which 75% appears at the rails.

Any locomotive operating above the base speed is carrying excess weight beyond that necessary to transfer the tractive effort being developed. Consequently, a four-axle light weight locomotive should perform as well as a heavy six-axle locomotive with the same power rating. Further, it can be seen that for the low specific train power of 1 hp per ton, the heavy locomotives are only a few percent less effective (i.e. requiring a slightly higher specific consist power) than the light, 3500 hp, 140 ton unit. At the higher specific train power of 3 hp per ton, the specific train power increases significantly and an 8% savings can be attained by using the four-axle, 3500 hp, 140 ton locomotive.

When the route and traffic specific needs are considered, it is concluded that the Diesel locomotives are already being used over their full range of capabilities to handle the existing traffic. The Diesel, in fact, is a most versatile unit and is used at both high and low speeds, on flat and hill grades, and in road and helper service. The heavy six-axle units are at a small, but significant, disadvantage in all service except possibly the iron-ore trains. In spite of the steepness of the Beaumont grade, it appears that this route is not really a low speed, heavy drag route. The current traffic demands some more moderate speed. The four-axle lighter weight, and particularly, the new higher horsepower locomotives have a definite advantage.

The high specific locomotive power of the all-electric locomotive is the key difference between it and the Diesel as far as train performance and operations are concerned. The light weight high speed passenger locomotives for the Northeast corridor will deliver 60 hp per ton on four-axles. The value of 50 hp per locomotive ton in Table 13-4 is a reasonable value. Four-axles and 140 tons were also assumed for this unit.

For level run operation, a desirable option is to reduce the number of driven axles and, thus, the number of locomotive units. At a power level of 3 hp per gross ton, a 210 ton Diesel unit will pull about 800 trailing tons using 16 hp per locomotive ton. At the same train power level, a 140 ton all-electric unit will pull about 2200 tons using 50 hp per locomotive ton. The locomotive-to-train weight ratio is about 0.19 for the Diesel and 0.06 for the electric. Both trains will do 70 mph on a level run. The Diesel train will go over the 1.9% hill at reduced speed without the addition of helpers. An all-electric locomotive train with fewer locomotives however, cannot go over the hill alone. The locomotive-to-consist weight ratio is a factor of two lower than required for adhesion on the grade. For safe operation on a 1.9% grade, the weight ratio must be increased to at least 0.125 to 1.0. If this is done by simply decreasing the trailing tons per locomotive, the single all-electric unit can pull only about 980 tons but will have a specific train power of 6.3 hp per train ton. This will give 50 mph on a 1.9% grade, over 60 mph on a 1.5% grade, and nearly 100 mph on level grade. This is more power than can be effectively used in either hill or level operation.

Also in the case of heavy trains the locomotive power must almost certainly be distributed within the train rather than at the head end to limit the maximum drawbar pull. Additional engines are needed regardless of the power capability of the head end units.

The selection of a desired speed over the hill is the key in defining the helper locomotive capabilities and the operational strategies. For example, assume that it is desired to maintain 30 mph on a 1.9% grade. This is about as high a speed as can be used on Beaumont Hill considering the grade and curve profile. The requirement is to increase the locomotive-to-consist weight ratio of the train above the value that was satisfactory for level grade operation. This is done by adding helpers. However, it is desirable to keep the total train power within the really useable power limits of the grade.

The train power becomes excessive when very high specif. power road engines suitable to level operation, are added as helpers. Consider now adding a 210 ton, 6-axle, helper unit for each 140 ton all-electric road locomotive that arrives at the bottom of the hill with its 2200 ton trailing load. To maintain 30 mph on the 1.9% grade requires 3.6 rail hp per train ton. The new locomotive weight is 140 + 210 or 350 tons and the new train weight is 350 + 2200 or 2550 tons. The weight ratio is 350/ 2550 or 0.137 which is slightly above the 0.125 selected as minimum. The required tractive effort at this speed and grade is about 115,000 lb which requires only 115,000/(350 x 2000) or 0.26 adhesion, below the 0.2 nominal value. Now the required locomotive power is about 26 rail hp per locomotive ton or about 35 nameplate hp per locomotive ton based on the nominal Diesel engine power rating.

Thus, the 50 hp per ton high specific locomotive power, all-electric, high speed, road locomotive must be operated at only 35 hp per ton at the 30 mph speed on the 1.9% grade. It is really designed for level rather than grade operation. On the other hand, the required specific power of the helper unit is the same 35 hp per ton. The present six-axle Diesel SD45 units operate at 17.5 hp per ton and the unit is power limited by the Diesel engine although the traction motors themselves are capable of higher ratings. The combination of the all-electric locomotive and the six-axle Diesel locomotive would not have enough power to climb the hill at 30 mph. It is feasible that an all-electric helper with a 35 hp per ton rating could be made from a six-axle Diesel locomotive by replacing the Diesel engine and fuel tanks by a transformer and a pantograph.

Another approach is the use of the dual mode or universal locomotives that has been proposed for all-electric operation on hills and for Diesel power operation over other sections of the route. This involves adding the required electrical equipment to existing units, hopefully displacing dead weight ballast. Even with the added equipment, these units would probably have the same hp per ton ratings as the unmodified Diesels in the Diesel mode and a somewhat higher rating in the electric mode. However, the primary purpose would be to replace oil as fuel on the heavy grades where maximum power and a large amount of energy is needed.

A third approach is to use power cars to condition the electrical power drawn from the catenary and deliver it to the motors of adjacent unmodified Diesel-electric locomotives. The power cars themselves could also have motors on their axles. Because a larger single transformer can be used to energize a number of locomotive units, the hp per locomotive ton of the combined configuration can be higher than that of the original Diesel units. Again the main purpose is to displace the use of oil on isolated sections of electrified territory.

In the long run, any of the above approaches could be part of a transition from Diesel to all-electric operations. As suggested, initial all-electric operation can be over a hill section where significant benefits would be immediately realized.

As the electrified territory is extended there would be adequate time to acquire the new high specific power electric locomotives that would minimize the locomotive weight and the number of locomotives required to handle the traffic on level grades. Helpers would be required on hills and ex-Diesels converted to full all-electric units could be very effective.

K. SUMMARY

The way locomotives are used is an important factor in their fuel consumption. Adhesion affects the number and kind of locomotives used in a train, which in turn, affects the locomotive-to-train weight ratio and the power-to-weight ratio. Both of these ratios directly impact fuel consumption. The adhesion is affected, in turn, by the axle loadings, the motor ratings, the train speed, and the rail conditions. Slip controls are an important factor in the dispatching of locomotives since they allow a higher level of adhesion to be used, and, therefore, permit fewer or lighter locomotives to be used in the train.

There are differences between ac and dc motors in terms of slip controls and usable adhesion. The ac motors appear to usable at adhesion values well above those of dc motors. They should also have lower maintenance than the dc motors. The main difficulty with an ac drive is the inverter that is required by the system. Its durability, cost, electromagnetic interference, and maintenance need to be improved.

Improvements in adhesion and motors may be less advantageous on some routes than on others because of the terrain. The fuel savings which may come from operations are very site and company specific. One site was analyzed to illustrate some operational problems. The quantitative effects must be carried out at a much more detailed level using actual operating data. The railroads are the only ones who can make these analyses for their own routes. New engines can improve the fuel economy of the railroads but they are years away. Fuel savings from operational changes can be effected in a very short time.

L. SECTION XIII REFERENCES AND NOTES

13-1. Garrett Airesearch, Inc., Torrance, CA, Personal Communication, Sept. 4, 1980.

13-2. Stallkamp, J. A., "Site Specific Study of Selected Traction Options for Locomotive Hauled Trains", Letter Report 22, Jet Propulsion Laboratory, California Institute of Technology, Pasadena, CA, Sept., 1980.

SECTION XIV
COST ANALYSIS

A. INTRODUCTION

Many of the new engines are technically feasible for use in line haul locomotives. Many fuels are also technically feasible for use in these engines. In this chapter, the cost of acquiring and operating these locomotives is examined. The purpose is to rank the various engines and fuels in terms of their life-cycle costs. In the earlier chapters, there have been brief notes about fuel or capital costs of the advanced locomotives. In this sectoin, the cost of the locomotives, their fuel costs, maintenance costs, and operational costs as well as taxes are taken into account.

The detailed method of calculation used in this life-cycle cost analysis is based on a costing methodology developed by the Jet Propulsion Laboratory and documented in The Cost of Energy from Utility-Owned Solar Electric Systems (Ref. 14-1). Although this model was originally developed for utility applications, it is actually a much more general approach. By using financial data from cooperating railroad companies and data from the open literature, the model will provide a present value life-cycle cost consistent with the accepted practices of engineering economics.

Because of the inflation experienced in the last 8 years, the cost of fuel, labor, and other goods and services have changed rapidly. The financial and cost data in the literature covers a wide range of time and therefore some means of converting this data to current dollars is needed. To project this information into the future requires the use of escalation factors for several different parameters, primarily maintenance, fuel, operations, and capital. Locomotives with different engines or using different fuels will have different escalation rates. The technique used in this study is to first predict the general inflationary or escalation rate, then to predict the specific rates as perturbations of the general rate depending on the details of the engine or fuel. The general rate used here is the Wholesale Price Index based on the year 1967. When constant dollar values are referred to in this chapter, they are 1967 dollars. Current dollars are for either a specific year or for the year 1980 if no year is given.

This section is arranged such that the methodology is presented first. Historical cost data is then presented followed by the predicted costs. The alternative Diesel engines, new engines, and alternative fuels costs are discussed in detail in the following sections. The sections are followed in turn by the comparison of the life-cycle costs and the final rankings of the fuels and engines. A brief discussion of the development costs concludes the chapter.

B. COST COMPARISON METHODOLOGY

A consistent cost methodology must be applied to the various alternative locomotive systems. The methodology helps to determine the relative economic desirability of the alternatives. The "life-cycle cost methodology" is one method of comparing alternative (in this case, locomotive) systems.

The life-cycle cost of a specific alternative locomotive represents the present value of all the costs incurred in purchasing and operating a locomotive, including a normal rate of return on equity. More specifically, the calculation of costs over the life-cycle of a locomotive implies that all capital, operations, maintenance, and fuel costs have been accounted for. In addition, such factors as taxes, depreciation schedules, rates of inflation, and system service life are also considered.

The fundamental equation of the life-cycle cost methodology is:

$$\underline{AC} = FCR \times CI_{pv} + CRF_{k,N} (OP_{pv} + MNT_{pv} + FL_{pv})$$

The terms are defined as follows:

$\underline{AC}$	Levelized annual life-cycle cost
FCR	Fixed charge rate
CI_{pv}	Present value of all capital investments
$CRF_{k,N}$	Capital recovery factor based on capital cost k and service life N.
OP_{pv}	Present value of operating costs
MNT_{pv}	Present value of maintenance costs
FL_{pv}	Present value of fuel costs

At a more detailed level, however, a number of parametric values must be specified in order to calculate life-cycle cost, including:

T	Combined effect of State and Federal income tax rate
N	System (engineering) lifetime
n	Accounting lifetime (for tax purposes)
B_1	Property tax rate (if applicable)
B_2	Insurance rate
f_d	Fraction of capital costs consisting of debt
f_e	Fraction of capital costs consisting of equity
k_d	Interest cost debt
k_e	Interest cost on equity
g_{CI}	Annual capital escalation rate
g_{OP}	Annual operations expense escalation rate
g_{MNT}	Annual maintenance expense escalation rate
g_{FL}	Annual fuel escalation rate
CI	Capital investment
OP	Annual operating expense
MNT	Annual maintenance expense
FL	Annual fuel cost

The method of depreciation (i.e., straight-line, accelerated etc.) must be specified. The terms in the life cycle equation are related to these parametric values by the following equations.

$$FCR = CRF_{k,N} \left(\frac{1 - T \times DPF_{m,k,n}}{1-T}\right) + B_1 + B_2$$

$$CI_{pv} = \sum_{t=1}^{N} CI \quad (1 + g_{CI})^t/(1 + k)^t$$

$$CRF_{k,n} = k/(1-(1 + k)^{-n})$$

$$CRF_{k,N} = k/(1-(1 + k)^{-N})$$

$$DPF_{m,k,n} = (n \times CRF_{k,n})^{-1}$$

$$OP_{pv} = \sum_{t=1}^{N} OP \ (1 + g_{OP})^t/(1 +k)^t$$

$$MNT_{pv} = \sum_{t=1}^{N} MNT \ (1 + g_{MNT})^t/(1 + k)^t$$

$$FL_{pv} = \sum_{t=1}^{N} FL \ (1 + g_{FL})^t/(1 + k)^t$$

where:

$$k = (1-T) \ f_d \ k_d + f_e \ k_e$$

Once the input values for each alternative locomotive system have been specified, the methodology will reveal which of the alternatives has the lowest cost over the life of the system. For instance, with this type of methodology, it is possible to compare a locomotive having higher capital costs but lower fuel costs with a locomotive having lower capital costs but higher fuel costs. The final values of $\underline{AC}$'s (taking into account all financial and economic parameters previously identified) will reveal the locomotive with the lowest life cycle cost.

The formulas used to calculate the life-cycle cost estimates made in this report were derived for a costing methodology developed by JPL. By using the tax environment and industry financial data of a representative railroad company, the model will produce a present value cost consistent with the accepted practices of engineering economics.

C. A CASE EXAMPLE

The following is a numerical example showing how the methodology compares the life-cycle costs ($\underline{AC}$) of two alternative locomotive systems. The two cases presented relate to the description presented in the above text: i.e., in both systems, all parametric values are identical except for the capital costs and fuel costs of the two locomotives. This numerical example is for illustrative purposes only and should not be construed as representative of actual system costs.

The straight line method of depreciation as well as the following constants are used in the two examples below.

$T = 0.5$	$f_d = 0.5$	$g_{OP} = 0.07$
$M = 15$	$f_e = 0.5$	$g_{MN} = 0.07$
$n = 15$	$k_d = 0.08$	$g_{FL} = 0.08$
$B_1 = 0.0$	$k_e = 0.08$	$OP = \$30,000/\text{year}$
$B_2 = 0.0025$	$g_{CI} = 0.06$	$MN = \$60,000/\text{year}$
		$k = 0.600$

CASE I:

FCR	= 0.1418	OP_{pv}	= \$485,504
CI_{pv}	= \$600,000	MNT_{pv}	= \$971,009
$CRF_{k,N}$	= 0.10296	FL_{pv}	= \$524,289
FL	= \$30,000/year		

Life-cycle cost = 0.1418 (600,000) + 0.10296 (485,504 + 971,009 + 524,289)
= \$289,023

CASE II:

FCR	= 0.1418	OP_{pv}	= \$485,504
CI_{pv}	= \$500,000	MNT_{pv}	= \$971,009
$CRF_{k,n}$	= 0.10296	FL_{pv}	= \$699,052
FL	= \$40,000/year		

Life-cycle cost = 0.1418 (500,000) + 0.10296 (485,504 + 971,009 + 699,052)
= \$292,837

The life-cycle cost calculation shows that even though the capital investment of Case I is greater than that of Case II, the higher fuel costs of Case II result in a higher life-cycle cost than Case I. It is interesting to note that a locomotive capital cost of \$473,103 for Case II would make the life cycle costs of the two systems equal.

Because the capital cost of a locomotive is a one-time event occurring in the first year of its life, the present value of the capital investment is equal to the capital investment. Further, because the annual capital escalation rate occurs only in the equation for the present value of the capital investment, no value for it is needed in the calculations. A value for the annual capital escalation rate would be needed if the capital expense is spread over a number of years.

D. COST DATA

Two sets of price and cost data are included in this section. The first data set is the historical and the second is the predicted prices and costs. The four cost areas are fuel prices, locomotive prices, maintenance costs and operating costs.

In many cases, the historical data is incomplete and, sometimes, non-existent. This is certainly the case with many of the new engines and with some of the fuels. In some cases, price trends can be established if a correlation can be established with another fuel or engine which does have a good historical base. The historical data is not used directly to predict the future price. It is used to establish a correlation with the

Wholesale Price Index. The index is predicted and the correlation is then used to generate the future price. The correlation is sometimes modified by including factors to account for resource depletion and increasing demand. In the case of Diesel No. 2, both factors must be taken into account. The influence of governmental decisions, such as sudden price hikes by OPEC nations, is not taken into account. These types of actions are not predictable. Price hikes which reflect the effects of inflation are taken into account.

The prices of the alternative fuels which have no historical base are based on the estimates given in Reference 14.17. This report by Exxon is one of the most recent documents on the costs of alternative fuels. It has been assumed that the correlation between the Wholesale Price Index and the individual fuels will be essentially the same as it is now.

E. HISTORICAL FUEL COSTS

The current dollar and the constant 1967 dollar wholesale prices of twelve fuels are listed in Table 14-1 for the period 1960 to 1980. The

Table 14-1. Historical Wholesale Prices of Fuels

	Gasoline, Ref.		Diesel No. 2		No. 6 Fuel Oil, L.S.	
Year	Current Dollars (per gal)	Constant 1967$	Current Dollars (per gal)	Constant 1967$	Current Dollars (per gal)	Constant 1967$
1960	.210	.221	.095	.100		
1961	.205	.217	.099	.105		
1962	.204	.215	.092	.097		
1963	.201	.213	.092	.097		
1964	.200	.211	.086	.091		
1965	.207	.214	.090	.093		
1966	.216	.216	.094	.094		
1967	.226	.226	.100	.100	.058	.058
1968	.229	.224	.103	.100	.058	.057
1969	.238	.224	.101	.095	.058	.054
1970	.246	.222	.108	.098	.071	.064
1971	.252	.221	.116	.102	.100	.088
1972	.245	.205	.117	.104	.113	.095
1973	.269	.200	.135	.100	.128	.095
1974	.404	.252	.213	.133	.257	.161
1975	.454	.260	.312	.178	.257	.147
1976	.474	.260	.331	.181	.290	.158
1977	.506	.261	.370	.190	.320	.165
1978	.516	.246	.374	.179	.304	.145
1979	.698	.296	.582	.247	.473	.201

Table 14-1. Historical Wholesale Prices of Fuels (cont'd)

	Methanol From Nat. Gas		Ethanol 190 Proof		Industrial Natural Gas	
Year	Current Dollars (per gal)	Constant 1967$	Current Dollars (per gal)	Constant 1967$	Current Dollars (Per 1000 ft^3)	Constant 1967$
1960	.30	.32	.52	.55		
1961	.30	.32	.52	.55		
1962	.30	.32	.52	.55		
1963	.30	.32	.52	.55		
1964	.30	.32	.52	.55		
1965	.27	.28	.52	.54		
1966	.27	.27	.52	.52		
1967	.27	.27	.52	.52		
1968	.25	.24	.52	.51		
1969	.25	.24	.52	.49	.40	.38
1970	.27	.24	.54	.49	.45	.40
1971	.25	.22	.54	.47	.49	.43
1972	.10	.08	.54	.45	.53	.44
1973	.14	.10	.54	.40	.52	.39
1974	.22	.14	.65	.41	.63	.35
1975			1.00	.57	.74	.42
1976					.95	.52
1977					1.34	.69
1978					1.54	.74
1979					2.01	.85

prices paid by the railroads for Diesel No. 2 is closer to the wholesale price of this fuel than to the retail price. Therefore, all the prices in this table are the wholesale prices.

The comparison between the current dollar price and the constant dollar price indicates that the present high prices are due more to inflation than to any other factor including OPEC. Different fuels react differently over a long period of time. Between 1960 and 1978, the constant dollar price of gasoline increased only 11.5% while the current dollar price jumped by a factor of 2.46. Diesel No. 2 during the same time period increased by a factor of 1.79 in constant dollars and 6.13 in current dollars. The faster rise in Diesel prices as compared to gasoline is due, primarily, to the increased demand for Diesel fuel and other middle distillate fuels. The increase in demand will probably continue in the future since Diesel passenger cars are becoming numerous. The price of Diesel fuel and gasoline will probably be equal sometime before 1985.

Table 14-1. Historical Wholesale Prices of Fuels (cont'd)

Year	Propane Current Dollars (per gal)	Propane Constant 1967$ (per gal)	N - Butane Current Dollars (per gal)	N - Butane Constant 1967$ (per gal)	Ammonia Current Dollars (per lb)	Ammonia Constant 1967$ (per lb)
1960					.070	.074
1961					.071	.075
1962					.067	.071
1963					.064	.068
1964					.063	.066
1965			.058	.060	.061	.063
1966			.058	.058	.060	.060
1967			.053	.053	.056	.056
1968			.053	.052	.046	.045
1969			.053	.050		
1970			.049	.044		
1971			.053	.046		
1972			.049	.041		
1973			.068	.051		
1974			.282	.176		
1975	.20	.11	.194	.111		
1976	.21	.11	.219	.120		
1977	.25	.13	.254	.131		
1978	.24	.12	.230	.110		
1979	.30	.12	.458	.194		

Some fuels have actually dropped in price during this 20 year period. The alcohols dropped significantly in price between 1960 and 1974. The price of methanol produced from natural gas dropped by a factor of three between 1960 and 1972 in current dollars and by nearly a factor of four when measured in constant dollars. The price has increased since then but is still low compared to the increase in Diesel fuel prices.

The sources of the historical fuel price data are primarily from References 14-2 through 14-7. For most of the fuels, more than one source was used when possible, because the price varies depending at the point in the distribution system for which it is quoted. It has been assumed that the wholesale price to a large quantity user is close to that charged to the railroads.

The real cost of fuels is not measured in terms of cost per gallon or per pound but in terms of energy. Table 14-2 presents the historical data of Table 14-1 in terms of the cost per million Btu's. On this basis, the

Table 14-1. Historical Wholesale Prices of Fuels

Year	Kerosene Current Dollars (per gal)	Kerosene Constant 1967$ (per gal)	Coal, Bituminous Current Dollars (per ton, del.)	Coal, Bituminous Constant 1967$ (per ton, del.)	Coal, at Mine Current Dollars (per ton)	Coal, at Mine Constant 1967$ (per ton)
1960	.112	.118	7.48	7.88	4.44	4.[illegible]8
1961	.115	.122	7.49	7.93	4.44	4.70
1962	.114	.120	7.47	7.88	4.45	4.69
1963	.115	.122	7.51	7.95	4.44	4.70
1964	.109	.115	7.56	7.98	4.45	4.70
1965	.113	.117	7.57	7.84	4.44	4.60
1966	.115	.115	7.55	7.56	4.54	4.55
1967	.120	.120	7.62	7.62	4.62	4.62
1968	.120	.117	7.68	7.49	4.67	4.56
1969	.120	.112	8.09	7.60	4.99	4.68
1970	.124	.113	9.67	8.76	6.26	5.67
1971	.129	.113	10.77	9.46	7.07	6.21
1972	.129	.108	11.33	9.51	7.66	6.43
1973	.141	.105	12.24	9.10	8.53	6.34
1974	.240	.150	20.46	12.78	15.75	9.84
1975	.274	.157	24.46	13.98	19.23	11.00
1976	.317	.173	25.18	13.76	19.43	10.62
1977	.358	.184				
1978	.372	.178				
1979			28.97	12.28	22.35	9.48

lowest cost fuel is coal, particularly coal at the mine mouth. These figures are the national averages for all bituminous coal. The growth of the national average coal and Diesel fuel prices is illustrated in Figure 14-1.

The ratio of the Diesel oil price to the coal price for 1979 is 4.8 to 1 at the mine mouth and 3.7 to 1 for coal delivered to a power plant or other major user. The price of coal and oil varies widely through-out the country as shown in Table 14-3. In the mountain states, the delivered price of coal averaged about 70 cents per million Btu in November 1979. The delivery cost by rail is about 30 cents per million Btu so the mine mouth cost for coal is about 40 cents per million Btu while Diesel No. 2 is about 490 cents per million Btu. In this part of the country, the energy from coal is only one-ninth to one-eleventh the cost of Diesel No. 2. In the South Atlantic states, the energy cost ratio of Diesel No. 2 to coal is about four to one. The economic advantage of coal compared to Diesel fuel is strongly dependent on the location of the railroad. This is also

Table 14-2. Historical Wholesale Energy Prices of Fuels

Year	Gasoline, Reg. Current Dollars (per million Btu)	Gasoline, Reg. Constant 1967$ (per million Btu)	Diesel No. 2 Current Dollars (per million Btu)	Diesel No. 2 Constant 1967$ (per million Btu)	No. 6 Fuel Oil, L.S. Current Dollars (per million Btu)	No. 6 Fuel Oil, L.S. Constant 1967$ (per million Btu)
1960	1.80	1.89	.73	.77		
1961	1.76	1.86	.76	.80		
1962	1.74	1.84	.70	.74		
1963	1.72	1.82	.70	.74		
1964	1.71	1.80	.66	.70		
1965	1.77	1.83	.65	.71		
1966	1.84	1.85	.72	.72		
1967	1.93	1.93	.77	.77	.42	.42
1968	1.96	1.91	.79	.77	.42	.41
1969	2.04	1.92	.77	.73	.42	.39
1970	2.10	1.90	.83	.75	.51	.46
1971	2.16	1.89	.89	.78	.72	.63
1972	2.09	1.76	.90	.80	.81	.68
1973	2.30	1.71	1.03	.77	.92	.68
1974	3.46	2.16	1.63	1.02	1.85	1.16
1975	3.89	2.22	2.39	1.36	1.85	1.06
1976	4.06	2.22	2.53	1.39	2.09	1.14
1977	4.33	2.23	2.83	1.46	2.30	1.19
1978	4.42	2.11	2.86	1.37	2.19	1.05
1979	5.97	2.53	4.46	1.89	3.40	1.44

true of the other fuels and will be for the new synthetic hydrocarbons and other alternative fuels.

F. LOCOMOTIVE COSTS

In 1980, the cost of a 3000 hp Diesel-electric locomotive is between $750,000 and $850,000. The lower figure applies to four-axle locomotives and the higher figure is for six-axle locomotives. These figures can vary by as much as $50,000 depending on the optional equipment selected by the railroad.

The total cost is broken down into component costs in Table 14-4. This table is not for any specific manufacturer or model of locomotive. The data was obtained from a number of sources and assembled into this table. The difference in cost between this four-axle locomotive and a six-axle unit is in the motors (two more), truck and axles (longer trucks and two more axles) and controls.

Table 14-2. Historical Wholesale Energy Prices of Fuels (cont'd)

Year	Methanol From Natural Gas		Ethanol, 190 Proof		Industrial Natural Gas	
	Current Dollars (per million Btu)	Constant 1967$ (per million Btu)	Current Dollars (per million Btu)	Constant 1967$ (per million Btu)	Current Dollars (per million Btu)	Constant 1967$ (per million Btu)
1960	5.27	5.55	7.16	7.55		
1961	5.27	5.59	7.16	7.58		
1962	5.27	5.55	7.16	7.56		
1963	5.27	5.59	7.16	7.58		
1964	5.27	5.57	7.16	7.56		
1965	4.75	4.92	7.16	7.41		
1966	4.75	4.75	7.16	7.18		
1967	4.75	4.75	7.16	7.16		
1968	4.39	4.29	7.16	6.99		
1969	4.39	4.13	7.16	6.72	.45	.43
1970	4.75	4.31	7.44	6.74	.50	.45
1971	4.39	3.87	7.44	6.53	.55	.48
1972	1.76	1.48	7.44	6.24	.59	.50
1973	2.46	1.83	7.44	5.54	.58	.43
1974	3.87	2.41	8.96	5.59	.70	.44
1975			13.78	7.88	.83	.47
1976					1.06	.58
1977					1.50	.78
1978					1.73	.82
1979					2.25	.95

G. MAINTENANCE COSTS

The locomotive maintenance cost on the railroads varies widely and the units in which the costs are reported vary widely as well. One recent report (Ref. 14-9) estimates the maintenance costs at $1.27 per thousand gross ton-miles. Reference 14-10 (1949) concludes that in comparable service on the New York Central Railroad, steam and Diesel engines cost about 35 cents/mi. Locomotive maintenance costs are reported in Ref. 14-11 at 20 cents/gal in 1974. One railroad reported its maintenance cost at $38,000 per year per locomotive in 1979. In 1976, the cost was 55 to 72 cents/mi according to Ref. 14-12. Shown here are the annual maintenance costs per locomotive reported by five railroads for 1977 (Ref. 14-11).

Table 14-2. Historical Wholesale Energy Prices of Fuels (cont'd)

	Propane		N - Butane		Ammonia	
Year	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$
1960					1.57	1.65
1961					1.58	1.67
1962					1.49	1.57
1963					1.42	1.51
1964					1.40	1.48
1965			.61	.63	1.36	1.40
1966			.61	.61	1.32	1.33
1967			.55	.55	1.26	1.26
1968			.55	.54	1.02	0.99
1969			.55	.52		
1970			.51	.46		
1971			.55	.48		
1972			.51	.43		
1973			.71	.53		
1974			2.95	1.84		
1975	2.33	1.34	2.03	1.16		
1976	2.43	1.34	2.29	1.25		
1977	2.95	1.52	2.65	1.37		
1978	2.84	1.36	2.40	1.15		
1979	3.48	1.48	4.79	2.03		

Railroad	1977 Maintenance Cost
Southern Railway	$32,818
Union Pacific	47,000
Southern Pacific	58,399
Denver, Rio Grande, Western	37,186
SOO Lines	31,432
Average	$41,367

As is obvious here, the maintenance costs vary widely from railroad to railroad. Using the data presented, the average cost of maintenance for a line haul 3000 hp locomotive is estimated to be $55,000 per year in 1980 dollars.

Table 14-2. Historical Wholesale Energy Prices of Fuels (cont'd)

	Kerosene		Bitum. Coal, Delivered		Bitum. Coal, At Mine	
Year	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$
1960	.87	.91	.312	.328	.185	.195
1961	.89	.94	.312	.330	.185	.196
1962	.88	.93	.311	.328	.185	.196
1963	.89	.94	.313	.331	.185	.196
1964	.85	.89	.315	.333	.185	.196
1965	.87	.90	.315	.326	.185	.191
1966	.89	.89	.315	.315	.189	.190
1967	.93	.93	.317	.317	.192	.192
1968	.93	.91	.320	.312	.195	.190
1969	.93	.87	.337	.316	.208	.195
1970	.96	.87	.403	.365	.261	.236
1971	1.00	.88	.449	.394	.295	.259
1972	1.00	.84	.472	.396	.319	.268
1973	1.09	.81	.510	.379	.355	.264
1974	1.86	1.16	.852	.532	.656	.410
1975	2.12	1.21	1.019	.583	.801	.458
1976	2.45	1.34	1.049	.573	.810	.442
1977	2.78	1.43				
1978	2.88	1.38				
1979			1.207	.512	.931	.395

H. OPERATING COSTS

As defined here, operating costs include miscellaneous supplies such as water and sand used on the locomotive as well as fuel handling costs. The cost of fuel is treated separately. The cost of right-of-way, supervision, yard, and train crews other than fueling crews are not considered here. The reported operating costs vary as widely as the reported maintenance costs. In addition, some railroads include only direct labor (train crews) and others exclude all labor costs.

The operating costs for this study include the following items on a annual basis.

Fuel Handling	$ 8,000
Misc. Supplies	5,000
	$13,000 per year

The train crews have been excluded because the crew size is independent of the number of locomotives used on a train and, thus, cannot be apportioned on a per locomotive basis. It should be noted that what is frequently called cost of operations or operating cost in the open literature is the sum of the labor costs, maintenance costs, and operating costs as defined in this study. These costs are divided up in this report so that the effects of engine and fuel changes can be better evaluated.

I. FINANCIAL COSTS

In addition to the material costs, the cost of capital must be included in the analysis. The two main costs on capital are the returns on equity

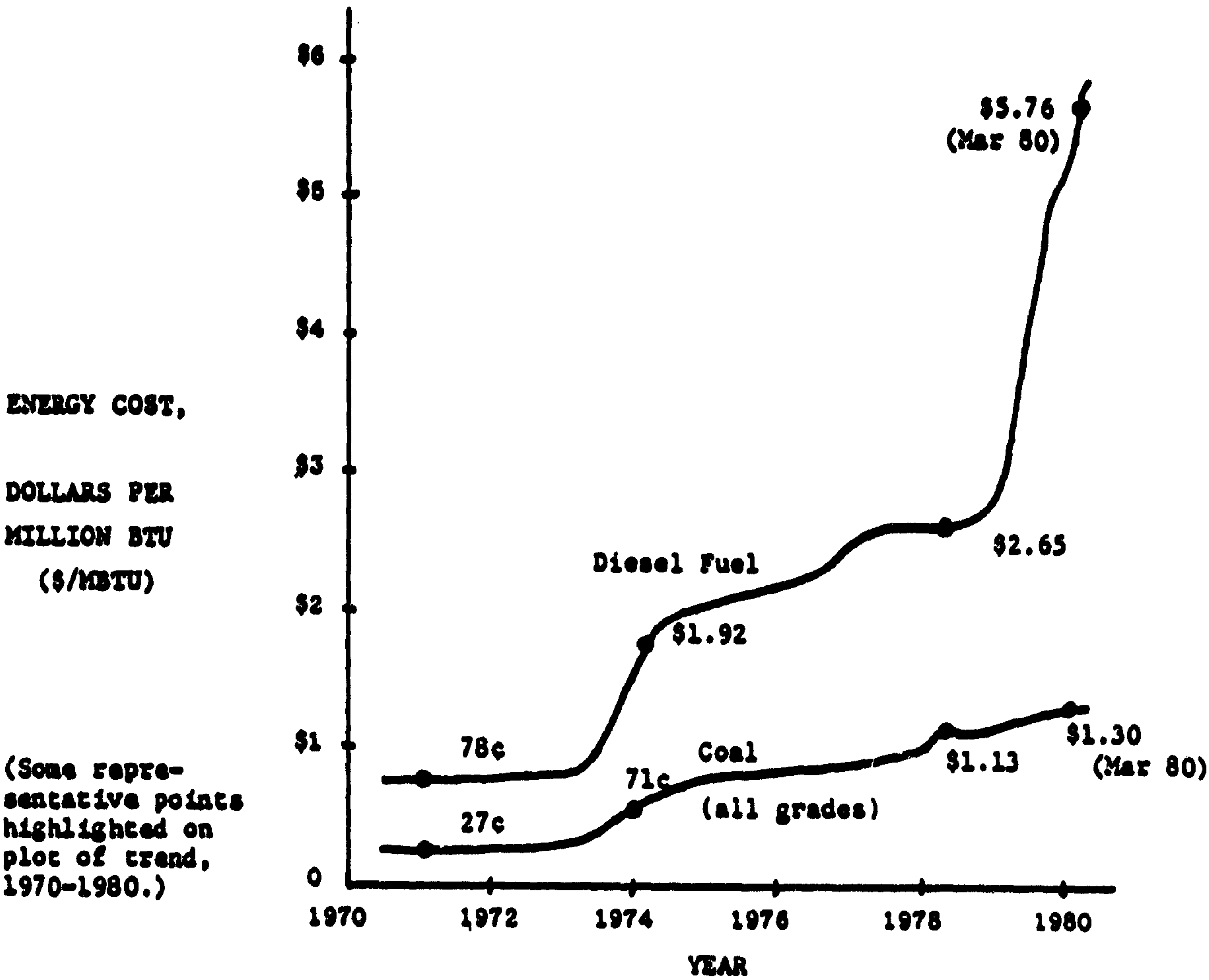

Figure 14-1. Average Energy Cost Comparison For Coal and Diesel Fuel (From Ref. 14-8)

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Table 14-3. Cost of Fossil Fuels Delivered to Steam-Electric Utility Plants (From Ref. 14-2)

All Fossil Fuels	1978		1979										
Region	NOV	DEC	JAN	FEB	MAR	APR	MAY	JUNE	JULY	AUG	SEPT	OCT	NOV
	Cents per million Btu												
New England	192.9	207.5	206.8	223.3	249.2	244.9	267.4	283.6	302.9	313.0	319.2	326.1	338.0
Middle Atlantic	159.6	163.5	170.2	180.5	174.4	168.2	176.7	184.3	212.0	204.7	202.5	196.8	215.2
East North Central	132.5	137.0	142.5	146.9	143.5	140.7	145.1	144.0	150.9	146.9	150.3	151.6	164.9
West North Central	100.7	105.9	121.6	124.3	106.9	107.3	110.9	114.4	110.3	112.1	107.5	108.0	110.0
South Atlantic	147.8	154.6	158.9	163.3	168.3	168.2	172.7	185.0	197.7	187.9	189.3	189.5	144.8
East South Central	125.4	128.3	129.7	128.1	131.7	132.4	137.5	136.9	144.0	143.3	142.8	142.4	146.5
West South Central	129.4	131.7	144.4	143.6	139.6	141.7	155.7	158.7	166.5	154.0	149.1	152.5	152.1
Mountain	82.3	82.8	89.3	91.4	92.3	99.7	120.3	101.6	100.8	100.8	102.2	105.2	101.2
Pacific	245.2	245.8	245.9	243.1	234.3	240.8	242.2	250.9	263.6	274.1	280.9	283.5	316.8
NATIONAL AVG.	138.8	142.9	150.4	154.3	152.3	151.4	158.0	161.2	168.7	167.1	167.9	167.3	171.5
Coal													
New England	147.0	146.8	147.1	150.3	149.9	150.9	152.7	156.2	155.5	155.7	156.9	156.7	155.8
Middle Atlantic	120.6	120.3	121.2	122.5	123.7	121.9	120.4	122.8	129.6	123.8	127.7	126.6	126.3
East North Central	123.9	123.8	124.3	123.7	126.7	129.0	131.4	130.6	137.0	134.3	138.4	140.9	139.1
West North Central	95.2	95.1	96.0	95.3	95.6	98.5	100.6	106.9	103.6	98.5	100.5	102.2	102.8
South Atlantic	134.1	138.8	136.6	136.4	136.0	137.8	139.0	[illegible]	142.9	142.7	144.1	145.1	145.9
East South Central	120.8	122.6	122.6	121.3	125.8	129.6	132.7	[illegible]	134.7	134.2	136.4	136.3	141.1
West South Central	73.4	81.4	88.2	89.3	92.9	94.9	89.9	99.8	99.0	100.2	98.0	104.4	113.6
Mountain	60.2	58.7	62.6	62.9	65.0	74.0	97.8	69.3	65.4	66.8	69.5	77.0	73.7
Pacific	78.2	78.6	84.3	82.9	83.4	82.7	83.0	84.6	84.2	82.0	90.2	81.7	82.1
NATIONAL AVG.	115.6	115.9	115.8	114.6	116.8	120.1	123.4	121.8	122.2	122.5	125.3	127.4	[illegible]
Residual Fuel Oil[1]													
New England	195.6	211.3	210.6	227.8	255.8	250.8	272.7	293.2	309.1	321.0	331.5	337.8	349.2
Middle Atlantic	224.2	226.0	232.2	243.4	266.4	273.7	279.9	305.0	325.2	338.1	347.3	357.7	[illegible]
East North Central	260.6	261.5	282.2	295.9	302.5	307.2	320.0	321.8	352.6	383.2	385.4	391.9	[illegible]
West North Central	217.6	212.6	233.9	265.4	246.4	277.0	384.5	244.7	373.0	[illegible]	451.0	391.6	406.4
South Atlantic	211.7	215.3	224.7	233.0	255.7	266.4	270.7	288.1	312.8	320.6	[illegible]	347.1	353.1
East South Central	168.8	177.4	174.7	198.3	211.6	212.1	231.8	218.9	240.2	266.3	281.0	[illegible]	289.0
West South Central	189.8	207.0	306.8	227.3	255.1	232.4	242.8	247.1	305.8	298.6	318.1	330.6	339.5
Mountain	252.0	228.2	237.3	233.6	246.4	276.5	284.3	287.8	337.2	350.0	383.2	405.9	405.0
Pacific	270.1	266.4	262.9	267.9	265.2	283.1	277.8	283.3	307.4	323.1	339.3	352.6	367.5
NATIONAL AVG.	225.6	228.7	231.8	245.6	261.4	268.0	277.7	289.3	314.7	328.0	337.8	351.4	367.1
Natural Gas[1]													
New England	187.6	193.7	208.4	219.1	224.0	233.9	250.1	263.1	[illegible]	277.5	295.4	308.0	317.3
Middle Atlantic	190.8	180.7	179.2	183.0	179.3	190.1	192.5	210.0	226.7	241.7	263.9	269.2	245.2
East North Central	201.6	209.8	217.2	241.7	242.3	244.3	247.1	231.2	222.9	258.3	278.9	253.3	261.0
West North Central	128.1	136.2	143.0	145.5	137.6	143.8	147.1	146.1	148.8	152.1	152.6	154.0	154.7
South Atlantic	109.2	105.1	94.1	103.0	118.5	119.7	123.5	126.5	155.5	155.3	160.0	158.1	138.5
East South Central	164.5	187.3	175.6	177.9	169.1	172.3	195.0	185.6	182.0	192.2	188.3	198.2	193.5
West South Central	134.8	133.9	146.2	147.6	142.5	149.2	169.2	168.5	161.3	160.4	157.1	161.3	152.9
Mountain	160.3	177.0	178.1	174.9	196.9	182.3	193.0	198.3	205.1	218.3	212.4	225.3	232.5
Pacific	222.1	227.7	231.0	224.9	222.0	221.6	225.8	238.7	245.3	246.3	248.9	255.6	283.5
NATIONAL AVG.	141.1	139.4	150.2	159.1	162.8	164.4	177.2	179.5	178.9	180.9	183.5	189.[illegible]	[illegible]

Table 14-4. Locomotive Costs (1980 Dollars)

Four-Axle, 3000 hp				
Component	Cost		% of Total	
Power Plant	$307,500		41	
Diesel Engine		$195,000		26
Cooling System		37,500		5
Fuel System		22,500		3
Auxiliary Equipment and misc.		52,500		7
Electrical	$255,000		34	
Alternator		$ 60,000		8
Rectifier		22,500		3
Motors, each		30,000		16
Controls and misc.		30,000		4
Dynamic Braking System		22,500		3
Chassis	$187,500		25	
Frame		$ 37,500		5
Trucks and Axles		82,500		11
Cab Equipment		15,000		2
Superstructure		37,500		5
Misc. (Sand box, Lights, etc.)		15,000		2
Total	$750,000			

and debt. The return on equity covers both the return on the common stock and on the preferred stock. The debt consists of both long- and short-term debts. Table 14-5 shows the trends in the prime rate for the 1960 to 1979 period. The interest rate that a railroad pays on a debt secured by a locomotive is usually lower than the prime rate. A locomotive is a movable asset and it is, therefore, easier to sell if necessary than a building, for example. The interest rate, of course, depends on other factors as well, such as the financial health of the railroad itself. Under normal conditions, the interest rate will be nearer the commercial paper rate than the bank prime rate.

J. WHOLESALE PRICE INDEX PREDICTIONS

In order to arrive at values for the various escalation rates, it is necessary to predict the actions of the economy over the 20 year lifetime of the locomotives. The measure of the economy adopted for this study is the Wholesale Price Index. Historically, the economy is characterized by alternating periods of slow and rapid changes. The period of these changes, rapid change to slow and back to rapid, is anything from 20 to 60 years. For this study, the period has been assumed to be about 40 years and that 1980 is about in the middle of a rapid change era. The Wholesale Price Index for the 1960 to 1979 20 year period is shown in Figure 14-2. This index changed slowly over the first 10 years and during the second decade it changed progressively faster as inflation soared. Three scenarios have been projected. All of them assume that the index will increase over the next decade but at a gradually reducing rate of change. The decade starting about 10 years from now is expected to be marked by a very low rate of change, comparable to that of the early 1960s.

The three scenarios assume that there will be no major disruptions such as all-out war, a depression comparable to that of the thirties, or inflationary rates over 20% per year. Figure 14-3 shows the predicted values of the Wholesale Price Index for a 20 year period using the 1960 to 1979 period as a base. The base year for the index is 1967.

Table 14-5. Historical Financial Rates

Year	Commercial Paper Rate (%)	Bank Prime Rate (%)
1960	3.85	4.82
1961	2.97	
1962	3.26	
1963	3.55	
1964	3.97	
1965	4.38	4.54
1966	5.55	
1967	5.10	
1968	5.90	
1969	7.83	
1970	7.72	7.91
1971	5.11	
1972	4.69	5.25
1973	8.15	8.03
1974	9.87	10.81
1975	6.33	7.86
1976	5.35	6.84
1977	5.60	6.83
1978	7.99	9.06
1979	10.91	11.75

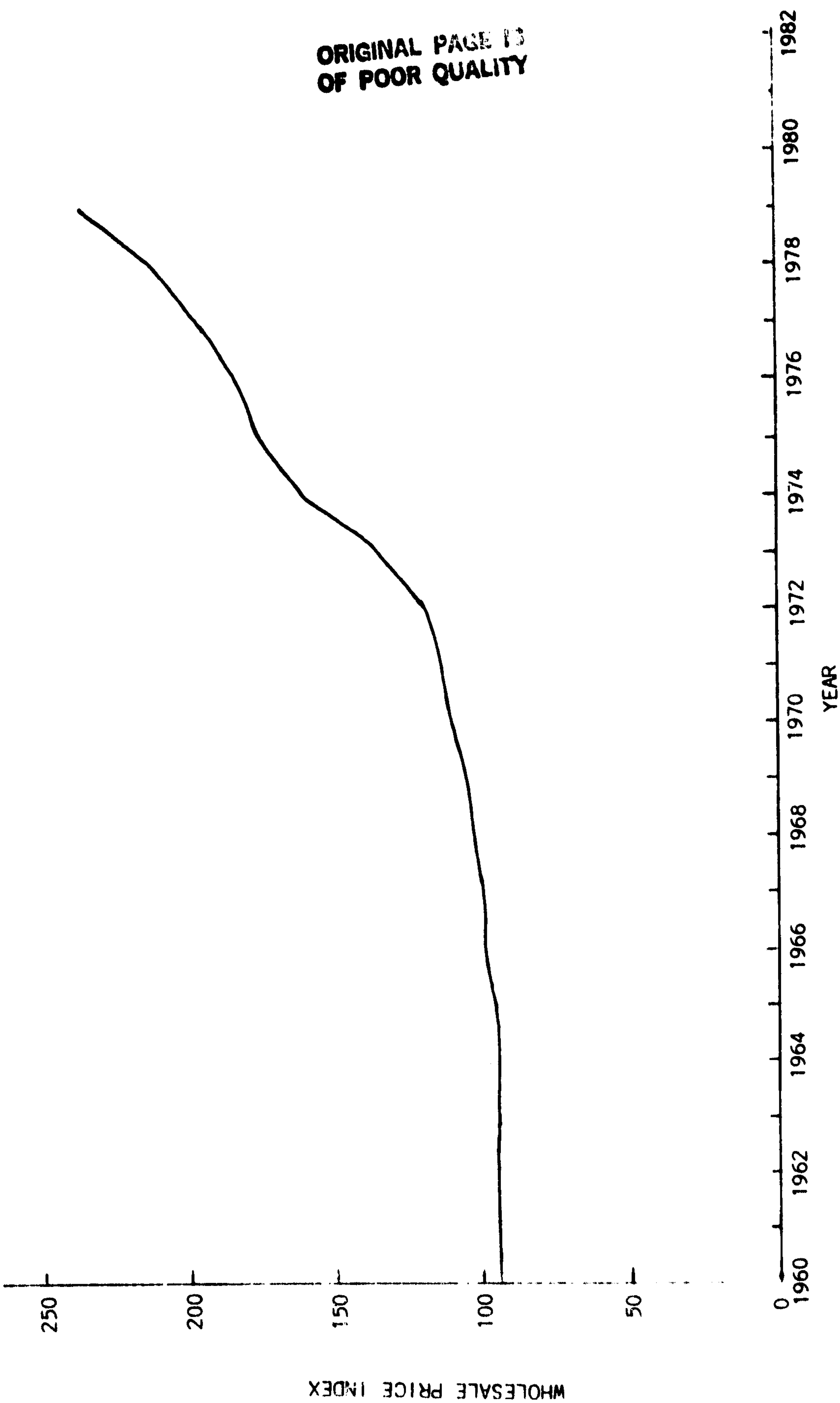

Figure 14-2. Historical Performance of The Wholesale Price Index

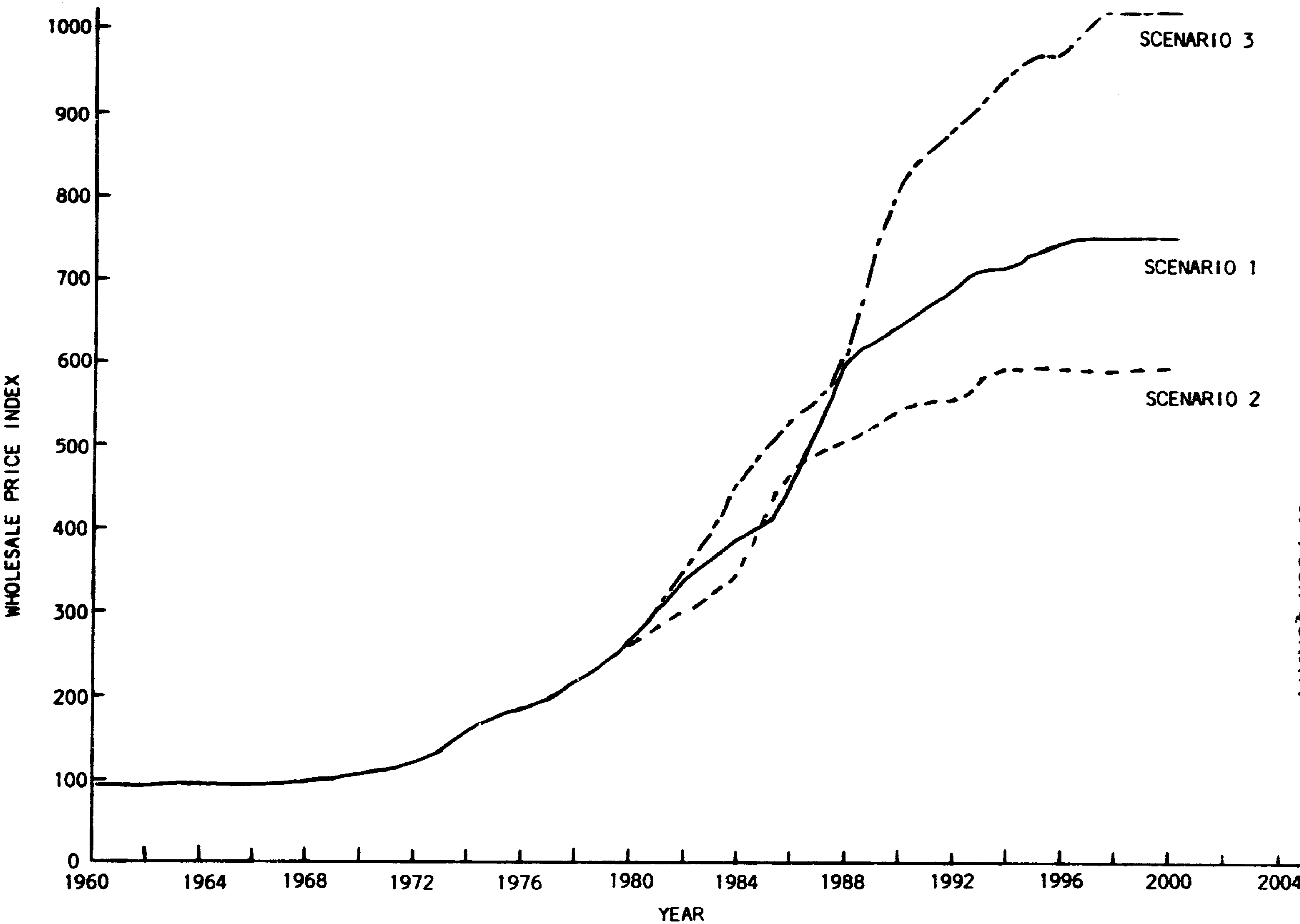

Figure 14-3. Predicted Performance of Wholesale Price Index

The first scenario assumes that 1980 is the mid-point of the economic cycle and that change between the index for 1981 and 1980 is the same as that estimated for 1979 to 1980. In other words, since there was an estimated change of 12.93% in the index between 1979 and 1980, there will also be a 12.93% change between 1980 and 1981. Between 1978 and 1979, there was on 11.59% change. The results of this calculation is shown in Table 14-6 and in Figure 14-3 for the period 1960 to 2000.

The second scenario uses the same data but assumes that the 1979 to 1980 change is the same as the 1978 to 1979 change. The 1980 to 1981 change will be 7.69%, the same as the 1977 to 1978 change. This scenario is also tabulated in Table 14-6.

The third scenario goes the other way. It assumes that the 1980 to 1981 change is 14.5% and the 1981 to 1982 change is this same amount. The 1982 to 1983 change would be 12.93%, the same as estimated for 1979 to 1980. This scenario is tabulated in Table 14-6 and shown in Figure 14-3.

Of the three scenarios, the first scenario appears to be the most probable. Events of 1980 indicate the change implicit in scenario 2 did not take place. The price index change for the 1979 to 1980 period is greater than the 1978 to 1979 period. At the other extreme, scenario 3 assumes double-digit inflation for six consecutive years during which time the index would have more than doubled. In reality, such a prolonged period of high inflation would be so disruptive that it would negate the assumptions on which these predictions are based. Figure 14-3 also shows how just a few years of inflation can profoundly affect prices many years in the future even when a return to a stable economy is assumed.

K. PREDICTION METHODOLOGY

The Wholesale Price Index is a measure of inflation for a particular area of the economy. If the price of a product rises at the same rate as the price index, then its constant dollar price does not change. If the constant dollar price increases, then the current price of the product is rising faster than the average price in the general economy. Conversely, if the constant dollar price decreases, then the current dollar price increases slower than the index. If the constant dollar price decreases fast enough, the current dollar price decreases as well. Electronic hand calculators are a good example of how both current and constant dollar prices can decrease in an inflationary period.

The prediction methodology used in this study is based on changes in the constant dollar price with time. For example, the use of Diesel fuel and the other distillate fuels are expected to increase faster than the use of gasoline. If it is assumed that the price of gasoline increased exactly at the inflation rate, its constant dollar price change is zero and the price of Diesel fuel will have a positive change. Some of the alternative fuels may have negative changes as the start-up costs are paid off and productivity improves. Volume production will tend to decrease the constant dollar costs.

Table 14-6. Wholesale Price Index Predictions

Year	Scenario 1		Scenario 2		Scenario 3	
	WPI	% Increase	WPI	% Increase	WPI	% Increase
1960	94.9		94.9		94.9	
1961	94.5	-0.42	94.5	-0.42	94.5	-0.42
1962	94.8	0.32	94.8	0.32	94.8	0.32
1963	94.5	-0.32	94.5	-0.32	94.5	-0.32
1964	94.7	0.21	94.7	0.21	94.7	0.21
1965	96.6	2.01	96.6	2.01	96.6	2.01
1966	99.8	3.31	99.8	3.31	99.8	3.31
1967	100.0	0.20	100.0	0.20	100.0	0.20
1968	102.5	2.50	102.5	2.50	102.5	2.50
1969	106.5	3.90	106.5	3.90	106.5	3.90
1970	110.4	3.66	110.4	3.66	110.4	3.66
1971	113.9	3.17	113.9	3.17	113.9	3.17
1972	119.1	4.57	119.1	4.57	119.1	4.57
1973	134.7	13.10	134.7	13.10	134.7	13.10
1974	160.1	18.86	160.1	18.86	160.1	18.86
1975	174.9	9.24	174.9	9.24	174.9	9.24
1976	182.9	4.57	182.9	4.57	182.9	4.57
1977	196.3	7.33	196.3	7.33	196.3	7.33
1978	211.4	7.69	211.4	7.69	211.4	7.69
1979	235.9	11.59	235.9	11.59	235.9	11.59
1980	266.4	12.93	263.2	11.59	266.4	12.93
1981	300.8	12.93	283.5	7.69	305.0	14.50
1982	335.7	11.59	304.3	7.33	349.2	14.50
1983	361.5	7.69	318.2	4.57	394.4	2.93
1984	388.0	7.33	347.6	9.24	457.1	11.59
1985	405.8	4.57	413.2	18.86	492.3	7.69
1986	443.3	9.24	467.3	13.10	528.4	7.33
1987	526.9	18.86	488.7	4.57	552.5	4.57
1988	595.9	13.10	504.2	3.17	603.6	9.24
1989	623.1	4.57	522.7	3.66	717.4	18.86
1990	642.9	3.17	543.1	3.90	811.4	13.10
1991	666.4	3.66	556.7	2.50	848.5	4.57
1992	692.4	3.90	557.8	0.20	875.4	3.17
1993	709.7	2.50	576.3	3.31	907.4	3.66
1994	711.1	0.20	587.9	2.01	942.8	3.90
1995	734.6	3.31	589.1	0.21	966.4	2.50
1996	749.4	2.01	587.2	-0.32	968.3	0.20
1997	751.0	0.21	589.1	0.32	1000.4	3.31
1998	748.6	-0.32	586.6	-0.42	1020.5	2.01
1999	751.0	0.32	587.2	0.11	1022.6	0.21
2000	747.8	-0.42	588.4	0.21	1019.3	-0.32

L. FUEL PRICE PREDICTIONS

There have been, in the last eight years, numerous attempts to predict the price of fuels, particularly petroleum fuels. Figure 14-4 shows the prediction for Diesel fuel made by Arthur D. Little, Inc. in 1976 (Ref. 14-12). Since the actual 1980 price is between 80 and 90 cents per gallon, the predicted price is far too low. Even predictions made in 1979 as shown in Figure 14-5 (from Ref. 14-14) are too low. The 1980 price of 30 to 32 dollars a barrel for oil is well above the highest predicted price on this figure by nearly 20%. However, if these prices are converted from 1978 to 1980 dollars, then the highest price line matches the actual prices. All of the other lines still fall well below the real prices.

Exxon Research and Engineering has made a number of studies into the availability, properties, and costs of alternative fuels. Figure 14-6 was taken from a 1974 study (Ref. 14-15). The costs are expressed in 1973 dollars so it is necessary to multiply them by 1.933. In terms of 1980 dollars, the initial price of shale distillate is $3.92 per million Btu and it is $7.35 per million Btu for methanol from coal. Exxon in a 1980 report (Ref. 14-17) lists shale distillate at $4.20 to $4.40 in 1980 dollars and methanol from coal at $6.65 to $11.15 per million Btu. The differences between the 1973 and the 1980 reports is just inflation with virtually no change in the constant dollar price. All of the curves in Figure 14-6 show a decrease in constant dollar prices due to increased productivity, improved processes, and increased volume. The drop in price varies from about 15% to 25% in the 20 year span. In general, the prices all decrease together so that the relative cost at the end of the 20 year period is very nearly the same as at the beginning.

The costs of a variety of alternative fuels are shown in Table 14-7. This table, taken from Ref. 14-17, illustrates the extremely wide range of costs involved in these fuels. There is a range of nearly eighteen to one between the low of $2.55 per million Btu to the high of $45.00. Even for a single fuel, the cost estimates vary over a range of two to one. This list agrees generally with other sources.

Table 14-8 presents the 20 year predictions made by JPL for 18 fuels with prices in both current dollars and constant 1967 dollars. The table is set-up in terms of dollars per million Btus. Many of these fuels are not now available at the prices quoted. The prices are present value prices, the price they are expected to be if they were in quantity production today. This is the only way to compare them in this type of analysis. Many of the fuels, except for petroleum-based ones, are not expected to be commercially available for 10 to 15 years and then, only if they are economically practical.

The prices shown in Table 14-8 are the predictions for the fuels delivered to the railroads. They are higher than the cost of production and distribution by the amount of the profits and by the amount due to market pressures. If there are two fuels competing for the same market, such as oil shale distillate and petroleum middle distillate, they will tend towards the same price even if the cost of production and distribution are different. The prices in the tables reflect this inter-fuel competition as well as can be predicted at this time. In general, the liquid fuels are

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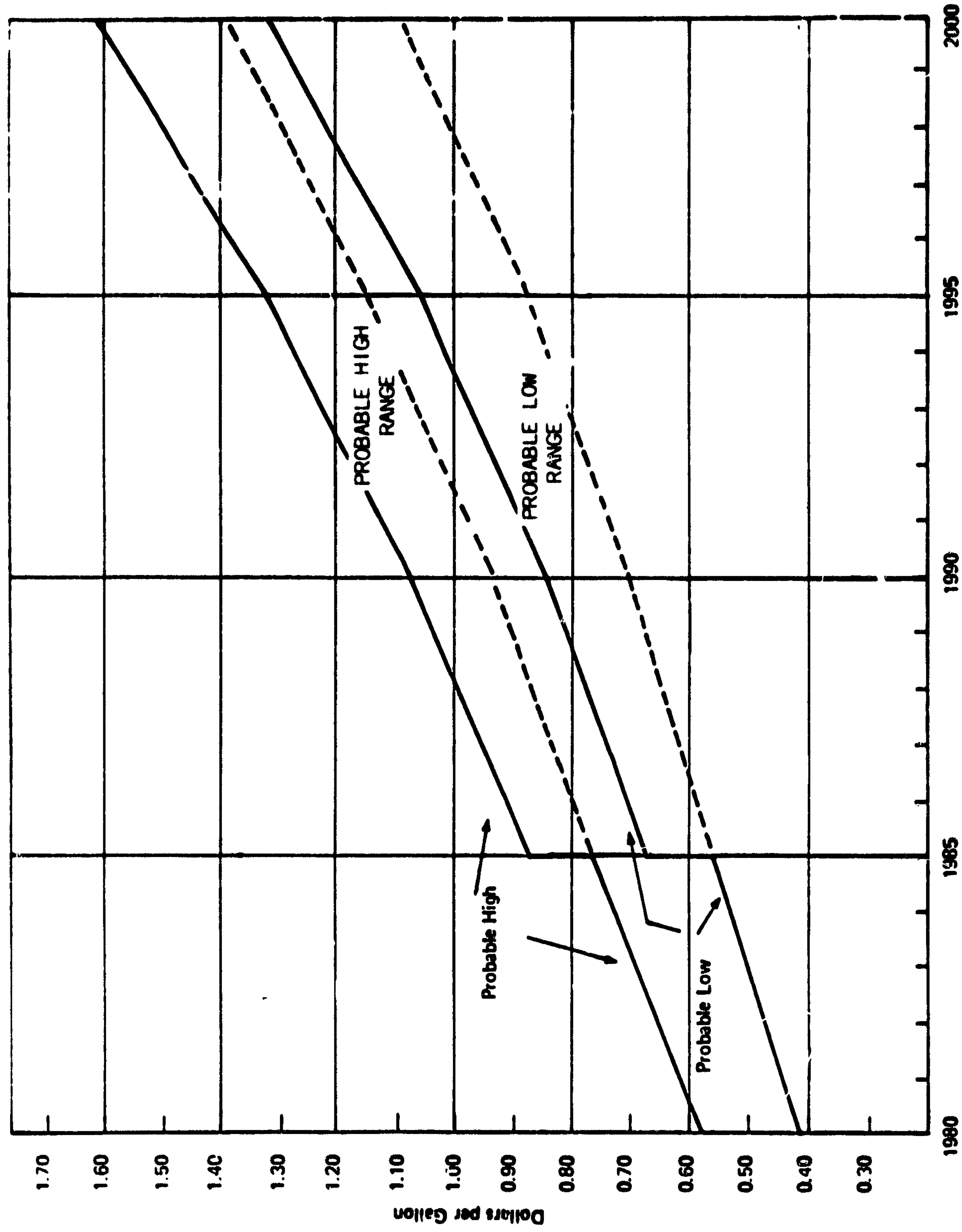

Figure 14-4. Forecast of Diesel Fuel Prices Delivered (From Ref. 14-12)

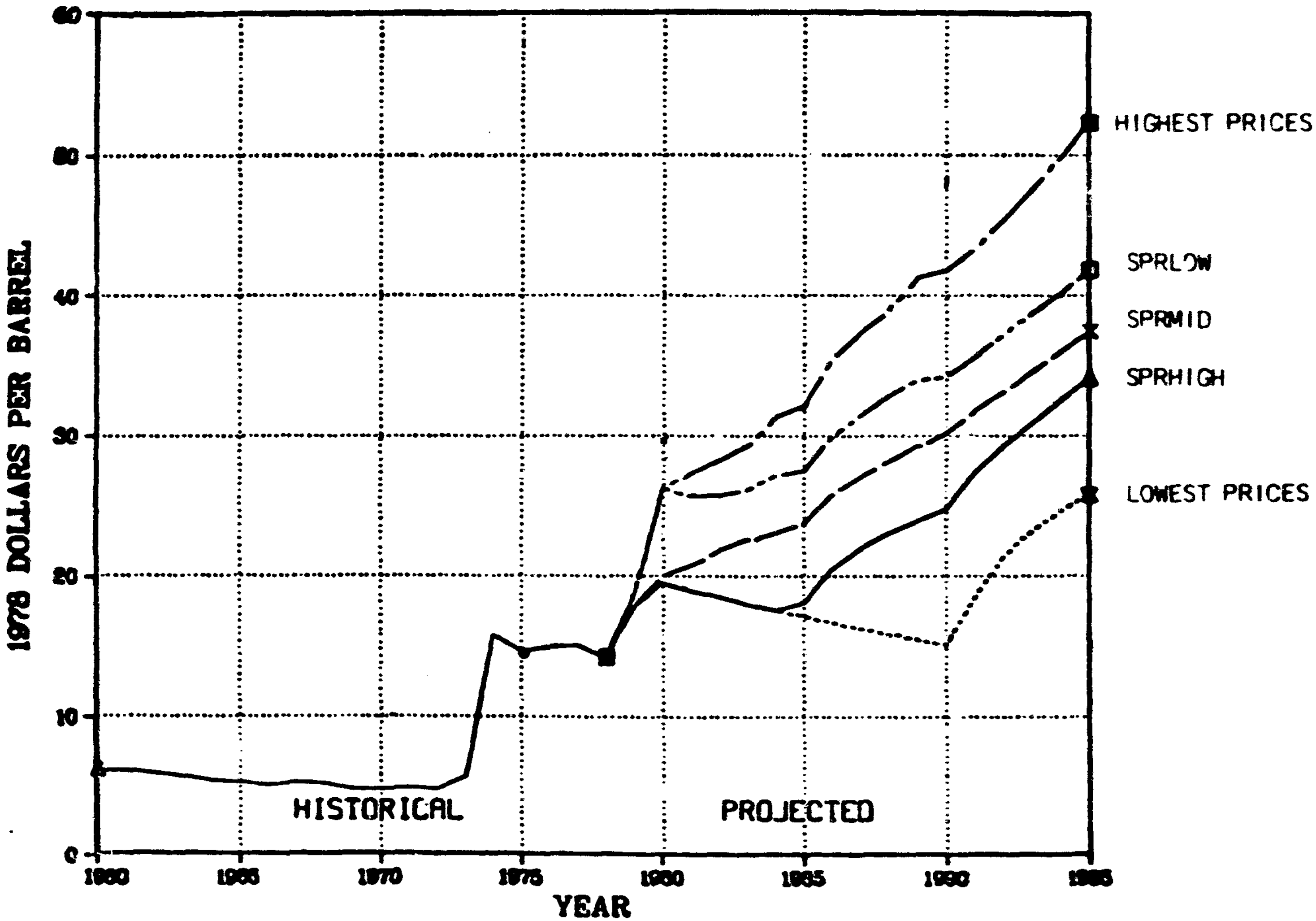

Figure 14-5. World Oil Prices, Historical and Alternative Projections 1960-1995
(From Ref. 14-14)

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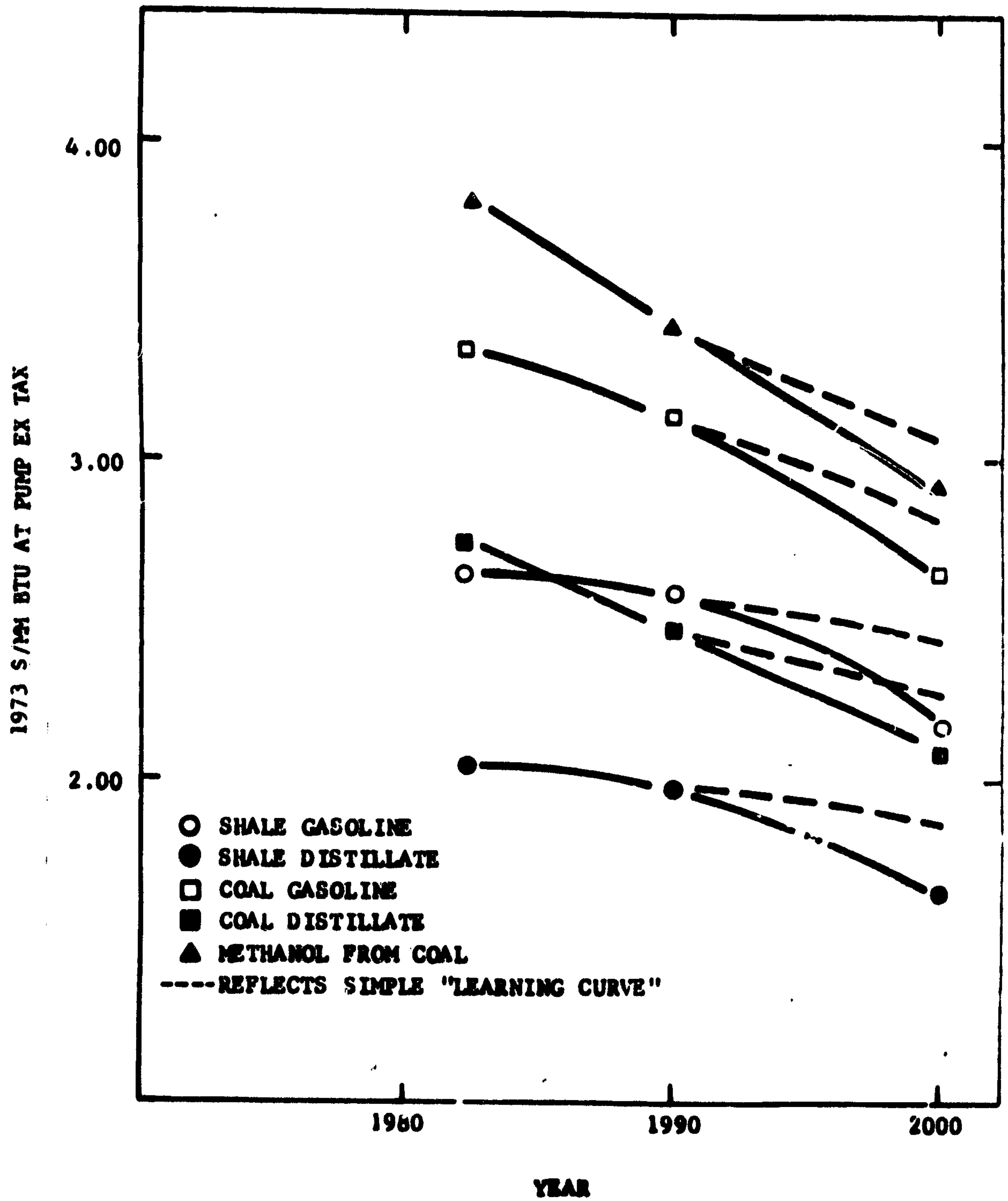

Figure 14-6. Cost Projections for Alternative Fuels (From Ref. 14-15)

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Table 14-7. Alternative Fuels Costs in Constant Dollars (From Ref. 14-17)

	Cost $/MBtu (1980$)
Shale gasoline (1)	4.90 - 5.20
Shale distillate (1)	4.20 - 4.40
Coal gasoline (2)	4.85 - 8.40
Coal distillate (2)	5.35 - 9.00
Coal (3)-in-Oil Slurry	2.55+
Methanol	
Coal (2)	6.65 - 11.15
Biomass (4)	14.80
Solid Waste Pyrolysis (4)	20.75
OTEC (5)	29.70
Hydrogen	
Coal (2)	7.90 - 9.60
Electrolysis (6)	19.65
Ammonia	
Coal (2)	12.30 - 12.90
Biomass (4)	17.55
Ethanol	
Fermentation (7)	16.75 - 21.20
Methane	
Coal (2)	5.45 - 7.95
Sewage Algae (4)	12.50
Oil from Organic Wastes (4)	8.60
Acetylene (8)	14.00
Methylamine (9)	>14.00
Hydrazine (9)	>45.00
Low Btu gas (10)	4.25

Notes:
(1) Produced by above ground retorting.
(2) Coal at 21 $/Ton.
(3) Coal at 8 $/Ton.
(4) Includes waste collection cost of 20 $/Ton.
(5) Ocean thermal energy conversion.
(6) Current potassium hydroxide technology; about 13 $/MBtu for developmental SPE process.
(7) Corn at 3.75 $/Bushel.
(8) Cost about the same for calcium carbide or coal arcing technology.
(9) Current raw material costs only; higher with addition of processing and investment costs.
(10) Coal technology; coal at 21 $/Ton.

Table 14-8. Twenty Year Prediction of Fuel Energy Prices

	Diesel No. 2		Gasoline		Broadcut Fuel Oil	
Year	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$
1980	6.10	2.29	7.73	2.90	5.73	2.15
1981	8.09	2.69	9.32	3.10	7.58	2.52
1982	10.04	2.99	10.58	3.15	9.33	2.78
1983	11.39	3.15	11.57	3.20	10.56	2.92
1984	12.53	3.23	15.53	3.23	11.60	2.99
1985	13.19	3.25	13.19	3.25	12.17	3.00
1986	14.54	3.28	14.45	3.26	13.34	3.01
1987	17.34	3.29	17.18	3.26	15.86	3.01
1988	19.67	3.30	19.37	3.25	17.94	3.01
1989	20.50	3.29	20.31	3.26	18.69	3.00
1990	21.15	3.29	20.96	3.26	19.29	3.00
1991	21.86	3.28	21.79	3.27	20.06	3.01
1992	22.78	3.29	22.71	3.28	20.91	3.02
1993	23.42	3.30	23.35	3.29	21.43	3.02
1994	23.40	3.29	23.40	3.29	21.40	3.01
1995	24.10	3.28	24.10	3.28	22.11	3.01
1996	24.58	3.28	24.58	3.28	22.48	3.00
1997	24.71	3.29	24.56	3.27	22.53	3.00
1998	24.55	3.28	24.55	3.28	22.53	3.01
1999	24.71	3.29	24.71	3.29	22.61	3.01
2000	24.53	3.28	24.53	3.28	22.58	3.02

expected to increase in constant dollar price by about 50% over the next 20 years. Coal prices are only likely to increase 25 to 30%. The 25 to 30% does reflect the increasing demand for coal in the future.

There were other price predictions made during the last year. One projection was made by the Department of Energy (DOE) in October 1980 (Ref. 14-16). Another projection was by Data Resources, Inc. (DRI) who regularly issue financial predictions for a variety of business sectors including energy (Ref. 14-17). The JPL, DOE, and DRI predictions are shown in Figure 14-7 in terms of constant 1980 dollars for the period from 1980 through to the year 2000. One problem with these predictions is that they are not very specific. All types of coal are handled under one heading with one price. For the petroleum fuels category, only residual, distillate, gasoline, and kerosine are included. Specific types of fuels are not listed. The JPL prediction used Diesel No. 2 for the distillate fuel

Table 14-8. Twenty Year Prediction of Fuel Energy Prices (cont'd)

	No. 6 Fuel Oil, Low Sulfur		Methane (From Coal)		Propane	
Year	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$
1980	4.58	1.72	6.10	[illegible]9	4.26	1.60
1981	6.15	2.04	7.58	[illegible]52	6.17	2.05
1982	7.63	2.27	9.03	2.69	7.39	2.20
1983	8.77	2.43	10.27	2.84	8.31	2.30
1984	9.77	2.52	11.56	2.98	9.12	2.35
1985	10.29	2.54	12.21	3.01	9.82	2.42
1986	11.49	2.59	13.52	3.05	11.08	2.50
1987	13.87	2.63	16.33	3.10	13.44	2.55
1988	15.74	2.64	18.71	3.14	15.49	2.60
1989	16.61	2.66	19.81	3.18	16.45	2.64
1990	17.34	2.70	20.83	3.24	17.23	2.68
1991	17.93	2.69	21.52	3.23	18.06	2.71
1992	18.91	2.73	22.43	3.24	19.18	2.77
1993	19.67	2.77	23.07	3.25	20.23	2.85
1994	19.66	2.76	23.11	3.25	20.48	2.88
1995	20.49	2.79	23.87	3.25	21.38	2.91
1996	21.14	2.82	24.28	3.24	21.73	2.90
1997	21.25	2.83	24.18	3.22	21.85	2.91
1998	21.36	2.85	23.81	3.18	21.78	2.91
1999	21.74	2.90	23.81	3.17	21.78	2.90
2000	21.59	2.89	23.56	3.15	21.69	2.90

projection in this figure. While the absolute cost dollars differ between the various estimates, the ratio of distillate fuel costs to coal costs are very close. The JPL estimates were made in mid - 1979 before the rapid increases in petroleum costs.

The current dollar prices increase much more rapidly because of the general inflationary trend. Using scenario 1, the price of petroleum-based Diesel No. 2 is expected to reach $3.21 per gallon by the year 2000. The constant dollar price will be 43 cents a gallon. Between the years 1967 and 2000, inflation will have raised the price of Diesel No. 2 fuel by a factor of eight over the increase due to increased demand and decreased supplies.

Table 14-8. Twenty Year Prediction of Fuel Energy Prices (cont'd)

Year	N-Butane Current Dollars (per million Btu)	N-Butane Constant 1967$ (per million Btu)	Methanol (From Coal) Current Dollars (per million Btu)	Methanol (From Coal) Constant 1967$ (per million Btu)	Ethanol Current Dollars (per million Btu)	Ethanol Constant 1967$ (per million Btu)
1980	5.85	2.20	6.65	2.50	21.20	7.96
1981	7.43	2.47	7.88	2.62	24.27	8.07
1982	9.03	2.69	9.06	2.70	27.43	8.17
1983	10.12	2.80	10.23	2.83	29.82	8.25
1984	11.06	2.85	11.42	2.94	32.36	8.34
1985	11.77	2.90	12.40	3.06	34.09	8.40
1986	13.12	2.96	13.32	3.00	37.19	8.39
1987	15.75	2.99	14.41	2.73	44.05	8.36
1988	17.88	3.00	16.09	2.70	49.76	8.35
1989	18.82	3.02	16.70	2.68	52.09	8.36
1990	19.67	3.06	17.35	2.70	53.55	8.33
1991	20.53	3.08	18.39	2.76	55.51	8.33
1992	21.12	3.05	19.21	2.77	57.47	8.30
1993	21.57	3.04	20.18	2.84	58.76	8.28
1994	21.55	3.03	21.02	2.96	58.81	8.27
1995	22.41	3.05	21.98	3.09	60.60	8.25
1996	22.93	3.06	22.86	3.05	61.83	8.25
1997	23.06	3.07	23.60	3.14	62.03	8.26
1998	22.98	3.07	23.58	3.15	61.61	8.23
1999	22.91	3.05	23.33	3.11	61.58	8.20
2000	22.81	3.05	22.25	2.98	61.32	8.20

M. LOCOMOTIVE COST PREDICTIONS

Since the entire cost of a locomotive occurs in the first year, the future prices of locomotives have no bearing on this cost analysis. However, locomotives using alternative engines or alternative fuels will not cost the same as the present Diesel-electric locomotive.

The difference between the present Diesel-electric locomotives using Diesel No. 2 and those using the alternative fuels is primarily in the fuel system and in the engine. Many fuels require the use of dual fueling. Locomotives using dual-fueled engines require a separate tank, pumps, lines, and injectors for the Diesel No. 2 fuel used for ignition. Some fuels have such low energy densities that tenders would be required to carry them. Table 14-9 presents the estimated costs of locomotives modified to use

Table 14-8. Twenty Year Prediction of Fuel Energy Prices (cont'd)

Year	Oil Shale Syn-Crude Current Dollars (per million Btu)	Oil Shale Syn-Crude Constant 1967$ (per million Btu)	Oil Shale Distillate Current Dollars (per million Btu)	Oil Shale Distillate Constant 1967$ (per million Btu)	Oil Shale Gasoline Current Dollars (per million Btu)	Oil Shale Gasoline Constant 1967$ (per million Btu)
1980	6.26	2.35	6.34	2.38	7.73	2.90
1981	7.19	2.39	7.34	2.44	9.32	3.10
1982	8.19	2.44	8.36	2.49	10.74	3.20
1983	8.97	2.48	9.18	2.54	11.75	3.25
1984	9.82	2.53	10.01	2.58	12.65	3.26
1985	10.35	2.55	10.59	2.61	13.23	3.26
1986	11.35	2.56	11.66	2.63	14.41	3.25
1987	13.54	2.57	14.07	2.67	16.[illegible]7	3.22
1988	15.43	2.59	16.09	2.70	19.07	3.20
1989	16.20	2.60	16.95	2.72	19.88	3.19
1990	16.91	2.63	17.68	2.75	20.25	3.15
1991	17.73	2.66	18.53	2.78	20.92	3.14
1992	19.11	2.76	19.46	2.81	21.60	3.12
1993	19.87	2.80	20.23	2.85	22.00	3.10
1994	19.91	2.80	20.27	2.85	21.90	3.08
1995	20.50	2.79	20.86	2.84	22.33	3.04
1996	20.83	2.78	21.21	2.83	22.71	3.03
1997	20.88	2.78	21.25	2.83	22.68	3.02
1998	20.74	2.77	21.11	2.82	22.53	3.01
1999	20.73	2.76	21.10	2.81	22.61	3.01
2000	20.56	2.75	20.94	2.80	22.43	3.00

alternative fuels. When a tender is required, it has been assumed that a single tender would serve two locomotives, so the cost charged against each locomotive is one-half the cost of the tender. Tenders are assumed to cost between $60,000 and $100,000 each depending on the type of fuel. Liquid hydrogen would be the most expensive to store and the tender for it would cost about $100,000. This tender is not intended for long-term storage, only one or two days, so it is not as well insulated as the main storage tanks at the refueling depot.

The locomotive costs range from a low of $750,000 for the four-axle 3000 hp locomotive using middle distillate fuels to a high of $869,500 for a liquid hydrogen fueled locomotive. It must be emphasized that these costs are for fully developed equipment in commercial production as expressed in 1980 dollars. The development and initial tooling costs are not included in this table.

Table 14-8. Twenty Year Prediction of Fuel Energy Prices (cont'd)

	Coal Derived Distillate		Coal Derived Gasoline		Bituminous Coal, Mine Mouth	
Year	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$
1980	6.93	2.60	6.45	2.42	1.03	.385
1981	7.97	2.65	7.55	2.51	1.19	.395
1982	9.10	2.71	8.59	2.56	1.36	.405
1983	10.05	2.78	9.47	2.62	1.49	.413
1984	11.10	2.86	10.48	2.70	1.63	.421
1985	11.77	2.90	11.08	2.73	1.74	.429
1986	13.08	2.95	12.28	2.77	1.94	.437
1987	15.75	2.99	14.75	2.80	2.33	.443
1988	18.17	3.05	16.98	2.85	2.68	.450
1989	19.32	3.10	18.07	2.90	2.84	.455
1990	20.19	3.14	18.97	2.95	2.96	.460
1991	21.19	3.18	19.86	2.98	3.10	.465
1992	22.16	3.20	20.91	3.02	3.25	.470
1993	23.07	3.25	21.72	3.06	3.37	.475
1994	23.40	3.29	21.97	3.09	3.41	.480
1995	24.24	3.30	22.92	3.12	3.56	.485
1996	24.84	3.31	23.61	3.15	3.67	.490
1997	24.78	3.30	23.88	3.18	3.72	.495
1998	24.63	3.29	23.88	3.19	3.74	.500
1999	24.63	3.28	24.03	3.20	3.79	.505
2000	24.53	3.28	23.93	3.20	3.81	.510

The capital costs of locomotives using alternative engines must also be considered. In some cases, such as the gas turbine, there is no cooling system needed. For the first generation new steam engine, the electrical transmission is eliminated but the boiler must be accounted for in the estimate of the capital cost. Table 14-10 is a list of the engines which are included in this cost analysis. The term "mature engine" in Items 4 and 9 refers to the engines discussed in Section V. Table 14-11 shows the total cost and the cost breakdown in 1980 dollars for 18 locomotives including a conventional turbocharged water cooled Diesel-electric locomotive. All locomotives are rated at 3000 hp (2700 hp at the rail) and have four axles ([illegible] configuration). The engine numbers correspond to the numbers on Table 14-10.

Table 14-8. Twenty Year Prediction of Fuel Energy Prices (cont'd)

	Bituminous Coal, Nat. Average		Hydrogen (From Coal)		Ammonia	
Year	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$	Current Dollars (per million Btu)	Constant 1967$
1980	1.28	.48	9.11	3.42	12.92	4.85
1981	1.47	.49	10.53	3.50	14.89	4.95
1982	1.68	.50	12.02	3.58	16.95	5.05
1983	1.84	.51	13.19	3.65	18.62	5.15
1984	2.02	.52	14.43	3.72	20.37	5.25
1985	2.15	.53	15.34	3.78	21.67	5.34
1986	2.39	.54	17.02	3.84	24.12	5.44
1987	2.90	.55	20.55	3.90	29.14	5.53
1988	3.34	.56	23.54	3.95	33.55	5.63
1989	3.55	.57	24.92	4.00	35.64	5.72
1990	3.66	.57	26.04	4.05	37.35	5.81
1991	3.87	.58	27.32	4.10	39.32	5.90
1992	4.09	.59	29.08	4.20	41.34	5.97
1993	4.19	.59	30.16	4.25	42.87	6.04
1994	4.27	.60	30.58	4.30	43.45	6.11
1995	4.41	.60	31.96	4.35	45.40	6.18
1996	4.50	.60	32.97	4.40	46.84	6.25
1997	4.58	.61	33.42	4.45	47.39	6.31
1998	4.57	.61	33.54	4.48	47.61	6.36
1999	4.58	.61	33.72	4.49	47.84	6.37
2000	4.56	.61	33.65	4.50	47.71	6.38

Most of the locomotives listed cost more than the present Diesel-electric unit. There are, however, five that cost less. These are the adiabatic Diesel engine powered locomotive and the first generation new steam locomotive. Their costs are within 10% of the costs of the current Diesel-electric locomotive. The adiabatic Diesel engine costs less simply because there are fewer cylinders and they do not have a cooling system. However, the cost per cylinder is higher for the adiabatic Diesel than for a conventional Diesel engine. The adiabiatic Diesel was costed at $14,000 per cylinder and the conventional Diesel at $12,000 per cylinder. Only ten cylinders are needed by the adiabatic Diesel to produce the same power level as sixteen cylinders in the conventional Diesel. This analysis assumes the same cylinder sizes and that bottoming cycles are used on the adiabatic Diesel. The cost of the bottoming cycles is estimated separately and is not part of the $14,000 per cylinder cost.

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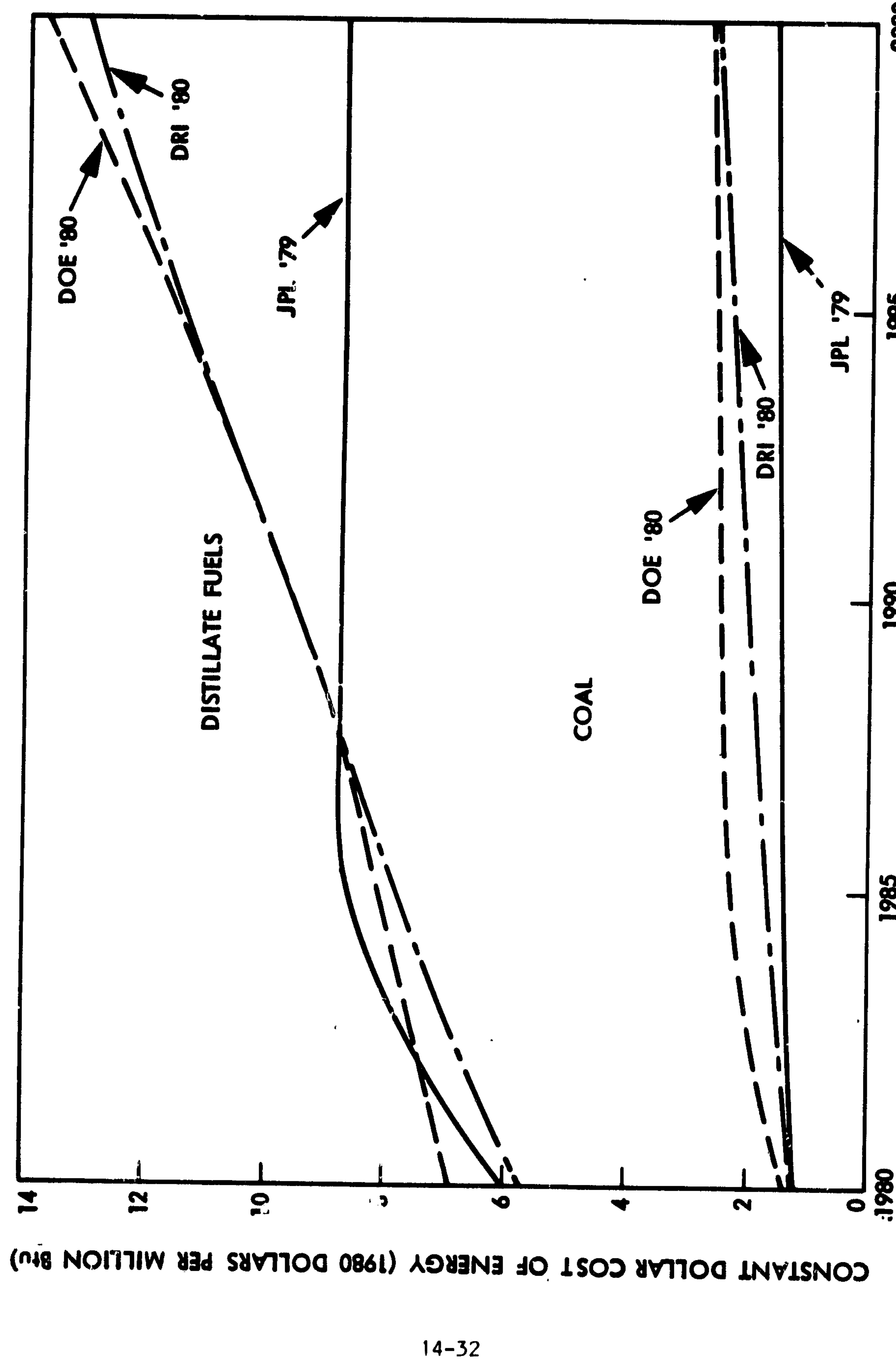

Figure 14-7. Predicted Fuel Prices to the Year 2000

Table 14-9. Capital Costs of Diesel-Electric Locomotives Modified to Use Alternative Fuels

Fuel	Diesel Engine (Cost)	Fuel Supply (System Cost)	Total Cost
1. Diesel No. 2	$195,000	$22,500	$750,000
2. Gasoline	220,000	37,000	789,500
3. Broadcut Fuel Oil	195,000	22,500	750,000
4. No. 6 Fuel Oil, L.S.[a]	200,000	37,000	769,500
5. Liquid Methane	240,000	79,000	851,500
6. Liquid Propane	230,000	69,000	831,500
7. Liquid N-Butane	230,000	69,000	831,500
8. Methanol	225,000	69,000	826,500
9. Ethanol	225,000	69,000	826,500
10. Oil Shale Syn-Crude	195,000	22,500	750,000
11. Oil Shale Distillate	195,000	22,500	750,000
12. Oil Shale Gasoline	220,000	37,000	789,500
13. Coal Derived Distillate	195,000	22,500	750,000
14. Coal Derived Gasoline	220,000	37,000	789,500
15. Liquid Hydrogen	250,000	87,000	869,500
16. Liquid Ammonia	235,000	79,000	846,500

Notes: Four-axle 3000 hp locomotives
Costs in 1980 dollars
[a] Low Sulfur

The most expensive locomotives are those using fluidized bed combustors. The most expensive, the Stirling engine with a fluidized bed combustor, is nearly 60% more expensive than the current Diesel locomotive. The justification for these expensive locomotives is that they use coal which is an inexpensive and abundant fuel.

N. MAINTENANCE AND OPERATING COST PREDICTIONS

It is assumed that the maintenance and operating cost will rise at the same rate as the Wholesale Price Index. This means that constant dollar cost will remain the same over the 20 year period although the current dollar price will rise. These costs will vary from engine to engine and from fuel to fuel.

Table 14-12 presents the estimates for both maintenance and operating costs for 17 fuels on a 1980 dollars per year basis. The variation in maintenance is the result of the need for dual-fuel operation for many fuels and for tenders in the case of fuels with low energy densities. The cryogenic fuels have special maintenance problems and also have the highest cost. Coal increases maintenance costs because of its abrasive nature and the ash disposal problem.

Table 14-10. Locomotive Engines Standard, Modified, and Alternatives

1. Turbocharged Water Cooled Diesel Engine (standard)
2. Four-Stroke Turbocharged Diesel Engine using Directly Decomposed Methanol
3. Turbocharged Water Cooled Diesel Engine with Rankine Bottoming Cycle
4. Short-Term "Mature" Diesel Engine[a]
5. Adiabatic Turbocompound Diesel Engine with Rankine Bottoming Cycle
6. Adiabatic Turbocompound Diesel Engine with Minimum Friction
7. Adiabatic Turbocompound Diesel Engine with Minimum Friction and Rankine Bottoming Cycle
8. Adiabatic Diesel Engine with Stirling Bottoming Cycle
9. Long-Term "Mature" Diesel Engine*
10. New Steam Engine with Reciprocating Drive
11. New Steam Turbine-electric Engine
12. Phosphoric Acid Fuel Cell
13. Open Cycle, Internal Combustion Regenerative Gas Turbine
14. Open Cycle, External Combustion Regenerative Gas Turbine
15. Closed Cycle, External Combustion Regenerative Gas Turbine
16. Advanced Stirling Engine using Liquid Fuels
17. Advanced Stirling Engine using a Fluidized Bed Combustor
18. Stratified Charge Rotary Combustion Engine

[a] See Section V for definition of "mature" engine.

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Table 14-11. Capital Costs of Locomotives Using Alternative Engines

	Engine No. 1[a]	Engine No. 2	Engine No. 3	Engine No. 4	Engine No. 5	Engine No. 6
Powerplant	$307,500	$381,000	$326,500	$350,000	$255,000	$252,000
Engine	$195,000	$205,000	$195,000	$210,000	$160,000	$177,000
Cooling System	37,500	42,000	41,500	41,500	4,000	0
Fuel System	22,500	79,000	22,500	22,500	22,500	22,500
Auxiliary Equipment and Misc.	52,500	55,000	67,500	76,500	68,500	52,500
Electrical	$255,000	$255,000	$255,000	$304,000	$255,000	$255,000
Alternator	$60,000	$60,000	$60,000	$60,000	$60,000	$60,000
Rectifier	22,500	22,500	22,500	22,500	22,500	22,500
Motors, each	30,000	30,000	30,000	30,000	30,000	30,000
Controls and Misc.	30,000	30,000	30,000	74,000	30,000	30,000
Dynamic Braking System	22,500	22,500	22,500	27,500	22,500	22,500
Chassis	$187,500	$187,500	$187,500	$187,500	$187,500	$187,500
Frame	$37,500	$37,500	$37,500	$37,500	$37,500	$37,500
Trucks and Axles	82,500	82,500	82,500	82,500	82,500	82,500
Cab Equipment	15,000	15,000	15,000	15,000	15,000	15,000
Superstructure	37,500	37,500	37,500	37,500	37,500	37,500
Miscellaneous	15,000	15,000	15,000	15,000	15,000	15,000
Total	$750,000	$823,500	$769,000	$842,000	$697,500	$694,500

[a]See Table 14-10 for list of engines
1980 dollars

Table 14-11. Capital Costs of Locomotives Using Alternative Engines (cont'd)

	Engine No. 7	Engine No. 8	Engine No. 9	Engine No. 10	Engine No. 11	Engine No. 12
Powerplant	$272,000	$245,000	$305,000	$473,000	$524,000	$567,500
Engine	$177,000	$140,000	$192,000	$175,000	$230,000	$225,000
Cooling System	4,000	6,000	6,000	75,000	65,000	100,000
Fuel System	22,500	22,500	22,500	129,000	112,000	197,500
Auxiliary Equipment and Misc.	68,500	76,500	84,500	94,000	117,000	45,000
Electrical	$255,000	$255,000	$304,000	$ 45,000	$265,000	$332,500
Alternator	$60,000	$60,000	$60,000	0	$70,000	0
Rectifier	22,500	22,500	22,500	0	22,500	0
Motor, each	30,000	30,000	30,000	0	30,000	30,000
Controls and Mis	30,000	30,000	74,000	20,000	30,000	280,000
Dynamic Braking System	22,500	22,500	27,500	25,000	22,500	22,500
Chassis	$187,500	$187,500	$187,500	$165,000	$197,500	$187,500
Frame	$37,500	$37,500	$60,000	$40,000	$47,500	$37,500
Trucks and Axles	82,500	82,500	22,500	52,000	82,500	82,500
Cab Equipment	15,000	15,000	15,000	15,000	15,000	15,000
Superstructure	37,500	37,500	30,000	38,000	37,500	37,500
Miscellaneous	15,000	15,000	22,500	20,000	15,000	15,000
Total	$714,500	$687,500	$796,500	$683,000	$986,500	$1,087,500

Table 14-11. Capital Costs of Locomotives Using Alternative Engines (cont'd)

	Engine No. 13	Engine No.14	Engine No. 15	Engine No. 16	Engine No. 17	Engine No. 18
Powerplant	$324,500	$549,500	$592,000	$450,000	$734,500	$312,500
Engine	$250,000	$385,000	$375,000	$300,000	$495,000	$200,000
Cooling System	0	0	50,000	75,000	75,000	37,500
Fuel System	22,000	112,000	112,000	22,500	112,000	22,500
Auxiliary Equipment and Misc.	52,500	52,500	55,000	52,500	52,500	52,500
Electrical	$265,000	$265,000	$265,000	$255,000	$255,000	$275,000
Alternator	$ 70,000	$ 70,000	$ 70,000	$ 60,000	$ 60,000	$ 80,000
Rectifier	22,500	22,500	22,500	22,500	22,500	22,500
Motors, each	30,000	30,000	30,000	30,000	30,000	30,000
Controls and Misc.	30,000	30,000	30,000	30,000	30,000	30,000
Dynamic Braking System	22,500	22,500	22,500	22,500	22,500	22,500
Chassis	$187,500	$197,500	$197,500	$200,000	$200,000	$187,500
Frame	$ 37,500	$ 47,500	$ 47,500	$ 37,500	$ 37,500	$ 37,500
Trucks and Axles	82,500	82,500	82,500	82,500	82,500	82,500
Cab Equipment	15,000	15,000	15,000	15,000	15,000	15,000
Superstructure	37,500	37,500	37,500	50,000	50,000	37,500
Miscellaneous	15,000	15,000	15,000	15,000	15,000	15,000
Total	$777,000	$1,012,000	$1,054,500	$905,000	$1,189,500	$775,000

Table 14-12. Maintenance and Operating Costs for Various Fuels

Fuel	Maintenance	Operating
1- Diesel No. 2	$55,000	$13,000
2- Gasoline	60,000	13,000
3- Broadcut Fuel Oil	55,000	13,000
4- No. 6 Fuel Oil, L.S.[a]	70,000	13,000
5- Liquid Methane	80,000	21,000
6- Liquid Propane	60,000	15,000
7- Liquid N-Butane	60,000	15,000
8- Methanol	70,000	14,000
9- Ethanol	70,000	13,000
10- Oil Shale Syn-Crude	60,000	13,000
11- Oil Shale Distillate	55,000	13,000
12- Oil Shale Gasoline	60,000	13,000
13- Coal Derived Distillate	55,000	13,000
14- Coal Derived Gasoline	60,000	13,000
15- Liquid Hydrogen	90,000	30,000
16- Liquid Ammonia	85,000	21,000
17- Bituminous Coal	80,000	16,000

Notes: Four-axle 3000 hp locomotives, Costs in 1980 Dollars, [a] Low Sulfur

The variation in operating costs is due to the costs of fuel handling. It is assumed that the miscellaneous supplies; sand, water, etc., are the same for all fuels and all locomotives regardless of the engine. Most liquid fuels will cost about the same to handle with the exception of the cryogenic fuels. The cryogenic fuels, particularly hydrogen, require very special handling. Coal is also more expensive to handle but not nearly as expensive as the cryogenic fuels.

Table 14-13 shows the variation in these costs for the 18 engines listed in Table 14-10. When a locomotive with a given engine can utilize a number of different fuels, the maintenance and operating costs are based on the principal fuel. For example, the principal fuel for an engine using a fluidized bed combustor is coal. Other fuels could be used but coal is the most likely one.

The lowest maintenance costs are associated with the internal combustion gas turbine and the phosphoric acid fuel cell. The highest costs occur with the fluidized bed combustor. The fuel handling costs are the highest with coal since none of the cryogenic fuels are principal fuels. There are a large number of fuel - engine combinations which are not analyzed in this chapter. However, some idea of the costs can be obtained by the cross-correlation of Tables 14-12 and 14-13.

Table 14-13. Maintenance and Operating Costs for Conventional and Alternative Locomotives Based on Primary Fuel

Engine No.[a]	Maintenance	Operating
1	$55,000	$13,000
	65,000	14,000
3	65,000	13,000
4	65,000	13,000
5	50,000	13,000
6	50,000	13,000
7	55,000	13,000
8	60,000	13,000
9	50,000	13,000
10	65,000	16,000
11	80,000	16,000
12	45,000	14,000
13	45,000	13,000
14	80,000	16,000
15	80,000	16,000
16	65,000	13,000
17	80,000	16,000
18	60,000	13,000

[a] Engine No. refers to the number on Table 14-10
Costs in 1980 dollars

D. COST ANALYSIS CONSTANTS

There are a number of cost parameters which are not affected by the type of fuel or the type of engine used. Some of these parameters are sensitive to the general economic conditions. In particular, the returns on debt and on equity are closely tied to the financial health of the railroads and the economy as a whole. It has been assumed that the current economic situation is a transitory period and that the economy will stabilize again. The period 20 years from now is projected to be similar to what it was about 20 years ago. The late 1950s and early 1960s were marked by a low inflation rate and low interest rates. It is anticipated that this general condition will prevail again but with rates slightly above those of 1960.

Predicted financial rates are listed in Table 14-14 for the 1980 to 2000 time period. The railroads are assumed to be increasingly healthy economically and this is reflected in the increasing return on equity from 2% in 1980 to 6% by the year 2000. Many railroads already have a return of 6% but a great number are nearer the 2% level. In fact, in 1978, the return on investment by the industry as a whole dropped to less than 1% as shown in Figure 14-8.

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Table 14-14. Predicted Financial Rates

Year	Return on Equity (%)	Current Return on Debt (%)	Average Return on Debt (%)
1980	2.0	14.1	8.5
1981	2.2	12.3	8.9
1982	2.4	11.2	9.6
1983	2.6	10.3	10.2
1984	2.8	9.6	11.1
1985	3.0	9.0	12.0
1986	3.2	8.5	12.2
1987	3.5	8.0	11.9
1988	3.8	7.7	11.1
1989	4.0	7.4	10.2
1990	4.2	7.0	9.6
1991	4.5	6.8	8.9
1992	4.8	6.5	8.5
1993	5.0	6.2	8.1
1994	5.2	5.9	7.7
1995	5.4	5.7	7.3
1996	5.6	5.6	7.0
1997	5.7	5.4	6.7
1998	5.8	5.2	6.4
1999	5.9	5.1	6.2
2000	6.0	5.0	6.0

The return on debt, instead of decreasing as the return on investment, has increased over the years, as shown in Figure 14-9. It is expected to decrease, reaching 5%, by the year 2000. In Table 14-14, predicted current return on debt is listed as well as an average return on debt. The current return refers to the rate prevailing during the particular year and it is the rate which would be paid by the railroads on new obligations. Since debts are incurred for some number of years, the average return depends on the rates over a number of years and the amount of debt at the various rates. As a result, the rate being paid in a given year tends to lag behind the current rate and also tends to change slower. The figures for average return on debt reflects the time lag and the leveling effect. The peak average rate is 12.2% in 1986 as compared to a peak current rate of 14.1% in 1980.

The service and accounting life, tax rates, and fractions of capital costs are shown on Table 14-15. These values combined with the financial predictions of Table 14-14 are used to determine the six financial constants at the end of Table 14-15. These constants were defined at the beginning of this cha er.

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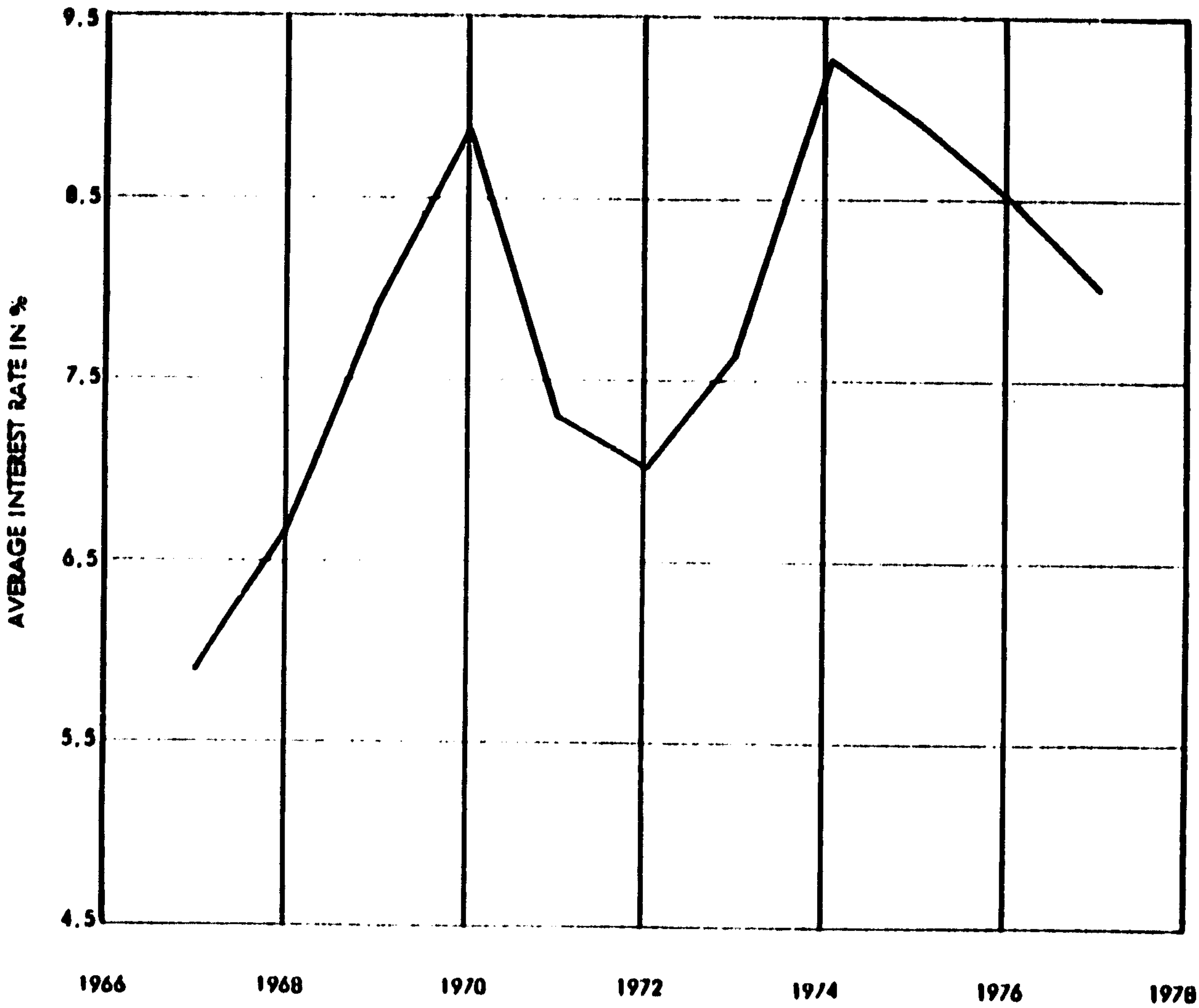

Figure 14-8. Railroad Industry Rate of Return on Investment
(From Ref. 14-16)

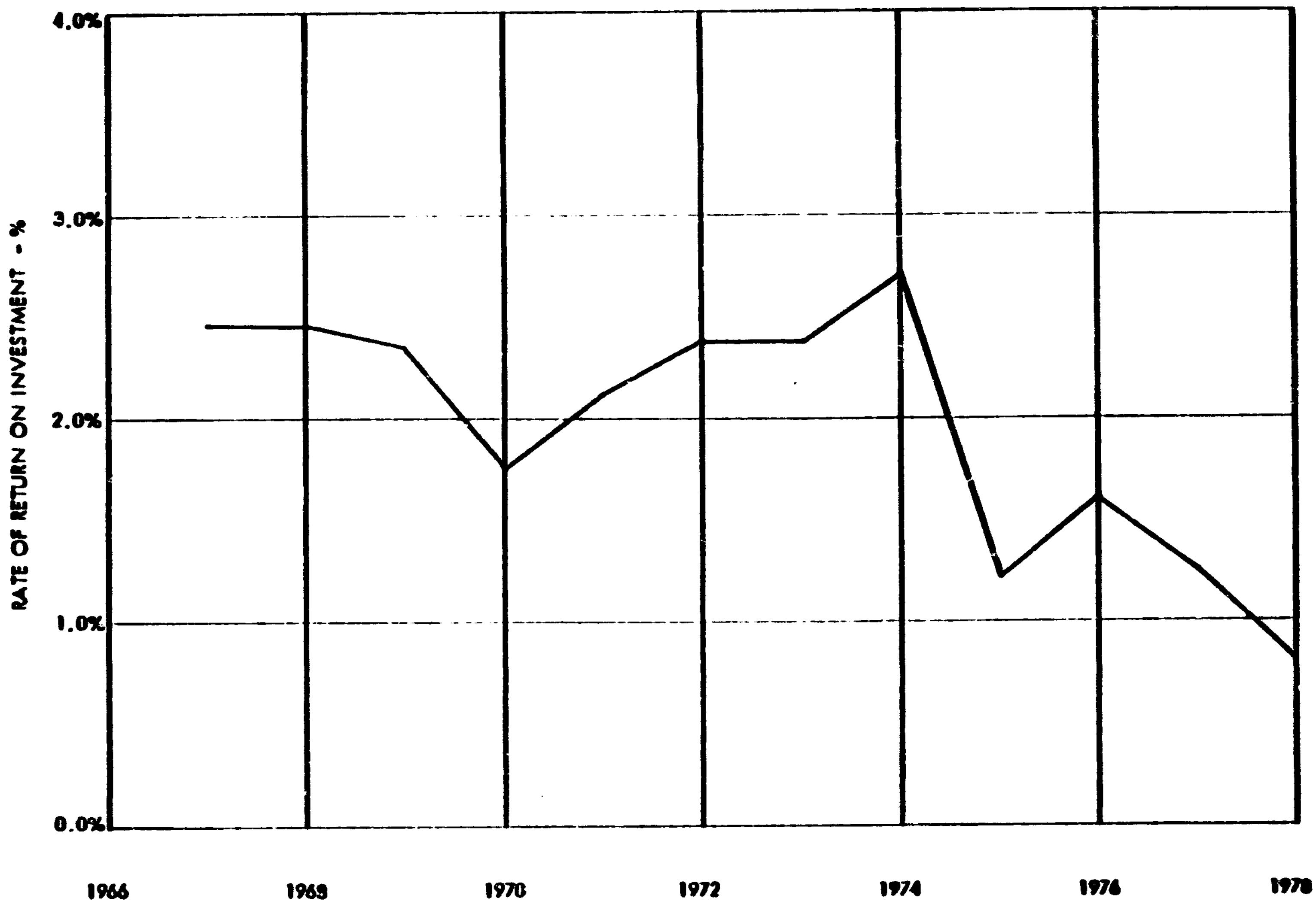

Figure 14-9. Railroad Industry Average Interest Rate on Equipment Obligations Issued During Year (From Ref. 14-16)

Table 14-15. Cost Analysis Constants

Constant	Symbol	Value
Service life	N	20 yrs.
Accounting life	n	11 yrs.
Effective corporate income tax rate	T	50%
Property tax rate	B_1	9.2%
Insurance rate	B_2	0
Fraction of capital costs consisting of debt	f_d	.80
Fraction of capital costs consisting of equity	f_e	.20
Average after tax cost of capital, N period	k_N	.05815
Average after tax cost of capital, n period	k_n	.06080
Capital Recovery Factor, service life	$CRF_{k,N}$	.08588
Capital Recovery Factor, accounting life	$CRF_{k,n}$	.1273
Depreciation Factor	$DPF_{m,k,n}$	.7141
Annualized Fixed Charge Rate	FCR	.1911

P. COST ESCALATION RATES

The methodology used in this cost analysis uses escalation rates in its predictions of the costs of operations, maintenance and fuel. Table 14-16 presents the escalation rates for 18 fuels and for operating as well as maintenance costs. Maintenance and operating costs escalate at a rate close to the general inflation rate. The escalation rates for fuels vary from a low of .0757 to a high of .1188. There are several reasons for the differences between fuels. One factor is the assumption that the energy content of the fuel will be the most important single factor in its cost in the future. In the past, the demand for a particular fuel such as gasoline made it more expensive on an energy basis than No. 6 fuel oil. In 1967, No. 6 low sulfur fuel oil cost 42 cents per million Btu while gasoline cost 193 cents per million Btu. As fuel prices increase, there will be an increased demand for the lower priced fuels so their prices will escalate faster than the average. As a group, the petroleum fuels will increase in price faster than the alcohols or coal. The increased demand for coal by utilities will keep its price moving upward. The alcohols will escalate at the lowest rates, comparable to general inflation, since they are already expensive fuels but they will become more cost competitive in the future. However, they are not likely to be cheaper than the hydrocarbons within 20 years.

Q. FUEL ENERGY USAGE

In order to estimate the annual fuel cost for a locomotive, it was first necessary to determine the energy used and then to multiply it by the fuel price and to divide the product by the lower heating value of the fuel. Using the JPL computer program "RAIL" to simulate the operations of locomotives, it was found that the energy usage at the rail is about 9×10^9 to 15×10^9 Btu/year for freight operations. The General Motors

 C-5

Table 14-16. Cost Escalation Rates

	Symbol	Rate
Operating Costs	g_{op}	.0718
Maintenance	g_{mnt}	.C718
Fuel costs	g_{FL}	
1- Diesel No. 2		.1045
2- Gasoline		.0839
3- Broadcut fuel oil		.1024
4- No. 6 fuel oil		.1136
5- Methane		.1006
6- Propane		.1188
7- N-Butane		.1004
8- Methanol		.0884
9- Ethanol		.0757
10- Oil shale syn-crude		.0833
11- Oil shale distillate		.0846
12- Oil shale gasoline		.0793
13- Coal derived distillate		.0882
14- Coal derived gasoline		.0898
15- Hydrogen		.0884
16- Ammonia		.0886
17- Bituminous coal, National average price		.0880
18- Bituminous coal, western mine mouth		.0876

Electro-Motive Division duty cycles gave a similar range of values. For medium-duty service using 3000 hp four-axle locomotives, the rail energy was chosen to be 11.965 x 10^9 Btu/year. This value was subsequently used for all fuels and all engines in this cost analysis. Starting with this figure, the losses in the drive system, accessories, auxiliary equipment and the engine were added to it to arrive at the input or fuel energy necessary to deliver the required energy at the wheels. These fuel energy input figures are shown in Table 14-17. The range of input energy is from 32 x 10^9 Btu/year for the long term "mature" Diesel to 102 x 10^9 Btu/year for the first generation new steam locomotive. This is little more than a factor of three to one. Each engine was analyzed for only one fuel in this table and the fuel is listed next to the engine number. The engine numbers are defined in Table 14-10. In general, the lowest energy usage is associated with the adiabatic Diesel engines and the highest usage is for the coal-fired steam engines.

R. LIFE-CYCLE COST RANKING OF FUELS

The substitution of one fuel for another in a locomotive involves many factors, each contributing to the life-cycle cost. In order to use a

Table 14-17. Fuel Energy Usage by Various Engines

Engine No.[a]	Fuel Used	Fuel Energy (millions of Btu/yr)
1	Diesel No. 2	51,060
2	Methanol from coal	43,557
3	Diesel No. 2	45,451
4	Broadcut fuel oil	40,906
5	Oil shale distillate	37,861
6	Oil shale distillate	39,366
7	Oil shale distillate	35,237
8	Oil shale distillate	40,121
9	Oil shale distillate	31,946
10	Bituminous coal	102,265
11	Bituminous coal	72,933
12	Methanol	41,722
13	Oil shale syn-crude	48,748
14	Bituminous coal	52,414
15	Bituminous coal	51,060
16	Oil shale distillate	44,484
17	Bituminous coal	52,414
18	Oil shale distillate	58,437

[a]Engine No. refers to the engine number on Table 14-10
Energy at rail is 11.965×10^9 Btu/year for all engines.

different fuel, it may be necessary to add or change injectors and pumps, to modify controls, and for some fuels to add a tender. All these things change the capital cost of the locomotive and they must be included in the cost analysis. The necessary changes to the locomotive affects the maintenance and operating costs as well. A dual-fueled engine is necessary for the use of fuels like methanol and the additional pumps, injectors, lines, and the fuel tank will require additional maintenance. Those fuels requiring the use of a tender, particularly the cryogenic fuels such as hydrogen, will have higher maintenance costs than those fuels that can be carried entirely on the locomotive.

In order to assess the realistic life-cycle costs of various fuels, a single locomotive was chosen as the base system. This was a 3000 hp, 16 cylinder, four-axle locomotive typical of the present rail fleet. It is not specified as to whether it is a four-stroke or a two-stroke engine since the fuel consumption of both types of locomotives is nearly the same. The energy usage at the rail (11.965×10^9 Btu/year) is the same for all fuels. The primary variable in the energy usage with different fuels is the slight difference in the thermal efficiency of the engine with each one.

The life-cycle cost is most sensitive to the price of the fuel and least sensitive to the operating costs. The operating cost is typically only 2 to 3% of the life-cycle cost. The capital and maintenance costs are about equal in annualized costs at 10 to 15% each of the total costs. The remaining 65 to 75% of the life cycle cost is the cost of fuel. The fuels are ranked in Table 14-18 with the oil shale syn-crude having the lowest life-cycle cost over the next 20 years. The highest cost is for ethanol which is and will be an expensive fuel.

Because of the expected escalation in the price of Diesel No. 2 resulting from increased demand and reduced supplies, the present Diesel-electric locomotive is in an economically undesirable position. It is ranked eleventh out of sixteen on this list. However, it should be noted that it is only 15% higher in cost than the lowest cost oil shale syn-crude. The fact that 11 fuels fall in a 15% range is the result of the assumption that the future price of fuel will depend primarily on its energy content. Since the thermal efficiency of the Diesel engine is only slightly affected by the type of fuel, the fuel cost is nearly the same for most of the fuels. The four fuels with the highest life-cycle costs are either very expensive in terms of energy or have special handling problems which increase capital and maintenance costs substantially.

From a cost standpoint, the oil shale syn-crude and distillate fuels are the lowest cost of the alternative fuels. Methanol and the coal derived liquids are the next higher cost fuels. Among the petroleum-based fuels other than Diesel No. 2, the lowest cost is for propane, No. 6 fuel oil, and broadcut fuel oil. The maintenance cost of using No. 6 fuel oil is high but the lower price of the fuel itself more than compensates for the increased maintenance. This was not the case some years ago when residual oils were used by some railroads who found maintenance costs were excessive. The high cost of energy requires a reevaluation of maintenance policies.

The life-cycle costs of Diesel-electric locomotives can be reduced by changes in the fuel used. However, the reduction is generally less than 15% and usually requires a substantial capital investment in the locomotive. In addition, and not considered in this cost analysis, it may be necessary to construct or modify existing fuel distribution, storage, and handling systems.

S. LIFE-CYCLE COST RANKING OF ENGINES

Eighteen different engines were life-cycle cost analyzed in this study. Each engine was assumed to use the fuel best suited for it. In addition, the fuel cell was analyzed using hydrogen as well as methanol. Five engines were analyzed using western mine mouth coal as well as a national average priced coal. Both coals were assumed to be a bituminous coal with a lower heating value of 12,000 Btu/lb. The western mine mouth coal is the lowest price fuel in commercial production. It is included to indicate the costs involved if western unit coal trains were to be coal-fired with coal loaded at the mine.

The twenty four engine and fuel combinations are ranked in Table 14-19 on the basis of their annualized life-cycle costs. The engine number

Table 14-18. Ranking[a] of Fuels for Conventional Diesel Engines by Annualized Life-Cycle Cost.

Ranking	Fuel	Annualized Life-Cycle Cost
1	Oil shale syn-crude	$1,028,000
2	Oil shale distillate	1,036,000
3	Propane	1,047,000
4	No. 6 fuel oil, low sulfur	1,090,000
5	Broadcut fuel oil	1,104,000
6	Coal derived gasoline	1,114,000
7	N- Butane	1,137,000
8	Coal derived distillate	1,139,000
9	Methanol	1,150,000
10	Oil shale gasoline	1,173,000
11	Diesel No. 2	1,175,000
12	Gasoline	1,215,000
13	Methane	1,232,000
14	Hydrogen	1,569,000
15	Ammonia	2,024,000
16	Ethanol	2,676,000

[a]All fuels used in conventional Diesels modified to accept the fuel and dual-fueled when needed.

referred to on this table is the number assigned to each engine on Table 14-10. The descriptions of the engines are too lengthy to be used on this table. The fuels used with these engines are also specified on Table 14-19.

The energy usage values used in the calculations for Table 14-19 are those on Table 14-17. The operating and maintenance costs are taken from Table 14-13. The cost escalation factors are listed in Table 14-16. The results of the calculations show a range of costs from a low of $523,000 per year to a high of $1,196,000 per year. This is a range of over two to one.

The lowest costs are for the coal-fired locomotives with the minimum prices for the western mine run coal delivered to the locomotive at the mine mouth. The use of coal-fired locomotives for unit coal train operation is economically very attractive. Even the first generation new steam locomotive using national average price coal is more economical than the most advanced adiabatic Diesel using oil shale distillate. This engine, No. 9, the long term "mature" Diesel, has a life-cycle cost 22% higher than

Table 14-19. Ranking of Engines of Annualized Life-Cycle Costs

Engine No.[a]	Fuel Used	Annualized Life-Cycle Cost
15	Bituminous coal, western mine mouth	$ 523,000
14	Bituminous coal, westerm mine mouth	527,000
17	Bituminous coal, western mine mouth	548,000
10	Bituminous coal, western mine mouth	549,000
15	Bituminous coal, NAP[b]	554,000
14	Bituminous coal, NAP	559,000
11	Bituminous coal, western mine mouth	564,000
17	Bituminous coal, NAP	576,000
11	Bituminous coal, NAP	608,000
10	Bituminous coal, NAP	615,000
9	Oil shale distillate	750,000
7	Oil shale distillate	792,000
5	Oil shale distillate	818,000
6	Oil shale distillate	840,000
8	Oil shale distillate	870,000
12	Methanol from coal	907,000
16	Oil shale distillate	931,000
4	Broadcut fuel oil	976,000
2	Methanol from Coal	996,000
13	Oil shale syn-crude	1,029,000
3	Diesel No. 2	1,101,000
12	Hydrogen	1,118,000
1	Diesel No. 2	1,175,000
18	Oil shale distillate	1,196,000

[a]Engine No. refers to engine number on Table 14-10
[b]National average price

the first generation new steam engine and 43% higher than the closed cycle externally fired regenerative gas turbine (No. 15) using western mine mouth coal.

The adiabatic Diesel dominates the engines with life-cycle costs above those of the coal-fired locomotives. They are considerably more efficient than the coal-fired engines but require much more expensive fuels. The fuel cell using methanol is comparable in cost to the adiabatic Diesel but the fuel cell using hydrogen is one of the most expensive locomotives.

The more conventional water cooled Diesel engines are among the most expensive engines with the standard turbocharged engine (No. 1) being ranked twenty-third out of twenty four. The very high price of Diesel No. 2

expected in the next two decades is the primary reason for this poor ranking.

It might be imagined that the engine with the highest thermal efficiency would be the least expensive in terms of life-cycle costs since the fuel costs are predominant. For coal, this is not the whole truth. For example, the second generation new steam engine has a thermal efficiency of 28% while that of the first generation engine is only 20%. It would be expected that there will be a sizable difference in life-cycle costs because of the eight percentage point difference. The results shown in Table 14-19 indicate that the life cycle cost for the second generation engine is only 1% less than that of the first generation engine. When energy is cheap as with coal, the capital and maintenance costs are very important. However, when energy costs are high as with fuel oils, the thermal efficiency is the dominant factor and it is economically sound to invest more in achieving the higher efficiency.

From the point of life-cycle costs, the best locomotives are the coal-fired ones, the adiabatic Diesel engined ones, and the fuel cell powered ones, in that order. The first generation new steam locomotive is the best choice the the near- to middle-term. There is little development risk, it could be in production in a few years, and it is economically attractive. In the longer term, the fluidized bed combustor utilizing a number of fuels is attractive as is the adiabatic Diesel.

T. FRACTIONS OF THE LIFE-CYCLE COSTS

The life-cycle cost of a locomotive is the sum of the costs in four separate areas. These areas are: capital costs, operating costs, maintenance costs, and fuel costs. Cost information for each area is contained in the tables presented earlier in this chapter. However, the contribution of each area to the life-cycle cost has not been delineated. In Table 14-20, the life-cycle costs for four engines are shown with the costs broken out for each area.

The conventional Diesel-electric locomotive is shown first. As shown in Table 14-19, the total life-cycle cost per year is $1,175,000. Of this total: $143,000 is the capital cost; $26,000 for operating costs, $112,000 for maintenance, and $894,000 for fuel. In the next 20 years, fuel will be the biggest single expense for this type of engine. A high efficiency adiabatic Diesel engine is shown next. Its capital cost is estimated to be $6,000 less than the conventional Diesel, and its operating and maintenance cost are projected to be the same. The fuel cost, however, is only $518,000 which is only 44% that of the conventional Diesel.

The steam turbine locomotive using national average price coal is more expensive, about 36% more, than the conventional Diesel in the capital cost, operating cost, and maintenance areas. The fuel cost is only $225,000 per year or 25% of the conventional Diesel fuel cost. Even with greater costs in three of the cost areas, the lower fuel cost results in a total life-cycle for this engine that is only 51% of the conventional Diesel. The closed cycle gas turbine locomotive shows the same general trend. The capital cost of this locomotive is 42% higher than the conventional Diesel locomotive but the total life-cycle cost is less than half as much.

Table 14-20. Fractions of the Annualized Life-Cycle Costs

Type	Capital	Operating	Maintenance	Fuel	Total
Closed Cycle, External Combustion Regenerative Gas Turbine Using Western Mine Mouth Bituminous Coal	201,000	32,000	163,000	127,000	523,000
Open Cycle, External Combustion Regenerative Turbine Using W.M.M. Bituminous Coal	194,000	32,000	163,000	138,000	527,000
Closed Cycle, External Combustion Regenerative Gas Turbine Using N.A.P. Bituminous Coal	202,000	32,000	163,000	157,000	534,000
Advanced Stirling Engine Using W.M.M. Bituminous Coal	228,000	32,000	163,000	125,000	548,000
New Steam Engine with Reciprocating Drive Using W.W.M. Bituminous Coal	131,000	32,000	132,000	254,000	549,000
Open Cycle, External Combustion Regenerative Gas Turbine Using N.A.P. Bituminous Coal	193,000	32,000	163,000	171,000	559,000
New Steam Turbine-Electric Using W.W.M. Bituminous Coal	188,000	32,000	163,000	181,000	564,000
Advanced Stirling Engine Using N.A.P. Bituminous Coal	228,000	32,000	163,000	154,000	576,000
New Steam Turbine-Electric Using National Average Price Coal	189,000	32,000	163,000	224,000	608,000
New Steam Engine with Reciprocating Drive Using N.A.P. Bituminous Coal	130,000	32,000	132,000	321,000	615,000

Table 14-20. Fractions of the Annualized Life-Cycle Costs (cont'd)

Type	Capital	Operating	Maintenance	Fuel	Total
Long-Term "Mature"[a] Diesel Engine Using Oil Shale Distillate	152,000	26,000	102,000	470,000	750,000
Adiabatic Turbocompound Diesel Engine with M.F. and R.B.C. Using Oil Shale Distillate	136,000	26,000	112,000	518,000	792,000
Adiabatic Turbocompound Diesel Engine with R.B.C. Using Oil Shale Distillate	133,000	26,000	102,000	557,000	818,000
Adiabatic Turbocompound Diesel with Diesel with M.F. Using Oil Shale Distillate	133,000	26,000	102,000	579,000	840,000
Adiabatic Diesel with Stirling Bottoming Cycle Using Oil Shale Distillate	131,000	26,000	122,000	591,000	870,000
Phosphoric Acid Fuel Cell Using Coal Derived Methanol	208,000	28,000	91,000	580,000	907,000
Advanced Stirling Engine Using Oil Shale Distillate	173,000	26,000	132,000	600,000	931,000
Short-Term "Mature" Diesel Engine[a] Using Broadcut Fuel Oil	161,000	26,000	132,000	657,000	976,000
Four-Stroke Trubocharged Diesel Using Directly Decomposed Methanol	157,000	28,000	132,000	671,000	996,000
Lima 2-8-4 Berkshire Steam Locomotive Original Materials and Design	124,000	34,000	223,000	625,000	1,006,000

Table 14-20. Fractions of the Annualized Life-Cycle Costs (cont'd)

Type	Capital	Operating	Maintenance	Fuel	Total
Open Cycle, Internal Combustion Regenerative Gas Turbine Using Oil Shale Syn-Crude	148,000	26,[illegible]	91,000	764,000	1,029,000
Turbocharged Water Cooled Diesel with R.B.C. Using Diesel No. 2	147,000	26,000	132,000	796,000	1,101,000
Phosphoric Acid Fuel Cell Using Hydrogen	178,000	61,000	1[illegible]3,000	696,000	1,118,000
Lima 2-8-4 Berkshire Steam Locomotive Original Materials and Design	153,000	34,000	335,000	625,000	1,147,000
Diesel-Electric Using Diesel No. 2	143,000	26,000	112,000	894,000	1,175,000
Stratified Charge Rotary Combustion Engine Using Oil-Shale Distillate	148,000	26,000	122,000	900,000	1,196,000

NOTES: All locomotives are assumed to be production versions
[a]See SECTION 5 for definition of "mature" engines
R.B.C. - Rankine bottoming cycle
M.F. - Minimum friction
N.A.P. - National Average Price
W.M.M. - Western Mine Mouth

The calculations shown here indicate that if the fuel cost per unit of energy is as high as is projected for liquid hydrocarbons, then the thermal efficiency must be as high as possible. Higher maintenance and capital costs could be tolerated if they reduce the fuel costs. For a low cost fuel such as coal; capital, maintenance, and fuel costs are of similar magnitude. The thermal efficiency is of less importance for these fuels than for engines using liquid hydrocarbons. Substantially higher capital and maintenance costs could be attached to these engines and still result in lower total life-cycle costs than the conventional Diesel.

U. LIFE-CYCLE COSTS USING DOE AND DRI FUEL COST PROJECTIONS

The life-cycle costs of the conventional Diesel engine and the closed cycle gas turbine engine were calculated using the JPL, the Department of Energy (DOE), and the Data Resources, Inc. (DRI) fuel cost projections shown in Figure 14-7. The JPL and the other two predictions differ markedly not only in the fuel costs but in the trends as well. The comparison of the life-cycle costs for these three projections will show how sensitive these costs are to the fuel price over the 20 year period.

The results of the calculations are shown in Table 14-21. For the conventional Diesel, the highest life-cycle cost, using DOE data, is 10% higher than the lowest cost using the JPL data. For the gas turbine using coal, the highest cost, using DOE data, is 17% higher than the lowest cost using JPL data. In each set of data, the coal and distillate prices tend to move upward together. The DOE predicts the highest prices for both fuels and JPL predicts the lowest.

The ratio of the life-cycle cost for distillate and for coal emphasizes the competitive relationship between the fuels themselves exclusive of their absolute prices. The ratios for all three sources of data are nearly identical, 2.00 to 2.12, a difference of only 6%. Regardless of the source of the fuel price projections, coal has a two-to-one edge over distillate fuel in this comparison.

The difference in the life-cycle cost between the two fuels indicates the possible savings accrued from using coal rather than distillate fuel. The JPL data indicates a savings of $621,000 per year while the DOE data projects a $645,000 savings and the DRI data shows a $655,000 savings. The DRI data shows a 5.5% greater savings than the JPL data.

There are marked differences in the absolute life-cycle costs using the three different fuel price projections, but the relative cost differences are small and within the probable error of the data. The JPL data is intended only to provide a relative measure of the life-cycle costs. These results indicate that this objective was effectively achieved.

V. LIFE-CYCLE COSTS FOR AN OLD STEAM LOCOMOTIVE

For comparison with both the present Diesel-electric and the alternative locomotives, a life-cycle cost analysis was made for a 2-8-4 Berkshire locomotive built by Lima Locomotive Works, Inc. This locomotive has been chosen because its tractive effort is similar to that of modern 3000 hp locomotives. Using only the main cylinders, its tractive effort

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Table 14-21. Comparison of Life-Cycle Costs Using JPL, DOE, and DRI Fuel Projections

Conventional Diesel Using Distillate Fuel

Data Source	Life-Cycle Cost
JPL	$1,175,000
DOE	1,293,000
DRI	1,263,000

External Combustion Closed Cycle Gas Turbine Using Coal

Data Source	Life-Cycle Cost
JPL	$554,000
DOE	648,000
DRI	608,000

Ratio of Life Cycle Cost for Distillate to That of Coal

Data Source	RATIO
JPL	2.12
DOE	2.00
DRI	2.08

Differences Between the Life-Cycle Cost for Distillate and That of Coal

Data Source	Difference
JPL	$1,175,000 - 554,000 = $621,000
DOE	1,293,000 - 648,000 = 645,000
DRI	1,263,000 - 608,000 = 655,000

is 69,000 lb and is 81,400 lb with the booster engine. In addition, its weight is 194 tons, close to that of a modern six-axle locomotive. Of all of the steam locomotives examined, it is most comparable to a modern Diesel-electric.

The capital cost of this locomotive is calculated in two ways. First, it is assumed that it is constructed exactly as originally designed with the same materials and labor. Second, it is redesigned to take advantage of modern materials and production techniques. For instance, the use of a welded boiler rather than a riveted one. The basic design, however, has not been changed. The first version is estimated to cost about $800,000 per locomotive in production quantities while the second version is costed at $650,000 each. The difference is due to the reduction of labor in the assembling of the locomotive. There is considerable hand work in the original design.

The operating costs and the fuel costs are the same for both versions of the locomotive. The fuel cost is based on a total output energy at the rail of 12,000 MBtu per year and on national average price coal. This output energy is the same value used for all of the alternative locomotives. The system efficiency is assumed to be 6%, typical of the steam engine in the 1920 to 1950 era.

Maintenance costs are considerably higher for these steam locomotives than for the Diesel-electrics. The original version of the Berkshire engine is estimated to have an average maintenance cost of $335,000 per year while the second version has an estimated maintenance cost of $223,000. The use of different materials and production methods reduces the amount of maintenance needed.

The life-cycle costs of these two locomotives are shown in Table 14-20. The annualized life-cycle cost of the original design Berkshire locomotive is within 3% of that of the Diesel-electric locomotive using Diesel No. 2 fuel. The second version using modern materials has a calculated life-cycle cost of just over one million dollars per year, or about 15% less than the Diesel-electric locomotive.

In the late 1940s and early 1950s, the cost of energy from coal was nearer to that from Diesel fuel. If the ratio of the diesel fuel energy price to that of coal for 1950 is used instead of the 1980 ratio, the effect is to raise the fuel cost for the steam locomotive. The fuel cost for the Berkshire locomotive is $834,000 and the life-cycle cost is $1,356,000 per year assuming a 20 year life for the original design version. This cost is about 15% higher than the cost for the Diesel-electric locomotive. The steam engine was displaced for good economic reasons and, even today, with relatively higher prices for Diesel fuel, the old steam locomotives would still not be economically justified.

W. COST SENSITIVITY

The life-cycle costs are the result of a number of assumptions concerning interest rates, inflation rates, property and income taxes, fuel costs, and maintenance costs. It is necessary to examine the way changes in these assumptions affect the relative life-cycle costs of the various engines investigated in this study. The costs can be considered as the financial costs (interest, taxes, and inflation), and the direct costs (capital, operations, maintenance, and fuel).

To examine the effect of interest on the life-cycle cost, two locomotives have been selected. One is the Diesel-electric locomotive using Diesel No. 2 fuel. This is an example of a low capital cost-high fuel cost locomotive. The other one is the closed cycle gas turbine locomotive using national average price coal. This locomotive has a high capital cost with a low fuel cost. Most of the other alternative locomotives fall between these two examples.

The tax weighed interest cost, k, is given by this equation:

$$k = (1-T)\ f_d k_d + f_e k_e$$

where the variables are as defined earlier in this section. The range of values for k is from zero (no interest loans) to about .20 (20% interest on an equity fraction of one). Both extremes are very unlikely. The most probable range is .04 to .08. The best investment strategy is one where the division between debt and equity financing results in the lowest value of k. The strategy changes continuously as interest rates change and details of the strategy are well beyond the scope of this study. However, it can be determined if the value of k has a significant effect on the relative costs of the two sample locomotives. The life-cycle costs for the two locomotives are calculated using values of k = .02, .10, and .20. The results are shown in Table 14-22. The income tax rate is assumed to be 50% for both locomotives. The numbers in the parentheses beside the dollar cost values are the fractions of the total cost of that particular item. The change in these fractions for a change in the value of k from .02 to .20 is up to sixteen percentage points for the coal-fired locomotive. The ratio of the total life-cycle costs of the Diesel locomotive to the coal locomotive varies from 2.59 down to 1.74, for the change in k from .02 to .20. Essentially, the interest rate has a very definite effect on the competitive relationship between these two fuels. The effect of low interest rates is in favor of the coal-fired locomotive while the high rates favor the Diesel locomotive.

In Table 14-22, only the tax weighed interest cost was varied. The fuel price for each year in the [illegible]e was held constant. That is, the fuel escalation rate was held constant. In reality, the fuel escalation rate is not independent of the interest costs. When interest costs are high, fuel escalation rates will tend to be high and the reverse is true. They do not follow each other exactly, only the trends are similar. If the assumption is made that the fuel escalation rate is exactly the same as the interest cost, then the (1+g)/(1+k) factor is exactly one. If the same assumption is made with respect to operations and maintenance, then the life-cycle costs shown in Table 14-23 are the result. These absolute costs are greatly different than those in Table 14-22 but the ratios of the coal-fired locomotive are very similar though favoring the Diesel engine slightly. In both tables, the ratios are very nearly 2 to 1 in favor of the coal-fired locomotive. The assumptions appear to have a significant effect on the absolute costs but, the ratios and competitive relationship of the two fuels are not significantly affected.

The income tax rate is a factor only in the fixed charge rate (FCR) and affects only the capital investment portion of the life-cycle cost equation. It is expected, therefore, that a higher tax rate would favor the Diesel locomotive and a low rate would favor the coal-fired locomotive. Table 14-24 presents the results for a change in income tax rate. These results indicate that the income tax rate has only a slight effect on life-cycle cost. A reduction in rate from 50% to 30% results in a 4.5% shift in the Diesel-to-coal cost ratio. It can be concluded that any reasonable change in the income tax rate has only a slight effect on the competitive relationship of the two fuels. The lower rate tends to favor the coal-fired locomotive.

Another term in the fixed charge rate parameter is the property tax rate. It is not applicable to locomotives in many states but is encountered in some states and may be imposed by others in the future. This tax may be known under other names and still affect the equation the same way. The

Table 14-22. Effect of Interest Rates on Life-Cycle Costs in Thousands of Dollars per Year

Diesel Locomotive Using Diesel No. 2

Interest Rate (k)	Capital	Operations	Maintenance	Fuel	Total
.02	51 (.044)	27 (.023)	116 (.101)	958 (.832)	1152 (100)
.10	124 (.114)	24 (.022)	99 (.091)	840 (.773)	1087 (100)
.20	247 (.229)	20 (.019)	85 (.079)	725 (.673)	1077 (100)

Closed Cycle Gas Turbine Using Bituminous Coal

Interest Rate (k)	Capital	Operations	Maintenance	Fuel	Total
.02	72 (.162)	34 (.077)	168 (.378)	170 (.383)	444 (100)
.10	175 (.353)	29 (.059)	145 (.293)	146 (.295)	495 (100)
.20	348 (.562)	25 (.040)	123 (.198)	124 (.200)	620 (100)

Ratio of Diesel Engine Total Cost to Coal-Fired Locomotive Total Cost

Interest Rate (k)		Ratio
.02	1152/444	2.59
.10	1087/495	2.20
.20	1077/620	1.74

Note: ( ) denotes fraction of total costs

Table 14-23. Effect of Interest Rates and Fuel Price Escalation* on Life-Cycle Costs in Thousands of Dollars per Year

Diesel Locomotive Using Diesel No. 2

Interest Rate (k)	Capital	Operations	Maintenance	Fuel	Total
.02	51 (.099)	16 (.031)	67 (.130)	381 (.740)	515 (100)
.10	124 (.122)	31 (.031)	129 (.127)	732 (.720)	1016 (100)
.20	247 (.137)	53 (.029)	226 (.125)	1279 (.709)	1805 (100)

Closed Cycle Gas Turbine Using Bituminous Coal

Interest Rate (k)	Capital	Operations	Maintenance	Fuel	Total
.02	72 (.267)	20 (.074)	98 (.363)	80 (.296)	270 (100)
.10	175 (.315)	38 (.068)	188 (.339)	154 (.278)	555 (100)
.20	348 (.344)	66 (.065)	329 (.325)	269 (.266)	1012 (100)

Ratio of Diesel Engine Total Cost to Coal-Fired Locomotive Total Cost

Interest Rate (k)		Ratio
.02	515/270	1.91
.10	1016/555	1.83
.20	1805/1012	1.78

Note: Fuel escalation rate (g_{FL}) is assumed equal to tax weighed interest rate (k) at all times.
( ) denotes fraction of total costs

Table 14-24. Effect of Income Tax Rate on Life-Cycle Costs in Thousands of Dollars per Year

Diesel Locomotive Using Diesel No. 2

Tax Rate (T)	Capital	Operations	Maintenance	Fuel	Total
.30	104 (.097)	24 (.022)	99 (.093)	840 (.788)	1067 (100)
.50	124 (.114)	24 (.022)	99 (.091)	840 (.773)	1087 (100)

Closed Cycle Gas Turbine Using Bituminous Coal

Tax Rate (T)	Capital	Operations	Maintenance	Fuel	Total
.30	146 (.313)	29 (.062)	145 (.312)	146 (.313)	466 (100)
.50	175 (.353)	29 (.059)	145 (.293)	146 (.295)	495 (100)

Ratio of Diesel Engine Total Cost to Coal-Fired Locomotive Total Cost

Tax Rate (T)		Ratio
.30	1067/466	2.29
.50	1087/495	2.20

Note: Interest rate (k) is 0.10
( ) denotes fraction of total costs

results shown in Tables 14-22, 14-23, and 14-24 were calculated using a property tax rate of zero. In Table 14-25, results from Table 14-22 with the zero tax rate are compared to life-cycle costs using a 10% property tax rate. The income tax rate in both cases is 50% and the tax weighed interest cost is 0.10. The property tax increases both life-cycle costs. However, it increases the coal-fired locomotive cost more due to its higher capital costs. The effect on the Diesel-to-coal ratio is a decrease from 2.20 to 1.94, or a change of about 12%.

The financial costs (taxes, interest, and inflation) have a significant effect on the absolute costs but a much smaller effect on the Diesel-to-coal cost ratio. This ratio has varied from a high of 2.59 to a low of 1.74. The competitive advantage of the coal-fired locomotive is not significantly affected by changes in financial parameters. They affect both, indeed all, locomotives very nearly equally.

In the direct cost area; the operations, maintenance, and fuel costs are directly scalable. If the price of fuel became double that predicted, then the fuel cost in tables such as 14-22 would double and the total cost would increase by the difference in fuel costs. A doubling of the Diesel fuel price would have a very large effect on total life-cycle cost. A similar change for coal is less significant. Similarly, changes in thermal efficiency or usage has a greater effect on the Diesel locomotive than on the coal-fired locomotive. The coal-fired locomotive is more sensitive to increases in operations and maintenance since these form a larger fraction of the life-cycle cost than they do in the Diesel locomotive. Even if all of these parameters were to be doubled, the life-cycle cost of the coal-fired locomotive would still be about 25% less than that of the Diesel locomotive.

The capital costs are not readily scalable. This is because all, but the first generation steam engine, use the same chassis, trucks, motors, brakes, and cab. As a result, a large fraction of the capital cost is identical for all of the locomotives. As is apparent in Table 14-26, the costs of the electrical and chassis items are nearly identical for the Diesel and the coal-fired locomotives. The difference is $20,000 out of over $400,000, or about 5%. The major difference between the two locomotives is in the powerplant. The cost of the powerplant in the coal-fired locomotive is nearly double that of the Diesel locomotive. If it is assumed that the powerplant of the coal-fired locomotive is triple the cost of the Diesel powerplant, then the total cost of the locomotive is $1,385,000. Using the 10% interest rate, the capital cost of the coal-fired locomotive shown in Table 14-22 increases from $175,000 to $229,000. The life-cycle cost is increased form $495,000 to $549,000. A 50% increase in powerplant cost results in an 11% increase in life-cycle cost. For the life-cycle costs of the two locomotives to be equal, the powerplant of the coal-fired locomotive would have to be twelve times as expensive as the Diesel powerplant. There is no reason to believe that a production powerplant, even in small quantities would be anywhere near that number.

This cost sensitivity study indicates that the two-to-one life-cycle cost advantage of the coal-fired locomotive is real and is not a result of the assumptions made during the study. No source of error has been found.

Table 14-25. Effect of Property Tax Rate on Life-Cycle Costs In Thousands of Dollars per Year

Diesel Locomotive Using Diesel No. 2

Tax Rate (B_1)	Capital	Operations	Maintenance	Fuel	Total
0	124 (.114)	24 (.022)	99 (.093)	840 (.773)	1087 (100)
.10	199 (.171)	24 (.021)	99 (.085)	840 (.723)	1162 (100)

Closed Cycle Gas Turbine Using Bituminous Coal

Tax Rate (B_1)	Capital	Operations	Maintenance	Fuel	Total
.30	175 (.353)	29 (.059)	145 (.293)	146 (.295)	495 (100)
.30	280 (.467)	29 (.048)	145 (.242)	146 (.243)	600 (100)

Ratio of Diesel Engine Total Cost to Coal-Fired Locomotive Total Cost

Tax Rate (B_1)		Ratio
0	1087/495	2.20
.10	1162/600	1.94

Note: Interest rate (k) is 0.10, income tax 0.50
( ) denotes fraction of total costs

Table 14-26. Capital Costs of Locomotives Using Alternative Engines

	Engine No. 1[a]	Engine No. 15
Power Plant	$307,500	$592,000
Engine	$195,000	$425,000
Cooling System	37,500	0
Fuel System	22,500	112,000
Auxiliary Equipment and Misc.	52,500	55,000
Electrical	$255,000	$265,000
Alternator	$60,000	$70,000
Rectifier	22,500	22,500
Motors, each	30,000	30,000
Controls and Misc.	30,000	30,000
Dynamic Braking System	22,500	22,500
Chassis	$187,500	$197,500
Frame	$37,500	$47,500
Trucks and Axles	82,500	82,500
Cab Equipment	15,000	15,000
Superstructure	37,500	37,500
Miscellaneous	15,000	15,000
Total	$750,000	$1,054,000

[a] See Table 14-10 for list of engines
1980 constant dollars

X. EFFECT OF USAGE ON LIFE-CYCLE COST

The costs computed in this study have been based on an energy output at the rail of 12×10^9 Btu/year or 4.7×10^6 hp-hr/year for all of the locomotives. If the locomotives are used less, then the fuel costs are reduced in direct proportion as are the operating costs. Maintenance costs are also reduced but not necessarily in direct proportion. Maintenance costs are assumed to be 50% related to usage and 50% independent of usage. Capital costs are not affected. The result is that the life-cycle costs change with usage as shown in Figure 14-10. It has been assumed that the locomotive operates on the GM-EMD medium duty cycle when in use and is shut down when not in use. Further, the medium duty cycle is used on a daily basis. That is, on a day when the locomotive is in use, the proportion of time in idle, notch 8 operation, etc., are defined by the medium duty cycle. On days when it is not used, it is completely shut down. As an example, 50% usage means that half of the days in the year the locomotive

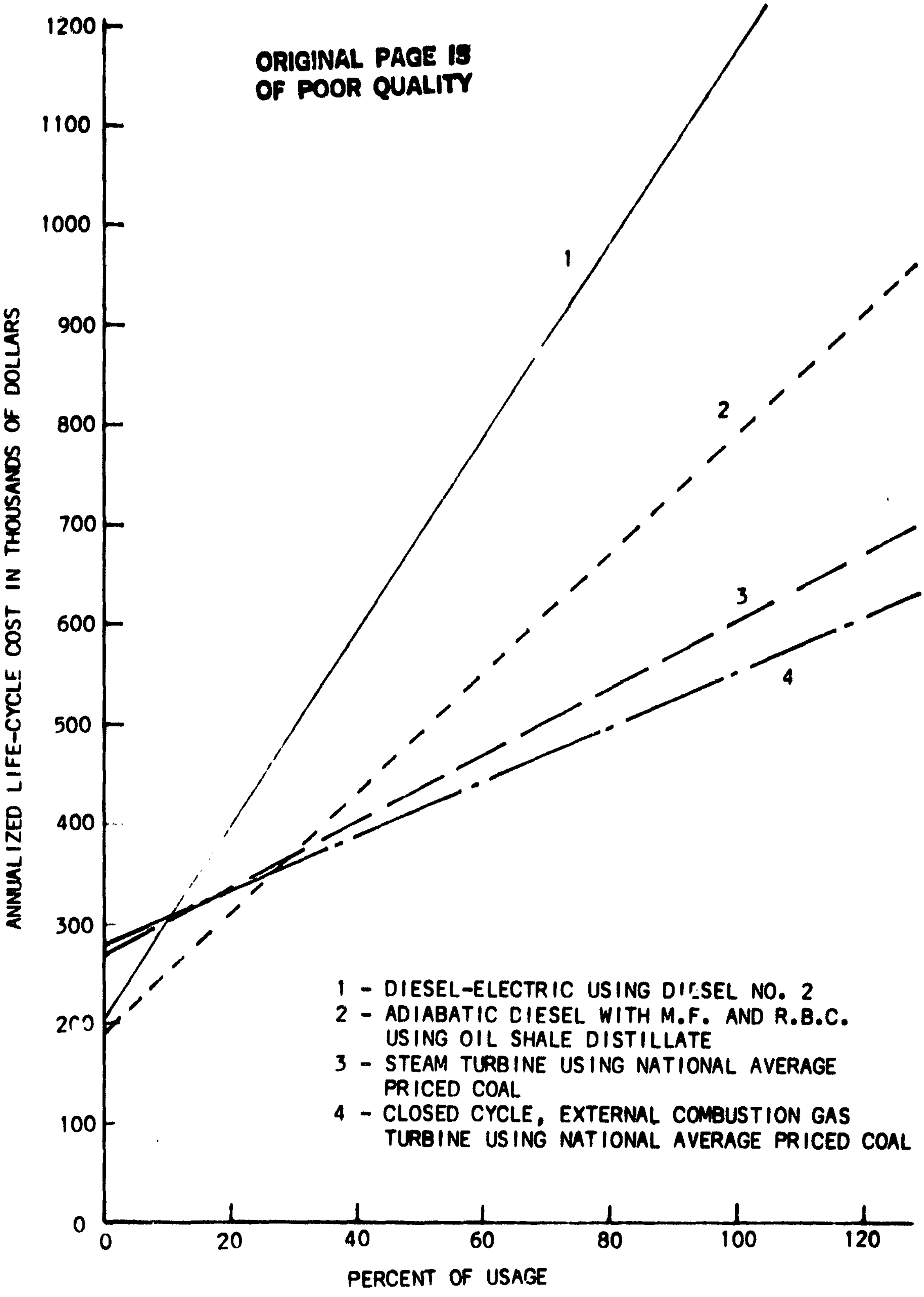

Figure 14-10. Effect of Usage on Life-Cycle Costs

operates on the medium duty cycle and the other half of the days, it is shut down. Ten percent usage is equivalent to having the engine used one day and out of service for nine days.

It is apparent that the usage has a significant effect on the competitive relationship between the various fuels and engines. At low usage rates, the capital costs become the dominant factor. At high usage rates, the fuel costs dominate. On this basis, it appears that the coal-fired locomotives are best suited to heavy duty line haul service and the Diesel is more suited to lighter duty branch service, short line haul service, and switch engines. A coal-fired switcher does not make sense.

No fuel or engine is best for all types of service today nor is it expected to be in the future. The individual railroads should decide the combination of the various locomotive options best suited to their needs on the basis of their usage patterns. Diesel engines, electric locomotives, and coal-fired locomotives all have a place. In addition there may be other options available in the future.

Y. END-TO-END ENERGY USAGE

All fuels require the expenditure of some amount of energy in their processing and distribution. Coal delivered to the locomotive at the mine mouth requires very little, less than 1% of the energy in the coal itself. At the other extreme, hydrogen made from coal requires 3.53 Btu of coal energy per Btu of hydrogen energy delivered to the locomotive. In the locomotive, the fuel energy is converted to energy at the rail. Combining these two conversion processes, the energy required at the source can be determined for each unit of energy delivered at the rail. The results of such energy calculations are presented in Table 14-27 for a variety of fuel sources, fuels, and engines. The engines are designated by the engine number listed in Table 14-10. The values for the source to fuel conversion process were taken from Ref. 14-17 while the fuel to rail conversion values are from JPL calculations.

Not surprisingly, the lowest end-to-end energy use is for petroleum which has been refined into Diesel No. 2 and used in the long term "mature" Diesel (No. 9) engine. The conversion efficiency of petroleum to Diesel No. 2 is high and the thermal efficiency of the long term "mature" Diesel engine is the highest of any of the heat engines investigated. By comparison, the present Diesel-electric locomotive is ranked No. 12 on this list with a value of 4.87 Btu at the source per Btu at the rail. The last ranked combination is for coal made into hydrogen for use in a fuel cell. This is not the worst case, only the lowest ranked on the list. Coal made into hydrogen for use in an internal combustion simple cycle gas turbine would require far more Btus at the source per Btu at the rail than any listed here. No attempt was made to cover all possible source-fuel-engine combinations. Such a list would include several hundred entries.

Those combinations which rank high on the life-cycle cost list (Table 14-19) and high on the energy usage list (Table 14-27) are the best ones for further investigation. They make the best use of both energy and financial resources. The first ranked combination on Table 14-20 was not analyzed for life-cycle cost but is likely to be similar to that of the

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Table 14-27. End-to-End Energy Usage

Rank	Source	Fuel	Engine No.	Btu at Source per Btu at Rail
1	Petroleum	Diesel No. 2	9	3.04
2	Coal	Electricity	Electrical	3.88
3	Coal	Coal, WMM[a]	17	4.22
4	Coal	Coal, WMM	15	4.28
5	Petroleum	Diesel No. 2	3	4.35
6	Oil shale	Distillate	9	4.35
7	Coal	Coal, Del[b]	17	4.35
8	Coal	Coal, Del.	15	4.41
9	Coal	Coal, WMM	14	4.66
10	Coal	Coal, Del.	14	4.80
11	Oil shale	Distillate	7	4.81
12	Petroleum	Diesel No. 2	1	4.87
13	Coal	Methane	9	4.95
14	Coal	Distillate	9	5.02
15	Oil shale	Distillate	5	5.15
16	Coal	Methanol	9	5.31
17	Oil shale	Distillate	6	5.36
18	Oil shale	Distillate	8	5.46
19	Coal	Distillate	7	5.55
20	Oil shale	Distillate	16	5.56
21	Oil shale	Distillate	4	5.59
22	Coal	Methanol	7	5.87
23	Coal	Ammonia	9	5.90
24	Coal	Distillate	5	5.94
25	Coal	Coal, WMM	11	6.11
26	Coal	Methanol	12	6.19
27	Coal	Distillate	6	6.19
28	Coal	Coal, Del.	11	6.29
29	Coal	Distillate	8	6.30
30	Coal	Distillate	4	6.43
31	Coal	Methanol	5	6.47
32	Coal	Methanol	6	6.55
33	Coal	Methanol	8	6.67
34	Coal	Methanol	2	7.24
35	Oil shale	Distillate	13	7.25
36	Coal	Ammonia	4	7.55
37	Oil shale	Distillate	18	8.33
38	Coal	Coal, WMM	10	8.72
39	Coal	Coal, Del.	10	8.98
40	Coal	Hydrogen	12	9.35

Notes: [a] Western Mine mouth
[b] Delivered, National Average Price

same engine using oil shale distillate which is ranked No. 11 on Table 14-19. No cost analysis was made for electric locomotives which are ranked second on Table 14-27. The third ranking in energy is the advanced Stirling engine using coal but it is ranked eighth in cost. The fourth ranked combination in energy is ranked first one in cost.

There are engines which are both cost and energy effective. Their ranking depends on the weight given to each of these two factors. Certainly, the closed cycle, external combustion regenerative gas turbine would be very near, if not, at the top of the list. From the standpoint of fuels, coal and oil shale distillate are the important alternative fuels.

Z. DEVELOPMENT COSTS

The costs of developing the new engines have not been mentioned so far in this section. These costs are not included in the analysis. Most of the new engines require a significant advance in technology and the relatively low production rate of locomotives (about 1500 per year) does not justify the large expenditure that will be necessary. However, the technology is applicable to more than just locomotives. Marine, pipeline compressors, and electrical utilities could share in the benefits which will accrue from the adiabatic Diesel engines, the coal-fired engines and the fuel cell. The development of new engines should be a shared program. The only engine which would not be a shared program is the first generation new steam engine which will use technology that is already well developed. Only the application is new.

The savings in life-cycle costs amount to $600,000 per year per locomotive. The yearly production of new locomotives is currently about 1200 units. If the savings for one year of operation amortized over one year of production was used for development funds, it would amount to $600,000,000. After the first year, the savings would be added profit. It should be possible to develop, test, and tool a new locomotive of any one of the three candidates; the adiabatic Diesel, the fluidized bed, or the fuel cell for the sum of $600,000,000. In production, the savings of $500,000 per year would pay for the locomotive in only two years.

SUMMARY

This life-cycle cost analysis has shown that locomotives with costs substantially lower than the present Diesel-electric units are economically possible. The coal-fired locomotives are the most attractive with life-cycle costs about half that of the present locomotives. The adiabatic Diesel using synthetic hydrocarbons is also an attractive candidate engine with life-cycle costs about 30% less than the present engines.

The cost of developing these new locomotives is not known since the cost could and should be shared with other programs using the same technology. However, the magnitude of the savings possible as compared to the present engines is great enough to justify the development of at least one new engine.

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