ALTERNATE MEANS OF POWER GENERATION AND

FUEL CONSERVATION IN SHIP OPERATIONS

by

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ABSTRACT

With the price of fuel rising continuously, fuel conservation remains a basic problem for all industries with savings in cost as an incentive and reward. Possible responses of the maritime industry to increases in fuel cost are undertaken. Alternate means of generating power using a high conversion efficiency and minimizing fuel consumption are descriped. These include bottoming cycles, COGAS plants and regenerative gas turbines used as propulsion machinery for marine power plants. Optimum steaming speeds are determined according to the objective of the shipowner or charterer. However, it has become clear that there is no painless method of saving fuel. The price that must be paid for fuel conservation may include cargos left behind, shippers receiving less frequent and slower service and higher operating costs.

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To Joan Dowd I wish to express my deepest thanks for her help and encouragement and for making this project possible.

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CHAPTER I

INTRODUCTION

The energy crisis was the cause for serious shortages of fuel during the winter of 1973-74. As a result, even though supply of fuel had reached an adequate level by the summer of 1974, the price of fuel went up three to four times when compared to the 1971-72 levels. Fuel conservation thus remains a basic problem for all industries, with savings in cost as an incentive and reward.

The purpose of this thesis will be to evaluate the potential for more effective use of fuel in ship operations. Accordingly, possible ways of reducing fuel consumption covering both technological improvements (power plants, hull form) and operating practices (ship speeds, ports of call) will be examined.

First, the thermodynamic conept of available useful work is discussed in an effort to gain an insight on the ultimate possibilities of saving fuel. By the use of this concept a calculation of the ideal or minimum fuel requirement of a given process is possible and thus maximum fuel savings which can be expected from entirely new processes that might be developed can be realized.

Second, possible savings of fuel with existing technology is estimated through the examination of combined

thermodynamic cycles. For this reason, the gas turbine and steam turbine combined plant for power generation is analyzed. Another attractive method is afforded by the use of an organic Rankine plant used as a bottoming cycle in converting waste heat of a diesel engine to mechanical power. Marine applications of such power plants in commercial ships is also discussed.

Finally, changes in shipping company operating practices are discussed in an effort to determine optimal strategies to be followed under various assumed fuel supplies. This way an attempt was made to determine possible ways that the shipping industry can respond to changing economic conditions and thus realize which responses will be most effective in maintaining acceptable levels of cost and service.

CHAPTER II

THERMODYNAMIC AVAILABILITY ANALYSIS

A. Available Useful Work

If one wants to evaluate the effectiveness with which fuel is used in various processes, consideration of properties other than energy alone is required. This becomes obvious when thinking of a cold but fully charged battery being more useful than a discharged battery that has the same total energy by its virtue of being hot.

The concept of available useful work is best illustrated by the use of an example. Consider the formation of CH_2 from its basic constituents. If CH_2 was to be formed in the best possible way, that is reversibly from carbon dioxide and water in the atmosphere, then the work required would be the available useful work of the fuel. This work would be identical with the work that would be recovered if the fuel were combined with oxygen in a reversible process which restores the carbon dioxide and water that were initially used to form the CH_2. Any oxidation process which resulted in the production of less work than the available useful work, would be a measure of the irreversibility of the process.

B. Fuel Oxidation Process

Figure 1 shows the curves calculated for one pound-mole of a liquid hydrocarbon fuel, namely CH_2, with a L.H.V. of 280,000 Btu. The available useful work for the reactants is about 291,000 Btu. per pound-mole. This work could be recovered by the following processes:

   a. The oxidation is carried out in a reversible fuel cell at a temperature T_A, while at the same time delivering electrical work to the environment.

   b. The products are then cooled to T_o as they provide heat to Carnot engines which produce further work.

   c. Each of the products CO_2, H_2O, and N_2 is separated from the mixture reversibly and also reversibly introduced in the atmosphere.

However, fuel cells are not currently available and the oxidation process occurs in a combustion chamber without production of electrical work. The resulting temperature at the end of the combustion process, which is shown by the line AB, is 4300° F. Because the process is irreversible, there is an increase of entropy and a loss of available useful work. This loss amounts to 80,000 Btu or 27% of the original available useful work. The remaining 211,000 Btu is the maximum work that can be obtained by transferring heat to Carnot engines while the products are cooled to the limit imposed by the environment at C.

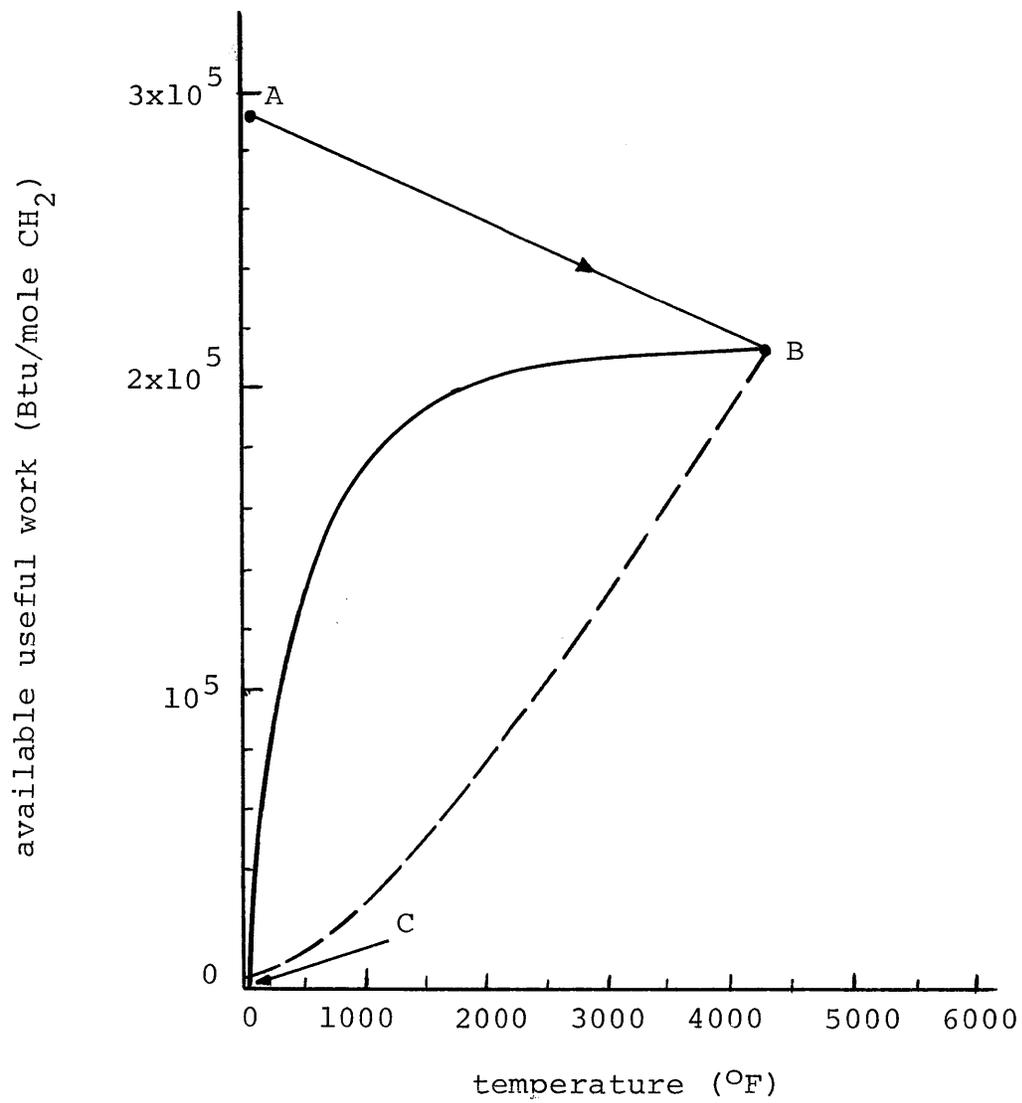Beginning with state B the available useful work can

FIGURE 1: Available useful work from hydrocarbon
oxidation process.

be altered in a number of ways. For example, by using a cooling process during which energy from the combustion products in the form of heat is transferred to any material at a temperature less than $4300°$ F. Because there is a finite temperature difference between the combustion products and the material, the process is irreversible and hence there is a loss in available useful work. The solid curve represents the available useful work contained in the combustion products plus that in a material at temperature $t°$ F, which has cooled the products to $t°$ F without itself changing temperature. The dashed curve shows the available useful work of the products of combustion at temperature t. The difference in the ordinates of the two curves is the available useful work from infinite heat capacity material, which has cooled the products from $4300°$ F to $t°$ F.

It is evident from the solid curve that as the temperature of the heat receiving material is lowered below $2000°$ F, the loss in the available useful work increases rapidly with decrease in temperature. Typical temperatures of the heat receiving water-steam working fluid in a steam power plant are between 600 and $800°$ F. As it can be seen from the graph at these temperatures the value of the available useful work is 48% and 53% of that for the fuel initially or 140,000 Btu and 155,000 Btu per mole of fuel respectively.

The measure of effectiveness of a process is found by dividing the net work produced by the process with the available useful work of the fuel consumed. On the other hand, the efficiency of a thermodynamic cycle is found by computing the ratio of net work output to the heat input and not the available useful work of the fuel used. The results are, however, the same. This can be readily seen from the following example. Suppose that we compare two Rankine cycles with the only difference occurring in the heat receiving water-steam temperatures, all other parameters being the same (same turbine and pump efficiencies and same operating pressures), one would expect that for the higher heat receiving water-steam temperature the efficiency would be better, since the available useful work of the fuel is bigger for the higher operating temperature. Indeed calculations carried out for a Rankine cycle with turbine isentropic efficiency of 0.8, pump efficiency of 0.7, condenser pressure of 2 lbf/in^2 and top pressure of 600 lbf/in^2 support this fact. For top temperatures of 650, 850 and 1150° F the respective thermal efficiencies were found to be 0.26, 0.29 and 0.31. This suggests using the highest top temperatures possible for such cycles. On the other hand, it should be a guide to research in finding working fluids other than steam, which for a given range of top operating temperatures, will give better net work output than steam does.

This has already been achieved for top operating temperatures between 500 and 1000° F through the use of organic fluids.

The effectiveness of use of fuel could also be improved in many processes by recovery of useful energy that is now lost as sensible heat of exhaust gases. Regenerators can help in reducing fuel consumption by returning some of this energy to the process. This is usually carried out by preheating the combustion air with heat transferred from the exhaust gases.

Another method of transforming this low grade heat into mechanical power is by using an organic Rankine cycle as a bottoming plant. A typical application of the bottoming cycle engine for diesel engine exhaust heat recovery, which is discussed in detail in the following chapter, helps raise the efficiency of the diesel engine from 39% to that of 48% for the combined cycle. This represents an increase of 24% with no increase in fuel consumption. The savings that can be realized by using such a combined cycle are then obvious.

CHAPTER III

BOTTOMING CYCLES

Many processes currently reject waste heat at relatively low temperatures of 300 to 700° F. In some circumstances this heat is fed back into the process by means of regenerators, while on others it is used to generate steam. More often, however, this low grade heat is not utilized because of marginal economics or because process steam is not needed at the site.

One of the methods of converting waste heat into mechanical power is illustrated in this chapter. The approach is based on the use of a diesel engine combined with a Rankine bottoming plant engine using organic working fluid and operating on the diesel reject heat so as to maximize the overall efficiency. The use of the organic Rankine cycle system as a bottoming plant for a diesel engine is justified when comparing it with a steam plant for the following reasons:

    a. Simpler turbine with fewer stages.

    b. Higher power output, thus maximizing the combined cycle efficiency.

    c. Higher turbine efficiency for low power systems.

    d. Turbine expansion solely in the superheated vapor region.

    e. Non-corrosive working fluid.

    f. Competitive capital cost.

A. Diesel Engine and Organic Rankine Cycle Performance

The diesel engine examined to be used in the combined cycle was the Colt-Pielstick PC2 and both its characteristics and performance are summarized in Table 1. The engine has a continuous power capability of 500 bhp/cylinder at a speed of 514 rpm. At this continuous power rating, it has an efficiency of 38.9% and an exhaust gas temperature of 815° F for use in the bottoming plant. Of particular interest regarding the coupling of the bottoming plant are both the high exhaust gas temperature and the high efficiency at part load operation.

Of particular importance regarding the Rankine cycle performance is the selection of the working fluid. Hundreds of fluids have been examined over the last 10 years and the one found to have desirable characteristics is known as Fluorinol 85. This working fluid is a mixture of 85 mole percent trifluoroethanol (CF_3CH_2OH) and 15 mole percent water. The system flow schematic of the combined diesel and organic Rankine cycle is shown in Figure 2. Figure 3 shows both the exhaust gas heated cycle and the steam heated cycle on T-S diagrams of Fluorinol 85. Energy from the exhaust gas recovered in the exhaust gas heated boiler is used to generate high pressure Fluorinol 85 vapor, which is then expanded in a high pressure turbine. By using the superheated vapor leaving the turbine, a regenerator transfers

TABLE 1

Characteristics and Performance of the Colt-Pielstick PC2 Diesel Engine

Speed, rpm   514
Horsepower/Cylinder
    Base Rated (100%)   500
    Peak Rated (110%)   550
Number of Cylinders   12

|	110	100	75	50	25
% of continuous power	110	100	75	50	25
BMEP, psi	241	219	164	109.5	54.8
Input (LHV)	6582	6528	6475	6553	7096
Useful work, Btu/bhp-hr	2545	2545	2545	2545	2545
Exhaust temperature, $^\circ$F	822	815	790	698	610
Air consumption, lb/bhp-hr	13.3	13.6	14.6	17.5	22.5
Fuel consumption, lb/bhp-hr	0.362	0.359	0.356	0.360	0.390
Exhaust Gas, lb/bhp-hr	13.66	13.96	14.96	17.86	22.89
η (LHV)	0.387	0.389	0.393	0.388	0.359
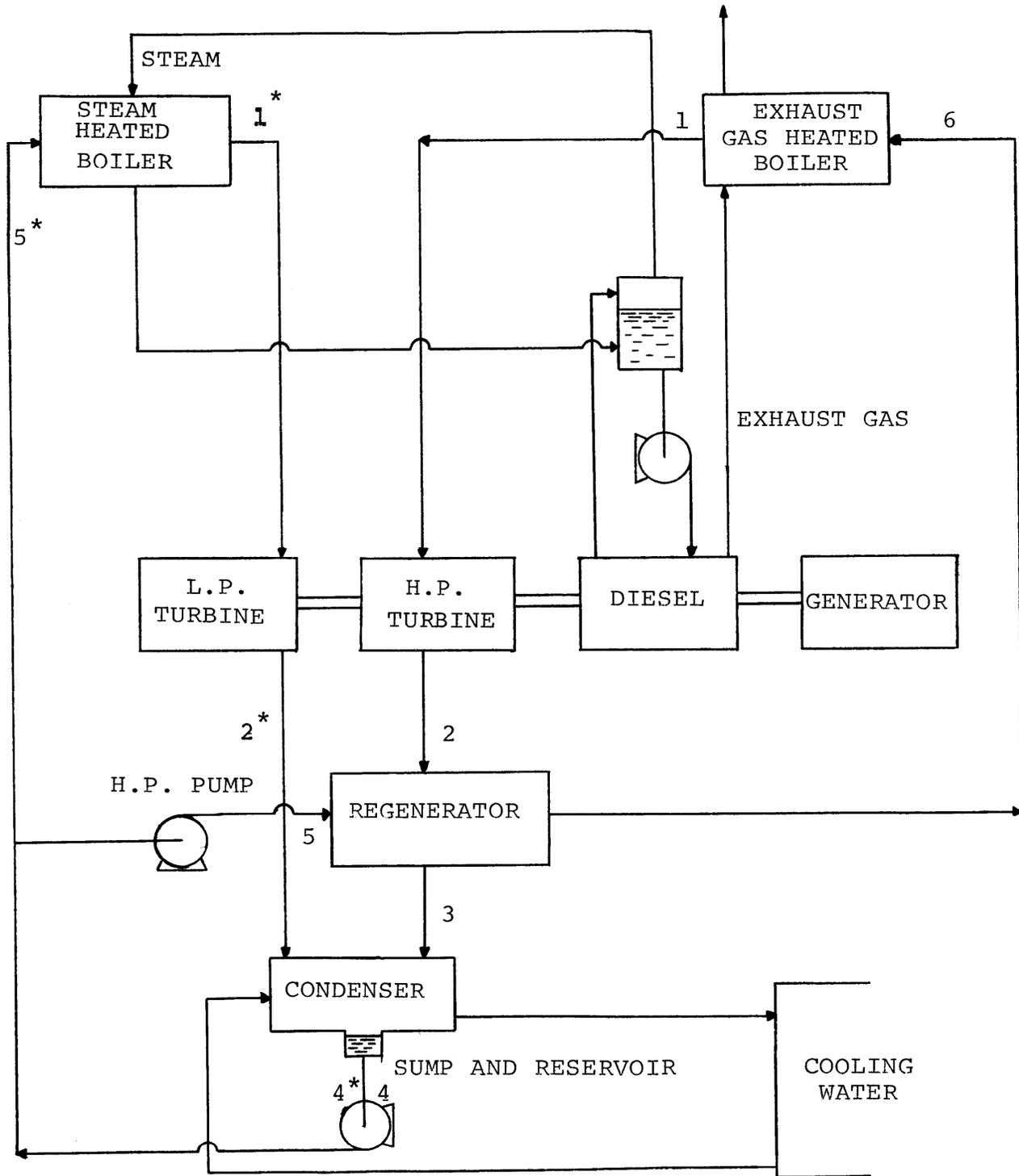
FIGURE 2: Flow schematic of combined diesel and
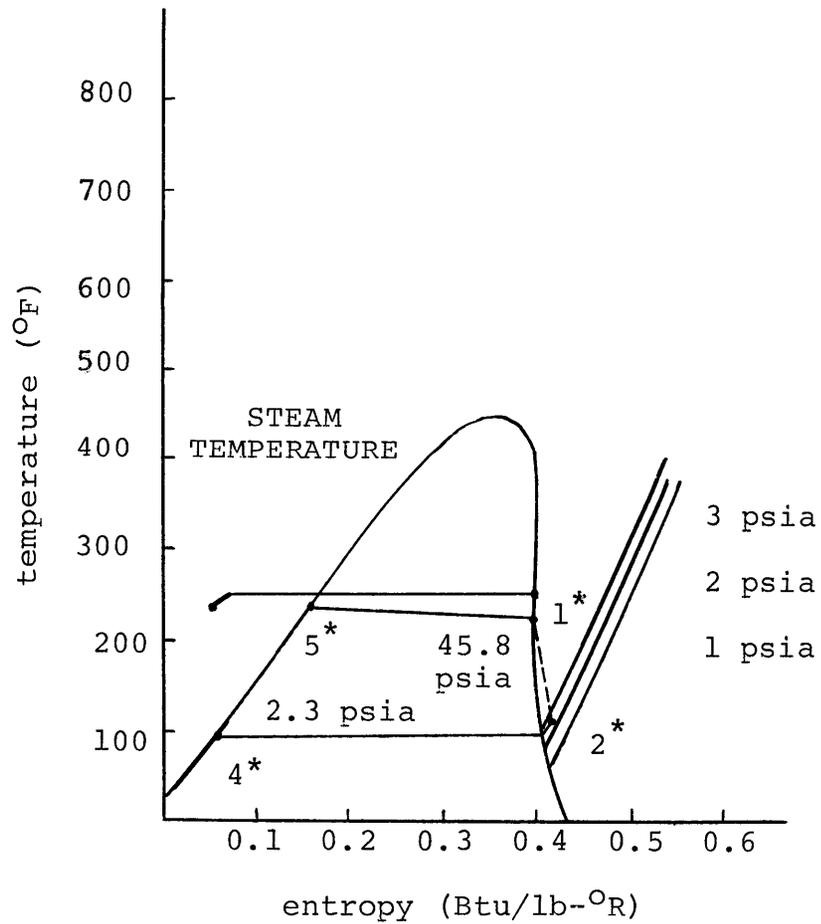organic Rankine cycle.

- 18 -
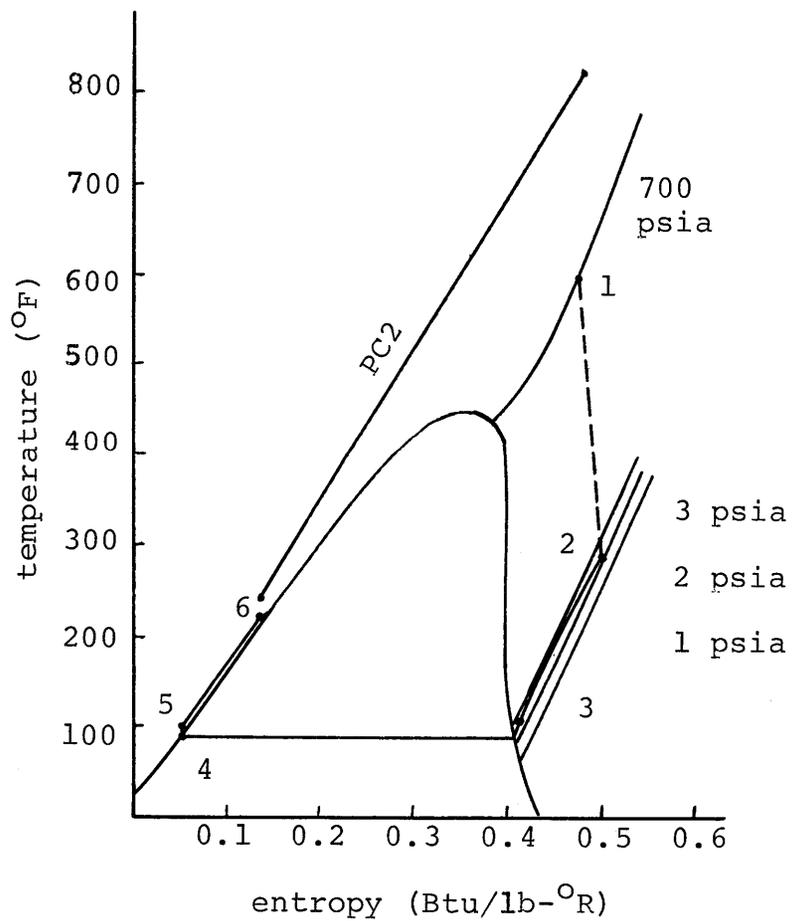
FIGURE 3: Exhaust gas heated cycle (left) and steam heated cycle (right) on T-S diagram of Fluorinol 85 working fluid.

heat to the feed liquid going to the exhaust gas heated boiler. The vapor from the regenerator goes to the condenser at near saturation conditions, where it is condensed to liquid by rejection of heat to cooling water. Part of the condensed liquid, pressurized to steam heated boiler pressure by a low pressure pump, goes to the steam heated boiler, while the rest goes to the high pressure pump which supplies pressure to the exhaust gas heated boiler. The vapor exiting the steam heated boiler is then expanded through a low pressure turbine. The exhaust vapor from the low pressure turbine, which is near saturation, goes directly to the condenser.

The specific cycle conditions of both the exhaust gas heated cycle and the steam heated cycle are given in Table 2. Note that with a peak cycle temperature of 600° F for the exhaust gas heated cycle, exhaust gas temperatures with a lowest value of approximately 650° F can be utilized. With reference to Table 1, it can be seen that the organic Rankine cycle system using Fluorinol 85 can be operated down to 50% continuous rated diesel power, at which point the exhaust gas temperature is 698° F.

TABLE 2

Specific Cycle Conditions of Both Exhaust Gas
Heated and Steam Heated Cycles

	Exhaust Gas Heated Rankine Cycle	Steam Heated Rankine Cycle
BOILER		
Outlet pressure (psia)	700	45.85
Outlet temp. (^{O}F)	600	225
Q_H (Btu/lb)	286.9	229.5
ΔP (psi)	70	10
TURBINE		
Exhaust temp. (^{O}F)	290.3	106
Exhaust pressure (psia)	2.6	2.3
W (Btu/lb)	84.1	28.5
$\eta_{O.A.}$	0.81	0.71
REGENERATOR		
Effectiveness	0.90	
Q_R (Btu/lb)	52.7	
Exit vapor temp. (^{O}F)	112.8	
Exit liquid temp. (^{O}F)	217.3	
CONDENSER		
Pressure (psia)	2.3	2.3
Temperature (^{O}F)	90	90
Q_C (Btu/lb)	200.8	199.7
FEEDPUMP		
W_P (Btu/lb)	2.49	0.17
$\eta_{O.A.}$	0.70	0.70
NET SHAFT WORK (Btu/lb)	81.6	28.3
CYCLE EFFICIENCY	28.44%	12.33%

B. Combined Cycle Performance

The combined cycle performance obtained by coupling the organic Rankine bottoming cycle to a Colt-Pielstick PC2 12 cylinder diesel engine operating at a continuous rated power is given in Table 3. The exhaust gas temperature leaving the boiler is 242° F and the exhaust gas vapor variation through the boiler is illustrated on the T-S diagram in Figure 3. By the use of this combined cycle the diesel power of 6000 bhp is increased to 7438 hp or about 24% for the same fuel input. As a result the efficiency of the combined engine cycle is increased from 38.9% to 48.3%.

TABLE 3

Combined Cycle Performance

DIESEL ENGINE

Rated power/cyclinder	500 bhp
Rated power	6000 bhp
BSFC	0.359 lb/bhp-hr
Exhaust gas temperature	815° F

EXHAUST GAS HEATED ORGANIC RANKINE CYCLE

Stack temperature	242° F
Exhaust flow rate	83.75×10^3 lb/hr
Heat transferred to organic	11.59×10^6 Btu/hr
Organic flow rate	40.3×10^3 lb/hr
Work (Net)	3.3×10^6 Btu/hr = 1293 hp

STEAM HEATED ORGANIC RANKINE CYCLE

Heat transferred to organic	2.99×10^6 Btu/hr
Organic flow rate	13.01×10^3 lb/hr
Work (Net)	3.69×10^5 Btu/hr = 145 hp

COMBINED CYCLE

Diesel	6000 hp
H.P. turbine	1293 hp
L.P. turbine	145 hp
Total	7438 hp
% increase in power	24
BSFC	0.289 lb/bhp-hr
η_{SHAFT}	48.3%

C. Economic Evaluation

In order to make an economic evaluation of the
entire combined plant there is a key question that has to
be answered. If one has decided upon installing a diesel
power plant, what is the breakeven cost of the organic
Rankine cycle bottoming plant at which the yearly fuel saving
is balanced by the yearly capital cost of the bottoming plant?

A detailed economic analysis of the costs of both
the diesel and the organic Rankine cycle plants will help
answer this question. The installation price of the organic
Rankine cycle bottoming plant is shown in Table 4. As
expected, the $/kwe cost of the steam heated portion is
substantially higher than that of the exhaust gas heated
portion because of the lower heat source temperature. It
should be noted that a cost optimization was not performed
on the system. Opportunities for substantially reducing the
cost exist in several areas, i.e. increasing the pinch point
ΔT in the exhaust gas heated boiler, redesign of some com-
ponents and combining the two turbines into a single unit.

It should be noted that the values calculated were
performed for a power plant producing electricity. The
savings that can be realized, however, by utilizing a similar
combined cycle plant as the propulsion plant of a ship are
obvious. A power plant cost summary for a plant comprising
nine PC2-12 cylinder combined with an organic Rankine cycle

TABLE 4

Installation Price of the Organic
Rankine Cycle Plant

	Power (Design Point),kwe	Inst. Price, $	Inst. Price,$/kwe
Exhaust gas heated portion	1009	316,200	313
Steam heated portion	117	85,100	727
Combined	1126	401,300	356

bottoming plant is given in Table 5. The cost in $/kwe
is given on the assumption that the plant is operating at
100% continuous rated power. The unit cost of $254/kwe
can be compared to current costs for 1000-MWe central station
plants of approximately $400/kwe for oil-fired steam,
$600/kwe for coal-fired steam and $700/kwe for light-water
nuclear.

Assuming now that the operation and maintenance cost
per unit of generated power will be the same for both the
diesel plant alone and the combined diesel and organic Rankine
cycle plant, the breakeven cost for the organic Rankine cycle
bottoming plant is given by the following relation:

$$I_{org. Rank. cycle} = I_{diesel} + \frac{29.9 \ (c_q) \ L}{\eta_{diesel} \ (r)}$$

where $I_{org. Rankine \ cycle}$ = breakeven installed cost for the
organic Rankine cycle bottoming
plant, $/kwe

I_{diesel} = installed cost for the diesel
providing the waste heat
utilized by the organic Rankine
cycle bottoming plant, $/kwe

η_{diesel} = diesel efficiency

r = capital charge rate per year

L = plant load factor

c_q = fuel cost for diesel, $/$10^6$ Btu

The breakeven cost is presented in Figure 4 as a
function of load factor with the ratio of c_q/r as a variable.

TABLE 5

Power Plant Cost Summary for Plant Comprising
Nine PC2-12 Diesel Engine Units Combined
With Organic Rankine Cycle Bottoming Plant

Component	Total Cost, $	Unit Cost, $/kwe total plant rating
Diesel/generator equipment	6,768,000	138.69
Installation of diesel	1,186,000	24.30
Organic Rankine cycle, Equipment and Installation	3,379,000	69.24
Cooling tower system installed	295,000	6.05
Building equipped	656,000	13.44
Land and improvements	48,000	0.98
Fuel storage	58,000	1.19
TOTAL		253.89

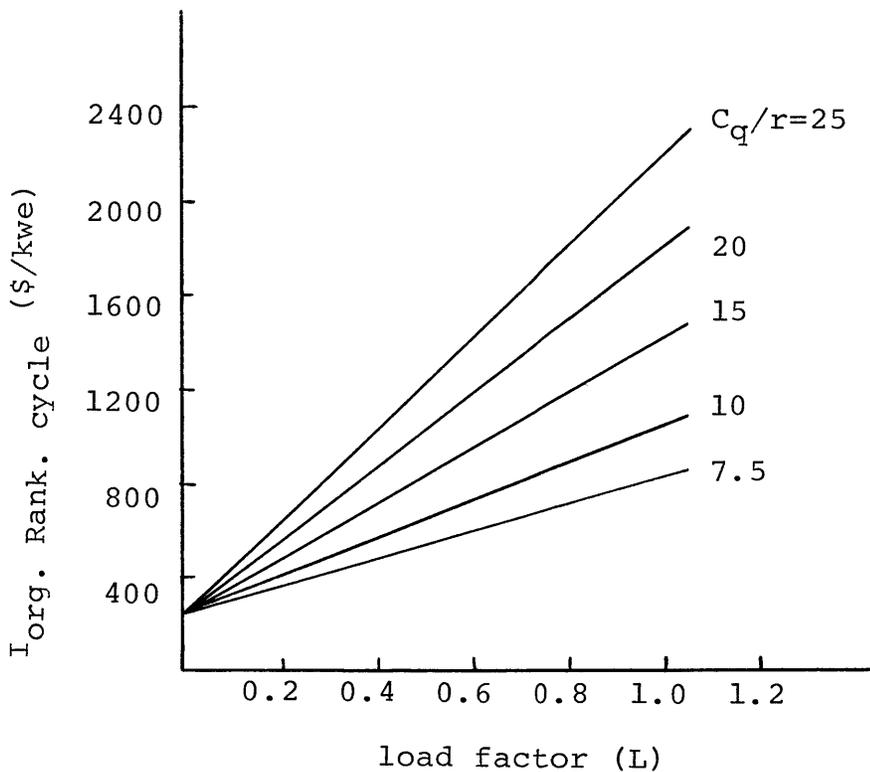| C_q,$/10^6$ Btu	C_q/r for		
	r=0.10	r=0.15	r=0.20
1.50	15	10	7.5
2.00	20	13.3	10
2.50	25	16.7	12.5

FIGURE 4: Breakeven capital cost for organic Rankine cycle bottoming plant for Colt-Pielstick PC-12 diesel.

Assuming an incremental cost for the organic Rankine cycle bottoming plant of $400/kwe including building and land requirements ($356/kwe installed equipment cost plus $44/kwe incremental cost), the load factor for economic feasibility of adding the organic Rankine cycle plant is:

High Fuel Cost/Low Capital Charge Rate, L > .08

Low Fuel Cost/High Capital Charge Rate, L > 0.29

Addition of the organic Rankine cycle plant to the diesel engine is thus justified at surprising low load factors. For high load factor applications, such as a propulsion power plant of a ship, there is very strong economic incentive for the use of such an organic Rankine bottoming plant.

CHAPTER IV

COMBINED GAS-STEAM ENGINE CYCLE

The gas turbine and the steam turbine can be combined to offer attractive power generation cycles. As a result, in recent years interest in the combined gas-steam turbine cycles has been growing rapidly. The advantage of the waste heat recovery or COGAS plant, as it is commonly called, is energy conservation. Because of its basic economy, the COGAS plant has already found significant application in the electric utility industry. It is for this reason that some of the examples examined in the sequel refer to electric power generation plants and are hence bigger plants than one could use as propulsion machinery for ships. However, there is currently a great amount of research in "marinizing" the COGAS cycle for commercial ship applications. In examining such a plant, the main interest is in savings that can be realized and hence a different approach than that which a design engineer would take in solving the problem is followed.

A. Basic Components

A combined gas-steam turbine cycle consists of a gas turbine, an exhaust waste heat recovery or conventionally fired boiler and a steam turbine, which may or may not be a condensing machine. However, these plants have demonstrated high efficiency particularly when the steam produced has been condensed in the process. The application of a condensing steam cycle presupposes that an adequate supply of cooling water is available. This is indeed the case for the COGAS plant used as a propulsion plant in ship operations.

In one form of arrangement, the gas turbine exhaust is supplied as preheated air to the combustion chamber of a conventional fired boiler steam turbine cycle. Because the gas turbine operates on full load, at an air fuel ratio of between 50:1 and 70:1 the exhaust gases are clean and only lightly contaminated by combustion products. The oxygen content of the exhaust may thus be up to 18%, which is adequate to support combustion. This form of cycle has been called the "High Efficiency Combined Cycle."

Another possible cycle is that in which a waste heat recovery boiler generates steam from the energy remaining in the gas turbine exhaust and subsequently expanding it in a steam turbine. This form of cycle has been called the "Regeneration Combined Cycle." In this combined form what is known as supplementary firing is permitted, during

which the waste heat boiler is additionally equipped to
burn fuel in the gas turbine exhaust, thus helping raise the
temperature of the gases before they enter the boiler.
Supplementary firing increases the quantity of steam generated
by the exhaust boiler and improves the flexibility of the
design by making the generated steam conditions independent
of the gas turbine's exhaust temperature.

B. Waste Heat Boiler

        In the recuperation cycle, the gas turbine exhaust
heat is recovered in a waste heat boiler, which in its
simplest form comprises of three components--a superheater,
an evaporator and an economizer--as shown in Figure 5. The
controlling feature is the pinch point temperature, which is
the smallest temperature difference between the exhaust gas
passing through the boiler and the steam saturation tempera-
ture at entry to the evaporator. It would thus appear
desirable to lower the pinch point temperature to the
smallest value possible so as to obtain the maximum heat
recovery. This, however, involves a disproportionately
large increase of the heat transfer surface area in the
boiler, as the pinch point temperature is reduced. A common
practice is to make this temperature difference around 80° F.
One American manufacturer (Vogt) states that the cost of
a boiler will increase approximately 4% for every 4° reduction
in the pinch point temperature. Today with fuel costs being
extremely high, economic justification may exist for such
reductions.

        Supplementary firing can also be adopted to increase
the temperature and pressure of steam for a given pinch point
temperature. A low pressure evaporator may also be of value
so as to avoid the possibility of dew point condensation of
exhaust gas, when the feedwater temperature is below the

Tg   gas temperature
TS   water-steam temperature
Tg_3-TS_3 pinch point,approximately
     82 OF
TS_3-TS_2 economizer approach
     temperature, typically
     approximately 72 OF
Tg_1-TS_4 superheater approach
     temperature, typically
     approximately 150 OF

FIGURE 5: Simple Recuperation cycle.
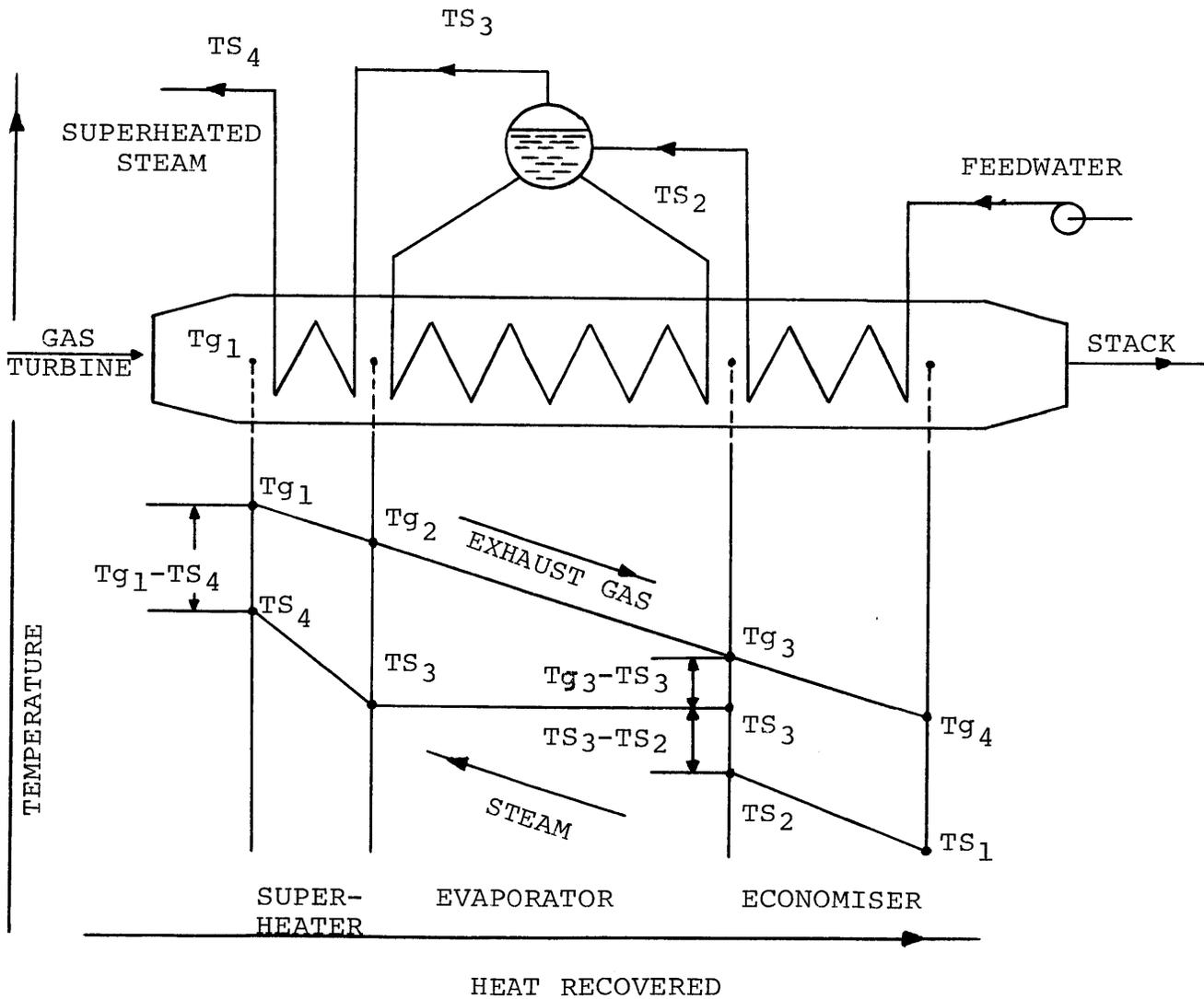
dew point. If a low pressure evaporator is not used, then a section of the economizer must be made corrosion resistant, which is usually more costly than the provision of the low pressure evaporator. Figure 6 portrays the temperature profile diagram including the features described above.

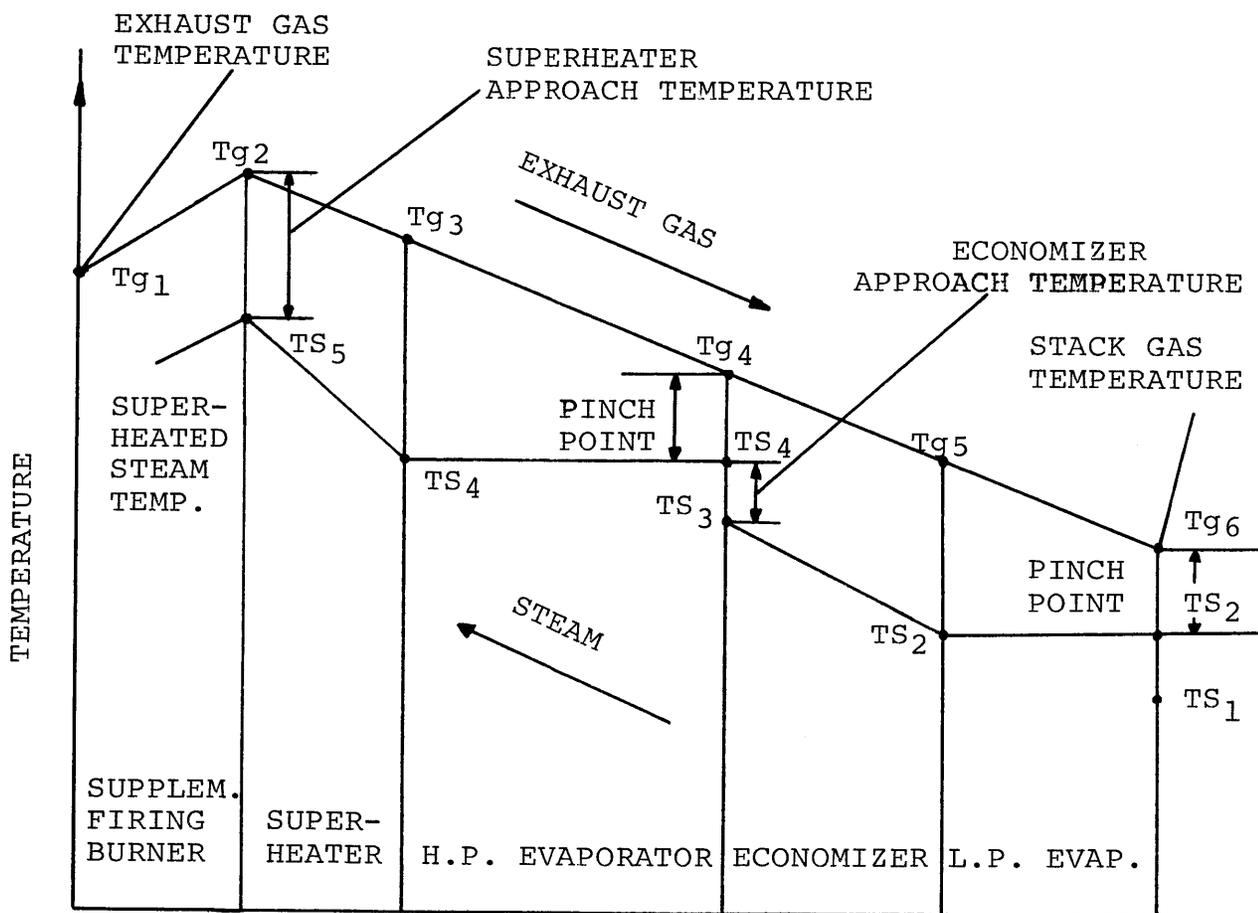Tg gas temperature
TS steam and feedwater
 temperature

FIGURE 6: Dual pressure Recuperation cycle.

C. Waste Heat Boiler Economics

The Vogt Company has published interesting waste
heat boiler performance data. For example, with 895,000 lb/hr
gas turbine exhaust at 880° F it is possible to produce
160,000 lb/hr steam at 1250 psig and 950° F with a supple-
mentary firing temperature of 1240° F and a natural gas fuel
input of 104 x 10^6 Btu/hr.

Changing the pinch point temperature for the unfired
case, results in variation of steam production. For example,
decreasing the pinch point from 111° F to 72° F gives a
13% increase in steam output but with an 18% increase in
boiler cost. A similar decrease in pinch point temperature
for the fired case, decreases the supplementary fuel input by
12% with a simultaneous increase in boiler cost of 15%.

Changes in steam pressure and superheat temperature
will also affect boiler performance and cost. For example,
in the unfired case, a change from 600 psig, 750° F to
250 psig, 500° F while keeping the steam production and the
pinch point constant, will reduce the boiler cost by 25%.
On the other hand, in the fired case, a change from 600 psig,
750° F to 1250 psig, 950° F with constant steam output will
increase boiler cost by 25%. For this case the supplementary
fuel input also increases by 20%.

It is therefore seen that considerable flexibility
exists in determining the operating conditions for the steam

part of the cycle, particularly with respect to the economics

of the system. In practice, however, this may not be the case

because of the necessity to utilize existing and readily

available marketed equipment rather than custom built units.

D. High Efficiency Cycle Performance

The performance of the high efficiency cycle has been examined carefully and systematically by Aguet. In this study a comparison is made between a classical steam cycle and an identical cycle of the combined gas-steam turbine type, with the same steam pressures and temperatures, same turbine efficiencies and the same preheating system.

The steam cycle operates at 180 bar pressure and 1004° F steam and reheat temperature. There are 4 low pressure and 3 high pressure feed heating stages and the net thermal efficiency is calculated at 40.7%, while the net output is 225.6 MW.

The gas turbine of the combined cycle is so chosen that there is an excess of combustion air of 5% and the supply of heat from the boiler to the steam cycle due to combustion of fuel in the boiler is exactly the same as for the power steam cycle. Radiation losses are the same as before. This way all differences that appear in the combined cycle are due to the supply of heat from the gas turbine. The turbine entry temperature is 1740° F and the exhaust temperature 920° F. The exhaust temperature leaving the boiler is 750° F.

The combined cycle efficiency was calculated at 43.86%, with a total net output of 316.12 MW of which 56 MW

is supplied directly by the gas turbine. Thus the addition of the gas turbine to the steam cycle gives a reduction of 7.12% in specific fuel consumption and an increase of 40.14% in net output.

E.   Regeneration Cycle Performance

Stal-Laval offer a range of complete combined cycle
plants. A diagrammatic arrangement for the largest plant
with heat balance figures for full load is shown in Figure 7.
Two 90 MW gas turbines each exhaust into a waste heat boiler
incorporating a superheater, high pressure evaporator and
economizer. At the cold end, the feedwater tank/deaerator
has for heating the condensate an associated low pressure
evaporator which can be oversized to generate low pressure
steam for admission to the steam turbine and allows the gas
turbine exhaust to be cooled down to the dew point. This
improves the overall thermal efficiency to between 45% and
47% at design load point, depending on the gas turbine fuel
and operating conditions.

	(°C)	(kg/s)
Station	T	m
A	452	362
B	114	362

Gross plant output: 252.2 MW
Auxilliary power   :   2.2 MW
Net plant output : 250.0 MW
Net thermal efficiency: 47.0%

	(bar)	(°C)	(kJ/kg)	(kg/s)
Station	p	T	h	m
1	0.068	38.8	161	87.2
2	2.95	133	559	39.9
3	28.10	409	3257	33.6
4	2.95	133	2724	10.0

FIGURE 7: Stal-Laval Recuperation cycle.

CHAPTER V

APPLYING THE HEAVY DUTY GAS TURBINE TO MARINE PROPULSION

During the course of the past five years, heavy duty industrial type marine gas turbines have been applied to marine propulsion systems on several ships. This is primarily a result of the advantages of this type of prime mover in the areas of weight, volume, acquisition cost, availability and manning requirements. The trend has appeared in both military as well as commercial vessels.

The heavy duty gas turbine that will be discussed in the sequel operates at cycle pressure ratios of between six and eight to one and turbine inlet temperatures from 1650 to 1730° F. At these cycle conditions, it is economical to use exhaust heat recovery in a regenerative cycle to improve specific fuel consumption. This type of gas turbine also has a high degree of fuels flexibility and has operated on treated marine residual fuels quite successfully.

A. Gas Turbine Description

The gas turbines described here operate on the Brayton cycle and use atmospheric air as a working fluid. In the regenerative gas turbine cycle, some of the heat added to the cycle may be recovered from the heat rejected to the atmosphere if there is a sufficient temperature differential between the compressor discharge temperature and the turbine exhaust temperature. Design-point performance characteristics of four models of heavy duty gas turbines are summarized in Table 6 for both distillate and residual fuel operation. The change from design output, design SFC and design airflow with changing compressor inlet temperature can be approximated from the following equations:

$$(\% \ HP)_{DES} = 127.46 - 0.4654 \ T_{AMB}$$

$$(\% \ SFC)_{DES} = 87.98 + 0.2038 \ T_{AMB}$$

$$(\% \ W_{AIR})_{DES} = 112.83 - 0.2174 \ T_{AMB}$$

Ambient temperature is in degrees Fahrenheit. These equations are valid from 120° F down to 35° F, at which point the demister anti-icing system becomes operative to hold constant compressor inlet temperature.

The effect of changes in atmospheric pressure are approximated by the following equations:

$$(\% \ HP)_{DES} = 3.3420 \ P_{AMB}$$

$$(\% \ W_{AIR})_{DES} = 3.3420 \ P_{AMB}$$

TABLE 6

Gas Turbine Performance

| | MS - 3002 | | MS-5002R"A" | | MS-5002R"B" | | MS - 7002R | |
	Dist.	Resid.	Dist.	Resid.	Dist.	Resid.	Dist.	Resid.
Output, bhp	12650	11550	23850	22300	29300	27450	59800	55450
SFC, lb/hp-hr	0.42	0.453	0.417	0.440	0.418	0.450	0.424	0.466
LP speed, r.p.m.	6500	6500	4670	4670	4670	4670	3020	3020
Comp. speed, r.p.m.	7100	6900	5100	5100	5100	5100	3600	3600
Air flow, lb/sec	111.7	108.9	213	213	254.5	254.5	525.3	525.3
Turb. exhaust, ^{O}F	1001	961	993	956	946	911	946	906
Regen. exhaust, ^{O}F	657	630	640	627	662	647	661	647
Firing temp., ^{O}F	1730	1650	1705	1650	1710	1650	1700	1650

Atmospheric pressure is expressed in inches of mercury.
Atmospheric pressure has no effect on specific fuel consumption, since compressor and turbine pressure ratios, and therefore temperature ratios and internal gas velocities, remain unchanged.

Design output and specific fuel consumption are also sensitive to pressure drops occurring in the inlet and exhaust ducts as shown by the following equations:

$$(\% \ HP)_{DES} = 101.58 - 0.5250 \ \Delta P_{inlet}$$

$$(\% \ SFC)_{DES} = 99.18 + 0.2750 \ \Delta P_{inlet}$$

$$(\% \ W_{AIR})_{DES} = 100.75 - 0.2500 \ \Delta P_{inlet}$$

$$(\% \ HP)_{DES} = 101.38 - 0.2750 \ \Delta P_{exhaust}$$

$$(\% \ SFC)_{DES} = 98.63 + 0.2750 \ \Delta P_{exhaust}$$

Pressure drops are expressed in inches of water gage. Exhaust pressure drops do not have any influence on cycle airflow.

Gas turbines display a very flat power curve for any given fuel flow. This is analogous to the steam turbine speed-power characteristic when operated at constant steam flow. In both cases the speed torque characteristic shows an increase in torque with a decrease in speed. As a result of this inherent characteristic, the gas turbine is capable of maintaining constant power output, even under overload conditions such as might be caused by a fouled hull or heavy seaway. When combined with the essentially negligible degradation of performance with time, this results in the

ability to operate for extended periods of time at the maximum continuous rating. In other words, for the heavy duty gas turbine, the maximum continuous rating and the continuous service rating can be the same.

The regenerator consists of one or more static plate and fin heat exchangers that transfer heat from the gas turbine exhaust to the compressor discharge air. The regenerator modules are located in the gas turbine exhaust system and are connected to the gas turbine by piping which runs from the regenerator to the combustion wrapper. Despite the recovery of a substantial portion of the exhaust heat of the gas turbine by the regenerator, a considerable amount of energy is still available for auxilliary purposes. Several marine installations to date have made use of this capability, through either exhaust heat recovery boilers or heat recovery systems using a thermal fluid, to provide energy for cargo heating, fuel oil heating, compartment heating and operation of evaporators.

B. Future of the Heavy Duty Marine Gas Turbine

The indicated direction or evolution of the U.S. merchant fleet is toward more productive ships, thereby increasing revenue and return on investments. This demand increases the relative importance of improvements and innovations in propulsion machinery systems, in order to minimize the effects of machinery and fuel oil weight on acquisition costs and operating costs.

Two major factors had initially contributed to the limited acceptability of the industrial regenerative gas turbine as a prime mover for marine propulsion machinery systems; specifically, its comparatively unproven capability to burn efficiently, economically and reliably the standard marine fuel oil, Bunker C, and its dependance upon an external device for reversing the direction of propeller thrust.

As a result of the above mentioned factors, a number of research and development task efforts were redirected. The net effect of this redirection was to place emphasis upon hardware development and marine applications rather than on research and development. On this basis, specific tasks were identified and defined. These include:

      a. Corrosion and deposition reduction: aimed at developing the technology for reducing hot corrosion and deposition rates at high firing temperatures.

b. Fuel oil conditioning system improvement:
    aimed at developing the technology and
    hardware necessary to permit automatic analysis
    and treatment of residual fuel oil.

c. Regenerator improvement: aimed at designing
    a highly reliable, low cost and low weight
    regenerator with substantial improvements
    in maintenance requirements.

d. Reversing gas turbine development: aimed at
    developing an efficient and economic, internally
    reversing gas turbine capable of meeting all
    maneuvering requirements of a shipboard
    propulsion machinery system, with a minimal
    effect upon the acquisition costs of the prime
    mover.

The current world-wide economic climate and fuels
situation has focused critical attention on power plant
performance, with emphasis in the area of specific fuel
consumption. Improvements in future marine gas turbine
installations will generally fall into two categories,
namely, upratings of the basic gas turbine through increases
in airflow and firing temperature and the use of binary
cycles to obtain improved plant efficiencies using present
configurations of gas turbines.

Combined cycles using steam as a secondary working
fluid to reduce the heat rejected to the atmosphere are
achievable using current state of the art technology. The
mechancial arrangement of such a plant can be accomplished in
several ways. In one form of arrangement known as the STAG
arrangement, the steam turbine is directly connected to a
separate high-speed pinion of the main reduction gear. The

potential performance for such combined cycles is shown in
Figure 8. It must also be pointed out that most combined
cycles confine the combustion process solely to the gas
turbine, which vastly simplifies the steam generator.
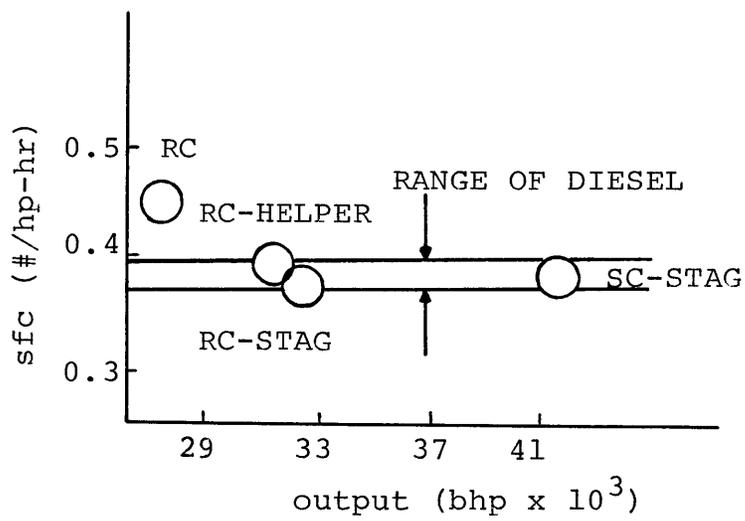
RC : Regenerative cycle

SC : Simple cycle

FIGURE 8: Expected combined cycle performance.

CHAPTER VI

OPERATING HINTS FOR FUEL CONSERVATION

Fuel conservation can be achieved by speed reduction. As it will be shown in subsequent chapters, fuel consumption can be cut by approximately 20% per voyage by reducing average speed from 22 knots to 20 knots. Although slowing down makes it possible to reach destination on less fuel, it takes longer to get there. Unless you can make up the lost time by faster turnarounds, you carry fewer payloads per year. This means that return on investment is reduced which is a very unsatisfactory solution for shipowners and operators.

Fuel can also be saved by studying tide charts and carefully planning most efficient speed. By reducing speed slightly over a long voyage, it may be possible to meet a favorable tide when coming into port, partially making up for lost time. A slight increase in speed may sometimes be justifiable for the same reason. A little extra fuel consumed to meet a favorable tide may result in a net fuel savings.

Keeping the hull clean is another way to save fuel. By reducing drag, the ship's average speed can be increased perhaps by one knot without an increase in hourly fuel consumption. On the other hand, you can maintain the same speed on less fuel.

Even when steaming at reduced speed in a ship with

a clean hull one may be wasting fuel consumed due to in-
sufficient combustion, poor maintenance and poor operating
practices. If the combustion is fuel rich, carbon monoxide
and hydrogen are blown up the stack and fuel is being wasted.
On the other hand, if it is fuel lean, an excessive amount
of hot flue gas goes up the stack and again fuel is being
wasted. Many a times improvements in combustion efficiency
can be made by analyzing the emissions from the stack.
Heavy black smoke signifies that combustion is incomplete so
that air supply should be increased. On the other hand,
white smoke indicates that excess air should be cut down.
The trouble with this sytem of combustion control is that the
clear stack range is from 15% to 300% excess air. While at
the low end, combustion efficiency is satisfactory, at the
high end fuel wastage may be 20% or even higher. These
problems can be eliminated by installing in the stack an
instrument known as oxygen analyzer. This instrument samples
flue gases continuously and measures their net oxygen
concentration so that air adjustments in combustion can be
made. Of course this presupposes that all equipment is in
good operating condition. Burners should be clean and
correctly adjusted. Fuel should be at the temperature
where it has the right viscocity to be atomized correctly
and boiler tubes should be free of soot.

Even when burning fuel efficiently, one may still be wasting a lot of fuel running electric and steam powered auxilliaries inefficiently. Boiler feedwater pumps and oil pumps should not be set to operate at higher pressures than necessary. Worn valves may also waste fuel, by reducing control over pressures and flow rates.

If all of the above have been taken care of, it is time to check for minor steam leaks. Even small leaks can waste major amounts of fuel. Fuel is burned to make steam that is wasted and then more fuel is consumed to make up the water losses with evaporators requiring even more steam.

In the search for further ways to conserve fuel, the turbine heat cycle should be studied. Plotting fuel consumption against load, one finds that there is a point at which the total turbine cycle is most efficient. This may be well below the peak load for which the plant was designed. On a practical basis, what is significant about an overall turbine load curve is that it is scalloped, rather than smooth. Each scallop represents the efficiency of a specific valve in positions from fully closed to fully open. As it is understood, the restricted orifice of a partially opened valve results in an energy loss. What this suggests is that one should learn which points on the load curve give peak efficiency (valves fully open) and try to always operate at those points on the curve.

Ultimately, one might install a new generation of control systems designed to optimize marine power plants performance automatically. These integrated control systems designed for maximum fuel economy will coordinate boiler and turbine much better than is possible with manual control. As fuel shortages continue and fuel prices escalate, an investment in more sophisticated instrumentation and controls should pay increasing dividends.

CHAPTER VII

DETERMINATION OF OPTIMUM OPERATING SPEEDS

In this part of the thesis an attempt to determine optimum speeds or ranges of optimum speeds for various prices of fuel will be made. The particular class of ships to be examined will be containerships having the North Atlantic as their trade route. Optimum speeds will be determined from the point of view of the charterer or owner based on two different criteria, namely:

      a. Maximizing cost per unit output
      b. Maximizing profit

In a real world situation, it is rather difficult for a single shipping company to operate at speeds different than those at which the conference operates. It will be assumed, however, that if it is economical for one operator to run at a reduced speed, it is also economical for the rest and hence the resulting policy will push the real life situation to have ships operate at that "more economical" steaming speed. The only reason the containership trade was chosen for study was because the data available for the analysis were most appropriate.

At the same time an attempt will be made to analyze specific strategies to be adopted by the individual shipping companies so as to reduce fuel consumption and hence cut operating costs. These strategies will include:

      a. Reduced sailings per year

b.   Increased number of ships
c.   Reduced number of ports served directly

A. Optimal Speed for Minimizing Fuel Consumption

It is well known that the rate at which a ship consumes fuel is nonlinearly related to its speed. Thus, the ship needs a disproportionately larger amount of fuel at high speeds than at low speeds. In addition, the amount of fuel in the ship's tanks, due to its weight, also has an effect on the fuel rate required to maintain a given velocity pattern. Thus, when the tanks are nearly full, the ship consumes a larger amount of fuel than if the tanks were nearly empty because of increased drag forces. To counterbalance this, one can argue that when the tanks are nearly empty, the ship consumes a larger amount of fuel due to decreased propeller efficiency. Hence, one expects the fuel consumption to be optimal when the tanks are more or less half full.

In Appendix A it has been shown that in order to minimize fuel consumption on a fixed time trip, it is best for the ship to travel at a constant velocity. This conclusion holds true independent of the initial fuel level, a fact which might come as a surprise. It is thus shown that savings of between 10.3% and 22.7% can be realized depending on the speed profile of the ship.

B. Fuel Consumption, Steaming Speed, Turn Around Time and
    Output for a Ship Operating on a Steady Route

Based on the conclusion that the optimal speed for a ship travelling on a fixed time trip is best for the ship travelling at constant velocity, a calculation is made of some interrelations between fuel consumption, steaming speed, turn around time and output.

For a ship operating on a steady route the steaming time, t_s, in days per round trip is:

$$t_s(V) = \frac{L}{V}$$

where   L = round trip distance, in miles
            V = steaming speed in miles per day

The turn around time, T, in days is then computed as:

$$T(V) = t_p + t_s(V) = t_p + \frac{L}{V}$$

where t_p = port time, in days per round trip.

Now let N be the number of containers that the ship carries per round trip. In this part it will be assume that N equals the containership capacity. A correction to account for the actual output of the ship will be discussed later. The average daily output of the ship, n, in number of containers carried daily is then found to be:

$$n(V) = \frac{Q}{T(V)} = \frac{Q}{t_p + \frac{L}{V}}$$

Empirical data supported by theory, show that there exists a relationship between the propulsion fuel consumption, C, and steaming speed, V, of the form:

$$C(V) = C_o \left(\frac{V}{V_o} \right)^K$$

where C_o = propulsion fuel consumption per steaming day at service speed, in bbls per day

V_o = service speed, in miles per day.

The value of the constant K may vary from one vessel to another and will also depend on weather conditions. In the absence of data, K = 3 may serve as a good approximation for steaming speeds between 10 and 25 knots.

Based on the above relationships the average fuel consumption, F, in bbls per day is:

$$F(V) = \frac{C \, t_s}{T} = C \left(\frac{V}{V_o} \right) \frac{\frac{L}{V}}{t_p + \frac{L}{V}}$$

Finally, the average output of the ship, P, in number of containers carried per bbl of propulsion fuel consumed is found to be of the form:

$$P(V) = \frac{n}{F} = \frac{N \, V_o^K}{C_o L V^{K-1}}$$

C. Minimizing Cost Per Output Unit

As it will become clear in a later chapter, the optimum steaming speed depends upon the criteria adopted by the charterer in operating his vessels for transporting commodities from one place to another. Thus the optimum steaming speed obtained when trying to minimize cost per output unit is usually different from that obtained when trying to maximize profits. In this section the optimum steaming speed for minimizing cost per container carried will be obtained.

First, the actual number of containers carried by the ship which may be different from the containership capacity N must be accounted for. Let a be the factor by which N has to be multiplied so as to obtain the actual number of containers carried. This is called the utilization fraction of the containership and is equal to the ratio of the actual number of containers carried over the capacity of the containership.

When computing the costs of producing the ships output the following categories have to be considered:

    a. Fixed expenses not dependent upon the duration of the trip, denoted by B, which include loading and unloading costs, canal fees, port dues, pilotage, etc.

    b. Expenses dependent upon the duration of the trip, such as propulsion fuel, A_1 in dollars per bbl, auxilliary fuel, A_2, in dollars per bbl and chartering costs per day, A_3 in dollars.

The total average expenses per day then become:

$$E = \frac{B}{T} = A_1 F + A_2 d + A_3$$

where d = auxilliary fuel consumption, in bbls per day.

The average cost of daily output, R, in dollars per number of containers carried, is then found to be:

$$R = \frac{E}{an} = \frac{B}{aN} + \frac{1}{aN} [A_1 C_o L \frac{V^{K-1}}{V_o^K} + (t_p + \frac{L}{V})(A_2 d + A_3)]$$

By differentiating the above relationship with respect to V and setting the derivative equal to zero, the optimal speed can be found as:

$$V^* = V_o \sqrt[K]{\frac{A_3 + A_2 d}{(K-1)A_1 C_o}}$$

This result is of great interest since it shows that the optimal speed is determined by only a few number of factors, that is daily chartering cost, the daily fuel consumption and the cost of the fuels. Other important parameters such as the ship capacity, round trip distance, time in ports and fixed expenses per trip do not influence the optimal speed. From the equation of the optimal speed a conclusion is made that if the daily chartering and auxilliary fuel consumption costs exceed K-1 times the daily cost of propulsion fuel at service speed, the ship should sail at service speed. Otherwise it should sail at the speed indicated by the formula. The ratio of the optimal to service speed (V^*/V_o) is given in Figure 9 for use when values of K other than K = 3 are desired.
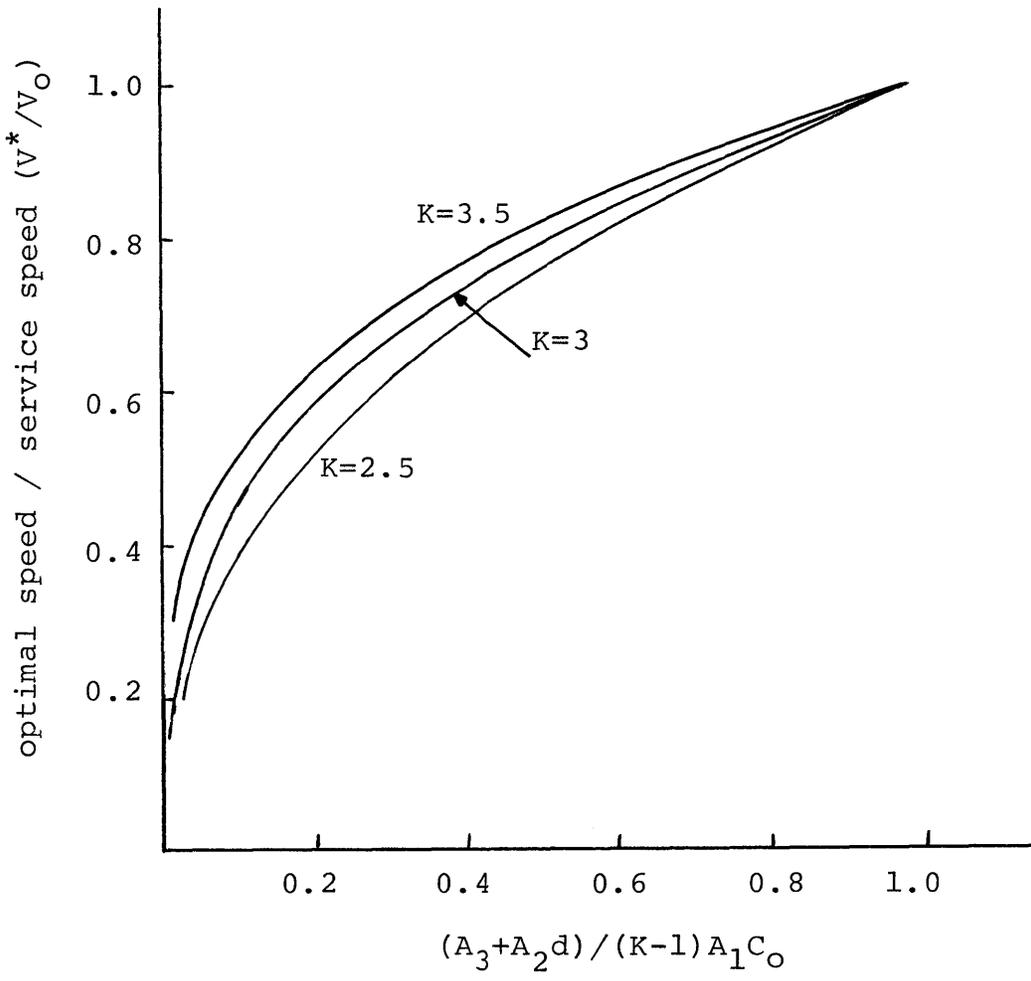
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FIGURE 9: Optimal speed to service speed ratio for
different values of K.

D. Example

In an effort to apply the previously determined optimum speeds and to suggest operating practices pertaining to a real life situation, the case of container ships in the North Atlantic trades was considered as the subject of this example. The characteristics of the hypothetical fleet studied are summarized below:

Number of ports served   8

Ship capacity, 20 ft. equivalent
   containers   2000 per round trip

Designed speed   22 knots

A detailed description of available information about the North Atlantic trade and the various assumptions made are discussed in Appendix B and the following data is provided as an input to the problem examined.

Service speed: V_o = 528 miles per day

Propulsion fuel consumption: C_o = 1088.0 bbls/day
                                    at service speed

Auxilliary fuel consumption: d = 54.4 bbls/day

Number of containers carried: N = 2,000 per round
                                    trip, hence a = 1

Round trip distance: L = 7,647 miles

Port time per round trip: t_p = 6.52 days

Chartering cost: A_3 = 22,241 dollars per day

Fixed expenses per round trip: B = 403,800 dollars

It should be noted that in all subsequent calculations K = 3 and $A_2 = 2A_1$, that is the cost of auxilliary oil is twice that

of fuel oil, was assumed.

Figure 10 shows the optimal speed, V^*, as a function of fuel oil costs for various chartering costs. Whenever the chartering cost increase makes $A_3 + A_2 d > (K-1) A_1 C_0$ the ship will sail at service speed, as is suggested by the nature of the diagram. Figure 11 shows the minimal shipping cost per container again as a function of fuel cost. As expected increasing the fuel cost increases the minimal shipping cost of the charterer, thus making him operate at a lower steaming speed.

Figure 12 shows the shipping cost in dollars per container as a function of steaming speed, for various assumed daily chartering costs. It is important to point out that R is rather flat near V^*, the optimal steaming speed. This is of great interest because it means that deviations from the optimal speed are not costly. In the extreme case where $A_3 = \$10,000$ per day, the optimal speed is 17.19 miles/hour for a fuel cost of $394 per container. If instead the ship steamed at either 22 miles/hour (its service speed) or 12.95 miles/hour, the shipping cost is $405 per container, less than 3% saving. This property is of special interest in the case of fuel shortages.
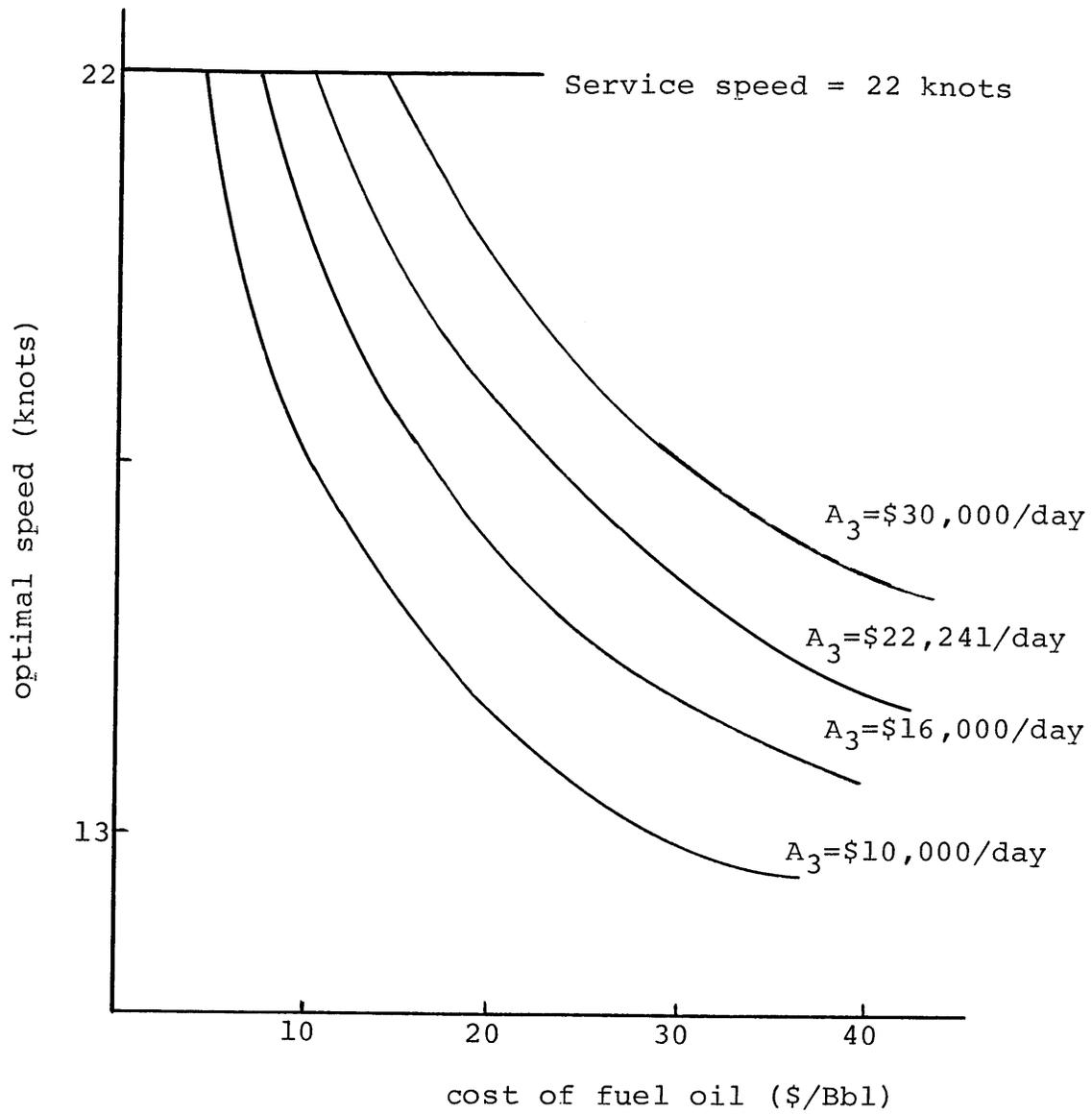
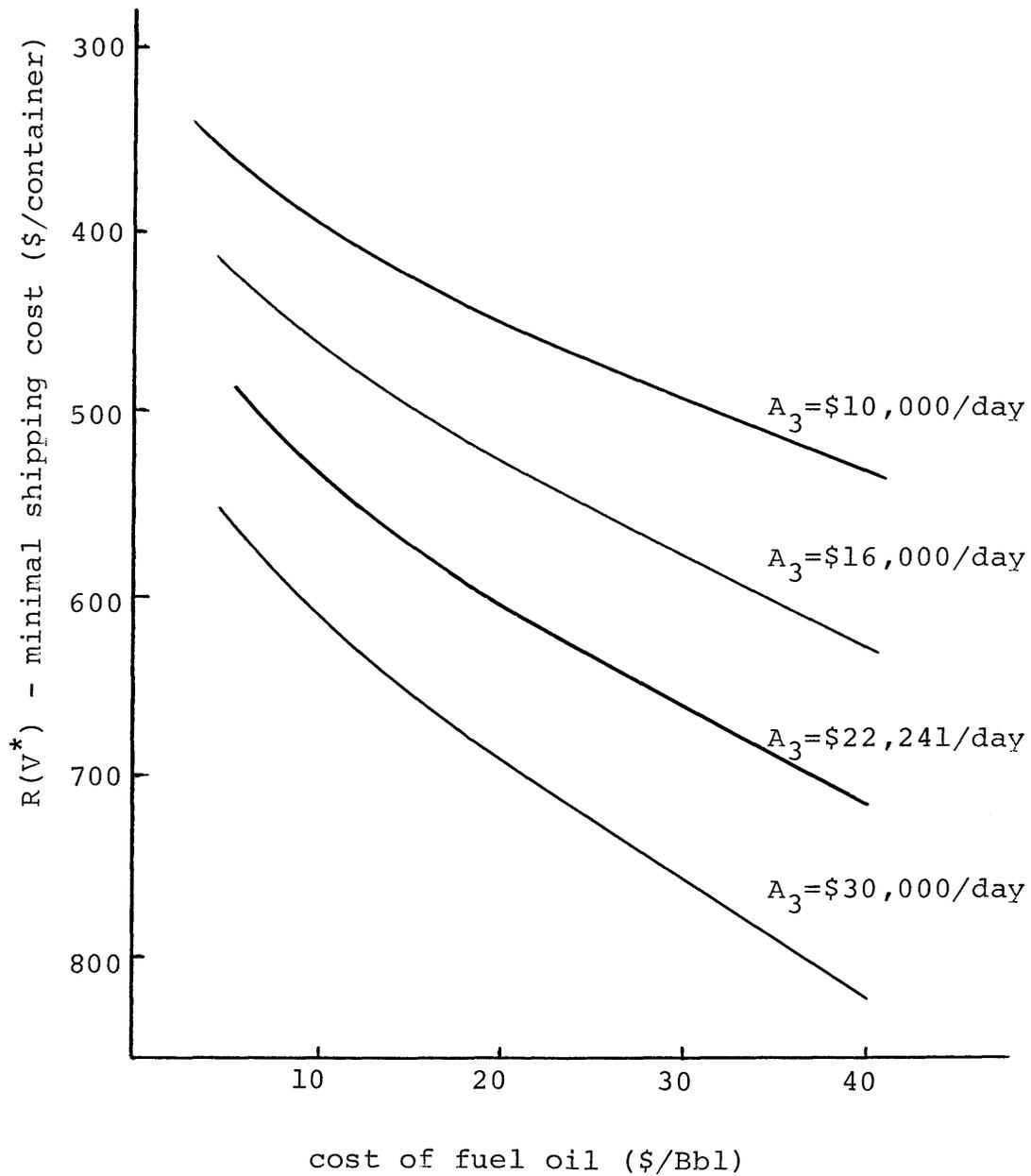FIGURE 10: Optimal speeds as function of fuel oil cost and chartering cost.

FIGURE 11: Minimal shipping cost per container as function
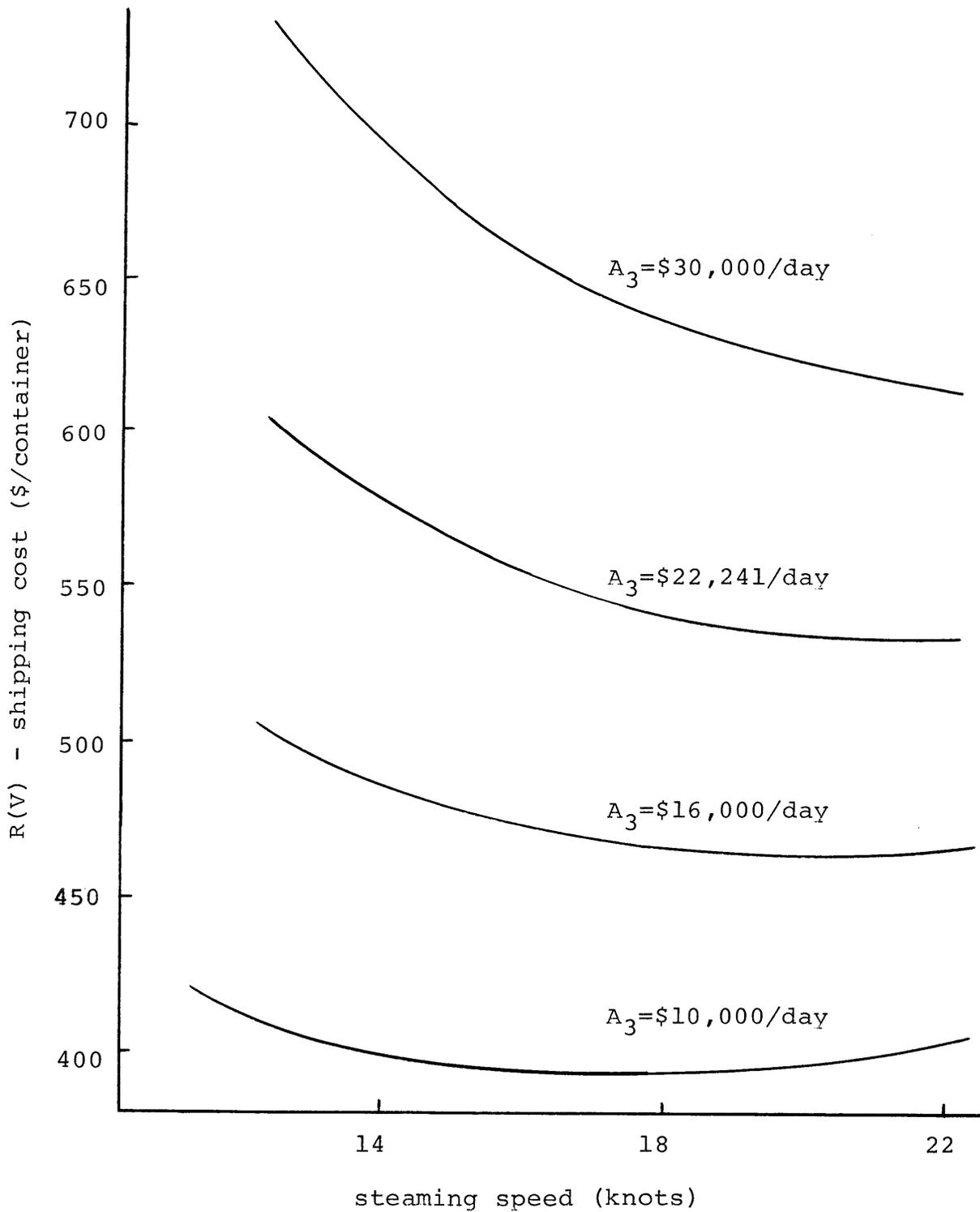of fuel oil cost and chartering cost.

FIGURE 12: Shipping cost per container as a function
of speed and chartering cost.

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E. Maximizing Profit

In the previous section, the main concern was to minimize the costs involved per output unit. In many cases, however, income based on payments per unit of output is the source of revenue for many shipping companies operating their own ships. In such cases the objective of the shipowner or the charterer is not to minimize cost per unit of output, but rather to maximize net income.

The average daily net income, denoted by I, in dollars is given by:

$$I = A_4 an - E = \frac{A_4 aN}{t_p + \frac{L}{V}} - \frac{B}{t_p + \frac{L}{V}} - \frac{A_1 C_o L V^{K-1}}{V_o^K (t_p + \frac{L}{V})} - A_2 d - A_3$$

where A_4 = average payment received per container in dollars. If the ship is owned, then A_3 stands for the average daily operating costs. The fixed cost per round trip, B, includes only costs covered by the shipping company (cargo handling and some other fixed costs may be paid directly by the shippers and receivers in many cases). For deriving the optimal speed the above relationship is differentiated with respect to V and equate the result to zero. For a value of K = 3 the following cubic equation was obtained:

$$V^3 + 1.5 \frac{L}{t_p} V^2 - \frac{A_4 aN - B}{2A_1 C_o t_p} = 0$$

For all practical cases revenue per round trip exceeds fixed cost per round trip, $A_4aN > B$, and there exists only one positive real solution. As was also the case in the previous consideration, whenever the optimal speed exceeds the service speed, the ship will sail at the service speed.

From the above equation, one sees that the determining factors have changed quite radically with the change of objective. While the cost of propulsion fuel is still an important factor, the average daily operating cost (or chartering cost) does not play any role in determining the optimal speed. On the other hand, the port time, the round trip distance and the fixed expenses per round trip, which did not play any role in the previous analysis are among the determining factors here. As was expected the size of the ship and the payment received per unit of actual output have their place among the determining factors.

F. Example

Using again the example of container ships in the North Atlantic trade, one finds from Figure 11 that the minimal shipping cost per container for a chartering cost of $10,000/day and fuel cost of $10.75/bbl is $394 obtained at an optimal speed of 17.19 miles/hour.

Suppose now that the shipowner (charterer), not being the owner of the goods carried, is paid A_4 dollars per delivered container. The optimal speed at which he should operate his vessel is given in Table 7.

Table 7

A_4	394	418	508	533
V^*	17.19	18.14	21.25	22

As is expected, the optimal speed of the vessel is increasing with A_4. If A_4 goes above $533 it is best to steam at service speed.

Having now examined the ranges of optimal speeds for vessels operating on a particular route according to the objective of the shipping company, an attempt will be made to gain an understanding of the savings resulting from fuel conservation.

CHAPTER VII

FUEL CONSUMPTION REDUCTION

The effect of the drastic changes in fuel oil prices since 1971 and 1972, was to increase the percentage of operating costs allocated to fuel. Thus fuel conservation in ship operations remains a basic problem with savings as a tremendous reward. Hence, the goal of the shipping industry will be to find all possible ways of fuel conservation, in an effort to respond to the changing economic conditions, and accept those that are most effective in maintaining acceptable levels of cost and service.

As explained earlier, the case of container ships in the North Atlantic trade was considered as the object of exploring possible fuel savings attainable from the use of different operating practices. The increased proportion of fuel costs to total operating costs (the figures have more than doubled since 1971 and 1972) results in changes in economic tradeoffs, as for example optimum steaming speeds. Ship operators have been noticing that the cost of high speeds exceeds the competitive advantage that they gain and as a result have begun to change their operating practices. In the long run it will be necessary to re-evaluate the tradeoffs between speed and cost, thus establishing new optimal design characteristics, based on less fuel intensive ships.

In addition to possible changes in marine power plants, employing weather routing, reducing port delays and maintaining power plant and hull surface in optimal condition there are certain other measures in reducing fuel consumption. These measures can be applied by individual shipping companies and include the following strategies:

    a. Reduced sailings per year
    b. Jumboizing
    c. Increased number of ships.

A. Reduced Sailings Per Year

Reduced sea speeds will save fuel without reducing the total cargo transported only if ships had not been operating full prior to speed reductions. Since this is hardly ever the case, the object of the shipping company should be in reducing fuel consumption subject to minimizing the inconvenience incurred to the shippers. Table 8 based on the analysis of Appendix B, shows how small speed reductions can produce large fuel savings with a relatively small increase in the average delivery time and a small decrease in the frequency of service.

Generally speaking, Table 8 shows that a slow down of 1 1/2 knots corresponding to a 95% throughput per year with a 15% savings in fuel would not be a bad policy to adopt if the decrease in the frequency of service is not crucial.

A more drastic reduction in speed can be effected without serious disruption of service if the fleet is initially operating at sufficiently low load factor. To illustrate this point, fuel calculations for an initial load factor of 0.75 are shown in Table 9. Potential fuel savings under this approach are greater than before, while only frequency of service and average delivery time are adversely affected, in amounts considerably more than before. However, the throughput per year remains at 100%. One also sees that with ships originally operating with less than full load, both the cost

TABLE 8

Effects of Speed Reduction on Fuel Consumption, Service and Operating Costs

Knots Speed Reduction	BBls Fuel Used	%fuel Saved	Annual Container Capacity	Frequency Of service (days)	Average Days Delivery Time	Annual Fleet Operating Cost ($million)	Average Cost per Container ($)
0	816,000	0	100,000	7.0	10.5	44.6	445.75
1	734,000	10.0	96,700	7.24	10.9	43.0	444.30
1 1/2	692,000	15.2	95,100	7.36	11.0	42.1	443.16
2	662,000	18.9	93,500	7.49	11.2	41.5	443.63
3	593,000	27.4	90,000	7.78	11.7	40.0	444.04

TABLE 9

Effects of Speed Reduction on Fuel Consumption, Service And Operating Costs with 75% Initial Load Factor

Ship Speed Knots	Bbls Fuel Used	%fuel Saved	Annual Container Throughput	Load Factor (%)	Frequency Of service (days)	Average Days Delivery Time	Annual Fleet Operating Cost ($million)	Average Cost per Container ($)
20.2	698,000	0	75,000	75.0	7.0	11.1	37.1	495
17.7	509,000	27.1	75,000	85.6	8.0	12.6	35.0	466
15.7	412,000	41.0	75,000	96.4	9.0	14.1	33.8	451

per container as well as the total fleet cost can be reduced,
while saving substantial amounts of fuel, in contrast to
Table 8 where the average cost per container started in-
creasing again after a reduction in ship speed of more than
1 1/2 knots. This can be explained because the additional
time between sailings allows the ship to pick up more cargo
and hence operate more efficiently.

It may be noticed that reducing speed by 3 knots (last
line of Table 8) results in almost the same frequency of
service as reducing it by 2.5 knots (second line of Table 9).
However, in the first case 90,000 containers are transported
each year and in the second only 75,000. Even though the
larger throughput of the former requires higher sea speeds,
more fuel consumption and higher fleet costs, on a per
container basis fuel consumption and costs are lower than the
8 day departure service carrying fewer containers per year.

B. Reduced Sailings with Jumboizing

Even though the reduced speed strategy may be adopted as an immediate fuel saving measure during a temporary shortage, solutions for more extended shortages or even greater price hikes should be found. These include the jumboizing and increased number of ships strategies to be described in the sequel.

The jumboizing approach provides a means of reducing speed and hence sailings per year, while at the same time reducing fuel consumption and without leaving any cargo behind. An attractive feature of adopting this approach is that parallel middle body sections can be constructed in advance, so that time out of service for jumboizing can be quite short.

As can be seen from Table 10 two lengthened variants of the basic ship were considered. Fuel consumption, frequency of service and operating costs are also shown for both the original and jumboized ships in Table 11. As it can be seen substantial fuel savings can be realized without reduction in total cargo throughput, while at the same time operating costs and hence average cost per container decline.

TABLE 10

Characteristics of Jumboized Ships

	Original	Jumbo No. 1	Jumbo No. 2
Length of added section, ft.	0	96	144
Container capacity, 20 ft. equivalent containers	1000	1312	1468
Sea speed required, knots	22.0	15.9	14.0
Cost of jumboization, $million	0	8.5	12.4

TABLE 11

Effects of Jumboization on Fuel Consumption, Service
And Operating Costs for an Annual
Throughput of 100,000 Containers

Ship	Ship Capacity, Containers	Bbls Fuel Used	%fuel Saved	Frequency Of service (days)	Average Days Delivery Time	Annual Fleet Operating Cost ($million)	Average Cost per Container ($)
Original	1,000	816,000	0	7.0	10.5	44.6	446
1	1,312	449,000	45.0	9.2	13.7	43.0	430
2	1,468	342,000	58.1	10.3	15.2	43.0	430

C. Increased Number of Ships

A better approach to long range fuel savings is to increase the size of the fleet by one or more ships and run all ships at the reduced speed necessary to deliver the same amount of cargo per year as previously. An alternative solution is to add a ship of the same capacity as the existing ones, but with a lower design speed. This approach would probably be the first step of a shipping company leading to a larger fleet consisting of slower vessels. Table 12 summarizes the effects on the original fleet of adding one ship of a. the same type; and b. designed for lower speed.

Adoption of this strategy may result in fuel savings of up to 40%. However, the operating cost increase because of introducing the extra ship adds the costs of crew, maintenance, amortization, insurance, etc. for the additional ship. For the four ship fleet consisting of the same type ships, this cost is nearly offset by the cost of fuel saved. On the other hand for the fleet consisting of the old ships and the new ship of lower design speed, the savings in fuel more than offset the increase in operating costs, thus resulting in smaller overall average cost per container. This occurs because the fuel rate for the new ship is more favorable, since the power plant would be operating closer to its designed condition. The construction cost for this slower ship would also be less than for the original ships of the fleet.

TABLE 12

Effects of Adding a Ship on Fuel Consumption, Service and Operating Costs for an Annual Throughput of 100,000 Containers

Number Of ships	Ship Speed Knots	Bbls Fuel Used	%fuel Saved	Average Days Delivery Time	Annual Fleet Operating Cost ($million)	Average Cost per Container ($)
a. Ships of the same design speed						
3	22.0	816,000	0	10.9	44.6	446
4	15.2	495,000	39.3	12.0	44.9	449
b. New ship of lower design speed						
4	15.2	466,000	42.9	13.2	43.6	436

It should also be remembered that cargo volume and
frequency of service have been maintained with this solution
and it is for this reason that this strategy appears to be
the most attractive of all the previous strategies examined.

CHAPTER IX

IMPLICATIONS OF POSSIBLE BUNKER SHORTAGES

The effects of possible bunker price hikes on the operating policies to be adopted by the firms was examined earlier. However, the oil crisis may take a different dimension, namely, bunker shortages, which in turn may result in real crisis or even disasters. In the sequel, an attempt will be made to estimate the effects of possible shortages on ocean transportation and examine what policies the shipping companies would have to adopt.

In continuation of what was said earlier about the interrelations between fuel consumption, steaming speed, turn around time and output for a ship operating on a steady route, $n(V,V_o)$ will denote the ratio between the daily outputs of a ship steaming at speed V and at speed V_o, that is:

$$n(V,V_o) = \frac{n(V)}{n(V_o)} = \frac{t_p + \frac{L}{V_o}}{t_p + \frac{L}{V}}$$

By substituting this relationship into a similar ratio for the propulsion fuel consumption denoted by $F(V,V_o)$ one gets:

$$F(V,V_o) = (\frac{V}{V_o})^{K-1} n(V,V_o)$$

This relation shows that when slowing the ship's speed, the propulsion fuel consumption is reduced by far more than the output of the ship.

Suppose now that for one shipping company, the fraction of part time to turn around time is ρ, that is:

$$\rho = \frac{t_p}{t_p + t_s(V_o)} \qquad \text{or} \qquad t_p = \frac{\rho}{1-\rho} t_s(V_o)$$

It should be clear that if the company consists of ships involved in different trades, i.e. tankers, general cargo, specialized trades, different values of ρ will exist for each trade. The turn around time can then be calculated from:

$$T(V_o) = \frac{t_s(V_o)}{1-\rho}$$

The turn around time at speed V can now be expressed as:

$$T(V) = t_p + t_s(V) = \frac{\rho}{1-\rho} t_s(V_o) + t_s(V)$$

By making the necessary substitutions one finally finds the relationships of the daily output, $n(V,V_o)$, and propulsion fuel consumption, $F(V,V_o)$, from:

$$n(V,V_o) = \frac{1}{\rho + (1-\rho)\dfrac{V_o}{V}}$$

$$F(V,V_o) = \frac{\left(\dfrac{V}{V_o}\right)^{K-1}}{\rho + (1-\rho)\dfrac{V_o}{V}}$$

By using these relationships estimates of the decline in total output as a function of fuel cuts can be calculated. Figure 13 shows the reduction in speed of the fleet as a function of fuel cuts, while Figure 14 shows the reduction in output again as a function of fuel cuts. The value of ρ depends on the particular trade and may be as high as 0.7 for general cargo liners, while approaching 0 for tankers involved in long trips.

For the containerships examined earlier, the value of ρ is approximately 0.3. This signifies that a 50% cut in supply of propulsion fuel would result in ships sailing at 77% their prior steaming speeds and a decline of output to 83%. More generally, for any trade, a 50% cut in supply of propulsion fuel would mean steaming speeds reduced to 74-79% of the present service speeds and a total yearly output decline to 79-91% of the present output. Even though shortages of up to 50% will not have a disastrous effect on ocean transportation, there arises the need of optimal allocation of the available fuel to specific routes and vessels.

Next an examination will be made of how possible fuel shortages would be faced by a shipping company involved in the North Atlantic containership trade, resulting in a strategy of reducing the number of ports served directly to be adopted in an effort not to leave any cargo behind.
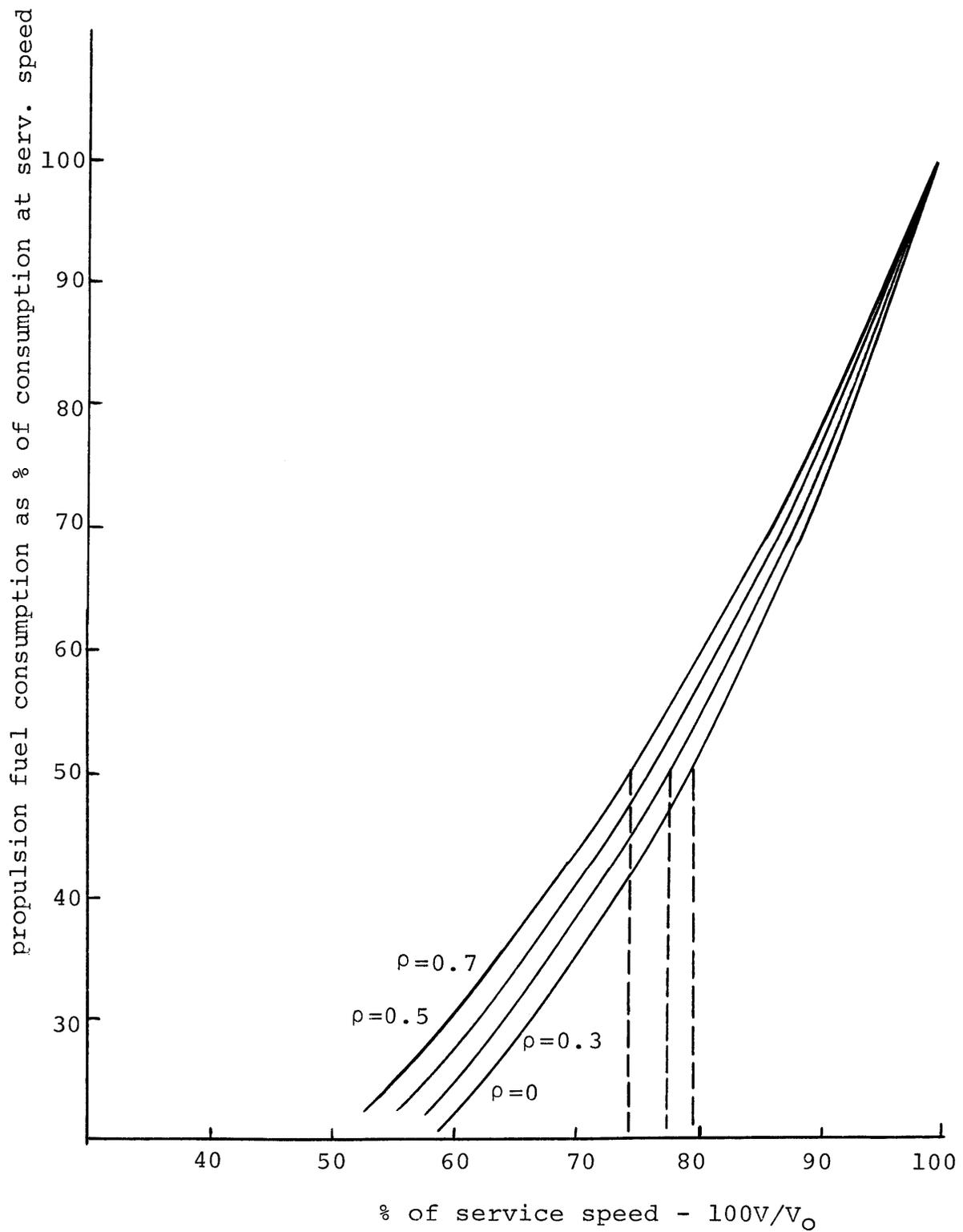
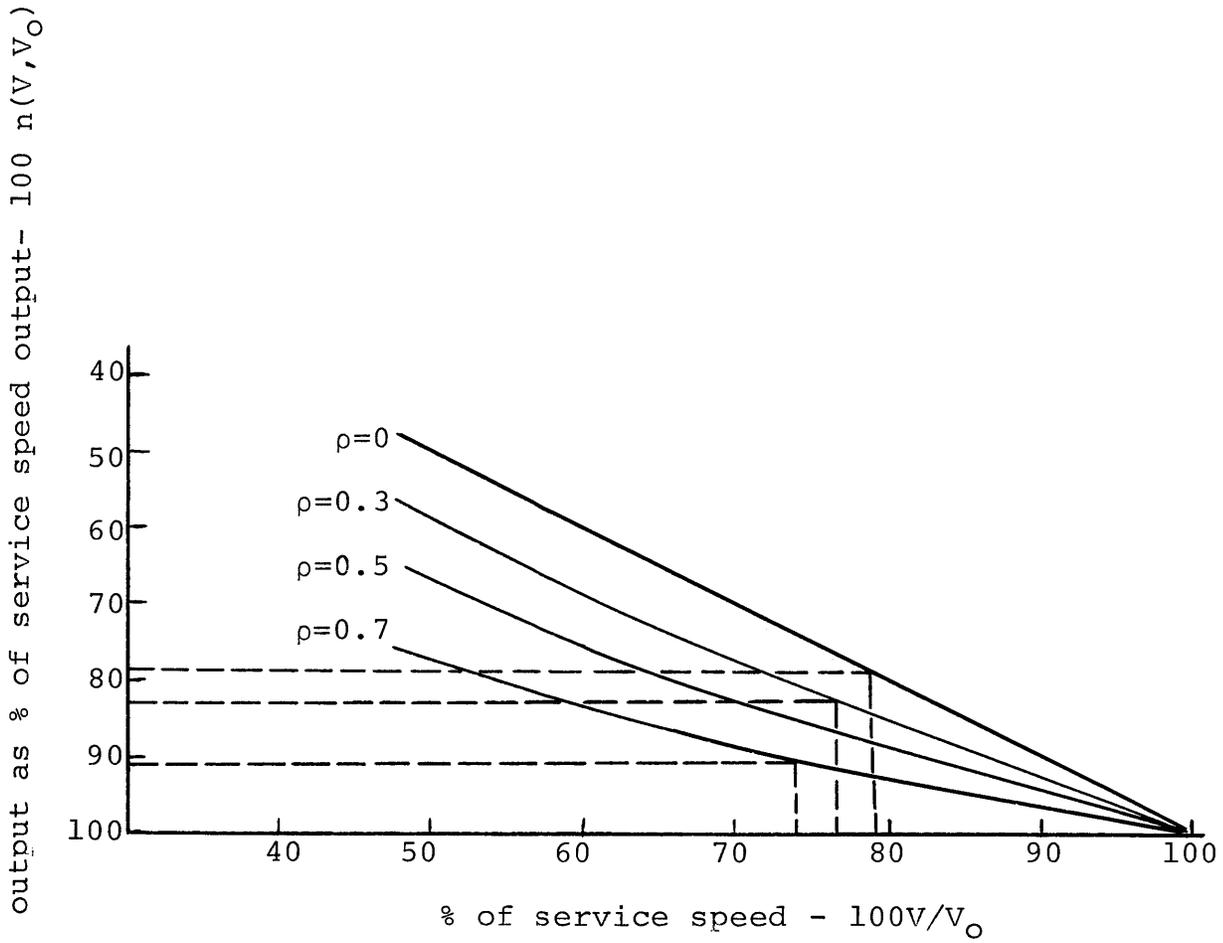FIGURE 13: Reduction in speed as a function of fuel cut.

FIGURE 14: Reduction in output as a function of
fuel cut.

A. <u>Reduced Number of Ports Served Directly</u>

The approach described here would be only adopted
in a fuel rationing situation, provided that there is no
time to adopt either the jumboizing or the increased number
of ships strategy. Another assumption about this approach
is that it is not desirable to upset the regularity or
frequency of ship departures.

One possible approach is available to those services
calling at many ports. When the cargo handled in one or more
ports is small, especially if the ports involve substantially
extra distance travelled, the elimination of those ports is
rational. The cargo can be transhipped by land or sea, with
little or no effect on service to shippers. Thus with
turn around time kept constant, the sea time is increased
and hence ships can run at lower speeds.

Whenever significant amounts of cargo are involved,
service to the ports dropped must be provided by feeder
service. Table 13 shows fuel consumption and operating
costs for servicing 8,6,4, and finally 2 ports via the
line-haul ships, the rest being served via the feeder.

The figures in Table 13 show savings in fuel up to
30%, while at the same time no cargo is left behind. This,
however, is not done at no extra cost, the result being that
average operating costs increase by 12.5%. In the above
calculations the costs and fuel consumption of the feeder

TABLE 13

Effects of Reduced Direct Port Calls on Fuel Consumption, Service and Operating Costs for an Annual Throughput of 100,000 Containers and Weekly Frequency of Service

Ports served Direct/Feeder	Ship speed (knots)	Bbls Fuel Used	%fuel Saved	% cargo Transshiped	Average Days Delivery Time	Annual Fleet Operating Cost ($million)	Average Cost per Container ($)
8/0	22	816,000	0	0	10.9	44.6	446
6/2	20.3	724,000	11.2	7.5	10.4	45.1	451
4/4	17.4	610,100	25.2	25	10.9	46.3	463
2/6	14.8	580,000	28.9	45	11.4	50.2	502

service are included in calculating the fuel consumption and operating costs for the fleet. On the other hand, average delivery times do not increase very much; in fact, they decrease when serving two ports by feeder ship.

CHAPTER X

CONCLUSIONS

In the previous chapters, a survey and evaluation of feasible responses of the maritime industry to increases in fuel cost was undertaken. Both short and long-term approaches to technological improvements and more effective operating practices were considered.

Alternate means of generating power using a high conversion efficiency and minimizing fuel consumption were described. Thus, a combined diesel and organic Rankine bottoming cycle with an overall conversion efficiency of 48.3% was examined. This combined cycle was especially attractive because the high conversion efficiency was retained at both full load and part load operations. On the other hand, because of the simple turbine design and the non-corrosive nature of organic working fluids which are compatible with carbon steel, the capital costs of the combined plant were low, resulting in a more advantageous power plant.

Energy conservation can also be achieved through use of the COGAS plant. Because of its basic economy, the COGAS plant has already found significant applications. There is a multiplicy of combination arrangements possible and most of them offer some thermodynamic advantage in performance over either type of prime mover operating independently. However, the criteria for an optimum technical

solution are not the same as those for an optimum economic solution. Thus, maximum fuel utilization is achieved with the gas turbine alone operating at low process steam pressure and maximum degree of supplementary firing. On the other hand, this cycle arrangement only gives the shortest payout time.

The current applications and experience gained from them have verified the advantage to shipbuilders and owners of using the gas turbine as a prime mover for marine propulsion systems. Ship designers have made use of the low volume, low weight, and flexibility of gas turbine systems in machinery arrangements that preclude the use of more conventional prime movers. The simplicity and ease of automation of the gas turbine are now being translated into reductions in manning requirements reducing not only the operating costs, but also the acquisition costs associated with crew accommodations. Since 1970, there has been a dramatic increase in the number of marine applications for heavy duty industrial gas turbines, ranging from roll-on/roll-off ships to LNG carriers and bulk carriers, utilizing both electric drive as well as controllable-pitch propellers.

From what has been discussed in the previous chapters it has become clear that there is no painless method of saving fuel. The price that must be paid for fuel conservation may include cargoes left behind, shippers receiving

less frequent and slower service and higher operating costs.

Optimum steaming speeds can be determined in accordance with the objective of either minimizing cost per output unit or maximizing profit. This will in turn depend on whether the company involved in a particular trade acts like a time charterer employing chartered vesses, in which case the first objective will be adopted, or like an owner, in which case the objective of maximizing profits will be adopted.

The reduced speed strategy is the one most likely to be used as an immediate (short-run) fuel saving measure during a temporary shortage and also the most likely to be discontinued in case of increased availability of fuel. This strategy will result in significant fuel savings and will reduce operating costs per container carried, increasingly so as previously unused capacity becomes bigger.

As longer range solutions to even greater price hikes jumboizing of existing ships is an economical way to reduce speed and fuel consumption without reducing cargo transported per year. An even more attractive approach is adding a new slower ship to the existing fleet resulting in substantial fuel savings, while at the same time keeping same frequency of service as before.

Finally, in cases of extended shortages the number of ports served directly can be reduced, resulting however in

increasing the operating costs, especially so when the feeder service is of sizeable amount. On the other hand, if cargo throughput per year is not a limiting factor, it has been shown that extended shortages will result in slower steaming speeds and cargo left behind.

APPENDIX A

Assuming that the fuel rate varies parabolically with velocity at fixed fuel levels and parabolically with fuel level at fixed velocities, the following relationship is established:

$$\frac{df}{dt} = -[\, c_1 + c_2(f-f_1)^2\,]v^2$$

where
     f = amount contained in ships tank
     f_1 = point of optimal fuel consumption
    df/dt = fuel rate
    c_1, c_2 = constants depending on particular ship.

The problem then is to determine the ship velocity as a function of time to transport goods from point x_1 to x_2, that is over a distance d, within a given time T. Expressed mathematically this becomes:

$$\min J = \int_0^T -\frac{df}{dt}dt = f(0) - f(T)$$

subject to
        $x(0) = x_1$

        $x(T) = x_2$

By using calculus of variations the result $v(t)$ = constant is obtained. This then requires that $v = d/T$. This result is true regardless of the values of c_1, c_2 and f_1, x_1, x_2 and the initial fuel level $f(0)$.

In order to realize the fuel savings involved consider the following example, where d = 1,000 miles and

T = 50 hrs.. First consider a class of linearly varying velocity functions such that:

$$v(t) = v_o + (0.8 - 0.04v_o)t$$

where $10 < v_o < 30$.

Each of these functions takes the ship from x_1 to x_2 in T = 50hrs.. These functions are portrayed in Figure 15. For each of these functions it is possible to calculate J, the amount of fuel consumed. The results for 3 such cases are shown in Figure 16. As expected, the minimum fuel consumption occurs in each case at $v_o = d/T = 20$ miles/hr.. The largest savings of fuel occurs when the initial fuel level is $f(0) = 0.6\ f_{max} = 120,000$ gallons and they amount to approximately 10,700 gallons or 10.3%.

As a second example, consider the class of parabolically varying functions such that:

$$v(t) = 30 - \frac{v_o}{2} + \frac{3}{625}\left(\frac{v_o}{2} - 10\right)(t-25)^2$$

where $0 < v_o < 30$.

These velocity functions are portrayed in Figure 17 and the amount of fuel consumed for three different cases is also shown in Figure 18. Again the minimum fuel consumption occurs at the theoretically optimal value of $v_o = 20$ miles/hr. The largest fuel savings in this case is approximately 27,300 gallons or 22.7% when $v_o = 20$ as compared with $v_o = 0$,
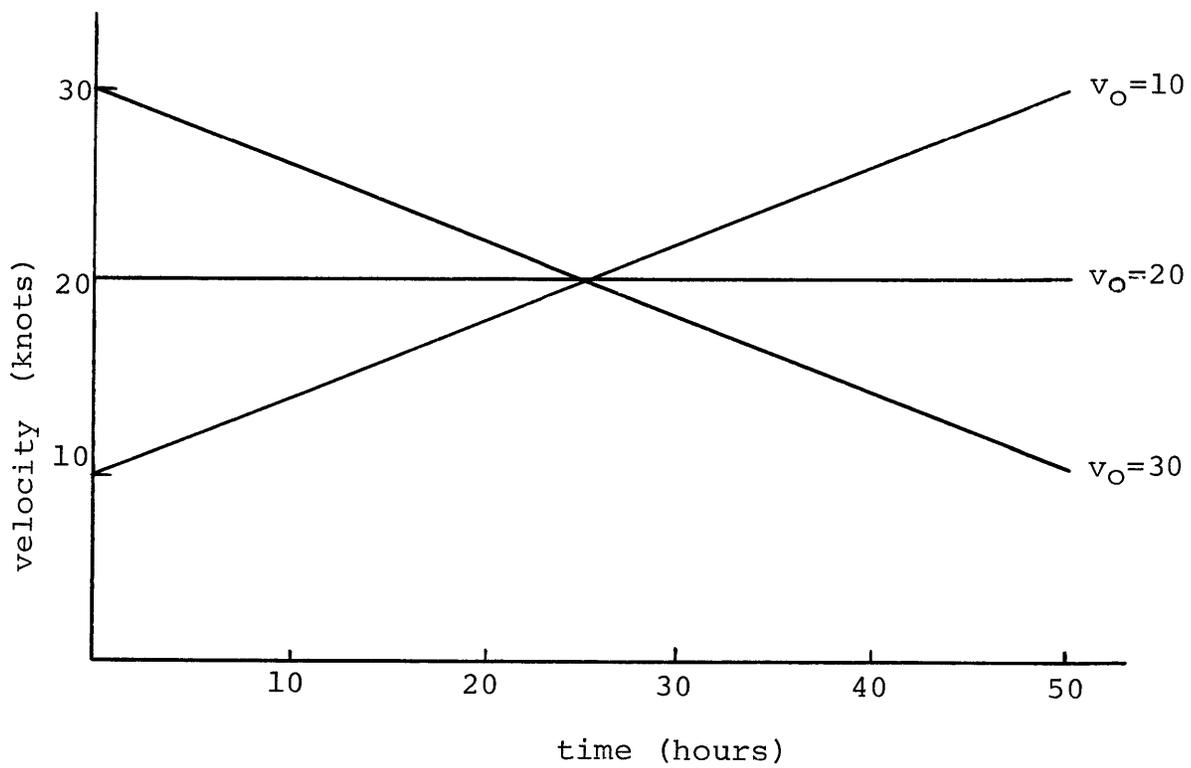
FIGURE 15: Linearly varying velocity functions for various values of v_o.
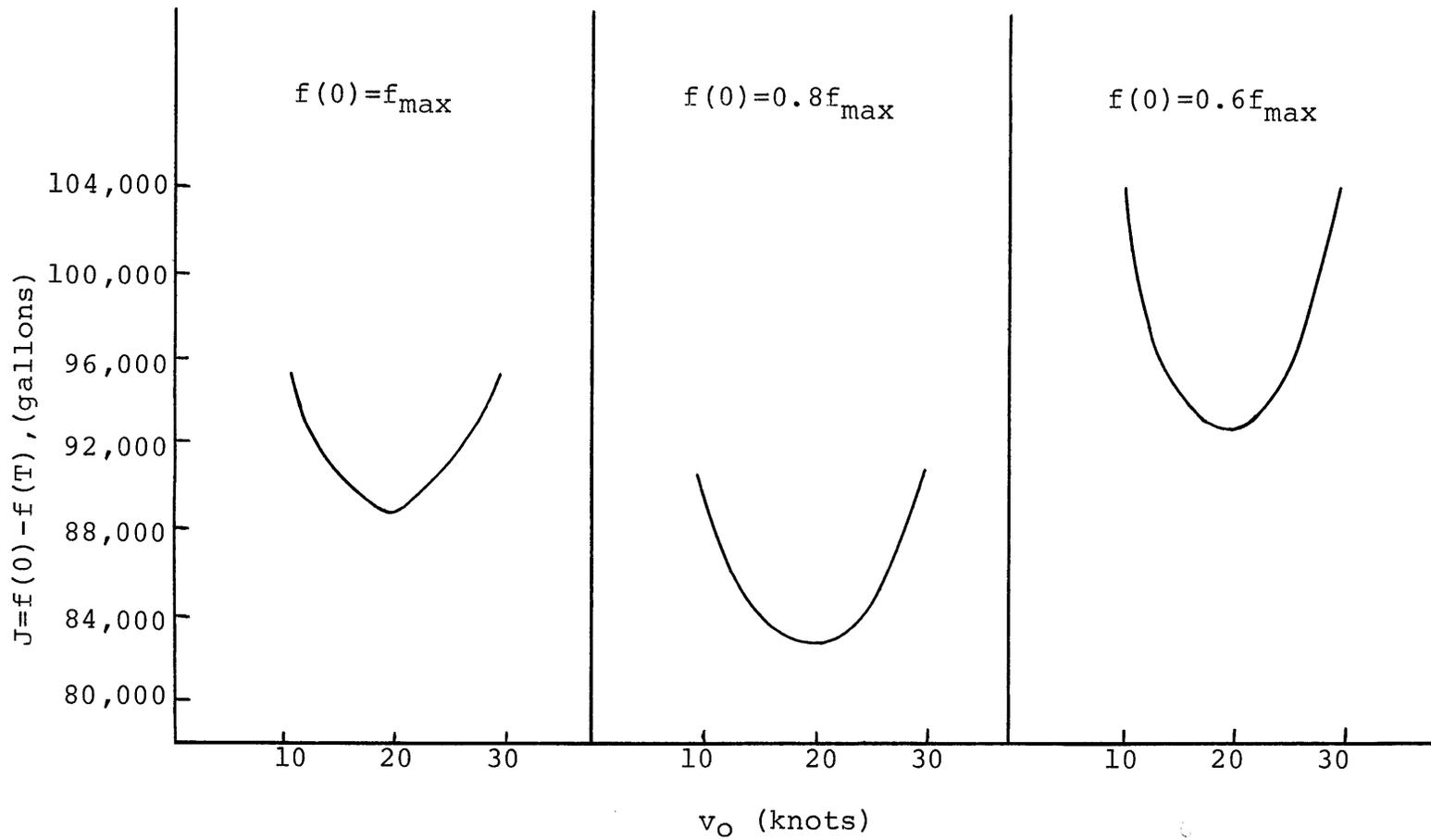
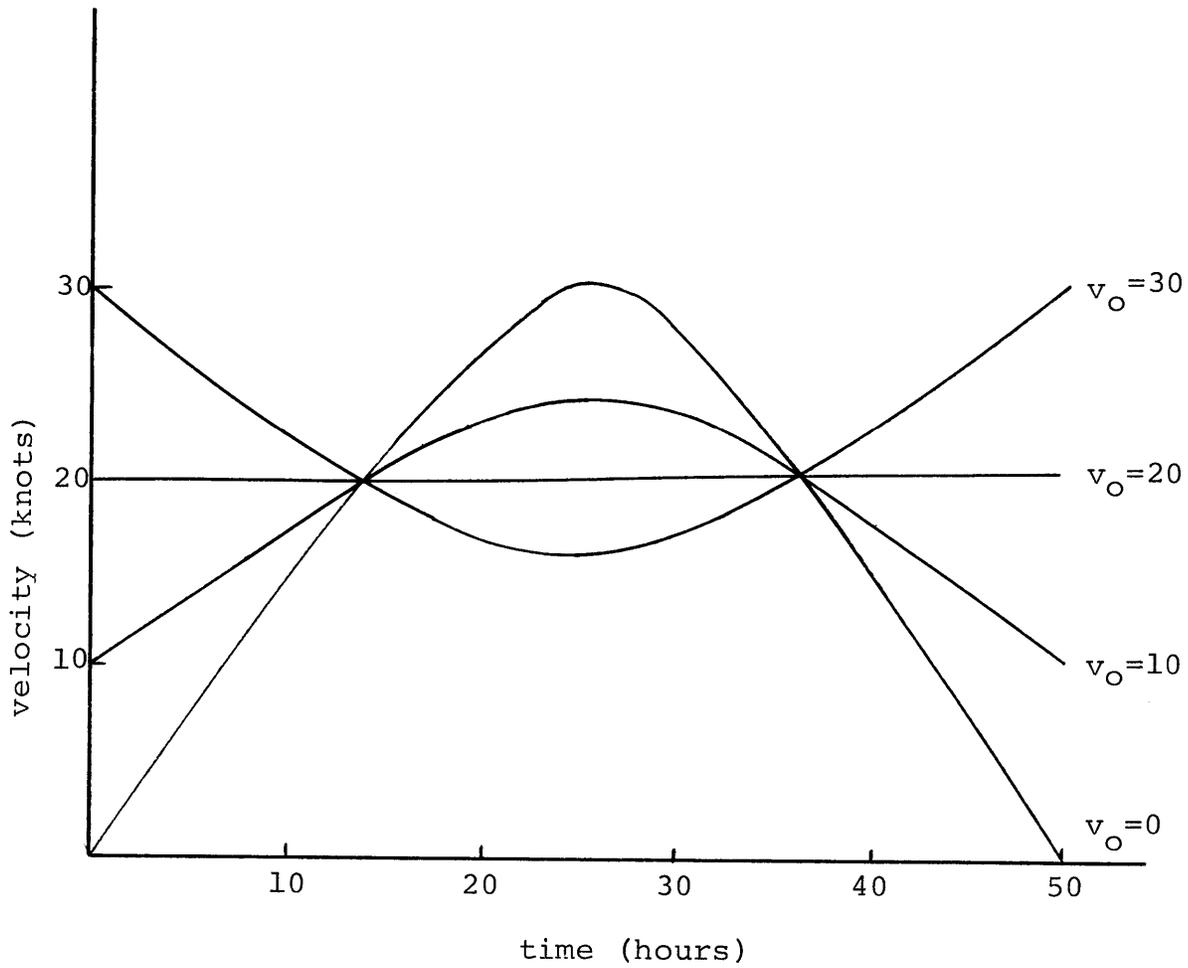FIGURE 16: Fuel consumption for various values of f(0) and v_o.

FIGURE 17: Parabolically varying velocity functions
for various values of v_0.
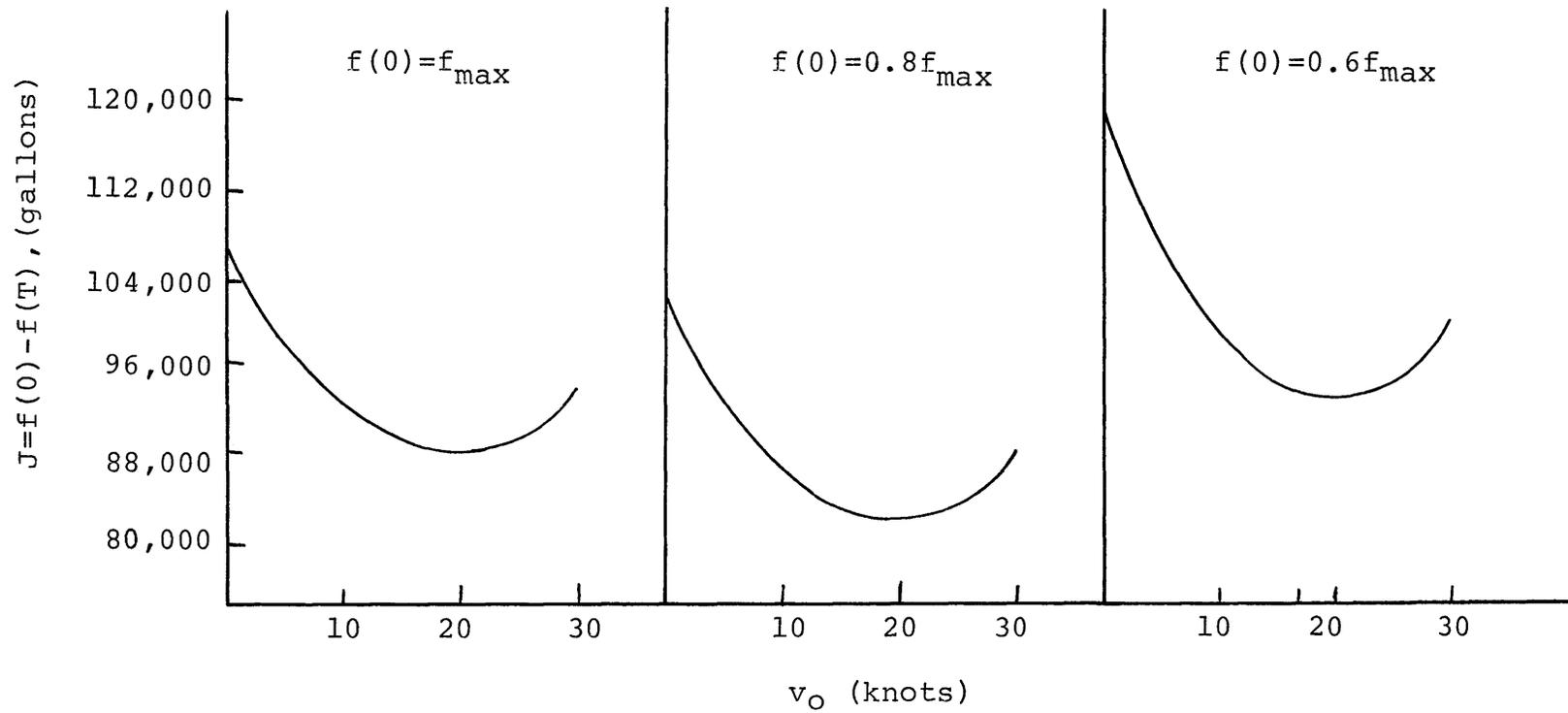
FIGURE 18: Fuel consumption for various values of f(0) and v_o.

and occurs when the initial fuel level is $f(0) = 0.6 \, f_{max} = $ 120,000 gallons.

It has thus been shown that the minimum fuel consumption on a fixed time trip is when the ship travels at a constant velocity. This conclusion also holds true when the final time and the initial fuel level are also considered as variables.

In arriving at this result, the following assumptions were made:

    a. The sea environment remains constant throughout the trip

    b. The time needed to accelerate and deccelerate from full velocity, $v_o = d/T$, is a small portion of the total trip and can be neglected.

APPENDIX B

Total Time Per Round Trip

Listed below are a few tables which will later help in the calculation of port and sea time per voyage and hence total time spent per round trip. Based on the North Atlantic trade, with the number of ports visited varying from 2 to 8, Table 14 gives the distribution of ports and respective sea miles:

TABLE 14

Number of Ports	U.S. Ports	European Ports	Round trip distance
2	New York	Rotterdam	5909
4	New York Norfolk	Rotterdam Southampton	6257
6	New York Norfolk Baltimore	Rotterdam Southampton Bremen	6778
8	New York Norfolk Baltimore	Rotterdam Southampton Bremen Goteborg Le Havre	7647

The assumed cargo distribution among the ports is shown in Table 15.

TABLE 15

No. of Ports	U.S. Ports			European Ports				
	N.Y.	Norf.	Balt.	Rott.	South.	Brem.	LeHavre	Got.
2	100	–	–	100	–	–	–	–
4	75	25	–	75	25	–	–	–
6	70	15	15	45	30	25	–	–
8	70	15	15	40	25	20	5	10

The number of cranes capable of working simultaneously is a function of ship size and consequently depends on the percent of total container capacity to be handled. Table 16 shows the number of cranes working the ship simultaneously as a function of both the ship capacity and percentage of cargo handled at each port.

TABLE 16

Ship Size 20 ft. equiv. cont.	Percentage Handled at a Given Port				
	(80-100)	(60-79)	(40-59)	(20-39)	(1-19)
500	2	2	1	1	1
1000	3	2	2	1	1
1500	4	3	2	1	1
2000	5	4	3	2	1

The frequency of service will be determined on the assumption that there are 350 voyage days per year (the equivalent of 50 weeks). The total voyage time per round trip is composed of sea time and port time.

A. Sea Time Per Voyage

The number of sea days per voyage is found by dividing the total round trip distance by the actual miles travelled per day at service speed.

B. Port Time Per Voyage

Time spent in port consists of loading/unloading time and port delays.

a. Loading/unloading time

Assuming a basic rate of 20 containers per hour

handled off and on the ship for each crane, that is that
20 containers are "put through" at each terminal, the loading
time can be found from:

$$\text{loading hours} = \frac{(\text{cont. capacity}) \times (\% \text{ cont. handled})}{20 \times (\text{number of cranes})}$$

b. Port delays.

Delays due to time spent for operations such as
pilot to pier, customs clearance, awaiting berth, etc. of
10.5 hours per port are assumed.

Example

For a sample calculation of the different aspects of
operation in the North Atlantic trade, a fleet having the
following characteristics will be used:

Number of ships	3
Service speed	22 knots
Ship capacity, 20 ft. equiv. cont.	2000 per roundtrip
Specific fuel rate, lb/shp-hr	0.48
Cubic number	2841

Sea time per voyage

$$\text{number of days} = \frac{\text{distance travelled(miles)}}{\text{miles per day}} =$$

$$= \frac{7647}{22 \times 24} = 14.48 \text{ days}$$

Port time per voyage

Loading/unloading time

New York : loading hrs = $\dfrac{1000 \times 0.70}{20 \times 2}$ = 17.5 hrs.

Norfolk : " = $\dfrac{1000 \times 0.15}{20 \times 1}$ = 7.5 hrs.

Baltimore : " = $\dfrac{1000 \times 0.15}{20 \times 1}$ = 7.5 hrs.

Rotterdam : " = $\dfrac{1000 \times 0.40}{20 \times 2}$ = 10 hrs.

Southampton : " = $\dfrac{1000 \times 0.25}{20 \times 1}$ = 12.5 hrs.

Bremen : " = $\dfrac{1000 \times 0.20}{20 \times 1}$ = 10 hrs.

Le Havre : " = $\dfrac{1000 \times 0.05}{20 \times 1}$ = 2.5 hrs.

Goteborg : " = $\dfrac{1000 \times 0.10}{20 \times 1}$ = 5 hrs.

Total = 72.5 hrs = 3.02 days

Port delays

For an eight port trip the port delays amount to

8 x 10.5 = 84 hrs = 3.5 days.

Total Time Per Round Trip

This time for the particular example examined above
is:

t_p = 14.48 + (3.02+3.5) = 21 days.

Hence the frequency of service is once every three weeks. Due
to the fact that there are 3 ships in the fleet, service can
be provided weekly.

Fuel Consumption

The main propulsion fuel consumed throughout the trip consists of fuel consumed at sea plus fuel consumed in port.

Sea Fuel

The amount of fuel consumed at sea by each ship can be found from:

at sea fuel, bbls: $0.071 \times (SFR) \times (SHP) \times (\text{sea days per year})$

where   SFR = specific fuel rate, in lb/shp-hr.

In the above relationship 6.52 bbl/ton of fuel were assumed.

Port Fuel

The amount of fuel consumed in port was found according to:

port fuel, bbls: $(0.043 \times CTRS + 38.1) \times (\text{port days per year})$

where   CTRS = number of 20 ft. equivalent containers carried.

Auxilliary Fuel

The auxilliary fuel consumption was taken to be 5% of the main propulsion fuel consumption.

Example

Returning to the particular example examined above, the main propulsion fuel can be computed as follows:

sea fuel, bbl/yr,: $0.071 \times 0.48 \times 31,000 \times 250 = 264,120.$

Note that 250 sea days were assumed instead of 241 as one would expect if he took into consideration the sea and port time calculations discussed earlier. This was done because

some of the delay time, i.e. pilot to pier time, can be considered as sea time.

port fuel, bbl/yr,: (0.043 x 1,000 + 38.1) x 100 = 8,110. Hence the total fuel consumption per year for the entire fleet is found to be 3 x (264,120 + 8,110) = 816,690 bbls. This corresponds to 1088.9 bbl/day per ship. Finally, the auxilliary fuel consumption is found to be 5% of the main propulsion fuel consumption, or 54.4 bbl/day.

Fixed Expenses Per Round Trip

These include expenses which are not dependent upon the duration of the trip, but which include loading and unloading, pilotage, port dues, agents fees, stevedoring, etc. It should be noted that these are not the operating expenses for the trip, but are rather costs dependent upon the nature of the trip.

Fixed expenses per round trip were computed as follows:

Terminal costs: $50.0 per cont. loaded or discharged

Stevedoring: $11.12 per cont. loaded or discharged

ILA Assessment: $33.5 per cont. put through the port of New York

Stuffing: $93.96 per cont. stuffed or stripped

Port, Wharf, Dock: (250 + 0.02CN) x (ports per year) + (20 + 0.01CN) x (port days per year)

```
Commissions and
Brokerage Agency:  $22.0 per 20 ft. equivalent container
                   loaded or discharged.
```

For the particular example examined above, the fixed costs

per round trip are calculated to be 403,795 dollars.

Chartering Cost

Chartering costs were initially chosen as

$22,241/day because it was for these costs that the optimum

speed for the ships involved in the North Atlantic container-

ship trade was 22 knots. However, other chartering cost

may also be employed and all calculations performed in the

previous chapters are portrayed as a function of chartering

costs which run as low as $10,000/day and as high as

$30,000/day.

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