The INSTITUTE of MARINE ENGINEERS
Transactions

1949, Vol. LXI, No. 9

Economic Selection of Steam Conditions For Merchant Ships*

DOUGLAS C. MacMILLAN and MARK L. IRELAND

The continued increase in steam conditions for central station and industrial power plants has spurred the marine industry to consider the possibilities of similar advances in marine power plants. The corresponding advances in naval machinery serve as a valuable background of operating experience for commercial operators.

How far the commercial operator should go is primarily an economic question with higher first costs and carrying charges to be weighed against the expected reduction in fuel costs.

The solution of the problem for any particular ship requires a determination of the fuel consumption and the initial cost of various steam plants of the desired power. Studies previously have been made of comparative steam cycles, and considerable information published, [1]† to [12] inclusive, showing the thermal efficiency that may be expected with various steam conditions. However, there has been comparatively little information presented, [6] [8], concerning the initial cost of comparative marine steam plants having different steam conditions. References [4] and [15] give some information on the increased cost of higher steam conditions for stationary power plants.

In the course of their normal work during the past four years, the authors have made such studies independently for particular applications. These were rather limited in scope and, due to the rapid change in the relation of machinery cost and fuel cost, as illustrated by Fig. 1, it was considered advisable to undertake a new and more comprehensive series of studies. It is the purpose of this paper to present the results of these studies in such a manner, and with sufficient supplementary information, that the results may readily be adjusted in the future for a change in the basic price ratios of fuel and machinery.

* Reprinted by permission of the Society of Naval Architects and Marine Engineers, New York.
†Numbers in brackets indicate references listed at the end of the paper.

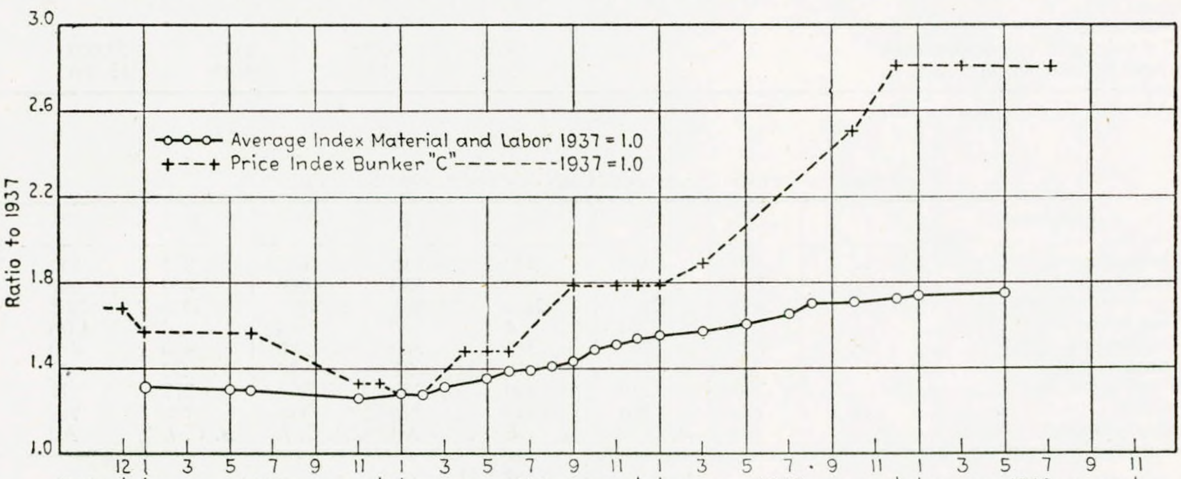

FIG. 1—*Increase in cost of machinery and fuel*

FUEL PERFORMANCE

The essential features of the various types of vessels covered by the present study are shown in Table 1. Only three power ratings are used but three quite different auxiliary steam and hotel requirements are applied to the single-screw 12,500-shaft-horsepower design, resulting in five separate studies.

A basic condition of 450 pounds per square inch gauge and 750 degrees F. at the superheater outlet has been used as the reference level for all five studies. Higher conditions studied range up to 1,250 pounds per square inch gauge and 950 degrees F. for the largest powered vessel, design *E*, as shown in Table 2. Reheat steam cycles have not been considered. The condenser vacuum used for these studies was 28·5 inches, corresponding to an optimum sea-water temperature of about 73 degrees F. A separate economic study was made to determine the optimum vacuum corresponding to average sea temperatures ranging from 60 to 80 degrees F. as shown later.

The number of feed heaters used was based upon the boiler pressure, alternate selections being tried in several cases as described later. In each case the boiler feed temperature was determined by the formula given at the bottom of Table 2. It will be recognized as similar in form to the approximate theoretical expression for the optimum feed temperature as a function of the number of heaters. In this expression the quantity 0·05, which is subtracted from $(n - 1)/n$, is an arbitrary value representing the economic factors involved. This correction results in a choice of feed temperatures which are in good agreement with current practice.

The deaerating feed heater pressure has been increased for the higher pressure studies to obtain a better division of temperature rise between heaters and to utilize more effectively the increased turbine cross-over pressure.

A fuel rate was calculated for each of the studies indicated in Table 2, and for a number of intermediate conditions. Details of the method are given in Appendix 1. It need be mentioned here only that the analysis takes into account the engine efficiency of the main units under extracting conditions, the actual feed temperatures and number of heaters, a constant boiler efficiency of 0·875, and an allowance for auxiliary steam requirements including turbo-generators, boiler feed pumps, low-pressure evaporators, air ejectors, and fuel-oil heating steam requirements. No allowance is made for ship's heating, as the sea temperature basis is 73 degrees F.

The reduction in fuel rate for higher steam conditions is shown in Fig. 2. It may be noted that a separate curve is required for each number of feed heaters as well as for each temperature and shaft horsepower per shaft.

These curves flatten off sooner than might be expected from previously published information, [1] [2] [3] [4] [5]. This is not due to the main units, as may be seen from Fig. 3, which indicates the reduction in fuel rate for main units only.

The explanation for the shape of the fuel rate curves lies in the turbo-generator and feed pump steam conditions selected for these studies. Evaluations showed that the turbo-generator steam conditions should be limited to a top figure of 590 pounds

TABLE 1.—CHARACTERISTICS OF SHIPS USED FOR ECONOMIC STUDIES.

Design identificaion	*A*	*B*	*C*	*D*	*E*
General type of vessel	C-2 cargo	Modified C-3 cargo	Tanker	Passenger-cargo	Passenger
Number of screws	1	1	1	1	2
Normal shaft horsepower per shaft	6,000	12,500	12,500	12,500	20,000
Propeller revolutions per minute	92	100	100	100	130
Number of passengers	0	12	4	188	653
Number of crew	48	54	57	162	452
Total persons	48	66	61	350	1,105
Air conditioning	None	None	None	Extensive	Extensive
Number compressors	—	—	—	2	2
Horsepower installed	—	—	—	1,000	800
Cargo refrigeration, cu ft	32,300	54,300	None	55,000	49,500
Refrigeration compressors, cargo and stores					
Number	5	15	2	(*a*)	3
Horsepower installed	37·5	90	15	(*a*)	180
A-c turbo-generators					
Number	2	3	2	3	4
Kilowatts	250	300	400	600	1,000
24-hour load, kw	195	360	300	900	1,650
Distilling plants					
Number and effects	1 × 1	1 × 1	1 × 1	2 × 2	2 × 2
Rating at 7·5 psia, gallons per day, each	6,000	6,000	6,000	20,000	40,000
Operating rate, gallons per day, each	3,000	4,500	4,200	13,000	27,000

(*a*) Included with air conditioning units.

TABLE 2.—STEAM CONDITIONS USED FOR ECONOMIC STUDIES.

Condition	1	2	3	4	5	6	7	8
Superheater pressure, psig	450	450	450	615	615	875	875	1,250
Superheater temperature, F	750	850	900	850	900	850	900	950
Exhaust vacuum, in. mercury	28·5	28·5	28·5	28·5	28·5	28·5	28·5	28·5
Number of feed heaters, *n*	3	4	4	4	4	4	4 (*b*)	5
Boiler feed temperature, F (*a*)	325	355	355	380	380	410	410	465
D-c heater pressure, psig	10	10	10	32	32	32	32	45
D-c heater temperature, F	240	240	240	277	277	277	277	293
Superheat control	No	No	Yes	No	Yes	No	Yes	Yes
Applied to vessels in study	All	*A, C, E*	*A, C, E*	All	*A, C, E*	*A, C, E*	All	*E*

(*a*) Heat added from condensate temperature to boiler feed temperature $=\left(\frac{n-1}{n} - 0{\cdot}05\right) \times$ (heat rise from condensate temperature to saturation temperature).
(*b*) Five stages used for 20,000 shp at this condition.

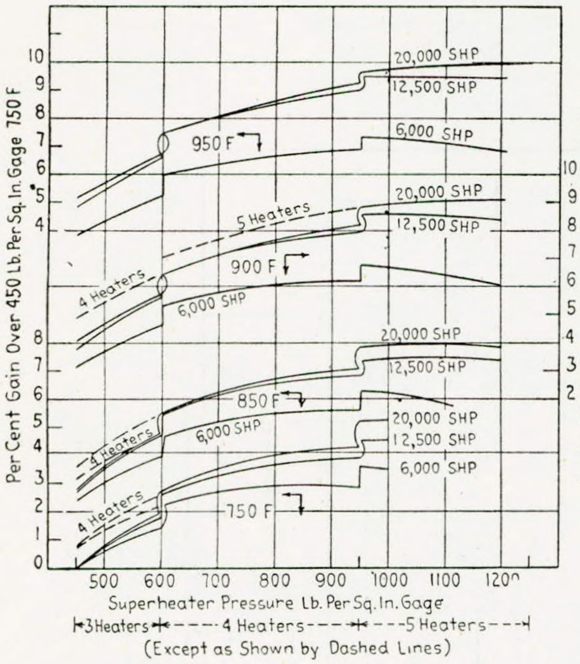

FIG. 2—*Per cent gain in ship fuel rate over 450 pounds per square inch gauge, 750 degrees F.*

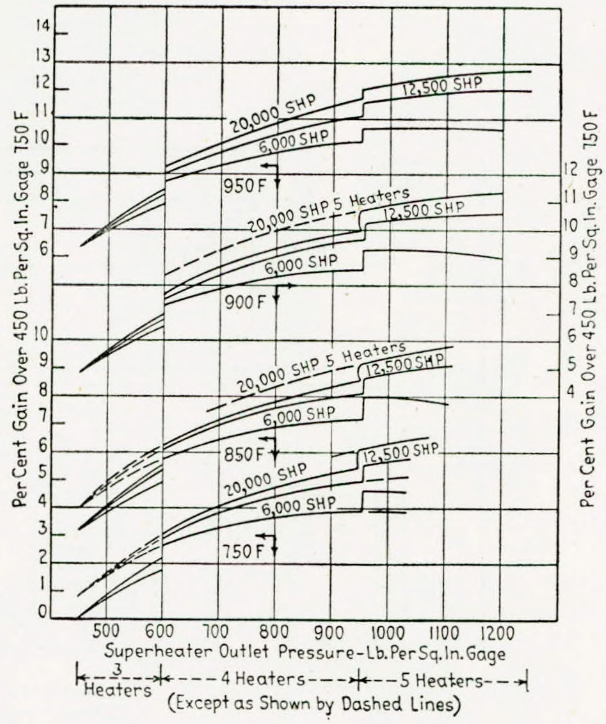

FIG. 3—*Per cent gain in main unit fuel rate over 450 pounds per square inch gauge, 750 degrees F.*

per square inch gauge and 825 degrees F. irrespective of higher main unit conditions. The corresponding limit for feed pump turbines was 590 pounds per square inch gauge, and 535 degrees F.

Although these limitations result in fuel rate gains which may appear somewhat disappointing, it will be shown later that the further gains to be obtained by using the main unit steam conditions cannot be justified on an economic basis.

The necessary desuperheating and pressure-reducing equipment to accomplish these limitations introduces no novel engineering features. As shown in the schematic flow diagram, Fig. 5, the turbo-generator steam is desuperheated in a small boiler drum coil and the feed pump turbine is supplied from an external desuperheater. The external unit is desirable in any case for pressures above about 600 pounds per square inch gauge as the other auxiliary desuperheated steam requirements are rather large for the amount of drum coil surface that can be used safely in a higher pressure boiler.

The use of motor-driven centrifugal feed pumps has been considered for the higher powers and steam conditions. Wound-rotor construction with speed control from an excess pressure regulator was selected as being the most suitable for control. Fig. 4 shows the fuel savings which are possible using motor-driven centrifugal feed pumps.

INITIAL COST

All those components of the machinery plant which may be influenced by the steam conditions should be included in the cost estimate. The studies prepared included the variation in cost of the following:

Main turbine gear units
Boilers, including erection
Turbo-generator sets
Feed pumps
Condenser plant
Feed heaters
Auxiliary steam desuperheaters
Reducing valves
Main steam piping and insulation
Generator steam piping and insulation
Bleeder steam and drain piping and insulation
Feed piping and insulation
Foundations
Installation
Overhead
Profit

In general, a cost estimate was prepared for each size and type of ship for each of the design steam conditions shown in Tables 1 and 2. In addition, supplementary estimates were prepared in sufficient number to determine the most economical selection of auxiliaries, particularly with reference to turbo-generators, feed pumps, number of stages of feed heating, etc. For all studies, estimating prices and engineering information

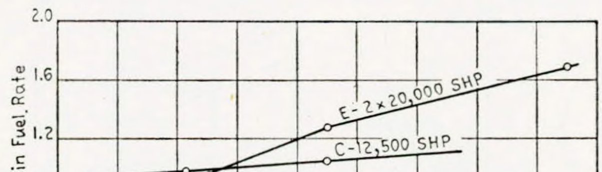

FIG. 4—*Per cent gain in ship fuel rate using motor-driven feed pumps, showing effect of shaft horsepower and steam pressure*

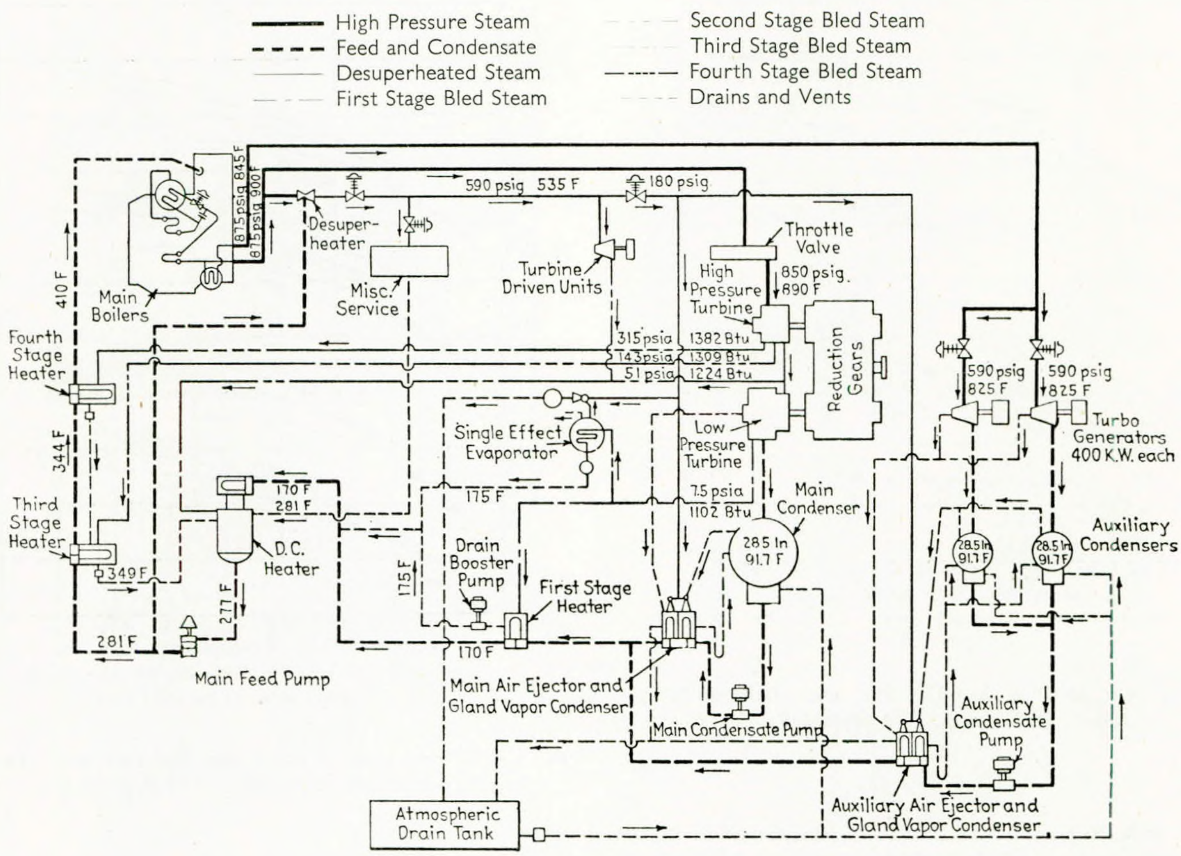

FIG. 5—*Schematic flow diagram: "C" study, 12,500 shaft horsepower, 875 pounds per square inch gauge, 900 degrees F.*

for each item of main machinery (including boilers, turbines, generators, feed pumps, condensers) were obtained from three manufacturers (except for conditions 2, 3, and 5, where price

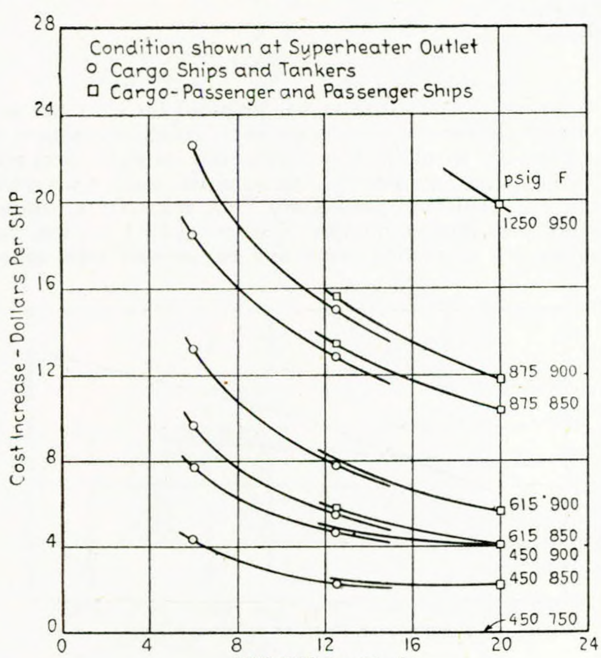

FIG. 6—*Increase in cost for various steam conditions*

information was obtained from only one manufacturer). Auxiliary machinery costs and performance data were obtained from at least one well-known manufacturer. Prices were based on one of three ships, and included American Bureau of Shipping classification and spare parts. Detailed pipe, valve, and fitting lists were prepared for each of the piping systems involved and cost estimates obtained. The results of the estimates are shown in Fig. 6, which shows the increased initial cost in dollars per shaft horsepower for the various designs considered.

Detail price data are not presented, but certain points of interest and importance are given in the following, together with the basis of price selection where alternate studies were prepared.

The prices received from the different manufacturers for the main turbine gear units were in substantial agreement, and were averaged for use in the main price estimates.

It should be noted that several of the turbine manufacturers have their own standard "book" or "published" prices for marine geared-turbine units. The base "book" or "published" price depends mainly upon the horsepower, revolutions per minute, and "K" factor for the reduction gears, the manufacturer's standard construction, and the steam conditions. This price is based upon a throttle steam pressure of 600 pounds per square inch gauge and steam temperature of 750 degrees F., or less. For higher steam pressures and temperatures, an addition is made at certain points, which occur at 601, 901, and 1,501 pounds per square inch gauge, and 751, 851, 901, and 951 degrees F. These price steps naturally influence the studies and it will be noted that the advanced design conditions used are just under the limits of each price group in order to secure maximum benefit from the increased cost. The percentage price increases are given in Table 3.

TABLE 3.—PERCENTAGE INCREASE IN COST OF PRINCIPAL ITEMS.

Steam conditions at superheater outlet				
Pressure, psig	450	615	875	1250
Temperature, F	750	850	900	950
Boilers	0	16	44	85
Turbines	0	4	12	25
Feed pumps	0	30–55	80–110	170
Feed heaters and bleeder piping ...	0	75	85	185
Condenser plant	0	(6)	(10)	(14)
Piping and insulation	0	12–(5)	25–50	62

Remarks :
( ) denotes decrease.
Range is given where change varies with size (6,000 to 20,000 shp).
Data for 1,250 psig 950 F condition obtained for 20,000 shp design only.

The authors find that the price increase for a step from 875 pounds per square inch gauge 900 degrees F. to 1,250 pounds per square inch gauge 950 degrees F. for a 20,000-shaft-horsepower geared-turbine unit is several times the corresponding increase for a 40,000-kilowatt stationary turbine-generator unit.

The prices received from the boiler manufacturers were in substantial agreement, and were averaged for use in the final studies. The percentage increase in cost for the higher pressure boilers is shown in Table 3.

Erection costs are higher for the high-pressure high-temperature boilers, due to the increased time necessary for rolling heavier tubes into thicker tube sheets and, in some cases, for welding in tube ends.

Prices and steam rate information for turbo-generator sets were obtained from the manufacturers for sets ranging from 250 to 1,000 kilowatts with various inlet steam conditions. Here again, several manufacturers have their own standard "book" or "published" prices with price changes occurring at certain points depending upon steam conditions. Unfortunately, all these points do not coincide with those used for the main turbine gear units, as they occur at 401, 601, and 901 pounds per square inch gauge, and 751, 826, and 901 degrees F. It will be appreciated that the steam rates for these small sets will not reflect the same gains for high-pressure and high-temperature that are obtained with the main units. Therefore, for each size, the economics of turbo-generator steam conditions and prices were studied on the same basis as the main economic analysis, described later.

In most cases it was found that turbo-generator sets having steam conditions in excess of 600 pounds per square inch gauge and 825 degrees F. do not pay; that is, the reduction in fuel cost for higher steam conditions and lower steam rate will not compensate for the additional fixed charges. This difference is rather pronounced, and will more than compensate for the cost of the additional desuperheating and pressure-reducing equipment previously described. Therefore, the main price estimates include generator sets having steam conditions not exceeding 590 pounds per square inch gauge and 825 degrees F. An example of an economic study for the turbo-generator steam conditions is shown in Table 6, and Fig. 7 shows typical data for increase in price and decrease in steam rate. It will be noted that data for both low and high efficiency units are shown for the 300-kilowatt size.

The main price estimates include the average cost of turbo-feed-pumps having steam conditions not exceeding 590 pounds per square inch gauge, 535 degrees F., which were found to be more economical than higher steam conditions. The percentage increase in turbo-feed-pump prices is also shown in Table 3.

The cost of the optimum size condenser plant for an exhaust pressure of 1·5 inches of mercury was selected from the economic study of exhaust pressure, described later. The cost reduction due to the smaller condenser plants required for

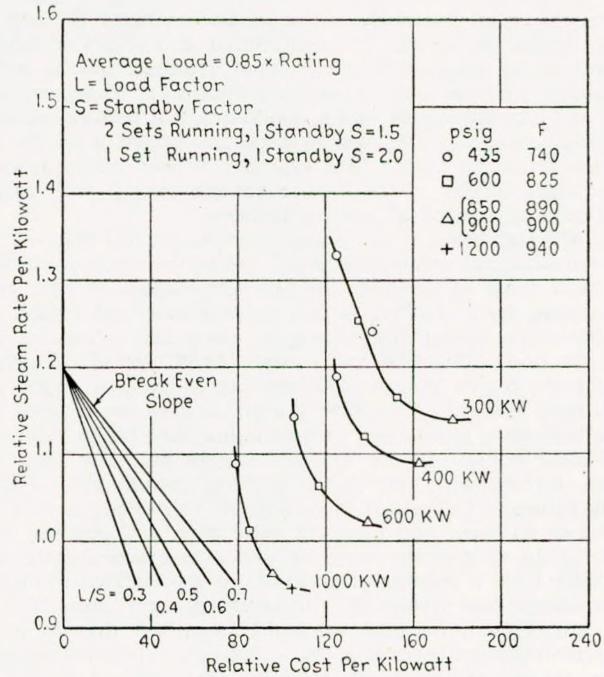

FIG. 7—*Turbo-generator sets: relative cost and steam rate for various steam conditions*

higher steam conditions is about equal to the increase for the high-pressure feed heaters. The average reduction is shown in Table 3.

Various alternative studies were made for a number of designs and conditions in order to arrive at the most economical number of feed heaters and bleeders, with final selection as shown in Table 2. A typical economic study is given in Table 7, and the percentage price increase is shown in Table 3.

One result indicated by the piping study is that the 615-pound 850-degree F. main steam piping costs less than the 450-pound 750-degree F. piping. This is mainly because a pressure of 450 pounds at the superheater outlet requires 600-pound carbon steel piping, valves and flanges, if standard facings are used. A reduction in steam pressure to 425 pounds would permit using 400-pound carbon moly piping and valves resulting in an appreciable cost saving, particularly since carbon moly valves are now standard. However, an economic study of 425-pound pressure and 750-degree F. made for the 12,500-shaft-horsepower tanker design with an 80 per cent load factor showed no saving in operating cost as compared to 450 pounds, since the increased fuel cost was about the same as the reduction in carrying charges. The percentage change in the cost of main and generator steam, and feed piping is shown in Table 3.

ECONOMIC FACTORS

The study so far has attempted to present values for the percentage reduction in fuel consumption obtainable by using higher pressures and temperatures, and the increase in first cost required to accomplish this result. While these may be considered the distinctive factors investigated, it appears desirable at this point to review briefly all factors which might be influenced by the selection of the steam conditions.

Operating costs which might be affected include crew wages, fuel, lubricating oil, water, machinery stores and supplies, and maintenance and repair. It is believed that the cost of lubricating oil, water, and machinery stores and supplies will be relatively unaffected by the selection of steam conditions for the cycles considered. Also, the crew cost should not vary, since the number required and the pay scale do not depend upon the steam conditions. The cycles compared for any one ship design are similar in arrangement and components, and therefore the amount of crew labour required should not be modified for the varying steam conditions. However, it is believed that the higher steam conditions will require increased alertness and more attention to good housekeeping on the part of the operating force. For example, higher pressure boilers are more sensitive to feed-water contamination, and internal and external fouling of heating surfaces.

Although there is no conclusive evidence, it is believed that maintenance and repair may be affected by the steam conditions. In some cases in the past, the first applications of new steam conditions have resulted in less maintenance and repair, due probably to careful engineering in designing, planning, and construction. The principal features which appear to indicate inherently higher maintenance are higher pressure joints and throttling devices. The first group includes boiler tube and handhole seats, piping and turbine joints, feed heater tube seats and head joints. In the second category are valve seats and disks, turbine and feed pump packing glands, etc. Higher temperatures also make it more difficult to provide satisfactory joints of all types and to avoid valve stem and bushing difficulties. In view of the foregoing and after examining the data available from a number of operators, it was decided to include maintenance and repairs as a fixed charge, and assess it at 1½ per cent of the initial cost. It will be seen later that, by including maintenance expense in this manner, it is possible to determine immediately its effect upon the result and, if desired, to vary it at will.

The remaining operating cost—fuel—has been considered as it is influenced by steam conditions, size and type of ship, and efficiency of components. In order to translate this fuel variation into dollars, it is necessary to apply the load factor (ratio of days at sea per year to 365) and the unit fuel cost for the particular application. Therefore, the results obtained herein are presented also for a range of values for each of these factors.

Fixed charges primarily depend upon the initial investment, which already has been considered, and the interest, amortization, and insurance rates. In order that the results may be examined for particular cases, these factors are also given a range of values.

However, in the selection of components, it was necessary to assume a rate for fixed charges, and also other factors. For this purpose, the rates and factors given in Table 4 were used.

TABLE 4.—BASIC CONDITIONS FOR ECONOMIC STUDIES AND SELECTION OF AUXILIARIES (UNLESS STATED OTHERWISE).

Unit fuel cost, *f*, dollars per barrel		3·00
Fuel density, pounds per barrel		340
Fuel heat content, Btu per pound		18,500
Fixed charge rate, *C*, per cent		11
Load factor, *L*, days at sea per year/365		0·6
For tanker studies, variable		0·85–0·95
Fixed charge rate, *C*, is based on the following :		
Interest	2·6	
Depreciation	4·9	
Insurance	2·0	
Maintenance and repair	1·5	
Total	11·0 per cent	

It is believed that the 9·5 per cent total for the first three items of fixed charges is in close agreement with usual practice and is intended to represent the average during a 20-year life.

Other items appearing in an operating statement, such as overhead, taxes, and profit, are not normally related to the propelling machinery. Overhead depends upon the level of operations and the efficiency of the operator, and is considered to have no influence on the present problem. Profit normally is based on revenue, but often is considered in relation to investment. On the latter basis, more expensive machinery would be required to show a profit on the increased initial investment. Therefore, the results are so presented that this may be determined, if required, by adding the profit percentage desired to the fixed charge rate. Taxes are based on profit, and consequently their effect should be considered in the same manner as profit.

Summarizing the foregoing, it is found that the results should provide for variations in the following:

Fuel saving, depending upon steam conditions, etc.
Increased cost, depending upon steam conditions, etc.
Unit fuel cost
Load factor
Fixed charge rate

It is believed that the method selected for the presentation

TABLE 5.—RESULTS OF

Shaft horsepower	6,000					
Superheater pressure, psig	450	450	615	615	875	875
Superheater temperature, F	850	900	850	900	850	900
Fuel saving, per cent	3·2	3·9	4·8	5·45	5·6	6·15
Comparison $f = 3{\cdot}0$, $L = 0{\cdot}6$, $C = 11$						
Fuel savings, dollars per year	5,300	6,450	7,940	9,020	9,270	10,200
Increased fixed cost, dollars per year	2,870	5,120	6,380	8,750	12,190	14,930
Net savings, dollars per year	2,430	1,330	1,560	270	−2,920	−4,730
Return on investment, per cent	9·3	2·9	2·7	0·3	Loss	Loss
Comparison $f = 2{\cdot}0$, $L = 0{\cdot}6$, $C = 11$						
Fuel savings, dollars per year	3,530	4,300	5,290	6,010	6,180	6,800
Increased fixed cost, dollars per year	2,870	5,120	6,380	8,750	12,190	14,930
Net savings, dollars per year	660	−820	−1,090	−2,640	−6,010	−8,130
Return on investment, per cent	2·5	Loss	Loss	Loss	Loss	Loss
Comparison $f = 4{\cdot}0$, $L = 0{\cdot}6$, $C = 11$						
Fuel savings, dollars per year	7,060	8,600	10,590	12,030	12,360	13,600
Increased fixed cost, dollars per year	2,870	5,120	6,380	8,750	12,190	14,930
Net savings, dollars per year	4,190	3,480	4,210	3,280	170	−1,330
Return on investment, per cent	16·0	7·5	7·3	4·1	0·2	Loss
Comparison $f = 3{\cdot}0$, $L = 0{\cdot}6$, $C = 7{\cdot}7$						
Fuel savings, dollars per year	5,300	6,450	7,940	9,020	9,270	10,200
Increased fixed cost, dollars per year	2,010	3,580	4,470	6,120	8,530	10,450
Net savings, dollars per year	3,290	2,870	3,470	2,900	740	−250
Return on investment, per cent	12·6	6·2	6·0	3·6	0·7	Loss
Comparison $f = 3{\cdot}0$, $L = 0{\cdot}9$, $C = 11$						
Fuel savings, dollars per yea	7,950	9,680	11,910	13,530	13,900	15,300
Increased fixed cost, dollars per year	2,870	5,120	6,380	8,750	12,190	14,930
Net savings, dollars per year	5,080	4,560	5,530	4,780	1,710	370
Return on investment, per cent	19·5	9·8	9·5	6·0	1·5	0·3

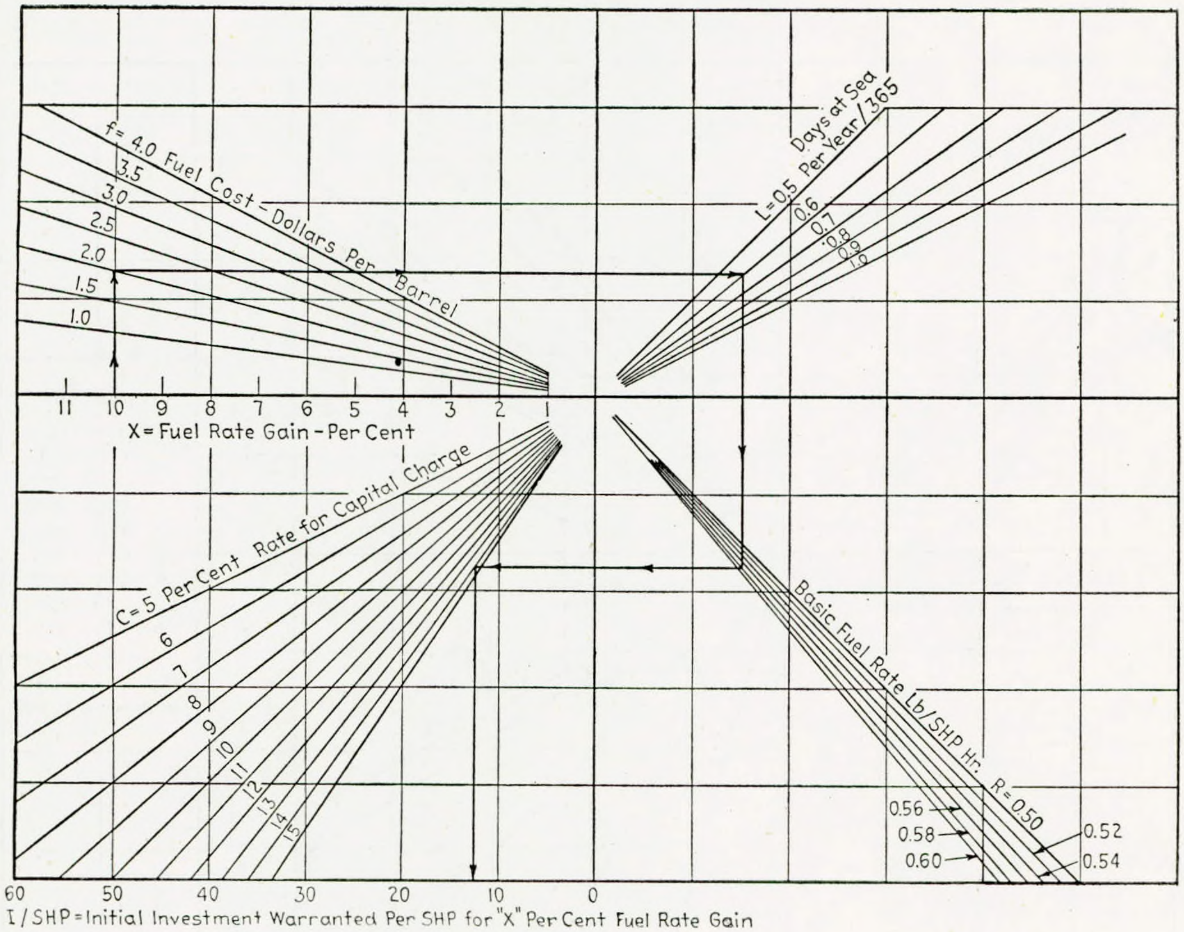

FIG. 8

ECONOMIC STUDIES.

12,500						20,000						
450	450	615	615	875	875	450	450	615	615	875	875	1,250
850	900	850	900	850	900	850	900	850	900	850	900	950
3·6	4·85	5·6	6·55	6·7	7·8	3·6	4·85	5·7	6·55	6·9	8·6	9·95
11,590	15,620	18,030	21,090	21,570	25,120	18,810	25,340	29,780	34,220	36,050	44,940	52,000
3,160	6,400	7,600	10,770	17,670	20,660	4,870	8,930	8,910	12,340	22,770	25,970	43,540
8,430	9,220	10,430	10,320	3,900	4,460	13,940	16,410	20,870	21,880	13,280	18,970	8,460
29·4	15·8	15·1	10·5	2·4	2·4	31·4	20·2	25·8	19·5	6·4	8·0	2·1
7,730	10,360	12,010	14,070	14,370	16,740	12,540	16,880	19,870	22,830	24,040	30,000	34,640
3,160	6,400	7,600	10,770	17,670	20,660	4,870	8,930	8,910	12,340	22,770	25,970	43,540
4,570	3,960	4,410	3,300	−3,300	−3,920	7,670	7,950	10,960	10,490	1,270	4,030	−8,900
15·9	6·8	6·4	3·4	Loss	Loss	17·3	9·8	13·5	9·3	0·6	1·7	Loss
15,460	20,840	24,040	28,120	28,750	33,500	22,080	33,780	39,700	45,650	48,100	59,900	69,400
3,160	6,400	7,600	10,770	17,670	20,660	4,870	8,930	8,910	12,340	22,770	25,970	43,540
12,300	14,440	16,440	17,350	11,080	12,840	17,210	24,850	30,790	33,310	25,330	33,930	25,860
42·8	24·8	23·8	17·7	6·9	6·8	38·9	30·7	38·2	29·7	12·2	14·4	6·5
11,590	15,620	18,030	21,090	21,570	25,120	18,810	25,340	29,780	34,220	36,050	44,940	52,000
2,210	4,480	5,320	7,560	12,370	14,460	3,410	6,250	6,240	8,640	15,940	18,180	30,480
9,380	11,140	12,710	13,530	9,200	10,660	15,400	19,090	23,540	25,580	20,110	26,760	21,520
32·7	19·2	18·4	13·8	5·7	5·7	34·8	23·5	29·0	23·0	9·7	11·3	5·4
17,390	23,430	27,040	31,640	32,360	37,680	28,220	38,010	44,670	51,330	54,080	67,410	78,000
3,160	6,400	7,600	10,770	17,670	20,660	4,870	8,930	8,910	12,340	22,770	25,970	43,540
14,230	17,030	19,440	20,870	14,690	17,020	23,350	29,080	35,760	38,990	31,310	41,440	34,460
49·6	29·3	28·2	21·3	9·2	9·1	53·9	36·8	44·1	34·7	15·1	17·5	8·7

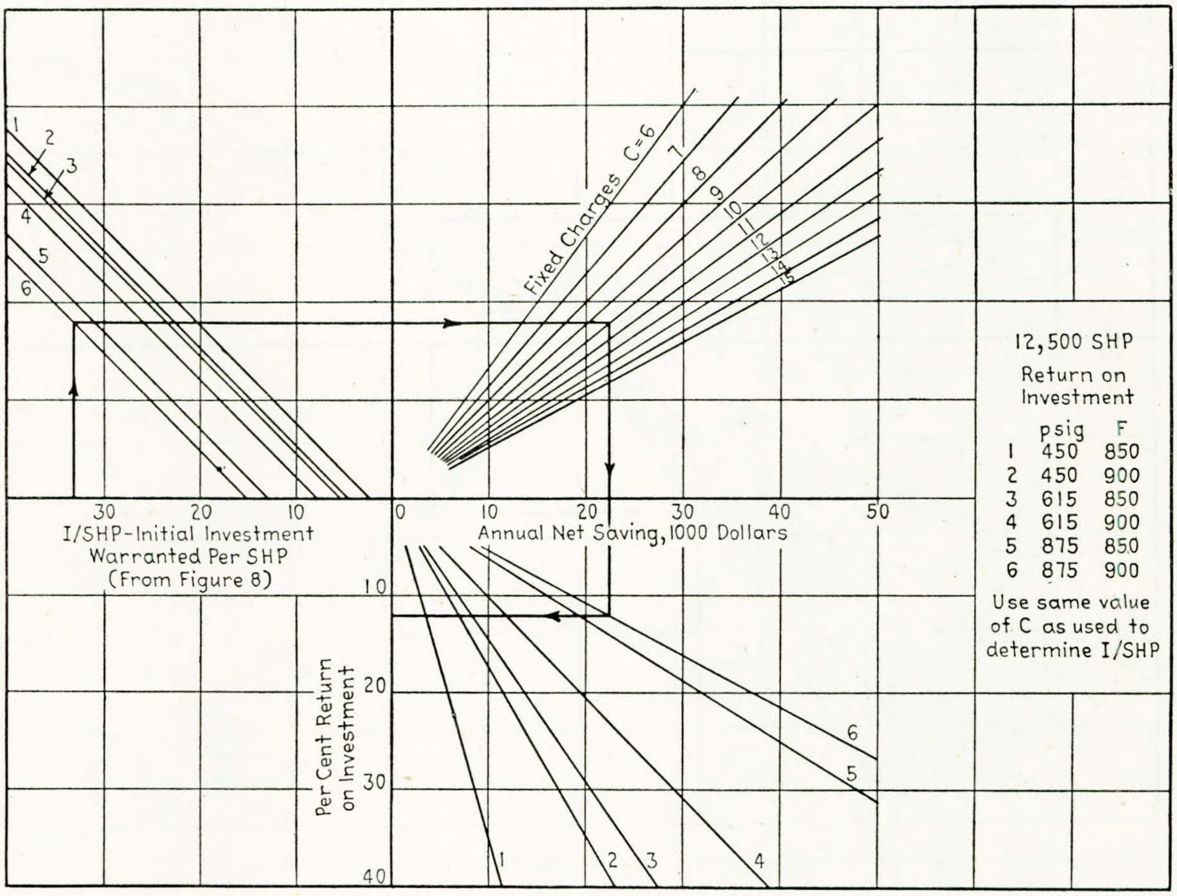

FIG. 9

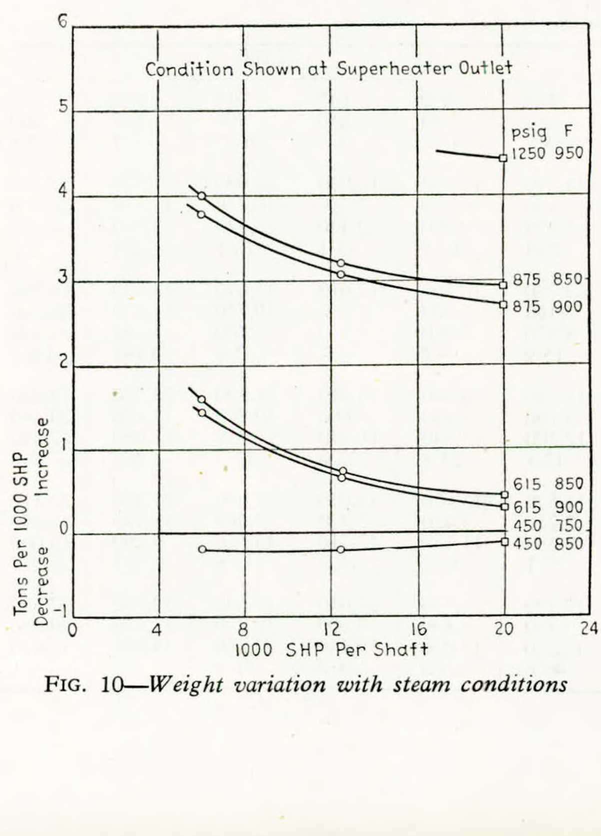

FIG. 10—*Weight variation with steam conditions*

of the first and second items is less controversial and less subject to individual interpretation than other methods which might have been used. The use of first cost of the entire basic design and percentage increase in cost has been avoided, since individuals do not always include the same components when determing machinery cost.

Of the many ways to present the economic results, two have been selected as being of greatest significance. They are:

(a) The annual saving due to higher steam conditions; i.e., the saving in fuel cost less the increase in fixed costs due to the increased capital investment.

(b) The per cent return on the investment as determined by the ratio of the annual saving to increased capital outlay.

It has been pointed out on numerous previous occasions that (a) is of particular interest in a highly competitive trade. However, to be attractive it would appear that the savings indicated must also be of an appreciable magnitude compared to the investment, and (b) is one method of representing this feature.

The effect of the factors mentioned in the foregoing is shown in Fig. 8 where starting with a particular fuel saving x in per cent one may find the increased initial cost in dollars per horsepower, *I/SHP,* which will just break even on an operating cost basis. The annual saving to be realized is simply the "break even" cost increase less the actual cost increase times the rate for capital charges which was used.

TABLE 6.—COMPARISON OF TURBO-GENERATOR SET STEAM CONDITIONS (Using Fig. 7).

Ship		C	
Superheater pressure, psig	615		875
Superheater temperature, F	850		900
Turbo-generator throttle pressure, psig ...	590		850
Turbo-generator throttle temperature, F ...	825		890
Generators, number and size, kw		2–400	
Load factor, *L*		1·0	
Standby factor, *S*		2·0	
Ratio, *L/S*		0·5	
Cost per kilowatt, Fig. 7	137		162
Cost increase	0		25
Cost increase warranted (*a*)	0		7
Difference	0		(18)
Difference, dollars	0		(14,400)
Difference, dollars per year	0		(1,585)
If f = 2·50			
Equivalent slope = 2.5/3·0 × 0·5, = *L/S* ...		0·417	
Cost increase warranted (*a*)	0		6
Difference	0		(19)
Difference, dollars	0		(15,200)
Difference, dollars per year	0		(1,670)
If C = 7·5			
Equivalent slope = 11/7·5 × 0·5 = *L/S* ...		0·733	
Cost increase warranted (*a*)	0		10
Difference	0		(15)
Difference, dollars	0		(12,000)
Difference, dollars per year	0		(900)

( ) denotes loss.

(*a*) Increase in cost warranted by reduction in relative steam rate from 1·12 to 1·09 using "break even" slope.

TABLE 7.—BLEEDER COMPARISON.
20,000 shp, 875 psig, 900 F
Comparison of five versus four heaters.

Number heaters	5
Increase in cost, dollars	13,870
Increase in cost, I/shp	0·69
Gain in fuel rate, per cent, Fig. 2	0·65
Increase in cost warranted, I/shp (*a*)	1·54
Saving, I/shp	0·85
Saving in annual cost, dollars	1,870
Profit on increased investment, per cent	13·5

(*a*) For basic conditions, see Table 4.

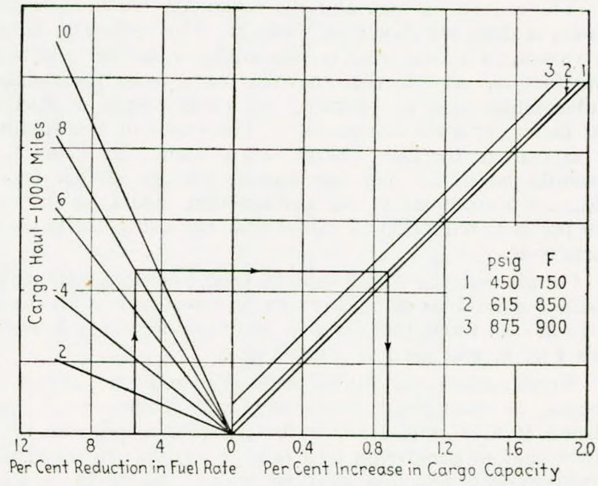

FIG. 11—*Tanker: increase in cargo capacity for weight change due to fuel and machinery*

TABLE 9.—CHARACTERISTICS ASSUMED FOR TANKER STUDIES.

Length, ft.		600
Displacement, tons		34,100
Shaft horsepower		12,500
Sea speed, kts		
Loaded		16·0
Ballast		16·7
Steam conditions, basic		
Pressure, psig		450
Temperature, F		750
Fuel rate, lb per shp per hr		0·565
Fuel consumption, tons per day		
At sea		75·7
In port		15·5
Port time, days per round trip		2
Overhaul days per year		20
Cargo capacity, 30 deg API	Bbl per trip	Bbl per year (10^6)
2,000-mile haul	185,000	5·24
4,000-mile haul	178,000 (*a*)	2·74
6,000-mile haul	170,000 (*a*)	1·80
10,000-mile haul	155,000 (*a*)	1·01

(*a*) Allows for increased fuel and stores.

TABLE 8.—COMPARISON OF MOTOR AND TURBINE DRIVEN CENTRIFUGAL FEED PUMPS.

Ship	C	D		E
Shaft horsepower	12,500	12,500		20,000
Superheater pressure, psig	615	615		875
Superheater temperature, F	850	850		900
Fuel gain for motor-driven pump, per cent ...	0·98	0·98		1·28
Gain corrected for part-time use	0·98	0·735 (*a*)		1·28
Increase in cost warranted, I/shp, (*b*) ...	3·51	1·8		3·02
Increased cost for motor-pump installation ...	38,720 (*c*)	6820 (*d*)	29,260 (*e*)	50,270 (*c*)
Increase in cost, I/shp	3·09	0·55	2·34	2·51
Net gain, I/shp	0·42	1·25	(0·54)	0·51
Saving in annual cost, dollars	580	1720	(750)	1120
Profit on increased investment, per cent ...	1·5	25	—	2·2

( ) denotes loss.

(*a*) Turbo-feed pump is assumed to be in operation 25 per cent of sea time to compensate for increased electrical hotel load under certain conditions.

(*b*) Based on Fig. 8 and Table 4. $L = 0{\cdot}9$ for tanker.

(*c*) Includes change in cost for one motor-pump, control, and increment for increasing size of generator plant.

(*d*) Includes change in cost for one motor-pump, control, cable, panel, etc.

(*e*) Pump charged with pro rata cost per kilowatt of generator plant.

The annual savings and the return on the investment for various designs are shown in Table 5. This table also indicates the variations in the results depending upon the selection of basic factors. A selection may be made from these data, or similar tables may be prepared for other values of fuel cost, load factor, or fixed charge rate. The result of a reduction of 30 per cent in the fixed charge rate is shown in Table 5, and it should be noted that the annual savings shown for this reduction correspond to the savings that would be shown for a 30 per cent reduction in initial cost but using the basic fixed charge rate.

Graphs such as Fig. 9 may be used to obtain a direct indication of annual saving and return on investment. The increase in initial cost warranted for any set of conditions is determined from Fig. 8, and used to enter Fig. 9.

Similar economic studies were prepared for various components, as mentioned previously, and Tables 6, 7, and 8 indicate typical results for turbo-generators, number of feed heaters, and motor-driven feed pumps. It will be noted that a turbo-generator selection may be made directly from Fig. 7, where the "break even" slope is indicated (for conditions given by Table 4 except as noted). The most economical set is indicated by the point on any curve where the tangent is the same as the "break even" slope selected for the particular condition being considered. A comparison using this method is given in Table 6.

The motor-driven feed pump results in an increase in first cost, which to a large extent depends upon whether an increase in the size of the generator set is or is not required. For example, ship *D* has a low normal generator load relative to installed capacity in order to provide for variation in the air-conditioning load due to weather conditions. With any of the steam conditions considered, it would be possible to use a motor-driven pump without changing the generator size, providing one pump is motor driven and one is turbine driven so that the steam unit may be used during periods of high air-conditioning load. For other designs such as *C* and *E*, it would be necessary to increase the size of the generator set, condenser plant, switchboard, etc. Typical results are shown in Table 8, and it appears that the gains shown represent a poor return on the investment.

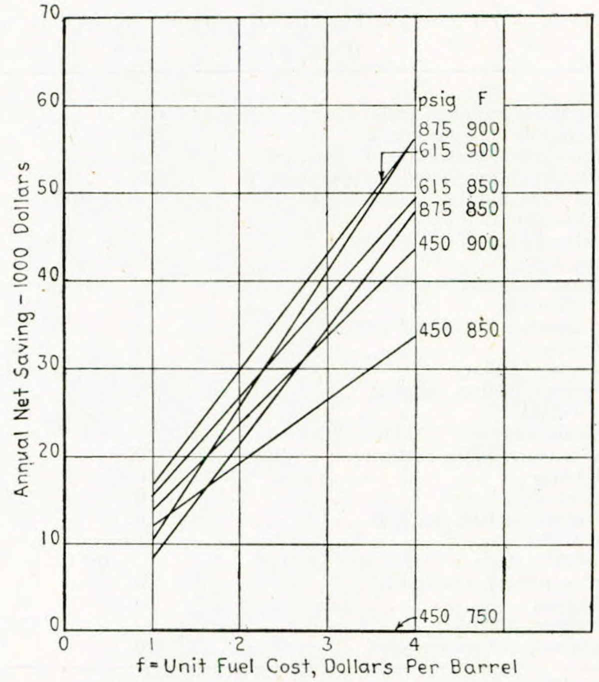

FIG. 12—*Tanker: annual savings for various steam conditions, cargo carried 10,000 miles (L = 0·91) (includes credit for weight)*

WEIGHT

As might be expected, there is a change in machinery weight due to higher pressures and temperatures, and the average results are shown by Fig. 10. Of the increase shown, 60 to 80 per cent is due entirely to the increase in boiler weights, although the boiler manufacturers' data were not as consistent as desirable, and not nearly as consistent as the price information. It will be noted that the weight changes are relatively unimportant, and would have little significance except as they

TABLE 10.—TANKER: EFFECT OF WEIGHT AND VARIOUS STEAM CONDITIONS.

Distance cargo carried, miles Sea time or load factor (*L*)	10,000 0·91					
Superheater pressure, psig	450	450	615	615	875	875
Superheater temperature, F	850	900	850	900	850	900
Investment and fuel cost change						
Per cent reduction in fuel rate, Fig. 2	3·6	4·85	5·6	6·55	6·7	7·8
Initial investment warranted, I/shp, Fig. 8 (*a*)	10·86	14·63	16·90	19·75	20·2	23·55
Increased cost, I/shp	2·3	4·66	5·53	7·83	12·85	15·0
Difference, I/shp	8·56	9·97	11·37	11·92	7·35	8·55
Annual savings, dollars	11,780	13,720	15,620	16,400	10,110	11,760
Saving, cents per barrel (*b*)	1·17	1·36	1·55	1·62	1·00	1·16
Weight change						
Cargo increase, Fig. 11, per cent	0·79	1·07	1·19	1·42	1·28	1·54
Credit for extra cargo, cents per barrel (*c*)	1·09	1·48	1·65	1·97	1·77	2·14
Credit for extra cargo, dollars per year	11,000	15,000	16,700	19,900	17,900	21,600
Total						
Total reduction, cents per barrel	2·26	2·84	3·20	3·59	2·77	3·30
Total reduction, per cent	1·6	2·0	2·3	2·6	2·0	2·4
Net saving per year, dollars	22,800	28,700	32,300	36,300	28,000	33,400
Net saving per year, per cent increased investment	79·4	49·3	46·8	37·1	17·4	17·8

(*a*) $f = 2{\cdot}50$; $C = 0{\cdot}11$.
(*b*) Cargo carried per year = 1,010,000 barrels.
(*c*) Basic transportation cost = 138·5 cents per barrel.

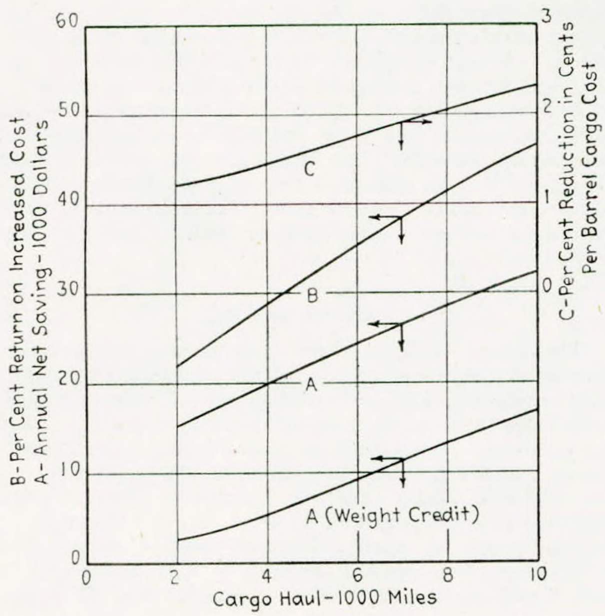

FIG. 13—*Tanker: effect of weight and distance, gain for 615 pounds per square inch gauge, 850 degrees F., over 450 pounds per square inch, gauge 750 degrees F.* (f = 2·50)

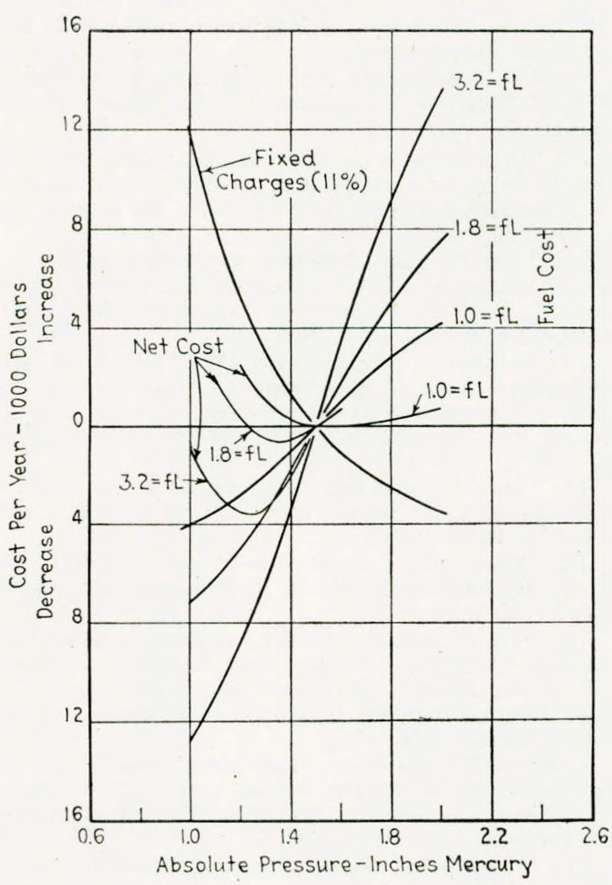

FIG. 14—*Economical vacuum, 20,000 shaft horsepower, 70 degrees F. sea temperature*

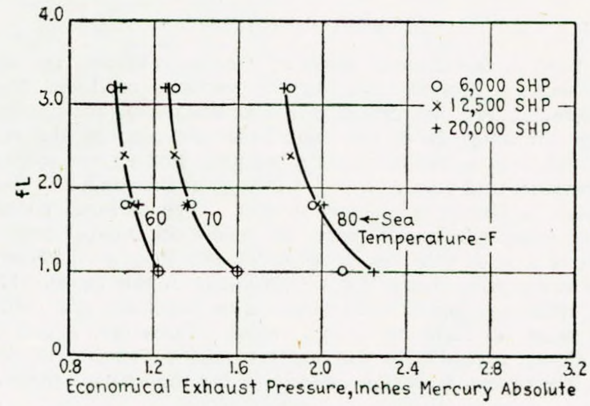

FIG. 15

might affect the economics of a deadweight carrier.

The fuel savings due to increased pressure and temperature will reduce directly the bunker requirements for a given voyage, and thereby result in an increased pay load for a deadweight carrier.

In order to assess the effects of the two items just mentioned, we have assumed a tanker design of normal characteristics as given in Table 9, and have estimated the increase in cargo capacity when cargo is transported over distances ranging from 2,000 to 10,000 miles. The per cent increase in pay load is shown in Fig. 11.

The combination of the increase in pay load plus the reduction in fuel cost obtained with the higher steam conditions has been investigated and typical results for the assumed tanker are given in Table 10 and Figs. 12 and 13. Table 10 compares a number of steam conditions at one distance, which is a probable maximum haul, and therefore indicates the maximum saving to be credited to weight reduction. Fig. 12 indicates the effect of unit fuel cost on the savings obtained with various steam conditions at the maximum distance. Fig. 13 indicates the effect of distance on the savings when the 615 pounds per square inch gauge and 850 degrees F. design is compared with the base.

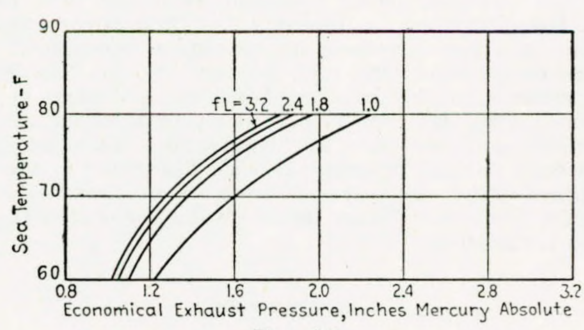

FIG. 16

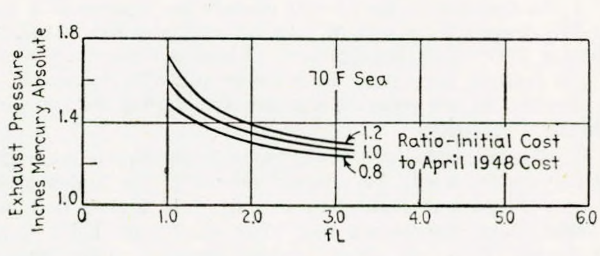

FIG. 17

EXHAUST PRESSURE

With a given set of throttle steam conditions, the steam consumption depends also on the vacuum produced in the condenser. The fuel consumption is affected by the change in steam consumption of the main unit, and also by the power required to drive the circulating pumps. For a given sea-water temperature and power output, a change in the condenser design vacuum results in a change of cost. The turbine manufacturers' standard "book" prices are based on exhaust pressures between 1 and 4 inches of mercury absolute, and, therefore, there is no price change for design vacua in this range. However, there is a considerable variation in the cost of the condensing plant for different design vacua. Therefore, it has been considered advisable to investigate exhaust pressure to determine the optimum design conditions for this end of the steam cycle.

The investigation covered a range of sea temperatures from 60 to 80 degrees F., exhaust pressures from 0·75 to 3·0 inches of mercury absolute, and exhaust steam flows from 38,000 to 104,000 pounds per hour per unit (roughly from 6,000 shaft horsepower to 20,000 shaft horsepower per unit). The costs of various sizes and proportions of condensing plants were estimated. The items included in initial cost were condenser, air ejector, circulating pump, pump motor and controls, electric generating capacity for pump motor, piping, installation, overhead, and profit. Yearly fixed charges of 11 per cent were used to determine the "break even" point, no allowance being made for profit on the additional investment, since no risk is involved. The variation in turbine performance was determined from the efficiency data presented in Appendix 2.

The result of the calculations was a series of nine plots of which Fig. 14 is a sample. The optimum points from the curves shown thereon were then plotted to give Figs. 15 and 16, which indicate the range of economical exhaust pressure. The results indicated are not at variance with present practice. Results for a stationary power plant are indicated by reference [14].

Appendix 3 presents sufficient information for recalculation, so that plots similar to Fig. 14 may be prepared if the price level changes or if a different fixed charge rate is desired. Fig. 17 indicates the effect of a change in initial cost for one set of design conditions.

The condenser design conditions on which these studies were based are given in Appendix 3. These exhaust pressure studies have not considered the economical selection of condenser design conditions, such as water velocity, tube length, size, material, etc., but have been based on usual values of these factors. Also, these studies have not considered the effect of fluctuations in sea-water temperature upon the selection of economical exhaust pressure. A wide fluctuation in sea temperatures during the year will result in an optimum design different from the optimum design obtained at the average sea-water temperature.

CONCLUSIONS

Based on the results presented, the following general conclusions are indicated:

1. Steam conditions of 450 pounds per square inch gauge and 750 degrees F. would be selected normally for installations of about 6,000 shaft horsepower. There is some improvement for 450 pounds per square inch gauge and 850 degrees F., but the incentive is not great even when considering the probability of higher fuel prices.

2. Steam conditions of 615 pounds per square inch gauge 850 degrees F. would be selected normally for higher powers for both cargo and combination ships. If higher fuel prices or lower fixed charges are used than shown in Table 4, then 900 degrees F. at the same pressure shows a slight improvement, which, however, hardly appears to be worth the risk. In any case there seems to be little incentive to a further increase in steam pressure. In this connexion it may be pointed out that steam conditions exceeding 600 pounds per square inch gauge 825 degrees F. are offered for standard stationary power plants only for unit ratings of 20,000 kilowatts or greater.

3. For a tanker of 12,500 shaft horsepower the use of higher fuel prices will show appreciable savings for various steam conditions higher than 615 pounds per square inch gauge 850 degrees F. In each case the expected savings should be weighed against the probable risk. There appears to be more incentive to increase the temperature than to raise the pressure but the risk also may be greater.

ACKNOWLEDGMENT

This paper would not have been possible without the co-operation of the manufacturers of marine equipment, who furnished engineering and cost information. The authors wish to acknowledge their indebtedness to the following firms and their representatives: Allis Chalmers Manufacturing Company, The Babcock and Wilcox Company, Combustion Engineering Company, DeLaval Steam Turbine Company, Elliott Company, Foster Wheeler Corporation, General Electric Company, The Griscom-Russell Company, Ingersoll-Rand Company, Leslie Co., Northern Equipment Company, P. S. Thorsen and Company, Westinghouse Electric Corporation, and Worthington Pump and Machinery Corporation.

REFERENCES

[1] "A 1,200-Pound Reheat Marine Installation", by B. Fox and R. H. Tingey, *Transactions* of The Society of Naval Architects and Marine Engineers, 1941.

[2] "High-Pressure Steam for Marine Propulsion", by R. H. Tingey, *Transactions* of The Society of Naval Architects and Marine Engineers, 1943.

[3] "The Ore Carrier S.S. *Venore*". by H. F. Robinson and E. P. Worthen, *Transactions* of The Society of Naval Architects and Marine Engineers, 1945.

[4] "Improvements in Thermal Efficiencies with High Steam Pressures and Temperatures in Non-Reheating Plant", by K. Baumann, *Proceedings,* Institution of Mechanical Engineers, 1946, page 125.

[5] "Considerations Affecting the Improvements in the Thermal Efficiencies of Non-Reheating Plants at High Steam Pressures", by J. R. Finniecome, *Proceedings,* Institution of Mechanical Engineers, 1946, page 134.

[6] "A Modern Marine Power Plant", by A. W. Wood, Philadelphia Section of The Society of Naval Architects and Marine Engineers, December 19 1947. Also, *The Log,* February 1948, and *Marine Engineering and Shipping Review,* April 1948.

[7] "Considerations Governing the Choice of Live Steam Conditions in Condensing Turbo-Sets", *The Brown Boveri Review,* December 1946.

[8] "Power Plants for Postwar Vessels", by A. C. Rohn and W. E. Hammond, *Marine Engineering and Shipping Review,* Volume 50, 1945.

[9] "The Possibilities of Applying Improvements Effected in Modern Land Power Plant to Ship Propelling Machinery", by W. T. Bottomley, E. W. Corlett, and Frank Piercy, *Transactions,* North East Coast Institution of Engineers and Shipbuilders, January 1934.

[10] "A Thermal Study of Available Steam Power Plant Heat Cycles", by G. A. Hendrickson and S. T. Vesselovsky, *Transactions,* The American Society of Mechanical Engineers, 1934, Paper No. FSP-56-4, page 203.

[11] "Marine Engineering", Seward, Volume I, Chapter I.

[12] "Marine Engineering", Seward, Volume II, Chapter I.

[13] "Marine Engineering", Seward, Volume II, Chapter III.

[14] "New Diagrams for Determining the Most Economical Size of Condenser and the Most Favourable Cooling Water Quantity in Condensing Steam Turbine Plants", *The Brown Boveri Review,* August 1945.

[15] "The 2,000 psi,—1,050 F.—1,000 F. Reheat Cycle at The Philip Sporn and Twin Branch Steam Electric Stations", by Philip Sporn, paper presented at December 1947, annual meeting of The American Society of Mechanical Engineers.

[16] "Heat Power Engineering", Part I, by W. N. Barnard, F. O. Ellenwood, and C. F. Hirshfield, John Wiley and Sons, third edition, 1935.

[17] "The Steam Turbine Regenerative Cycle—An Analytical Approach", by J. K. Salisbury, *Transactions* of The American Society of Mechanical Engineers, April 1942.

[18] "The High-Pressure High-Temperature Turbine-Electric Steamship *J. W. Van Dyke*", by Lester M. Goldsmith, The American Society of Mechanical Engineers, December 1938.

APPENDIX 1

FUEL RATE CALCULATIONS

The basis for the majority of the fuel rate calculations made for this study is the "short form" shown in Table 11 with typical results.

The selection of feed temperature using formula (1) of the table already has been discussed. The next step is to calculate the energy required for feed heating as given by formula (2).

TABLE 11.—"SHORT FORM" HEAT BALANCE CALCULATIONS

Item	Units	Symbol	*C*				For Curves		
Ship total	—	*shp*	12,500	12,500	12,500	12,500	12,500	12,500	12,500
Superheater outlet pressure	psig	*P*	450	615	850	875	450	—	—
Superheater outlet temperature	*F*	*t*	750	850	850	900	850	900	950
Superheater outlet enthalpy	Btu/lb	*H*	1,386·4	1,434·0	1,424·6	1,452·1	1,440·5	1,467·2	1,49·93
Throttle pressure	psig	P_1	435	600	825	850	435	435	435
Throttle temperature	F	t_1	740	840	840	890	840	890	840
Throttle superheat	F	ts_1	284	351	316	363	384	434	484
Throttle enthalpy	Btu/lb	H_1	1,381·8	1,429·1	1,420·0	1,447·4	1,435·6	1,462·4	1,489·0
Condenser pressure	in. Hg	P_0	1·5	1·5	1·5	1·5	1·5	1·5	1·5
Condenser temperature	F	t_0	91·7	91·7	91·7	91·7	91·7	91·7	91·7
Condenser enthalpy	Btu/lb	H_0	59·7	59·7	59·7	59·7	59·7	59·7	59·7
Condenser entropy	Btu/lb.-F	S_0	0·1147	0·1147	0·1147	0·1147	0·1147	0·1147	0·1147
Superheater outlet, saturation enthalpy ...	Btu/lb	*Hf*	440·9	477·8	516·6	525·0	440·9	440·9	440·9
Heat added, condenser to saturation ...	Btu/lb	$Hf - H_0$	381·2	418·1	456·9	465·3	381·2	381·2	381·2
No. of stages of feed heating	—	*n*	3	4	4	4	3	3	3
Heat added, condenser to boiler inlet (1) ...	Btu/lb	$H_6 - H_0$	235·1	292·5	319·8	325·7	235·1	235·1	235·1
Feed inlet enthalpy	Btu/lb	H_6	294·8	352·2	379·5	385·4	294·8	294·8	294·8
Feed inlet temperature	F	t_6	324	379	404	410	324	324	324
Feed inlet entropy	Btu/lb.-F	S_6	0·4690	0·5400	0·5716	0·5783	0·4690	0·4690	0·4690
Heat added in boiler	Btu/lb	$H - H_6$	1,091·6	1,081·6	1,045·1	1,066·7	1,145·8	1,172·5	1,199·2
Available energy, Rankine cycle (throttle)	Btu/lb	h_1	478·9	523·3	536·2	553·8	508·9	524·5	540·6
Energy for extraction feed heating (2) ...	Btu/lb	*hf*	52·5	72·4	84·6	87·6	52·5	52·5	52·5
Available energy for extraction cycle ...	Btu/lb	$h_1 - hf$	426·4	450·9	451·6	466·4	456·3	471·9	488·0
Engine efficiency (turbines and gears) ...	—	*eT*	0·788	0·777	0·763	0·767	0·788	0·793	0·798
Net used energy (per lb throttle flow) (3) ...	Btu/lb	*hu*	331·7	350·3	344·6	357·7	359·6	374·2	389·4
Boiler efficiency	—	*eB*	0·875	0·875	0·875	0·875	0·875	0·875	0·875
Allowance for auxiliaries	—	*a*	0·0725	0·0798	0·0852	0·0908	0·076	0·081	0·087
Ship fuel rate (4)	lb/shp-hr	*R*	0·5547	0·5239	0·5172	0·5111	0·5387	0·5322	0·5260
Per cent gain over 450 psig 750 F ...	—	—	0	5·55	6·76	7·86	2·89	4·05	5·17
Total evaporation (5)	lb/hr	*W*	102,800	98,000	100,200	97,100	95,100	91,900	88,800
Exhaust heat flow ratio (6)	—	*Re*	0·723	0·698	0·672	0·685	0·744	0·754	0·764
Condenser heat removal (10') (7) ...	Btu/hr	*W'e*	69,300	63,400	62,100	61,000	65,800	64,000	62,400
Fuel rate, main unit only	lb/shp-hr	*R'*	0·5172	0·4852	0·4766	0·4686	0·5007	0·4924	0·4893
Per cent gain over 450 psig 750 F ...	—	—	0	6·19	7·85	9·40	3·19	4·79	6·44
Internal work (8)	Btu/lb	*hi*	405·0	442·0	444·7	461·7	435·9	452·1	468·9
Exhaust enthalpy	Btu/lb	$(He = H_1 - hi)$	976·8	987·1	975·3	985·7	999·7	1,010·3	1,020·1
Exhaust wetness	—	—	12·0	11·0	12·1	11·1	9·8	8·75	7·8
Engine efficiency at 435 psig, 740 F, 28·5 in. Hg	—	*eST*	0·778	—	—	—	—	—	—
Efficiency correction pressure	—	—	0	−1·01	−2·40	−2·54	0	0	0
Efficiency correction temperature	—	—	0	+0·85	+0·40	+1·01	+1·30	+1·93	+2·55
Efficiency correction vacuum	—	—	0	0	0	0	0	0	0
Efficiency correction net	—	—	0	−0·16	−2·00	−1·53	+1·30	+1·93	+2·55

Formulae :

(1) $H_6 - H_0 = \left(\frac{n-1}{n} - 0{\cdot}05\right)(Hf - H_0)$.

(2) $hf = \left(1 + \frac{1}{n}\right)[(H_6 - H_0) - (460 + t_0)(S_6 - S_0)]$.

(3) $hu = eT(h_1 - hf)$.

(4) $R = 2{,}544/18{,}500 \times (1 + a)(H - H_6)/eBhu$.

(5) $W = 2{,}544 \text{ shp } (1 + a)/hu$.

(6) $Re = \frac{He - Ho}{1{,}000} \times \frac{He - H_0}{(He - H_0) + (H_6 - H_0)} \times \frac{k}{1 + a}$. ($k$ at 6,000 shp = 1 004 ; $k_{12{,}000} = 0{\cdot}99$; $k_{20{,}000} = 0{\cdot}973$.)

(7) $WJe = WRe$.

(8) $hi = eTh_1/0{\cdot}92$.

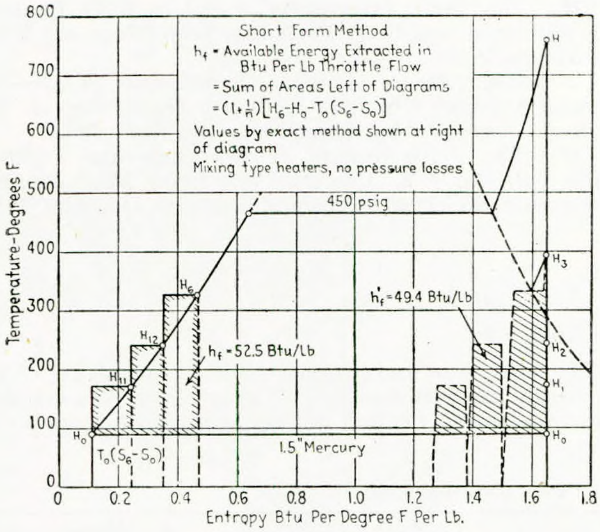

FIG. 18—*Temperature-entropy diagram to show energy used for extracting feed heating*

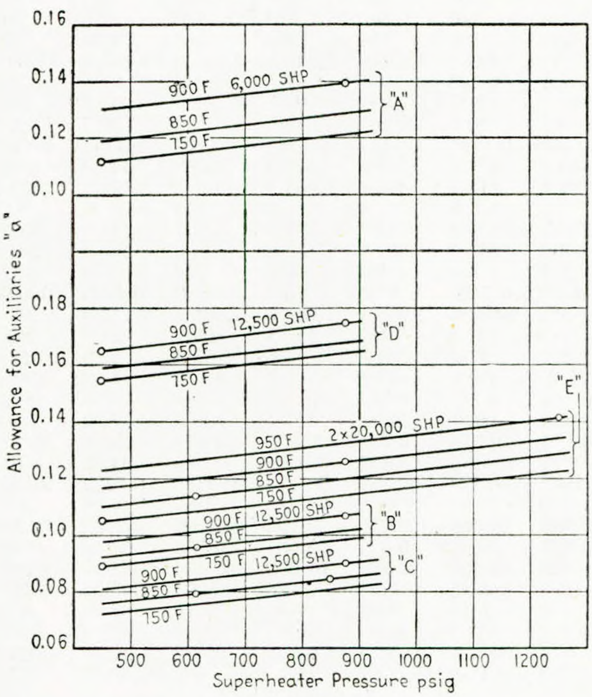

FIG. 19—*Allowance for auxiliaries "A" versus pressure for various steam temperatures*

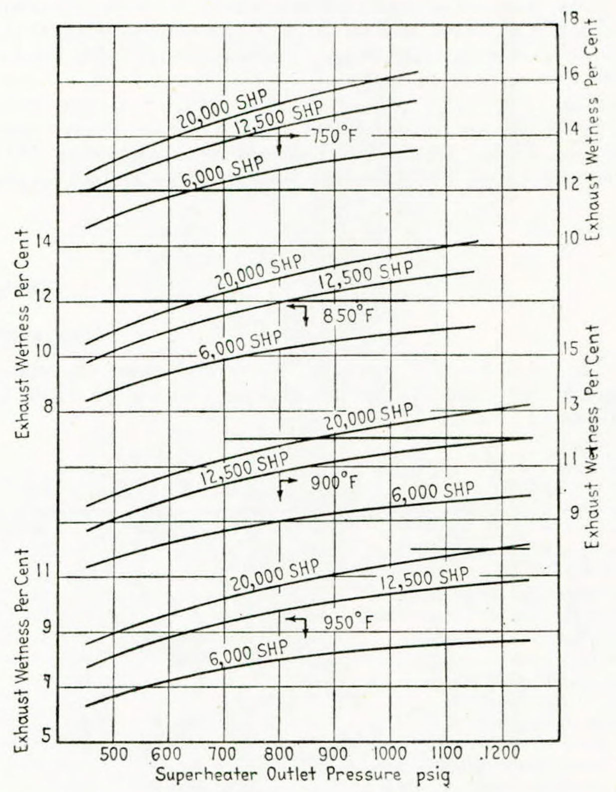

FIG. 20—*Per cent moisture in exhaust*

As indicated by the temperature-entropy cycle diagram in Fig. 18, this formula accounts for the energy subtracted from that available to the turbine on the straight Rankine cycle basis. The losses associated with a specific number of heaters are included but no account is taken at this point of pressure losses, temperature differences, or cascading losses.

It should be mentioned that this way of accounting for the available energy extracted for feed heating is not as exact as some methods which are available [16] [17]. The error as compared to the "exact" result was found to be consistent and is shown in Fig. 18 for a typical case. The short method was used for convenience, due to the large number of studies required to plot Figs. 2 and 3. The resulting error was absorbed in the allowance for auxiliaries along with many similar items, such as the heat recovered in the feed pump turbine exhaust, air ejector condensate and other drains, and the losses due to cascading drains from the higher pressure heaters.

The remaining energy available to the turbine is then multiplied by the "engine efficiency" of the turbine to obtain the net used energy per pound of throttle flow. The basis for determining the engine efficiencies is described in Appendix 2. The fuel rate for the main propulsion unit only is calculated directly from the net used energy and the heat supplied to the boilers, as shown by formula (4) [with $(1 + a)$ omitted].

The auxiliary steam requirements previously mentioned are lumped in the factor a which also includes the previously mentioned feed-heating system losses. The turbo-generator and feed-pump requirements were calculated from quoted performances applying to each particular study. The feed system losses were obtained from several detailed heat balance calculations covering the range of steam conditions and powers under consideration. Values of the factor a are shown in Fig. 19 as a function of the steam pressure. It will be noted that the increments reduce the gains for higher temperatures and the slope, which is essentially the same for all studies, affects the gains for higher pressure. The ship fuel rate is greater than the main unit rate by the factor a and the total evaporation follows directly from the ship fuel rate.

It was found that the turbine exhaust heat flow could be expressed as a fraction of the boiler evaporation as shown by formula (6). The small empirical correction noted was required to bring the result into agreement with detail heat balance calculations. The exhaust wetness is determined from the turbine internal work assuming 8 per cent external losses. It was not practical to plot exhaust wetness curves in Fig. 2, so these values have been shown in Fig. 20.

APPENDIX 2

ENGINE EFFICIENCY OF THE MAIN TURBINE

The standard engine efficiency under extracting operation at the basic steam conditions is shown in Fig. 21 as a function of the shaft horsepower per shaft. This curve was referred to the leading stationary and marine turbine manufacturers and found to represent closely the average practice of all concerns within the limits used herein.

Corrections to the basic engine efficiency for superheat, pressure, and vacuum are shown in Figs. 22, 23, and 24. Again the agreement between manufacturers was found to be very close. The total spread at the extreme conditions used here would not exceed 1 per cent.

The superheat correction actually takes into account only the change in moisture loss on the basis of about 1 per cent loss in stage efficiency for each per cent of average moisture in the turbine stages below the saturation line. The pressure correction takes into account the net increase of losses in the initial stages due to reduced volume flow at higher pressures. A very close approximation to the pressure correction curves can be obtained by assuming that the basic efficiency curve, Fig. 21, really represents the effect of reduced inlet volume flow. This basis for determining the engine efficiency implies that velocity ratios and exhaust losses are essentially constant for the various conditions studied.

The results obtained using these figures are in good agreement with the steam rates which have been quoted by the manufacturers for the various studies.

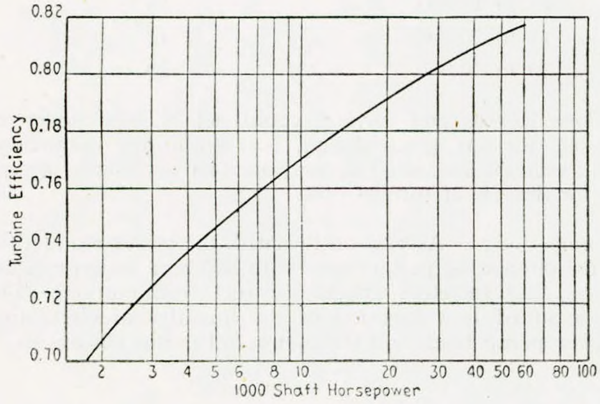

FIG. 21—*Efficiency of geared-turbines for various designed powers corrected to 450 pounds per square inch absolute, 740 degrees F., 28·5 inches mercury*

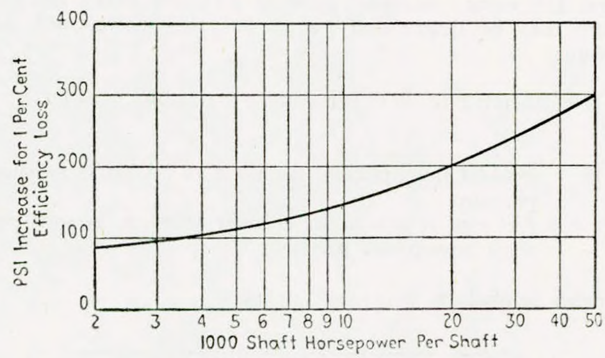

FIG. 23—*Pressure-efficiency correction*

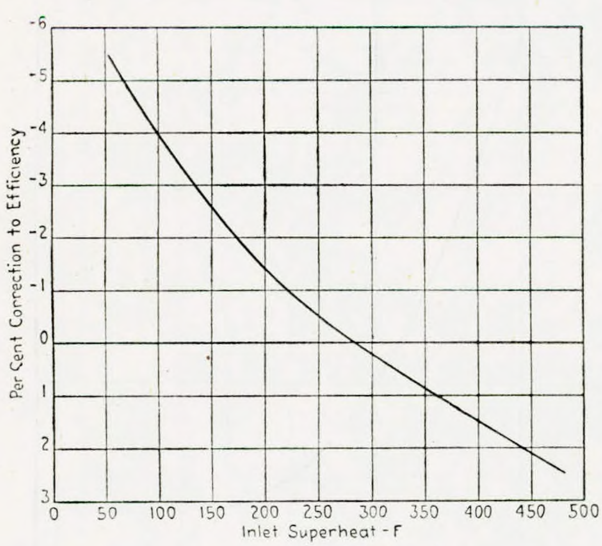

FIG. 22—*Superheat-efficiency correction. Basis: 450 pounds per square inch absolute, 740 degrees F., 284 degrees F. superheat, 28·5 inches mercury. Also good for pressures up to 1,500 pounds per square inch and approximately correct for any power*

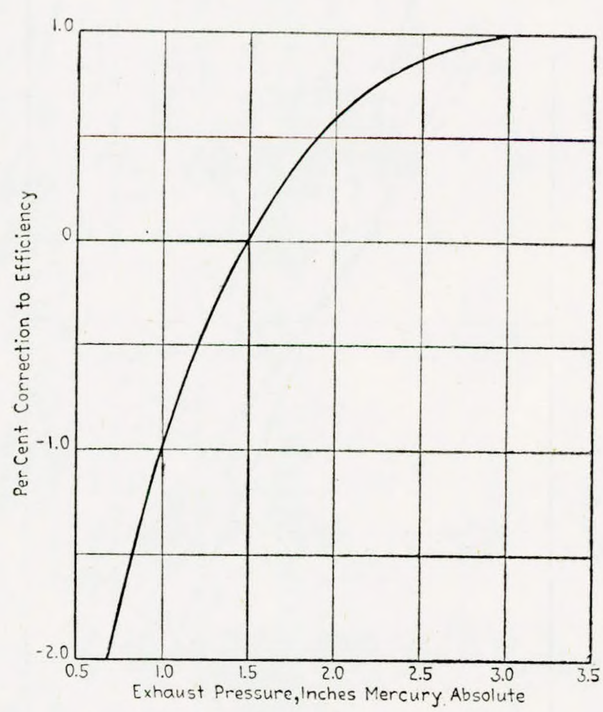

FIG. 24—*Per cent correction to turbine efficiency for different exhaust pressures*

APPENDIX 3

EXHAUST PRESSURE

Turbine Performance. Discussion with the turbine manufacturers indicated that there would be practically no change in the size of the exhaust end for different design vacua within a range of 1 to 4 inches of mercury absolute. It was possible, therefore, to calculate the changes in performance for various design vacua from the standard efficiencies and the pressure, temperature, and vacuum correction given in Appendix 2. This resulted in the non-bleed curve shown in Fig. 25. A check will show a slight variation in this curve for different inlet steam conditions and horsepowers, and therefore, when plotting, the points given the most weight were those for steam conditions appropriate for the size plant.

The curve corrected for bleeding provides for the change in output due to bleeding for heating the condensate from the condenser temperature to a uniform temperature, and also provides for the change in exhaust flow due to this bleeding.

For the main turbine, the change in fuel cost per year between 1·5 inches mercury absolute and any other exhaust pressure may be determined by use of this curve, and the following:

Cost difference (dollars per year) $= 0{\cdot}0258\, W_e{\cdot}w_e{\cdot}r{\cdot}f{\cdot}L$

where

W_e = exhaust flow, pounds per hour at 1·5 inches exhaust pressure

r = fuel rate at 1·5 inches exhaust pressure, pounds per shaft horsepower per hour

with other symbols as previously defined

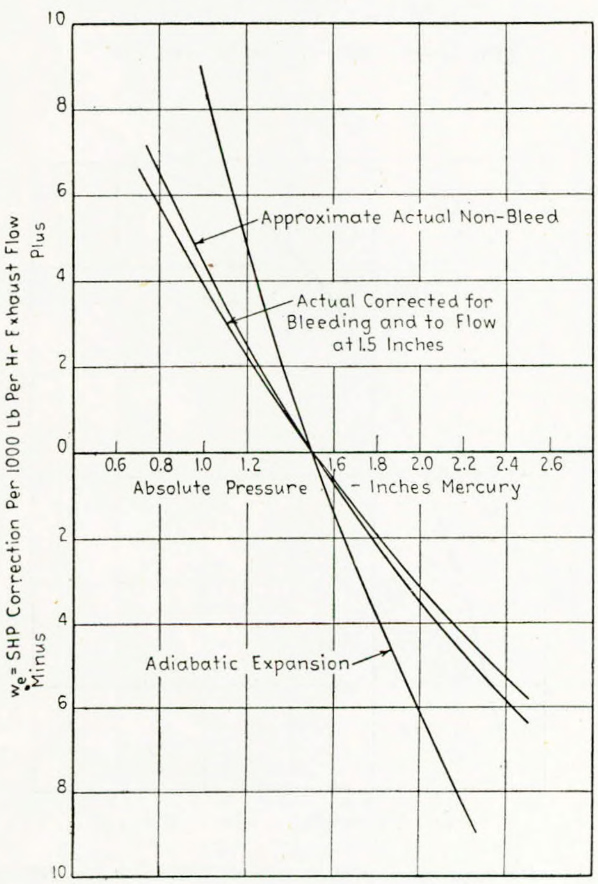

FIG. 25

Initial Cost. The change in initial cost of condenser plants, including those items previously mentioned, is shown by Fig. 26, and may be used as a basis for recalculation of economical exhaust pressure if material and labour costs should change, and also if different fixed charges are assumed. These costs were based on condensers designed in accordance with Fig. 27. The actual tube length used depended on the condenser size, as follows:

Condenser size, sq. ft.	Length between tube sheets, ft.
to 5,000	12
5 to 10,000	14
10 to 15,000	16
15 to 20,000	18
20 to 30,000	20

These lengths and surfaces could not be used under all conditions for any given design, but within the economical range of vacuum they result in condensers having lengths appropriate for the size of turbine unit.

Pump Power. As previously stated, the power required to drive the circulating pump varies with different design exhaust pressures, and therefore affects the fuel consumption. The power required is a function of the quantity of circulating water, the pump head, and the pump and motor efficiencies.

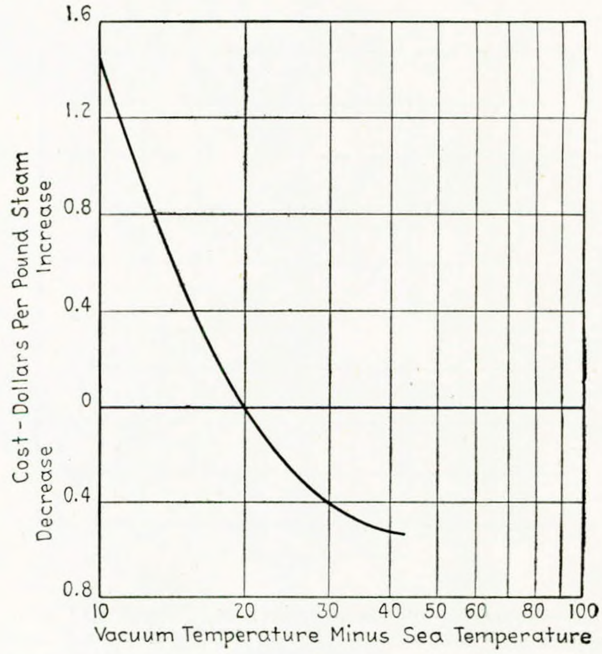

FIG. 26—*Change in condenser plant cost*

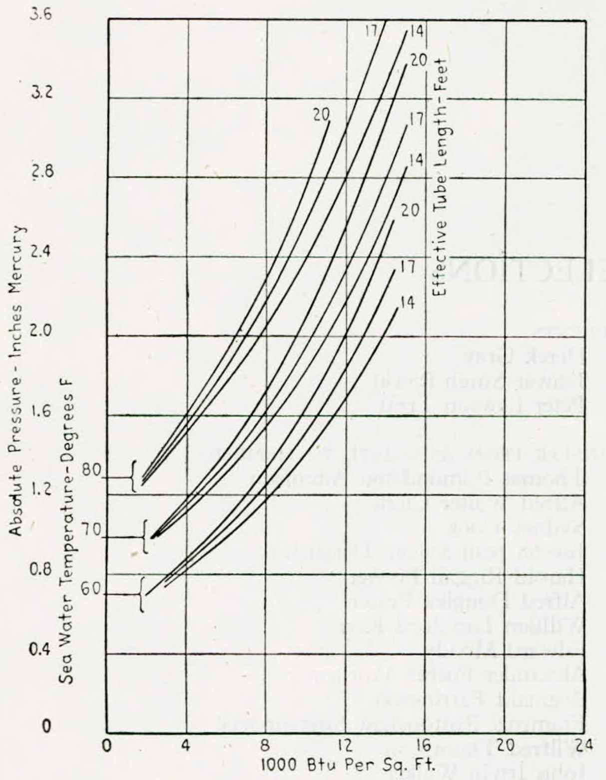

FIG. 27—*Condenser performance. Basis: velocity, 6·5 feet per second; tubes, 70-30 copper-nickel, ¾ inch 18 gauge; cleanliness factor, 85 per cent; two-pass*

The circulating water quantity, for the given condenser design conditions, is a function of tube length and surface, as follows:

$$GPM = \frac{17{\cdot}2 \times \text{square feet surface}}{\text{length between tube sheets, ft.}}$$

The pump pressure is the sum of the pressure losses in the condenser and piping. The condenser pressure loss was based on the Heat Exchange Institute Standard, Fig. 16 of reference [13]. The piping friction was estimated for a typical system and varied for all others by the usual pressure drop formulæ.

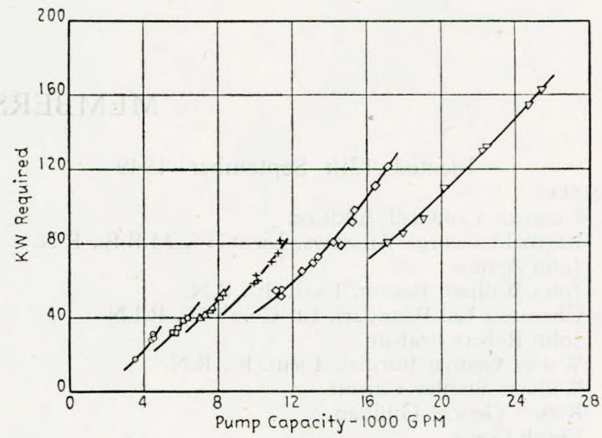

FIG. 28—*Circulating pump: power required for various designs*

The power required is shown by Fig. 28, for the range of designs used in the studies. The breaks are caused by the shift from one pipe size to another, and would not always occur at the same capacity due to differences in the basic piping system assumed.

The fuel cost per year due to the circulating pump power requirements may be found from the following:

$$\text{Fuel cost per year} = 22{\cdot}KW{\cdot}f{\cdot}L$$

where

KW = difference in power between base design (1·5 inches absolute) and any other design (the other factors are as defined previously). The constant in this formula is based on average values of generator set steam rate and appropriate steam conditions.

MEMBERSHIP ELECTIONS

Elected 13th September 1949

MEMBERS

Kenneth Crawhall Addison
Reginald George Andrews, Lieut.(E), M.B.E., R.N.
John Arthur
John William Batson, Lieut.(E), R.N.
Charanjit Lal Bhandari, Lt.-Com'r(E), R.I.N.
John Robert Braham
Walter George Burden, Lieut.(E), R.N.
William Stanley Carson
Robert George Gilfillan
Frank Green
Reginald Albert Sydney Vivian Huzzey
Thomas Wells Kirby
Henry Rawlinson Laycock, Com'r(E), O.B.E., R.D., R.N.R.
Robert Lindsay
James McDonald
Frederick Alexander Mincher
Alexander Morton
Thomas Forbes Russell
Clarence Reginald Peers Simkins
Joseph Lowe Smith
Walter John Twining
Bruce Anthony White
Adam Will

ASSOCIATE MEMBERS

Stewart Bellas
James Herbert Clugston
John Edward Levine
Leslie Ronald Morris

ASSOCIATES

John Dennis Ashton Bright
Donald Campbell, B.Sc.
George William Church
George Codd
Harold Frederick Cook
Archibald Thomas Cooper
John Francis Crane
Kenneth Harcourt England
Jack Greenwood
Eaon Edgar Henry Harron
Malcolm Francis Heslop
James Whyte McCowatt
Thomas Mooney
Reginald Thomas Moore
John Nelson
Reginald Wilfred Pigram
Dudley John Rees
Harry Sawkill

GRADUATES

George Edward Bowie
Edwin James Thomson Caie
John Hicks
Thomas Orr Leith
Donald Sinclair Weir Martin
Colin Francis Schneider

STUDENTS

Derek Gray
Ishwar Singh Rawat
Peter Lawson Trott

TRANSFER FROM ASSOCIATE TO MEMBER

Thomas Edmondston Aitchison
Alfred Walter Clark
Sydney Cook
Joseph Jean Sylvan Desjardins
Harold Riggall Fowler
Alfred Douglas Fraser
William Langford Kerr
Edward Moody
Alexander Forbes Murison
Reginald Partington
Framroze Ruttonshaw Singanporia
Wilfred Thompson
John Irwin Walker
John Norman Young

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBER

John Reginald Hobman
George Miller

TRANSFER FROM GRADUATE TO MEMBER

John Leslie Valentine Whittle

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER

Terence Edwin Hannan
Jan Boleslaw Krajewski
Robert Morrison
John Thomas Gosling Pereira, Lieut.(E), R.I.N.
James Sloan, M.Sc.
Shyam Uttamsingh, Lieut.(E), R.I.N.

TRANSFER FROM STUDENT TO ASSOCIATE

George Percy Reason

Elected 3rd October 1949

MEMBERS

Rustom Noshirwan Antia
Leslie Bewsher
Paul Brook
Henry Thomas Butt
George Alistair Campbell
Cecil Young Gilroy
William George Hawkins
Norman Irvin
William Henry Lambelle
Ernest Hugh Maddock
Waclaw Jan Trzebinski

ASSOCIATE MEMBERS

John Barclay Hill, B.Sc.
Ronald Stanley Paradise, B.Sc.

ASSOCIATES

Derek King Baguley

George Bruce
Raymond Carter
George Cato
George William Collins
William Harris Joslin
James Arthur Sanderson
Charles Herbert Stevens

GRADUATES
Provash Chandra Bhattacharjee
Leslie Ronald Hyett

STUDENTS
Allen Bellingham
Ishwar Singh Rawat, E.R.A., R.I.N.
Alexander Oswald Tucker

TRANSFER FROM ASSOCIATE MEMBER TO MEMBER
George Moir Christie

TRANSFER FROM ASSOCIATE TO MEMBER
James Stanley Mather
David Price

TRANSFER FROM ASSOCIATE TO ASSOCIATE MEMBERSHIP
Alexander Sim Allan
Charles Johnston

TRANSFER FROM GRADUATE TO ASSOCIATE MEMBER
Jagdish Mitter Bazaz, Lieut.(E), R.I.N.

TRANSFER FROM STUDENT TO GRADUATE
Edward Bowes

OBITUARY

JAMES ALLISON (Member 5925) was born in 1889 and served his apprenticeship with Fleming and Ferguson, Paisley. His first sea-going appointment in 1909 was as engineer in a bucket dredger built by Fleming and Ferguson and delivered to owners in Sydney. He obtained a shore appointment in Sydney and later sailed on the coast. He returned to the U.K. in 1911 and obtained his 2nd class certificate. In 1913 he went to China and served as an engineer with vessels of Butterfield and Swire and for a time also held a post in the naval dockyard in Hong Kong. In 1917 he returned again to the U.K. and joined the Inland Water Transport and was appointed engineer in train ferry running between England and France. He was demobilized in 1919 and obtained his first class certificate. In 1920 he joined the staff of the Asiatic Petroleum Co. and was assistant manager of the Bombay Office until 1923. He was elected a Member in 1928. From 1923-6 he was manager in Madras (Oil Installation) and from 1926-39 he was manager at Cochin (Oil Installation) retiring in 1939 owing to ill health. He died on the 26th January 1948 and leaves a widow.

JOSHUA MILLER BUCHANAN (Member 2302) was born in 1871 and served his apprenticeship with Messrs. Hutson and Son, Glasgow, and afterwards worked as journeyman fitter with the same firm. He later joined the staff of Messrs. Muir and Houston. He spent three years at sea as fourth and third engineer and obtained his first class B.O.T. certificate. In 1896 he was appointed assistant to Mr. George McFarlane, Superintending and Consulting Engineer and Naval Architect. In 1900 he was appointed to Lloyd's Register of Shipping as an engineer surveyor and served in London, Glasgow and Cardiff. In 1908 he was transferred to the United States and took up his duties in New York. In 1910 he was moved to New Orleans and in 1921 to Philadelphia where he served as a senior ship and engineer surveyor until his retirement from active service on the 31st December 1932. He was elected a Member in 1909. He died on the 4th January 1945.

WILLIAM GEORGE RUSSELL COATES (Local Vice-President and Member 8744) was born in 1904 at Gateshead-on-Tyne and received his professional training at the Rutherford Technical College, Newcastle, and with Clarke, Chapman and Co., Ltd., of Gateshead. On the completion of his apprenticeship he went to sea and served with British Tankers, the Federal Line and the King Line, and ultimately became a Chief Engineer. He held an Extra First Class Certificate. He was elected a Member of the Institute in 1938 and Local Vice-President for Hong Kong in September 1948. He was also a Member of the Institution of Naval Architects. On leaving the sea he spent some time in the drawing offices of Vickers, Ltd., Barrow, and with Insurance Engineers, Ltd., London, as Engineer Surveyor. Mobilized with 2nd Battery Corps Artillery, H.K.V.D.F., on 7th December 1941, he was imprisoned on the fall of the Colony at North Point and Shamshuipo. There his skill as an engineer in the adaptation of material for kitchen and other essential plant helped much in improving the lot of his fellow prisoners. From Shamshuipo, he was transferred to Nairumi Camp in Japan, and on his release in 1945 was repatriated. He returned to the Colony in the *Otranto* in June 1946. In February 1948 he proceeded home on leave and undertook an intensive course in radar, obtaining radar certificates. He returned to the Colony again as Acting Senior Engineer Surveyor (Ship Surveys) Marine Department and died suddenly at the age of forty-four at the Queen Mary Hospital on the 2nd June 1949. As an experienced and highly qualified engineer, Mr. Coates's death is a great loss to his brother officers in the Marine Department and to the shipping community with whom he had many intimate contacts. He is survived by a widow in the Colony.

JOHN THOMAS CORNEILLE (Member 6203) was born in 1872 and served his apprenticeship with Messrs. Latimer, Clark, Muirhead and Co., Ltd. He was for twenty-six years manager of the marine department of Messrs. Verity's, Ltd., designing electrical auxiliaries. In 1929 he went into business on his own account but later rejoined Messrs. Verity's Ltd., as technical director. He was elected a Member in 1929. He died on the 24th April 1949, in South Africa.

WILLIAM F. HICKS (Member 5165) was born in Liverpool, in 1893. He was educated at St. Laurence School and at Liverpool Technical College and served his apprenticeship with Messrs. Ellerman Lines, Ltd. He was elected a Member in 1924. He began his sea-going career in 1914 and served mostly with Messrs. Ellerman Lines, Ltd., but he was also employed by Messrs. Hain and Co., Louden Connell and Co., and the Booth

Line. He obtained his second class certificate in 1917 and his first class certificate in 1919. From 1921 to 1923 he took a shore appointment with Marine Motors and from 1923 to 1929 he was assistant engineer with Messrs. Lambert Brothers, London, coaling and salvage contractors. From 1929 to 1941 he served with Messrs. Hicks and Parkes, Liverpool, marine surveyors and consulting engineers. From 1941 to the time of his death on the 31st July 1949, at the age of fifty-six, he was Assistant Shipyard Labour Supply Officer for the Ministry of Labour and National Service, North Western Region.

H. G. How (Member 1858) was born in 1872 and served his apprenticeship with the L.N.E.R. at Straford. He joined the staff of the B.I.S.N. Co. and commenced his sea service with a voyage to America. He was later on the Bombay to Durban service. During the 1914-18 war he served on troopships plying between India and Italy and was torpedoed in the Bay of Naples in 1917. After returning to India on the Calcutta to Rangoon coastal run he went back to the Bombay to South Africa service, to which he was attached when he retired from the S.S. *Karagola.* Among his other ships were the *Avoca, Umbala* and the *Varsova.* He retired from the B.I.S.N. in May 1924 with the rank of Chief Engineer. He was elected a Member in 1925. He died on the 19th July 1947 aged seventy-five.

THOMAS HUGHES (Member 1786) was born in 1867 and was educated at Penpark National School and the University College of Wales, and later, at the Engineering College, Glasgow. He served his apprenticeship with Messrs. George Green and Sons, Aberystwyth, and commenced his sea-going career with a coasting firm transporting explosives. In 1889 he obtained his second class certificate and in 1892 obtained his first class certificate. He was elected a Member in 1905 and was also a member of the Free Mason's Lodge, Aberystwyth. He served with several Glasgow firms including the Strath Line and the Tower Line and finally joined the staff of the King Line for several years until his retirement in 1929. In 1934 he was elected member of the Aberystwyth Town Council. He died on the 5th April 1943, aged seventy-five, after a short illness and leaves a widow and three daughters.

JOHN ELEY KODE (Member 8942) was born in Durban in 1884 and educated at Durban High School. In 1889 he visited Norway and attended Bergen Technical High School, returning to Durban in 1901 and serving his apprenticeship with Messrs. James Brown. In 1907 he joined the Mercantile Marine and obtained his first class certificate in 1911 and returned to Durban in 1912 after a short period in the shipbuilding yards at Shields. In 1912 he gained an appointment as maintenance engineer at the Pietermaritzburg electrical power station and in 1918 he joined the staff of Pietermaritzburg Technical College as a lecturer in marine construction. In 1930 he was transferred as a lecturer to Glenwood High School, Durban. He was elected a Member in 1939. He retired in 1947 at the age of sixty-three having been retained for an extra three years service because of the war but during 1948 and the early part of 1949 his services were accepted by Howard College, University of Natal. He died on the 13th July 1949 aged sixty-five and leaves a widow.

D. S. POLLOCK (Member 1903) was born in 1877 and joined the staff of the P. and O. Steam Navigation Co., in 1899. After serving in various steamships as assistant engineer, 4th and 3rd engineer, he was promoted to 2nd engineer on the s.s. *Manila* in 1907. He had obtained his first class B.O.T. steam certificate a month earlier. He served as relieving chief engineer in the s.s. *Nubia* and *Palma* for coastal voyages in 1914 and was finally promoted to Chief Engineer in 1922. From this date until he retired on pension in 1931, he was chief engineer of s.s. *Nagpore, Plassy, Padua, Delta,* and *Kashmir,* remaining in the last named from 1925 until he retired. He was elected a Member in 1913. He died on the 13th August 1946 aged sixty-nine.

WILLIAM PORTEOUS (Member 2024) was born in London in 1872 and educated at Stoke Newington. He served his apprenticeship with Messrs. Appleby Brothers, Greenwich, and Messrs. Rait and Gardener, London, and commenced his sea-going career with the s.s. *Matatua.* In between his various voyages he also spent periods in shore appointments in Australia, New Zealand and Canada, returning to London in 1907. He saw service in France during the 1914-18 war attached to the Red Cross for repair work and in the 1939-45 war he was a Collector on behalf of the Red Cross for the Duke of Gloucester's Fund. He retired to Herne Bay and spent his last eleven years there, dying at the age of seventy-five on 14th March. He had first class B.O.T. certificates and was elected a Member in 1929.

CHARLES WALLACE SAUNDERS (Member 5575) was born in 1884 at Dunedin, New Zealand, and served his apprenticeship with Messrs. A. and T. Burt of the same town. He went to sea in 1906 and obtained his 1st class B.O.T. certificate in London in 1909. He then came ashore and studied for two years at the City and Guilds College, South Kensington, completing a three years course in two years, and thereafter he spent two years obtaining practical experience in various electrical works and at the Twickenham Power Station. In the 1914-18 war he enlisted in the British section of the New Zealand Armed Forces. After training he was sent to the N.Z. Expeditionary Force then assembling in Egypt and as a 1st Corporal in the 1st Field Company, Royal New Zealand Engineers, he took part in the epic landing on Gallipoli on April 25th 1915. He was awarded the D.C.M. for gallantry, granted a commission in the field, and later mentioned in dispatches. Early in 1916 he was invalided home, but returned to England before the end of the year, and in 1917 was seconded to the Explosives Department of the Ministry of Munitions. After demobilization in 1918 he joined the General Electric Co., Ltd., and formed their marine department, of which he remained manager until his retirement late in 1947. During that period he was associated with the building of the following electrically propelled ships: *Monarch of Bermuda, Queen of Bermuda,* and the four "Bel" type heavy-lift ships *Belocean, Empire Byng, Empire Marshall* and *Empire Wallace,* all with turbo-electric a.c. propelling machinery; H.M.S. *Adventure* (Diesel electric a.c.); *Cementkarrier, Loch Nevis,* Clyde Vehicular Ferry No. 4 and tugs *Acklam Cross* and *Sir Montagu* (Diesel electric d.c.) and also with the auxiliary installations of many more vessels. He was elected a Member in 1925 and was also a Member of the Institution of Naval Architects, and an Associate Member of the Institution of Mechanical Engineers and also of the Institution of Civil Engineers. In 1947 he was also appointed a member of the Technical Committee of Lloyd's Register, representing the British Electrical and Allied Manufacturers' Association. He retired to Jersey two years ago and died on the 21st September 1949 at the age of sixty-five.

W. E. SHARP (Member 1111) was born in 1860 and began his training at Newcastle-on-Tyne assisting in the building and trials of the first ocean going cable-laying ship, *The Great Eastern.* After a period in China he became chief advisory engineer to the newly created Siamese Navy and it was during trials at sea that he lost two fingers and an eye in an explosion. In addition to his other duties, he became advisory chief to the works and shipyards of the Bangkok Dock Co., and Messrs. Howarth and Erskine Co., at Bangkok and Singapore. He returned to England just prior to the 1914-18 war and founded in London a firm of engineers and shippers. As consultant for many interests, he rendered to his country valuable service, when, during the early critical days of the war, the merchant ships had to be war conditioned. He returned to settle down at River, Dover, and it was there that he died on the 20th March 1949, at the age of eighty-nine. He was elected a Member in 1928 and also was the oldest surviving Member of the Institution of Naval Architects and the Institution of Mechanical Engineers.