Fundamentals of Steam Turbine Thermodynamics*

W. L. COVENTRY, C.G.I.A., M.I.Mech.E., A.M.R.I.N.A. (Associate Member)†

INTRODUCTION

Since 1945 there has been a wealth of papers on steam turbine thermodynamics and design presented by British and American authors. The most remarkable of these is probably the Twenty-sixth Thomas Lowe Gray lecture "High Temperature Machinery for Marine Propulsion" delivered by Dr. T. W. F. Brown at the invitation of the Institution of Mechanical Engineers in 1945. This paper is notable, not only for its content, but for the detailed appendix on the behaviour of metals at elevated temperatures and the extensive catalogue of references and bibliography. Recently there have been several papers read before the Institute of Marine Engineers, principally on the subject of turbine cycles peculiar to marine propulsion and, pre-war, several papers on power station practice, which are classical both for their subject material and presentation.

All the above papers have been presented by engineers at the very top of their profession and acknowledgement has been made of the assistance of men who have, or eventually will, succeed them. Because of the very specialist nature of these papers, and the necessity to condense vast quantities of information within the compass of a very few pages, they have only been fully appreciated by the relatively few who enjoy day to day contact with the subject and then only after considerable concentration.

For many engineering subjects, standard series studies have been prepared which, by their fundamental simplicity and extent, have sought to provide a secure foundation upon which more detailed arguments may be developed. To a turbine designer such basic studies are an everyday mode of thought, almost a second nature, and it is therefore surprising that so elementary a survey of steam turbine thermodynamics has not been previously published.

It is the purpose of this paper to attempt such a fundamental survey, based upon the elementary laws of thermodynamics, with external imponderables reduced to an absolute minimum compatible with reliability and unrestricted by peculiar practice and opinion.

During the past forty years the progress in steam turbine thermodynamics is indicated by advance in the initial conditions of steam pressure and temperature, the exploitation of feed heating and resuperheating, and the steady reduction of specific fuel consumption. Typical advances in power station and marine propulsion practice are illustrated in Table I.

The progress in land power station practice has been remarkably steady except for the wartime and emergency period 1940-1950, when little outward advance was displayed in the construction of larger units. The fact is that, in 1940, turbines of 60,000 kW. capacity were the largest which could normally be economically employed by local power authorities and during the emergency period 1940-1950, when power demand threatened to outstrip supply, only drastic national action in standardizing new equipment resolved the desperate situation. Units of 30,000 and 60,000 kW. capacity were chosen as standard. Demand having been temporarily satisfied, and with the simultaneous consolidation of the Central Electricity Generating Board, larger units have been feasible and advances have since been rapid. Although British advances in pressure have not been so spectacular as on the Continent or in America, a consistent lead in higher temperature has been maintained.

Advances, pre-war, in marine propulsion practice did not keep pace with those on land, principally because of the adherence to the Scotch type boiler, which limited steam pressure to 240lb./sq. in. gauge. The shattering impact of the American wartime cargo vessel and tanker, fitted with medium pressure watertube boilers, caused the greatest revolution ever experienced in the British Merchant Service. Pre-war, the author cannot recall a single British ocean-going cargo vessel fitted with watertube boilers, which was not influenced by strong North American opinion. It is thought provoking to consider what would have been the present situation, when coal is no longer available as a maritime fuel, had it not been for this profusion of successful American vessels.

SCOPE OF THE STUDY

A review of notes, compiled over many years, of studies of the many steam cycle variables upon cycle efficiency, using initial pressure and temperature conditions which appeared significant at the time, indicated that a standardized series of conditions would need to be considered. Accordingly standard series conditions at combinations of 500, 700, 900lb./sq. in. abs. and 600, 800, 1,000 deg. F. are adopted. Whilst some of these combinations of pressure and temperature do not conform with established practice (see Fig. 17), they do afford a useful basis for comparison.

Variables used in the several studies are detailed below:

1 a) Variation of simple cycle efficiency (η_c) with varying initial pressures and temperatures, 200-1,200lb./sq. in. abs. and corresponding saturation temperatures to 1,000 deg. F. respectively. Constant exhaust back pressure of 28½in. Hg.

b) Variation of simple cycle internal efficiency (η_{ci}) (dry and wet) with various initial steam pressures and temperatures 500, 700, 900lb./sq. in. abs. and corresponding saturation temperature to 1,000 deg. F. respectively. Exhaust back pressures constant, 28½in. Hg.

2 a) Variation of simple cycle efficiency (η_c) with varying exhaust back pressure 27½in. Hg—29½in. Hg. The initial steam pressure and temperatures being 500, 700, 900lb./sq. in. abs. and 600, 800, 1,000 deg. F. respectively.

b) Variation of simple cycle internal efficiency (η_{ci}) (dry and wet) with varying exhaust back pressure 27½in. Hg.—29½in. Hg. The initial pressures and temperatures being 500, 700, 900lb./sq. in. abs. and 800 deg. F. and 1,000 deg. F. respectively.

3 a) Variation of single gas reheat cycle efficiency (η_o) with varying intercept pressures up to initial pressure,

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† Senior Mechanical Engineer, Messrs. Balfour, Beatty and Co. Ltd.

reheating always to initial temperature. The initial steam pressures and temperatures being 500, 700, 900lb./sq. in. abs. and 800 and 1,000 deg. F. respectively. Exhaust back pressure 28½in. Hg.

b) Variation of single gas reheat cycle internal efficiency (η_{ci}) (dry and wet) with varying intercept pressures up to initial pressure, reheating always to initial temperature. The initial steam pressures and temperatures being 500, 700, 900lb./sq. in. abs. and 800 and 1,000 deg. F. respectively. Exhaust back pressure 28½in. Hg.

4 a) Variation of single gas reheat cycle efficiency (η_c) with varying reheat temperatures, intercept saturation temperature to 800 deg. F. with initial steam pressure and temperatures of 700lb./sq. in. abs. and 800 deg. F. respectively. Intercept pressure 150lb./sq. in. abs. Exhaust back pressure 28½in. Hg.

b) Variation of single gas reheat cycle internal efficiency (η_{ci}) (dry and wet) with varying reheat temperatures, intercept saturation temperature to 800 deg. F. with initial steam pressure and temperature of 700lb./sq. in. abs. and 800 deg. F. respectively. Intercept pressure 150lb./sq. in. abs. Exhaust back pressure 28½in. Hg.

5 a) Variation of regenerating feed heating cycle efficiency (η_c) with varying final feed temperature from condensate temperature to boiler saturation temperature. Number of heater stages 1, 2, 3, 4 and infinite. The initial steam pressure and temperatures being 500lb./sq. in. abs./800 deg. F. and 900lb./sq. in. abs/1,000 deg. F. respectively. Exhaust back pressure 28½in. Hg.

b) Variation of two-stage regenerative feed heating cycle internal efficiency (η_{ci}) (dry and wet) with final feed temperatures varying from condensate temperatures to boiler saturation temperature. The initial steam pressures and temperatures being 500lb./sq. in. abs./800 deg. F. and 900lb./sq. in. abs./1,000 deg. F. respectively. Exhaust back pressure 28½in. Hg.

METHOD AND BASIC ASSUMPTIONS

Simple cycle efficiencies are calculated for adiabatic* expansion of steam from initial conditions of pressure and temperature to final exhaust pressure. In the case of the reheat cycle, adiabatic expansion is from initial pressure and temperature to intercept pressure and from intercept pressure and reheat temperature to final exhaust pressure. The heat gained by the steam in the boiler (except in regenerative feed heating cycle) is the difference between total heat of steam at initial conditions of pressure and temperature and the total heat of condensate at final exhaust pressure. For the reheat study, the heat of resuperheat must be included, and for the regenerative feed heat cycle, the liquid heat supplied in feed heaters must be deducted.

Particular is the simple cycle, without reheating or feed heating, which is the Rankine Cycle, whence $\eta_c = \eta_r$.

Internal cycle efficiencies are calculated using steam condition lines drawn on the total heat entropy chart by the step by step method, employing 10 stages and assuming constant stage efficiencies of 0·8. Internal cycle wet efficiencies are obtained by deducting the product of the mean steam wetness and the internal heat drop in the wet region from the total internal heat drop. This approximates to the long established practice due to Baumann, of allowing a reduction in stage efficiency of 1 per cent for each 1 per cent wetness of steam in that particular stage. No allowance is made for super-saturation and no correction is made to the condition curve on account of wetness. A point of historical interest is that super-saturation corrections are made to a line, approximating* to that of 0·94 dryness, on the Φ – I chart, known as the "Wilson Line" after the late C. T. R. Wilson, who investigated the super-saturation of water vapour and employed this knowledge in devising the "cloud chamber" which first enabled, in 1912, the tracks of alpha particles (the helium nucleus) to be photographed.

No external losses due to radiation or leakage have been allowed. The steam conditions at turbine stop valves have been considered to be identical to those at superheater outlet. The condensate temperature, equal to the exhaust saturation temperature, has been considered as the boiler feed temperature in non-feed heating cycles. Reheat cycle intercept pressures have always been chosen in the superheat zone; no reheater pressure loss has been allowed. Many first stage and several second stage heater steam bleed pressures are subatmospheric and, whilst this condition may be most undesirable in practice, the conventions of thermodynamics have been observed, in so much as the bled-steam saturation temperature is always equal to the heater outlet temperature.

TABLE I—STEAM TURBINE DEVELOPNT.

LAND POWER STATION

Year	Initial steam conditions		Output kW.	Coal rate lb./kW. hr.
	pressure lb./sq. in. gauge	temperature deg F.		
1920	250	650	25,000	2·2
1930	450	750	30,000	1·6
1940	600	850	60,000	1·2
1950	900	900	60,000	1·0
1955	1,500	975	100,000	0.9
1960	1,500	1,050/1,050	275,000	0.85
Building	2,300	1,050/1,050	550,000	0·80?

MARINE PROPULSION (Cargo vessels)

Year	Initial steam conditions		s.h.p.	Oil rate lb./h.p. hr.
	pressure lb./sq. in. gauge	temperature deg. F.		
1920	200	500	3,000	0·900
1930	220	550	4,500	0·880
1940	240	600	6,000	0·790
1945	450	750	8,000	0·600
1950	600	850	12,000	0·520
1955	650	950	18,000	0·495

ACCURACY OF PROGRAMMING OF CALCULATIONS

The accuracy of this study is governed by the use of the 1939 Callendar Steam Tables reading B.t.u. to the nearest whole number, the Edward Arnold half size Total Heat Entropy chart (1 cm. – 20 B.t.u. – 0·02 units of entropy) and a 25 cm. slide rule.

Cycle efficiencies (adiabatic expansion) are wholly calculated from steam tables. At least seven, and sometimes nine, points per curve are used and any deviation of a point by

* The term adiabatic, as used in this paper with reference to expansion of steam means an expansion with no external heat loss (adiabatic) and at constant entropy (isentropic). All isentropic processes are also adiabatic but certain adiabatic processes are not also isentropic; for example, the throttling of steam at constant total heat is an adiabatic process, but as the entropy is thereby increased it is not therefore isentropic.

Modern teaching of thermodynamics terms a process at constant entropy and, *ipso facto*, with no external heat loss as isentropic. However, as most marine engineers are more accustomed to the more familiar, if less correct, term adiabatic in this connexion, consequently this latter term has been used throughout the paper.

It is noteworthy that, if somewhat confusing at first contact, the term "adiabatic furnace temperature" is commonly used, in boiler design practice, to denote the theoretical maximum furnace temperature, which would be attained, assuming no heat losses by conduction, convection, and radiation and with no dissociation during perfect combustion. Whilst combustion is indisputably a thermal process it is not, as an isolated process, thermodynamic.

* More nearly the "Wilson Line" corresponds to 0·98 dryness at 1·82 entropy value and 0·96 dryness at 1·70 entropy value.

more than 0·002 from a fair curve has been investigated. This gives an error of plus or minus 0·8 per cent, which is greater than that of slide rule calculations periodically checked with seven figure logarithms.

Cycle internal efficiencies are determined by measurement, checked when possible against steam table values, from plotted condition curves using 10 expansion stages. The consistency of the many condition curves, verified by graphing corresponding reheat factors, are reasonably fair and moreover several condition curves which are necessarily duplicated prove agreeable within 2 per cent. Detailed checking reveals no point deviation by more than 0·002 from a fair curve, thus giving a probable error of 1 per cent. As previously, seven to nine points are plotted per curve.

In all, over 550 principal calculations of efficiency for various conditions of initial pressure and temperature, exhaust back pressure, reheat intercept pressure and temperature and final feed temperature are involved, several of which are necessarily duplicated. The resolution of such a large number of calculations is only possible by careful programming such as would be applied to a digital computer. The extensive use of tabular calculations minimizes error and facilitates rechecking.

Some series of curves are complimentary. For example, curves of efficiency plotted to a temperature base for various pressures are the cross curves of efficiency plotted to a pressure base. Several series of curves follow the same pattern, thus any wholesale divergence can be detected.

PRESENTATION OF RESULTS

Cycle efficiency is plotted as ordinate to a standard scale (originally 0·02 cycle efficiency = 1in.) with the variable under consideration — initial steam temperature, intercept pressure, final feed temperature, etc., as abscissae.

The practice of many investigators is to plot "per cent increment of cycle efficiency" against the relevant variable and, whilst this method of presentation has definite advantages, especially for the detailed economic study of steam turbine cycles, it is considered that the fundamental principles tend to be obscured.

An increase of cycle efficiency for an already highly efficient cycle shows a lesser percentage increase than would an identical increase for a low efficiency cycle, and thus the smaller percentage increase may be compared unfavourably with the larger. The law of diminishing returns applies generally to steam turbine cycle design and the former smaller percentage increase may well represent a more creditable, although perhaps more expensive, achievement than the latter larger gain.

For these reasons, in this fundamental theoretical study only true efficiencies are considered.

BASIC THERMODYNAMICS

It is familiar that the most highly efficient thermodynamic cycle is the Carnot Cycle, a reversible cycle, which is purely theoretical in conception and whose efficiency is given by: $\eta_{carnot} = \frac{T_1 - T_2}{T_1}$ where T_1 and T_2 are the upper and lower limiting absolute temperatures respectively.

Fig. 1 shows the variation of Carnot Efficiency with upper temperature for two conditions of fixed lower temperature viz. 90 deg. F. (corresponding to 28½in. Hg.) and 212 deg. F. (corresponding to 1 atmosphere) respectively. It will be observed that with a higher temperature of 1,050 deg. F. (the higher steam temperature at present used in power station practice) the highest attainable efficiency for this best theoretical cycle is 63·5 per cent. By way of contrast, the best steam turbine overall efficiency, including auxiliaries, on a kW. output basis is 32 per cent.

The theoretical cycle of comparison for the steam engine is the Rankine Cycle which assumes adiabatic expansion of steam from initial conditions of pressure and temperature to final exhaust pressure. The heat added in the boiler being equal to the total heat of steam at initial conditions of pressure and temperature less the total heat of condensate at the temperature corresponding to the exhaust pressure. The Rankine Cycle is a simple cycle, that is, without feed heating and resuperheating. It is also, except with certain modifications, applicable in theory alone, irreversible and therefore is not so efficient as the Carnot Cycle.

Modified to include feed heating and for resuperheating and assuming adiabatic expansion, a theoretical cycle efficiency will exceed the Rankine Cycle efficiency. In fact with fully regenerative feed heating, using an infinite number of heater stages, raising the feed temperature to initial steam saturated temperature, the theoretical cycle efficiency becomes equal to that of the Carnot Cycle, provided that initial steam conditions are dry saturated.

In practice, adiabatic expansion of steam in the turbine

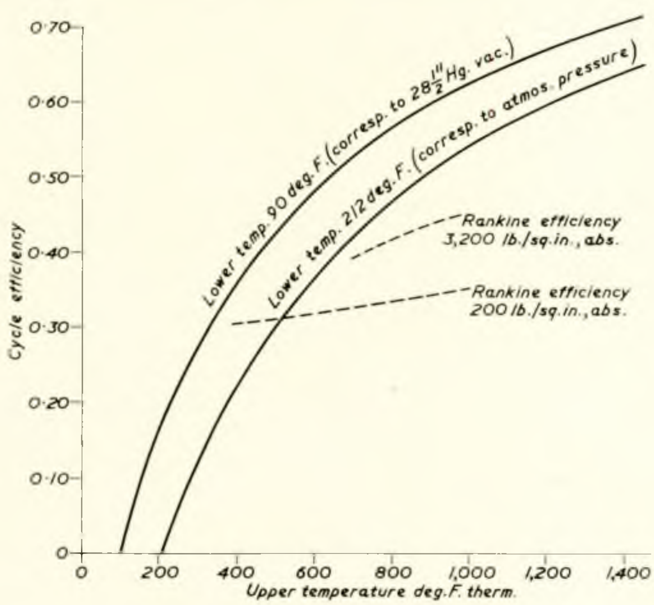


FIG. 1—*Variation of Carnot efficiency*

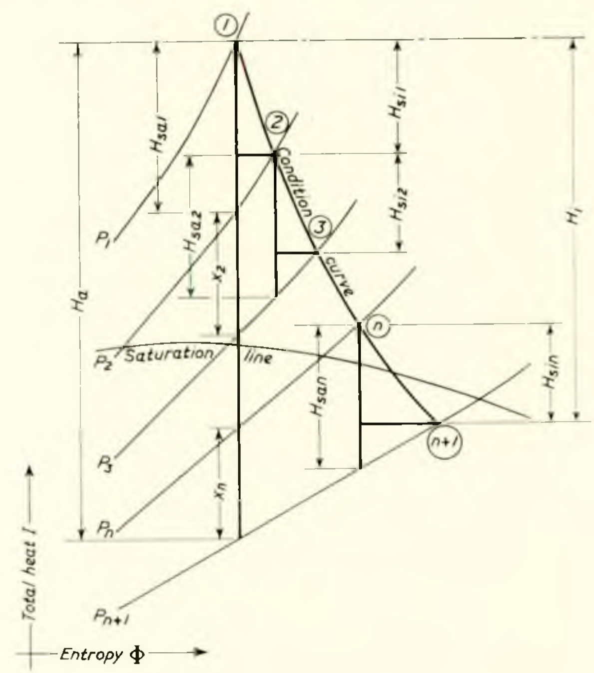


FIG. 2—*Condition curve—illustrating Table II*

TABLE II.

H_a = turbine adiabatic heat drop from initial pressure "p_1" to exhaust "p_{n+1}"
H_i = „ internal „ „ „ „ „ „ „ „
H_{sa_1}; H_{sa_2}; ... H_{san} = stage adiabatic heat drop for each of "n" number of stages.
H_{si_1}; H_{si_2}; ... H_{sin} = „ internal „ „ „ „ „ „ „ „ „

(i) Work done on rotor per stage $= H_{si_1};\ H_{si_2};\ \ldots\ H_{sin}$.

(ii) Losses per stage $\left[\begin{array}{l}\text{nozzle}\\ \text{velocity}\\ \text{leakage, etc.}\end{array}\right] = (H_{sa_1} - H_{si_1});\ (H_{sa_2} - H_{si_2});\ \ldots\ (H_{san} - H_{sin})$.

(iii) Total work done on turbine rotor $= \Sigma(H_{si_1} + H_{si_2} + \ldots H_{sin}) = H_i$.

(iv) Cumulative adiabatic heat drop $= \Sigma(H_{sa_1} + H_{sa_2} + \ldots H_{san}) = H_a \times R$.

(v) Total internal turbine losses $=$ (iv)—(iii) $= (H_a \times R - H_i.)$

(vi) Reheat Factor $= R$ $= \dfrac{\Sigma(H_{sa_1} + H_{sa_2} + \ldots H_{san})}{H_a} = 1{\cdot}03$—$1{\cdot}09$ in practice.

stage reheat factor $= H_{sa_2}/x_2 \ \ldots\ H_{san}/x_n$

(vii) Stage internal efficiency $= \dfrac{H_{si_1}}{H_{sa_1}};\ \dfrac{H_{si_2}}{H_{sa_2}};\ \ldots\ \dfrac{H_{sin}}{H_{san}} = {\cdot}69$—${\cdot}825$ in practice.

(viii) Turbine internal efficiency (mean stage efficiency η_{sm}) $= \dfrac{\Sigma(H_{si_1} + H_{si_2} + \ldots H_{sin})}{\Sigma(H_{sa_1} + H_{sa_2} + \ldots H_{san})} = {\cdot}69$—${\cdot}825$ in practice.
$= \dfrac{H_i}{H_a \times R}$ ... from (iii) / ... from (iv)

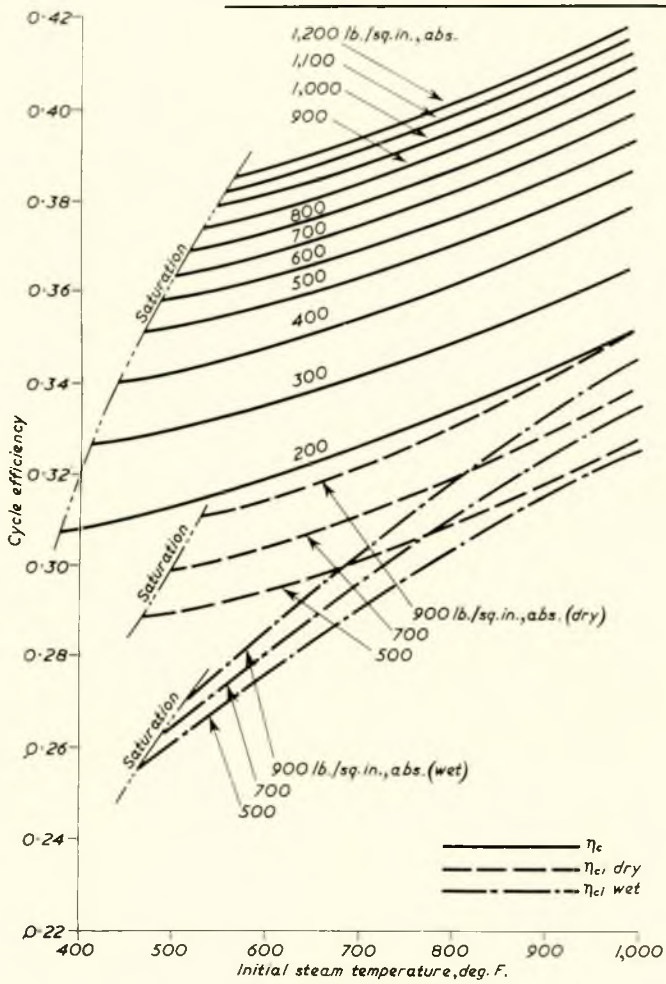
(ix) Turbine internal adiabatic efficiency (loosely termed, Turbine efficiency) $= \eta_i = \dfrac{H_i}{H_d} = {\cdot}75$—${\cdot}85$ in practice.

(x) $= \eta_i = \eta_{sm} \times R$ ... from (viii) and (ix).

Let H_b be heat to steam generation gained in boiler.
Then:—

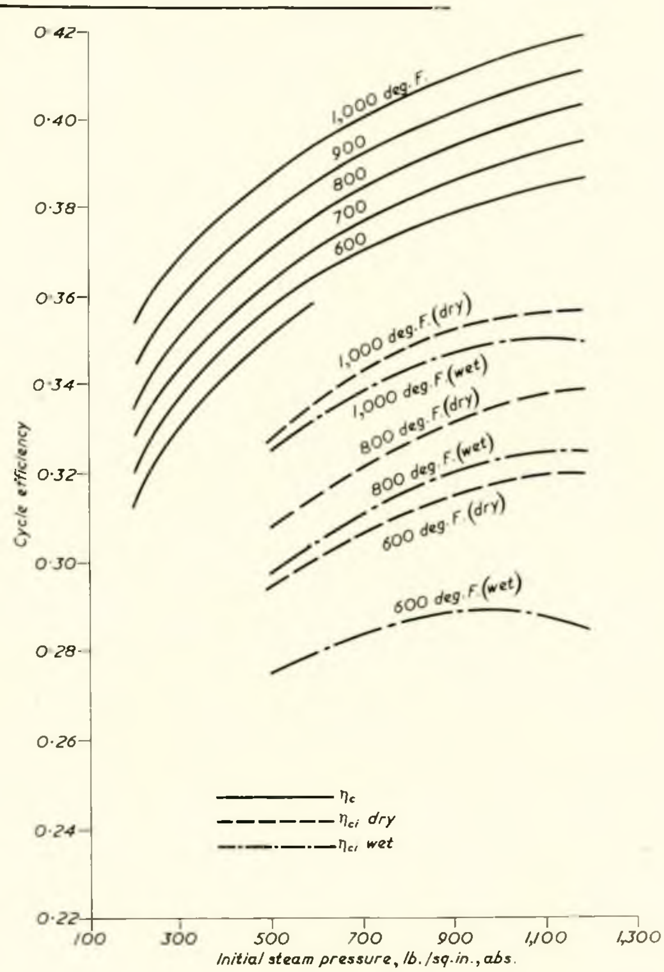
(xi) Cycle efficiency, η_c $= \dfrac{H_a}{H_b}$ = Rankine efficiency for simple cycle.

(xii) Cycle internal efficiency $= \dfrac{H_i}{H_b} = \eta_{ci}$.



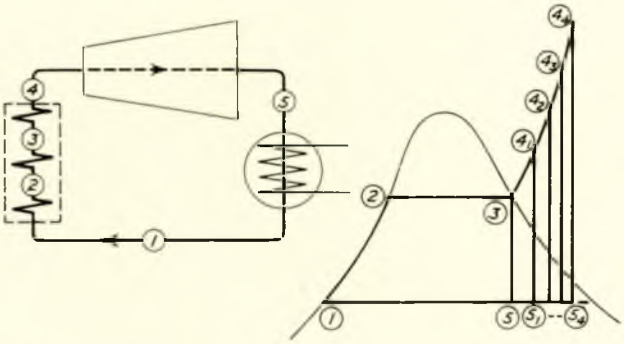
Stage efficiency 0·80
Exhaust vacuum 28½in. Hg

FIG. 3—*Variation of simple cycle efficiency with initial steam temperature*



Stage efficiency 0·80
Exhaust vacuum 28½in. Hg

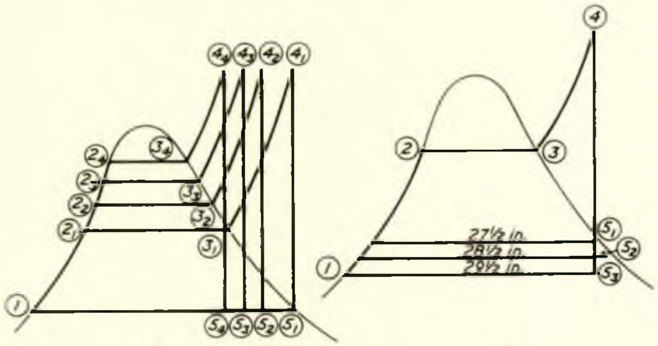
FIG. 4—*Variation of simple cycle efficiency with intial steam pressure*



(a)

$$\eta_C = \frac{(I'_4 - I_{W_1}) - (I_{S_5} - I_{W_1})}{(I'_4 - I_{W_1})} = \frac{I'_4 - I_{S_5}}{I'_4 - I_{W_1}}$$

(b)
Variation of cycle efficiency with initial steam temperature



(c)
Variation of cycle efficiency with initial steam pressure

(d)
Variation of cycle efficiency with exhaust back pressure

FIG. 5—*Simple cycle, variation of initial steam pressure, initial steam temperature and exhaust back pressure*

is not obtained because of frictional losses and the efficiency of the Rankine Cycle can never be equalled by the actual internal cycle efficiency, even when adopting such artifices as feed heating and/or resuperheating.

Table II and accompanying Fig. 2 define the common terms used in steam turbine thermodynamics.

a) The Influence of Initial Pressure and Temperature

Figs. 3 and 4 show the variation of cycle efficiency (η_c) and cycle internal efficiency (η_{ci} dry and wet) for simple cycles at varying initial steam temperatures and pressures for various constant initial steam pressures and temperatures respectively with constant exhaust back pressure.

The simple cycle and the temperature-entropy diagrams are shown in Fig. 5.

It will be observed (Fig. 3) that the cycle efficiency at the various constant initial pressure, increases more rapidly at higher temperatures in apparent contradiction to the law of diminishing returns. It will also be observed (Fig. 4) that the cycle efficiency, at the various constant initial temperatures, increases less rapidly at higher pressures.

At ultra-high temperatures, when steam behaves more nearly as a perfect gas, and the latent heat of generation becomes insignificant compared with superheat, the Rankine Cycle efficiency curves of Fig. 3 increasingly diverge from those of the Carnot Cycle. The reason is that the bulk of the heat supplied is under conditions of varying increasing temperature in the Rankine Cycle rather than at constant temperature (isothermal conditions) as in the Carnot Cycle. Likewise at high pressure, above the critical pressure, heat is supplied, in the Rankine Cycle, wholly under conditions of varying increasing temperature.

The variation of cycle internal dry efficiency follows closely the pattern of that of the cycle efficiency, but at much lower values due to extra heat loss to condenser because of non-adiabatic expansion.

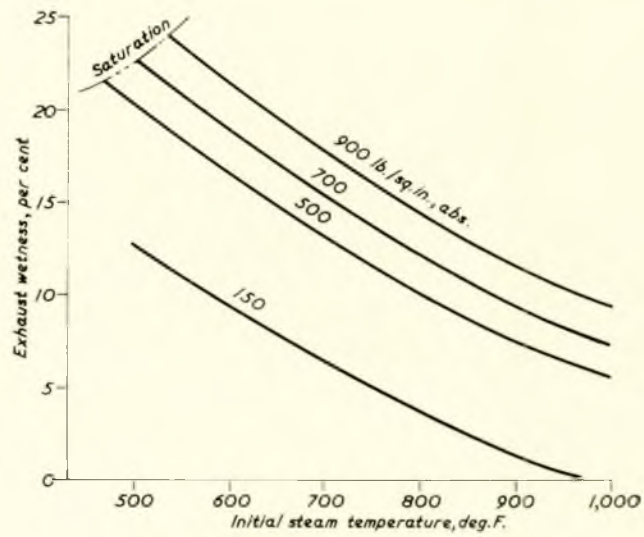
The cycle internal wet efficiency curves follow their own peculiar pattern. Greater efficiency losses at lower temperatures are apparent due to increased wetness of the exhaust steam. From Fig. 3 the cycle internal efficiencies, dry and wet, at 500lb./sq. in. abs. and 1,000 deg. F. are nearly equal because the exhaust for this particular initial condition is almost dry. For initial temperatures above 1,000 deg. F. the exhaust eventually becomes dry whence there is no efficiency loss due to wetness.

Fig. 6 gives the exhaust wetness resulting from various initial conditions of pressure and temperature. If a practical limit of say 12½ per cent exhaust wetness is not to be exceeded, then the minimum initial steam temperatures associated with initial steam pressures of 500 and 1,000lb./sq. in. abs. are approximately 750 and 1,000 deg. F. respectively.

There is ample evidence that, for any given initial steam temperature, there is a certain initial pressure at which the internal cycle wet efficiency becomes a maximum. Any increase of pressure above this optimum value results in the wetness correction loss exceeding the thermodynamic gain, with consequent decrease of efficiency. This is shown in Fig. 4.

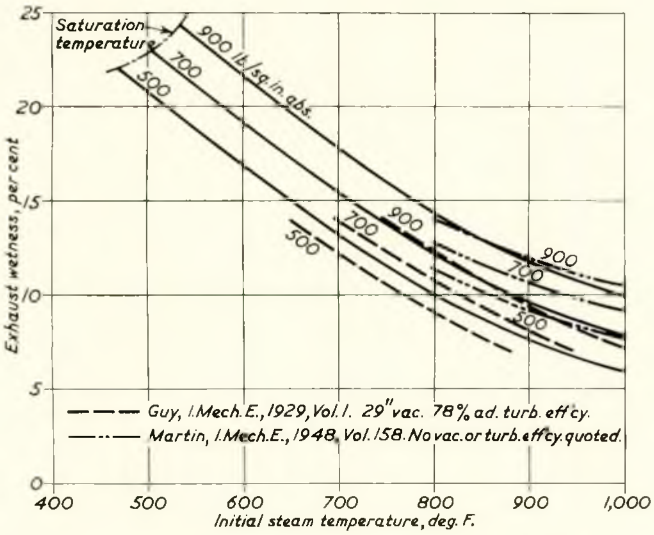
Although the simple cycle is rarely used in practice, a study of it enables the effect of initial temperature and pressure and later, exhaust vacuum to be observed, completely uninfluenced by other considerations. Summarizing, the lessons to be learned from the simple cycle internal wet efficiency curves, which represent steam expansion conditions obtaining in turbine practice are:

1) That advances of initial temperature result in greater thermodynamic gains than advances of initial pressure, whilst simultaneously reducing wetness losses in the low pressure stages.
2) For a given initial steam temperature there is a definite initial pressure which results in maximum efficiency, beyond which optimum pressure the efficiency falls, due to increasing wetness loss. In practice, the maximum exhaust wetness, which can be tolerated for mechanical reasons, limits the initial pressure which can be associated with any given initial temperature.



Stage efficiency 0·80
Exhaust vacuum 28½in. Hg

FIG. 6—*Variation of exhaust wetness with initial pressure and temperature*



Adiabatic turbine efficiency 0·80 vacuum 28½ deg. F.

FIG. 7—*Variation of exhaust wetness with initial steam pressure and temperature*

Fig. 7 is Fig. 6 with the results of two eminent investigators added thereto and which support the validity of the author's calculations. The higher exhaust wetness, for given equal initial pressures and temperatures, associated with the results of Martin[1], when impulse-reaction steam turbines were considered, over those of Guy[2] for purely impulse turbines, do not suggest any possible superiority of one type over the other. Rather does it reflect the greater blading efficiency achieved during the intervening years, resulting in greater turbine internal heat drops, H_i (see Table II), which approach more closely the theoretical ideal adiabatic heat drops, with consequent greater exhaust wetness.

b) *The Effect of Exhaust Vacuum*

Fig. 8 shows the effect of varying exhaust pressures upon cycle efficiency (η_c) and cycle internal efficiency (η_{ci}), dry and wet, for various combinations of initial pressure and temperature.

The simple cycle and temperature entropy diagrams are shewn in Fig. 5.

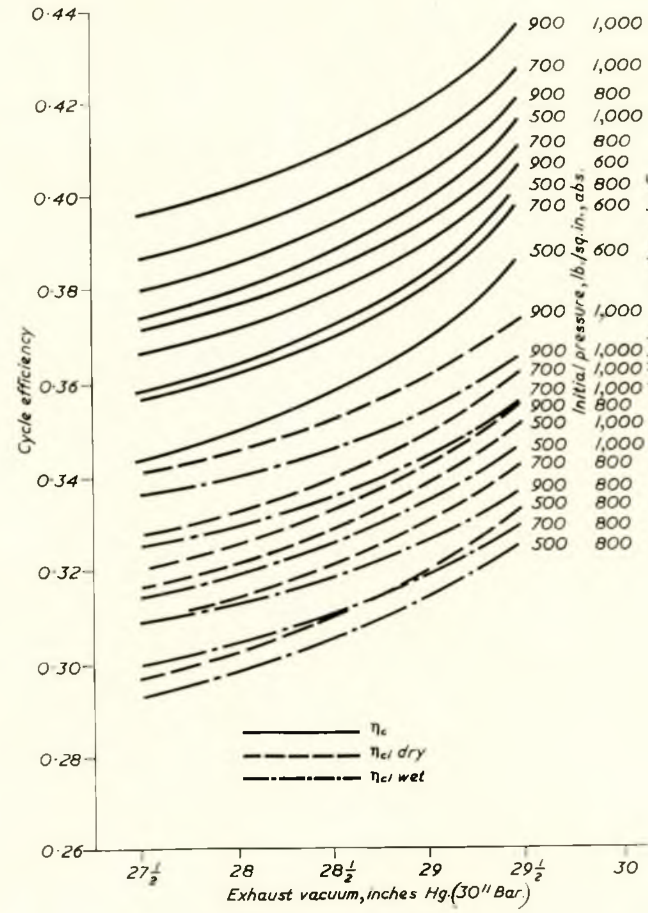
It will be observed that the cycle efficiency increases rapidly with exhaust vacuum, i.e. inversely as the saturation temperature corresponding to the exhaust back pressure. This is in accord with the Carnot Cycle efficiency variation, which, if plotted to a base of lower temperature for a given constant upper temperature, assumes a hyperbolic form, i.e. the curve of an inverse function.

The cycle internal efficiency (dry) curve is less steep than that of the cycle efficiency curve, which is explained by the fact that entropy, and consequently the amount of heat rejected to the condenser, increase more rapidly with vacuum.

Even less steep is the cycle internal efficiency (wet) curve, which is further influenced by the greater wetness, and therefore wetness correction loss, at high vacuum.

For initial steam conditions 700lb./sq. in. abs./800 deg. F., exhaust vacuum 28½in. Hg., it will be observed, from Figs. 8 and 4, that an increase in exhaust vacuum of 1in. from 28½in. to 29½in. Hg., produces an approximately equal cycle internal wet efficiency increment to that resulting from an advance in initial steam temperature of 150 deg. F. from 800 deg. F. to 950 deg. F. It will also be noticed that the improvement in efficiency, resulting in an increase of vacuum from 27½in. to 28½in. Hg., is only just over half that produced by an increase from 28½in. to 29½in. Hg.

The maximum vacuum obtainable in practice is limited



Stage efficiency 0·80

FIG. 8—*Variation of cycle efficiency with exhaust vacuum*

by the cooling water, the temperature of which, at inlet, requires to be some 15 deg. to 20 deg. F. less than the temperature corresponding to the condenser pressure. The smaller temperature difference is associated with higher performance condensing apparatus, the additional cost of which must be considered, together with the temperature of cooling water available, when deciding the maximum economical vacuum to be maintained.

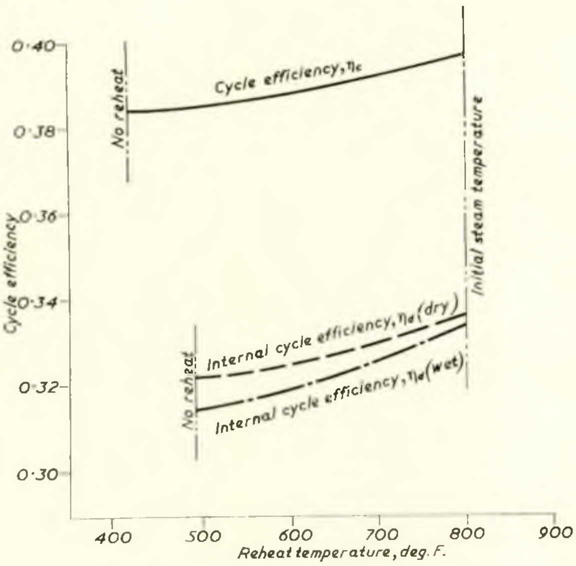
In marine practice a vacuum of 29in. Hg and over is only possible when in latitudes in excess of 40 deg. North and South. At the equator, with sea temperature just over 80 deg. F., only 28in. to 28¼in. Hg can be maintained.

Estuarine power stations have advantage over those situated inland, which may be obliged to cool the condenser circulating water in atmospheric evaporation cooling towers. The anomaly may then obtain that an inland station be overloaded during warm, humid summer weather, due to loss of power occasioned by the inability to maintain the designed vacuum.

It is also apparent that the tube or diaphragm vacuum gauge and mercury manometer, which depend on atmospheric pressure, are entirely unreliable for accurately measuring vacuum. Atmospheric pressure may vary from 2½in. below to 1½in. above that of the standard atmosphere of 30in. Hg (more correctly 76 cm. Hg) and engine room pressures may be further depressed by as much as ¼in. Hg. The only true method of directly determining the condenser pressure is by means of a Toricellian mercury tube, which is independent of atmospheric pressure.

c) and *d*) *Resuperheating*

Resuperheating, or more colloquially, "reheating", is the artifice of wholly intercepting the steam during its passage



Initial steam pressure 700lb./sq. in. abs.
Initial steam temperature 800 deg. F.
Intercept pressure 150lb./sq. in. abs.
Exhaust vacuum 28½in. Hg
Stage efficiency 0·80

FIG. 9—*Variation of gas reheat cycle efficiency with reheat temperature*

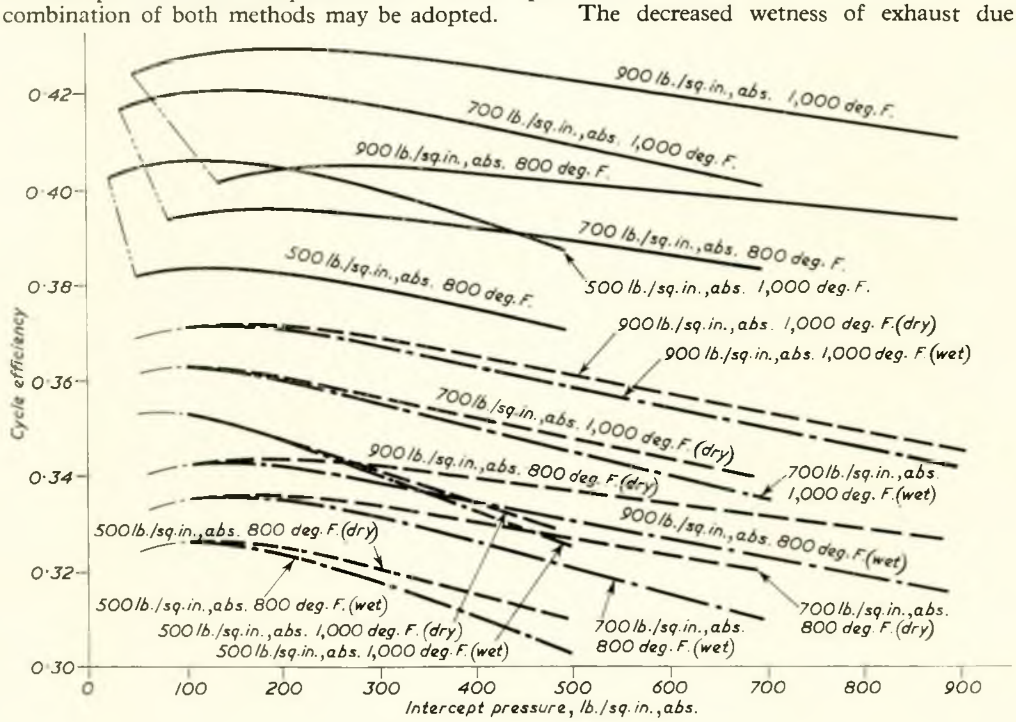
through the turbine, at a point before the dry saturated state is attained, and resuperheating to a temperature equal, or nearly equal, to the initial steam temperature, and then returning to the turbine for final expansion through the lower pressure stages. The original, precise reasons for reheating are obscure, but the ultimate object then, as now, was the improvement of cycle efficiency. Reheating by flue gas in the boiler, or by steam at initial pressure and temperature in a heat exchanger, or a combination of both methods may be adopted. The former method is known as gas reheating; the latter as steam to steam reheating.

Gas Reheating: Gas reheating was first proposed by S. Z. de Ferranti about 1910, the first practical application was the River Don Steelworks followed by the North Tees Power Station plant in 1920. The first marine propulsion application was the *Beaver* class of turbo-electric vessels in 1946, followed by the *Empress of Britain* class in 1956. These latter vessels have three boilers, one of which is arranged for controlled reheating, and it is interesting to note that a similar scheme was adopted for the Powerton Station, Illinois, in 1927.

Fig. 9 shows the variation of cycle efficiency (η_c) and cycle internal efficiency (η_{ci}), dry and wet, with reheat temperature, at constant intercept of 150lb./sq. in. abs. for initial steam conditions 700lb./sq. in. abs./800 deg. F. The cycle efficiency (η_c) rises slowly but increasingly with reheat temperature, but the maximum improvement in efficiency is only 3·4 per cent (thermodynamic gain). The improvement of cycle internal efficiency (wet) due, both to increased thermodynamic gain and increased exhaust dryness, is 7 per cent, which is substantial. It will be noted that, for reheat temperatures above 800 deg. F., the wet cycle internal efficiency curve corresponds with that of the dry cycle internal efficiency when the exhaust becomes dry. It is apparent from Fig. 9 that, for maximum gain, reheating should be carried up to the initial steam temperature.

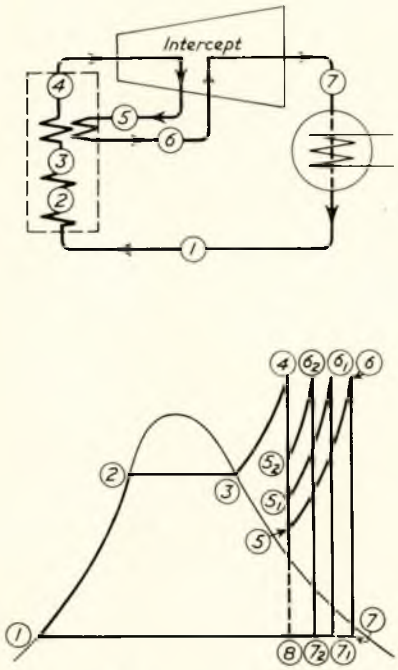
Fig. 10 shows the variation of cycle efficiencies with intercept pressure, reheating always to initial steam temperature, for various combinations of initial pressure and temperature. Fig. 11 shows the simple gas reheat cycle and temperature entropy diagram. Commenting briefly on these curves, which deserve a detailed study, it is apparent that the optimum intercept pressure is between 23-27 per cent of the initial pressure value; that there is a fairly wide range of intercept pressure over which cycle efficiency is sensibly constant and that these optimum intercept pressures are in the superheat zone. It is also apparent that, for any given initial temperature, the gains of reheating are more pronounced with higher initial pressures.

The decreased wetness of exhaust due to reheating at



Stage efficiency 0·80
Exhaust vacuum 28½in. Hg
Reheat always to initial steam temperature

FIG. 10—*Variation gas reheat cycle efficiency with intercept pressure*



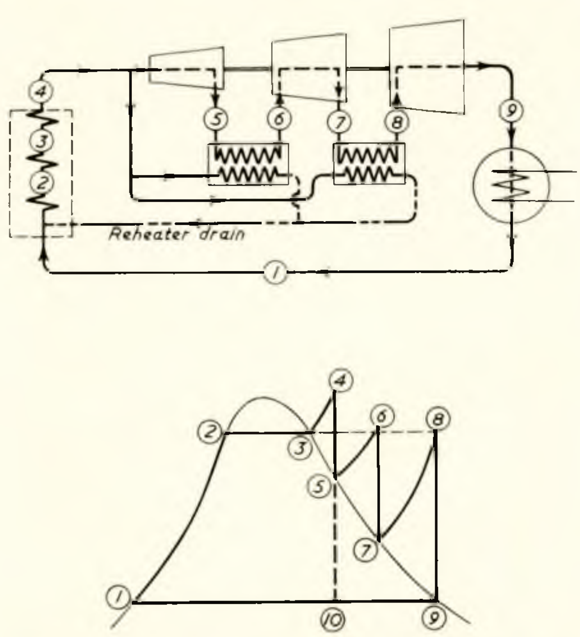
$$\eta_c = \frac{(I^1_4 - I^1_5) + (I^1_6 - I_{s7})}{(I^1_4 - I_{w1}) + (I^1_6 - I^1_5)}$$

FIG. 11—*Variation of cycle efficiency with intercept pressure for gas reheat cycle*

an intercept pressure of 150lb./sq. in. abs. is shown in Fig. 6 and, in practice, exhaust wetnesses of less than five per cent are obtained. There is no point in reheating to give a highly superheated exhaust, as heat is only wasted to the condenser.

Comparative figures for the gains of gas reheat cycle, over a simple cycle having same initial pressure and temperature, are given in Table III.

Steam to Steam Reheating: The first detailed proposal for steam to steam reheating appears that of Dr. H. L. Guy[2]



Initial steam conditions 1,500lb./sq. in. gauge 750 deg. F.

FIG. 12—*Double steam/steam reheat cycle (simplified) as used in* Venore *class vessels*

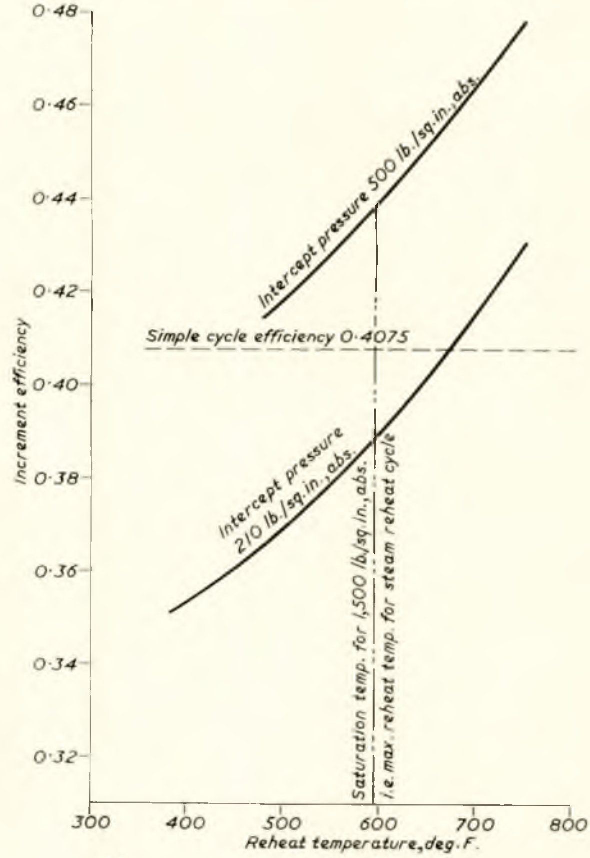
TABLE III.—IMPROVEMENT ON SIMPLE CYCLE GAINED BY REHEATING TO INITIAL TEMPERATURE.

Initial Steam conditions	700 lb./sq. in. abs. 800 deg. F.
Per cent increase of cycle efficiency (η_c)	3·4
Per cent increase of cycle internal efficiency (η_{ci}) dry	5
Per cent increase of cycle internal efficiency (η_{ci}) wet	7·7
Per cent reduction of steam rate for equal power	18
Per cent reduction of heat to condenser for equal power	12·1
Per cent improvement of exhaust dryness	9·2

but no application seems to have been made to power station practice except as an adjunct to gas reheating. The only application to marine propulsion is the American ore carrying *Venore* class vessels, 1945, in order to obtain a low fuel consumption with an initial steam pressure of 1,500lb./sq. in. gauge, whilst limiting the temperature to 750 deg. F., thus permitting the exclusive use of plain carbon steel.

The simplified *Venore* steam to steam reheat cycle and temperature entropy diagrams are shown in Fig. 12.

The reheat temperature of *Venore* is slightly less than that corresponding to 1,500lb./sq. in. gauge, 590 deg. F. approximately, as the reheating steam is condensed and returned to the boiler by means of separate pumps. Fig. 13 shows the reheat increment efficiency, with adiabatic expansion, for an intercept pressure of 210lb./sq. in. abs., from which it will



Initial steam pressure 1,500lb./sq. in. abs.
Initial steam temperature 750 deg. F.
Exhaust vacuum 28½in. Hg
Adiabatic expansion

FIG. 13—*Variation of reheat increment efficiency with reheat temperature*

be observed that a reheat temperature of 680 deg. F. must be attained before the reheat increment efficiency is equal to that of the simple cycle. Thus, at only 590 deg. F. reheat, the *Venore* cycle shows a thermodynamic loss. However, this is a double reheat cycle, such that the steam remains superheated throughout expansion, and the absence of any wetness loss is reflected in a 9·7 per cent gain of steam reheat cycle internal efficiency (which is dry) over a corresponding simple cycle internal efficiency (wet). This is a very creditable achievement.

Reheating Summarized: The advantage of reheating, in addition to improved cycle efficiency due to purely thermodynamic gains, are:

a) For a given limiting maximum initial steam temperature, the thermodynamic and constructional advantages of higher initial pressure may be exploited, without incurring cycle loss and mechanical erosion due to excessive steam wetness in the lower pressure stages.

b) For a given physical size of turbine, a greater output may be obtained, because of lower steam rate due to increased heat drops per pound of steam. Also, the size of condenser and auxiliaries are reduced with consequent increase of overall efficiency and reduction in first costs. In fact, for the largest of units reheating becomes a practical necessity.

Recently the technical press* described a 300,000 s.h.p. turbo-electric proposal, for a mammoth liner, based on current land power station practice, having initial steam conditions 1,500lb./sq. in./1,000 deg. F.; 1,000 deg. F. reheat and a phenomonally low propulsion fuel rate of less than 0·38lb. oil/s.h.p./hr.

The engineers of counter-proposals†, with orthodox reduction geared turbines were compelled to adopt the reheat cycle, having initial steam conditions 1,000lb./sq. in./1,000 deg. F.,; 1,000 deg. F. reheat, in order to approach the turbo-electric proposal. This fact speaks for itself.

e) Multi-stage Feed Heating

Feed water heating was first proposed by James Weir in 1876 and multi-stage feed heating by Ferranti in 1906, but it was some years before the latter proposal was adopted commercially.

Regenerative feed heating, strictly speaking, implies progressive heating in an infinite number of stages by means of steam, bled from the turbine, the temperature of which equals the feed temperature at that stage. The simple cycle, with initially dry saturated steam, and arranged for fully regenerative feed heating corresponds exactly to the Carnot Cycle. Truly regenerative feed heating is only a theoretical concept, as a finite number of heaters, two to seven in practice, are used.

Fig. 14 shows the simplified three stage feed heating cycle and temperature entropy diagrams.

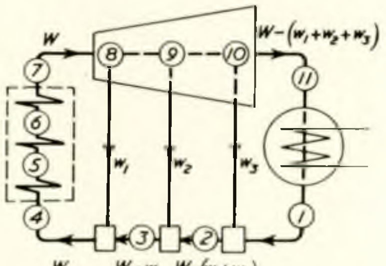
Fig. 15 shows the variation of cycle efficiency (η_c), with final feed temperature using various heater stages. It will be observed that, for a finite number of heaters, there is an optimum final feed temperature, which produces maximum cycle efficiency, to exceed which temperature only results in a reduction of efficiency. With an infinite number of heaters it would be possible to feed heat up to boiler pressure saturation temperature and produce maximum efficiency. It is also apparent that with an increasing number of stages the higher will be the optimum final feed temperature and that as each successive stage is added the improvement in efficiency becomes progressively smaller.

Shown also is the variation of cycle internal efficiency (η_{ci}) dry and wet, for two stage feed heating. It will be observed that these curves follow the same pattern as those above, but as much lower values and the optimum final feed temperature for non-adiabatic expansion is but a few degrees higher than that for adiabatic expansion.

The total quantities of bled steam, at optimum conditions, may be considerable and amount to approximately 15 per cent of the total steam supply with single stage feed heating to 25 per cent with four stage feed heating. Although the

* Marine Engineer and Naval Architect. April 1960; Vol. 83, p. 146.

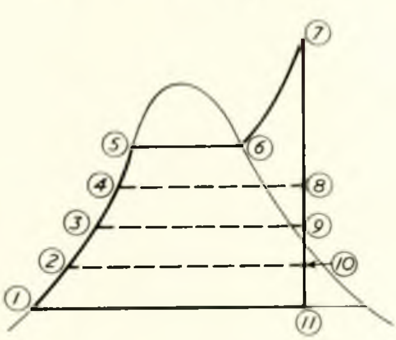
† Marine Engineer and Naval Architect. June 1960; Vol. 83, p. 233.



Final stage

$$I_{w_4} = w_1 I'_8 + (1 - w_1)\, I_{w_3}$$

$$\therefore w_1 = \frac{I_{w_4} - I_{w_3}}{I'_8 - I_{w_3}}$$



Second stage

$$(1 - w_1)\, I_{w_3} = w_2 I'_9 + (1 - w_1 - w_2)\, I_{w_2}$$

$$\therefore w_2 = \frac{(1 - w_1)(I_{w_3} - I_{w_2})}{(I'_9 - I_{w_2})}$$

First stage

$$(1 - w_1 - w_2)\, I_{w_2} = w_3 I_{s_{10}} + (1 - w_1 - w_2 - w_3)\, I_{w^1}$$

$$\therefore w_3 = \frac{(1 - w_1 - w_2)\,(I_{w_2} - I_{w_1})}{(I_{s_{10}} - I_{w_1})}$$

$$\eta_c = \frac{(I'_7 - I'_8) + (1 - w_1)(I'_8 - I'_9) + (1 - w_1 - w_2)(I'_9 - I_{s_{10}}) + (1 - w_1 - w_2 - w_3)(I_{s_{10}} - I_{s_{11}})}{(I'_7 - I_{w_4})}$$

FIG. 14—*Cycle efficiency η_c with three stage feed heating*

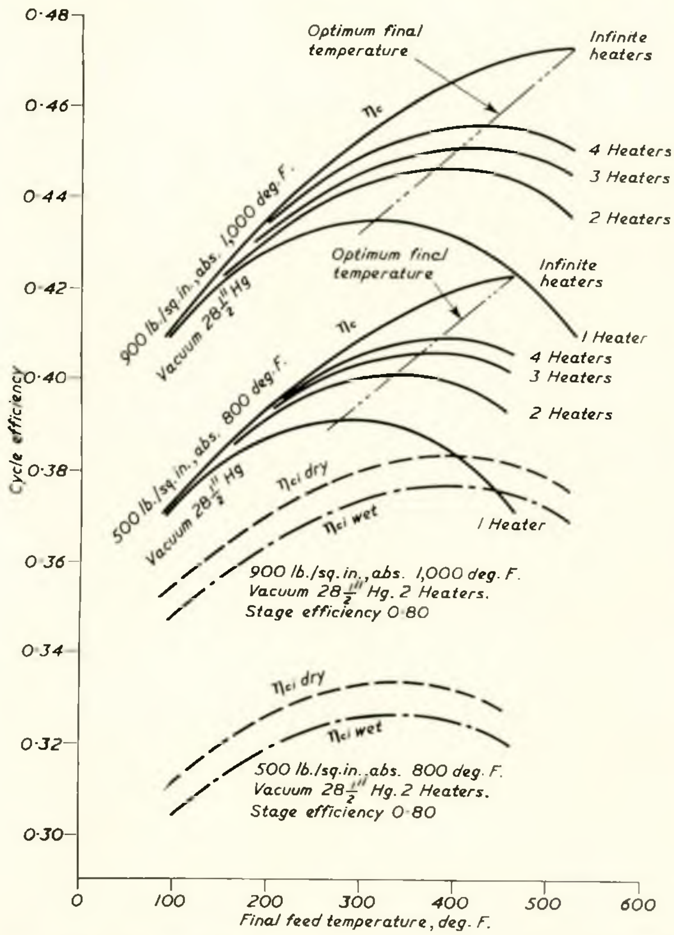
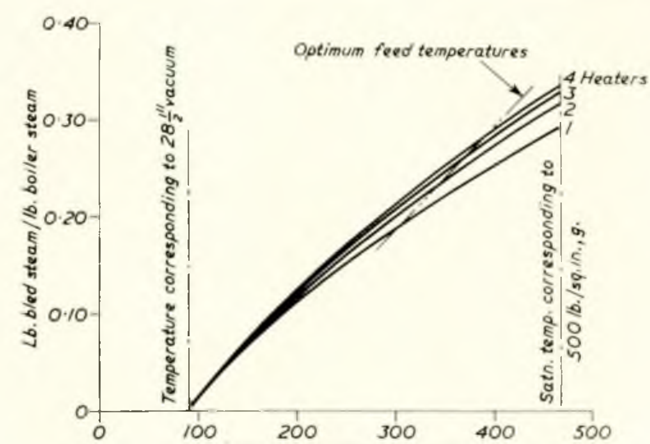


FIG 15—*Variation of feed heating cycle efficiency with final feed temperature*

quantity of steam evaporated in the boiler increases with feed heating, the amount of heat supplied is actually less, because of the high feed temperature, thus resulting in an efficiency gain. Fig. 16 shows the variation of bled heating steam with final feed temperature and number of heaters.

Because of the large quantities of steam which may be



Initial steam pressure 500lb./sq. in. abs.
Initial steam temperature 800 deg. F.
Exhaust vacuum 28½in. Hg

FIG. 16—*Variation of bled heating steam with final feed temperature and number of stages*

bled from the turbines, for feed heating and other purposes, it is the practice of turbine manufacturers to quote non-bled steam rate when describing the performance of their machines.

STEAM GENERATION

No account of steam turbine thermodynamics can omit at least a brief reference to the boiler or, more properly, steam generator, since the functions of superheaters, air heaters, economizers and reheaters, are tending to eclipse the simple means of ebullition. Fig 17 shows the effect of advancing steam conditions upon the allocation of boiler heating surface and the increasing importance of the superheater. One boiler designer and one cycle designer have considered the furnace essentially as a superheater, the former achieving actual evaporation by injecting superheated steam into the boiler drum.

The boiler unit is a highly efficient heat exchanger, the principal loss being the heat content of the exit gases, the minimum temperature of which is limited by the dew point of corrosive constituents. Fig. 18 shows the variation of boiler efficiency with exit gas temperature for various excesses of combustion air.

It will be seen that for a gas exit temperature of 300 deg. F. and ten per cent excess air (which gives 14·5 per cent CO_2 at furnace exit) the boiler efficiency is 88 per cent which is probably the maximum attainable in practice.

The objection is sometimes made that boiler efficiency being calculated upon the higher calorific value of the fuel, the effect of moisture loss unfairly depreciates the efforts of the boiler designer. Fig. 12 has been prepared to illustrate the influence of moisture loss, which is proportional to the hydrogen content of the fuel, upon boiler efficiency. It is apparent that in the absurd extremity of using pure hydrogen as fuel, a boiler efficiency of 79·5 per cent calculated on h.c.v. basis, could never be exceeded, whereas burning pure carbon, an efficiency of 94·2 per cent is possible as there is then no moisture latent heat loss. It is becoming increasingly common to calculate efficiencies of oil fired plant using a standard fuel h.c.v. of 18,500 B.t.u./lb. oil.

THE SYNTHESIS OF TURBINE AND BOILER

The product of cycle internal efficiency and boiler efficiency is fundamental for any steam turbine installation and gives the best possible overall efficiency for any particular cycle in association with a particular boiler, i.e. $\eta_{ci} \times \eta = \eta_{overall}$.

Boiler efficiency, given complete combustion, with minimum excess air and casing radiation losses of reasonable amount, is governed by the heat content of exit flue gases. This in turn depends upon the exit temperature and moisture content of the exhaust. It has been previously shown (Fig. 19) that moisture content and therefore moisture latent heat loss is entirely dependent upon the hydrogen content of the fuel. The only factor over which the boiler designer has control is therefore the final exit temperature of exhaust gas. This may vary from slightly below 400 deg. F. for boilers utilizing high sulphur content fuels, when corrosion troubles may be experienced on low temperature heating surfaces, if cooled much below 350 deg. F., to as low as 200 deg. F. with boilers burning low sulphur content fuel and where the gain due to improved fuel rate is sufficient to justify more expenditure on corrosion control and maintenance.

Flue gas from generating surface outlet to final boiler exit is capable of preheating all combustion air and supplying about 15 per cent of the total sensible heat of generation, which latter may amount to as much as 30 per cent of the heat supplied by bled steam feed heating.

The turbine cycle internal efficiency, for given initial steam conditions is dependent upon the turbine internal efficiency, i.e. the cumulative resultant of all stage efficiencies, and the amount of heat which can be usefully diverted, from rejection and loss in the condenser, to the preheating of feed water and/or combustion air.

Clearly then it is wasteful to provide for both full sensible

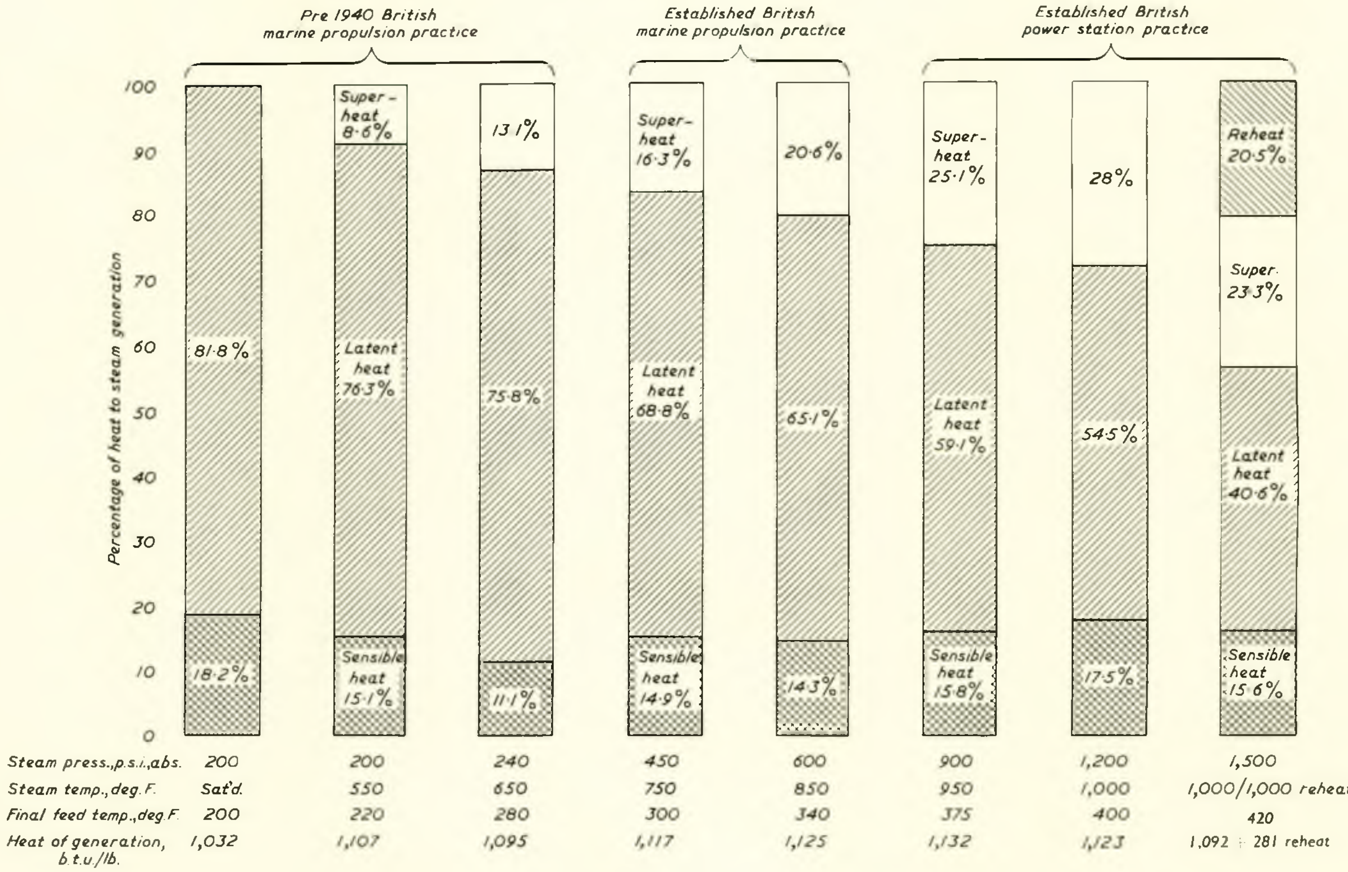
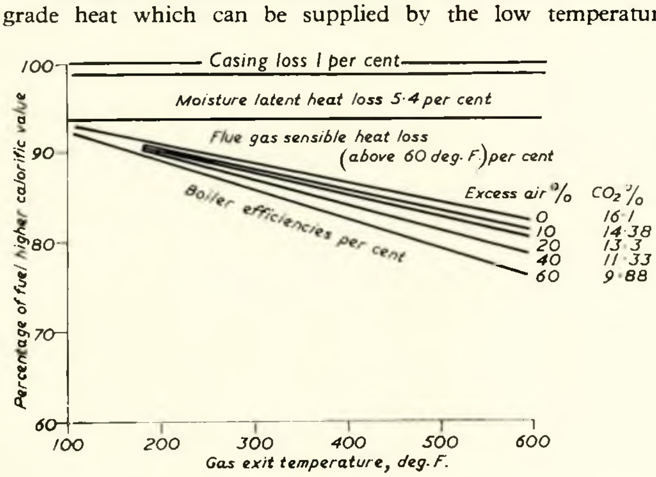


FIG. 17—*Analyses of heat to generate steam at various conditions of pressure and temperature*

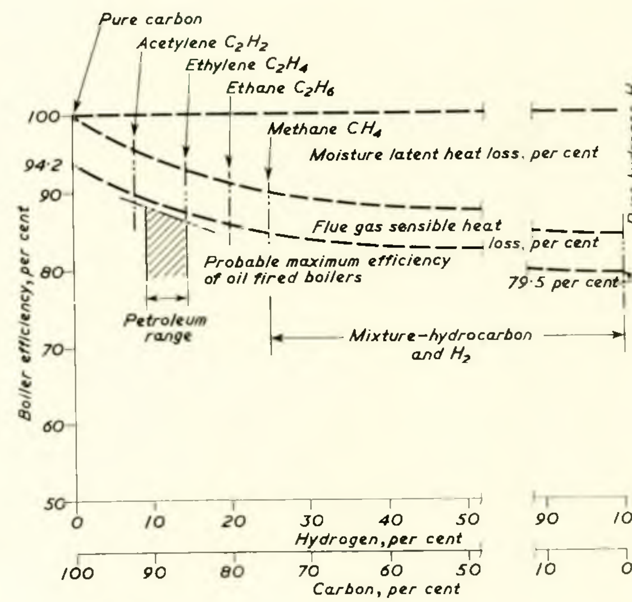
heating of feed water and combustion air heating by means of bled steam, at the expense of low temperature heating surfaces and consequently boiler efficiency.

It must be remembered that there is a limit to the low grade heat which can be supplied by the low temperature



Oil Fuel H.C.V. 18,500 B.t.u./lb.
Oil Fuel L.C.V. 17,500 B.t.u./lb.
Oil Fuel Composition C, 87·8 per cent; H_2 10·5 per cent. S, 2·0 per cent; O_2 0·7 per cent.
Boiler room temperature 90 deg. F.
Fuel and air assumed dry
No allowance for air leakage

FIG. 18—*Variation of boiler efficiency with gas exit temperature and excess air*



Assumed:—
No casing loss
Dry air and fuel
Complete combustion; no excess air
Stokehold temperature 60 deg. F.
Gas exit temperature 350 deg. F.

FIG. 19—*Probable maximum boiler efficiencies attainable burning various hydrocarbon fuels*

gas pass in the boiler as described above, and if more heat is required for low grade purposes, e.g. sensible heat of feed water, then this heat can only be supplied in the boiler, by sacrificing higher grade heating surface for this purpose, with consequent reduction of boiler rating. Also it must be remembered that there are limits to the quantities of steam which may be bled, for purposes of feed heating, and that there are optimum final feed temperatures to give maximum cycle efficiency as shown in Figs. 15 and 16.

To obtain maximum overall efficiency, therefore, the turbine cycle and boiler system must be carefully matched and bled-steam feed heaters, economizers (whether single or split feed), gas air heaters and/or bled-steam air heaters must be arranged to give the best result, whilst bearing in mind the relative initial and maintenance cost of these various combinations of components.

The theoretical overall plant efficiency figure will be greatly reduced in practice by the power requirements of various auxilliaries such as feed, circulating and extraction pumps and forced draft fans, also there will be pressure and temperature losses in piping and heaters. Careful design will minimize such losses without involving undue expense and it is important that further losses due to maloperation are avoided, particularly with marine installations, where the cycle is complicated by auxiliary power generation, heating, sea water evaporation and hotel load, it is essential that a heat flow and pressure and temperature diagram be available for reference by the operating staff.

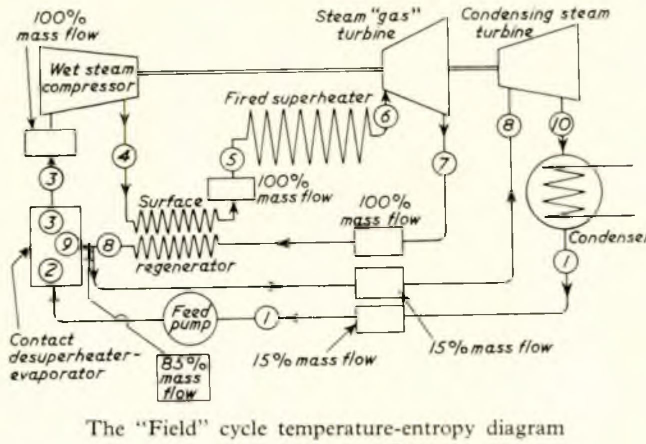
CONCLUSION

The choice of initial steam conditions, within the limits of reliably acceptable pressures and temperatures, and cycle modifications, is governed principally by operating economics related to capital cost. Occasionally the availability of materials may be decisive. The example of *Venore* has already been cited; more recently the initial steam temperature of the American standard *Mariner* class vessels was limited to 865 deg. F. in order to avoid the necessity of 1 cwt. of chromium, an extremely strategic material in wartime, for steel alloying. The consequence of this latter consideration is evident from the necessity of using 18 per cent Chromium – 12 per cent Nickel – 1 per cent Niobium (Columbium in America) alloy steel for superheater tubes and 3 per cent Chromium – 1 per cent Molybdenum alloy steel for rotors, when steam temperature exceeds 1,000 deg. F.

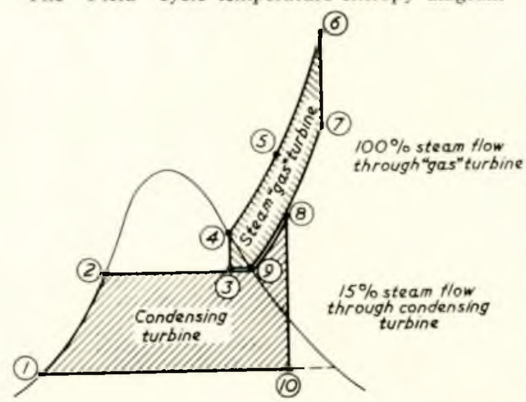
The important lesson to be learned from this study, as from any other, is that opinion based on prejudice and personal preferences can be very misleading and only by detailed investigation can the solution to any problem be resolved.

POSTSCRIPT

Over 40 years ago Riall Sankey[3] remarked that it was difficult to foresee any improvement in steam turbine plant efficiencies, then obtaining, of 18 per cent. In the interval, due to superheating, multi-stage feed heating, steam reheating, improved boiler and turbine performance and superior materials of construction, plant overall efficiencies of nearly 40 per cent and sustained annual efficiencies of over 33 per cent have been obtained.



The "Field" cycle temperature-entropy diagram



Simplified condensing "gas" turbine cycle adapted from the late J.F. Field, B.Sc.*

FIG. 20—*The "Field" cycle gas turbine technique applied to the steam turbine cycle: after late J. F. Field*

The proposed "Field" cycle after the late J. R. Field[4], promises even greater returns, but many practical difficulties have yet to be resolved. The simplified "Field" cycle is shown in Fig. 20. Enhanced efficiency is obtained by treating the high temperature/pressure portion of the cycle as for the closed circuit combustion gas turbine and bleeding off a proportion of the steam, 15 per cent in the particular cycle illustrated, to a low pressure condensing turbine, thus taking advantage of the low pressure and condensing properties of steam.

For prime movers of over 10,000 h.p. on land and 20,000 h.p. at sea the turbine is supreme, in fact there is no alternative

* I.Mech.E. Proceedings 1950; Vol. 162.

TABLE IV.—COMPARISON OF ADIABATIC HEAT DROPS OBTAINED WITH GIVEN EQUAL PRESSURE DROPS (A) AND TEMPERATURE DROPS (B).

Working fluid	Pressure lb./sq. in. abs.	Temperature deg. F.	Specific Vol. cu. ft./lb.	Adiabatic heat drop B.t.u./lb.	Ratio of ht. drops per lb.	per cu. ft.
STEAM (A and B)						
Before expansion	500	1,000*	1·698	329·4	1·00	1·00
After expansion	30	300*	14·82			
AIR (A only)						
Before expansion	500	1,000	1·08	198	0·602	0·946
After expansion	30	194	8·07			
AIR (B only)						
Before expansion	500	1,000	1·08	168	0·512	0·800
After expansion	48·9	300	5·76			

* Denotes steam in superheated condition.

available, but whether steam turbine or combustion gas turbine is another matter. Both steam and gas turbines are more or less equally handicapped by the economical upper temperature limit of readily available materials; the condensing properties of steam give this working fluid an advantage in the low temperature range. Moreover as indicated in Table IV (a precise comparison is not readily obtainable) steam has a slightly greater energy capacity than air or combustion gases, per unit weight or volume.

This study was originally undertaken as part of an overall investigation as to the future of the heat engine, or rather as to the heat engine of the future, a study as uncertain as it it fascinating and, although these purely theoretical results can be of little practical value, they are, by their fundamental nature, a reliable and basic guide to what can or cannot be achieved in practice.

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3) Sankey, H. Riall. Nov. 1917. Thomas Hawksley Lecture, "Heat Engines". Proc.I.Mech.E., Oct.-Dec., p. 703.
4) Field, J. F. 1950. "The Application of Gas-turbine Technique to Steam Power". Proc.I.Mech.E., Vol. 162, p. 209.

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From left to right: Mr. C. C. Pounder (President) and Mrs. Pounder with Mrs. Ingamells and Mr. B. P. Ingamells, C.B.E. (Chairman of Council)

INSTITUTE ACTIVITIES

Annual Conversaziones

A total of 1,727 members and their guests attended the Annual Conversaziones held at Grosvenor House, Park Lane, on 1st and 15th December 1961. At least one hundred were prevented from attending the second Conversazione by reason of the dense fog which prevailed throughout Southern England on that day.

On both occasions music for dancing was provided by Sydney Jerome and his Ballroom Orchestra and the following artists were included in the cabarets: The Musical Campbells, The Seven Volants, Joan Turner, The Barrie Manning Song and Dance Show, The Barantons, The Tiller Dancers and the Cuban Cossack Dancers. Once again members enjoyed the carols which were sung immediately after dinner on the second occasion.

Section Meetings

Devon and Cornwall

A Senior Meeting was held on Tuesday, 28th November 1961 at the Royal Naval Engineering College, Manadon, Plymouth, at 7.15 p.m.

A paper entitled "The Automatic Control of Naval Boilers" by Commander J. P. H. Brown, R.N. (Member) and Lieut. Commander W. J. P. Thomas, R.N. was presented by the authors.

The meeting was very well attended, about ninety members and visitors being present. At the conclusion of the most interesting lecture a discussion followed and a further hour passed before the meeting closed.

The arrangements for presenting the paper and for receiving the visitors were most efficiently carried out by the officers of the R.N.E.C.

Eastern United States

A meeting of the Section was held on Thursday, 30th November 1961, at the U.S. Merchant Marine Academy, Kings Point, New York, at 5.30 p.m. Following a reception at the Officers Club fifty members and guests dined with the Regiment of Cadets. Dinner was followed by a tour of the Nuclear Engineering Laboratory, conducted by Lt. Cdr. M. E. Hirschkowitz, U.S.M.S.

A Regimental Assembly was then attended by members and a large number of Engineering Cadets, at which Lt. Cdr. M. J. Gross, U.S.M.S. and Lt. Cdr. M. E. Hirschkowitz, U.S.M.S., Professors of Nuclear Engineering, presented a joint lecture entitled "Engineering Officer Training Programmes for the n.s. *Savannah*".

A quarter scale model of the control console for the n.s. *Savannah* was exhibited. The meeting also presented the first opportunity for marine engineers to preview the nuclear engineering officer training programmes being conducted at Kings Point.

Mr. J. H. Thomas (Chairman of the Section) presided at the meeting and, after introducing the speakers, was joined by Captain L. S. McCready, U.S.M.S., Head of the Department of Engineering, U.S. Merchant Marine Academy, to moderate the lengthy discussion period following the lecture.

The members acclaimed a vote of thanks to the authors, other Academy faculty and the Regiment of Cadets for their hospitality throughout the evening.

After the meeting closed, at 9.30 p.m. the members were invited to visit the Engineering Laboratory.

Kingston upon Hull and Humber Area

A meeting of the Section was held on Thursday, 14th December 1961 at the Royal Station Hotel, Kingston upon Hull at 7.30 p.m.

Mr. Bryan Taylor, B.Sc.(Eng.) (Chairman of the Section) presided and a paper entitled "Looking Back on Nuclear Propulsion" by R. P. Williams, B.Eng. (Associate Member) was presented by the author.

The lecturer is engaged on marine reactor development and his paper ranged over such subjects as the simple fission process, types of marine reactor and the economics of building and running a nuclear powered tanker. Mr. Williams used a number of slides to illustrate his paper and these included some depicting the economics of such a vessel in a graphical form which not only showed the present stage of development but also what the designer of such a plant must aim for. It was this part of the paper that provoked an interesting discussion period.

A vote of thanks to the author was proposed by Mr. H. F. Hesketh (Member).

South Wales

A meeting of the Section was held on Monday, 20th November 1961 at the South Wales Institute of Engineers, Park Place, Cardiff, at 6 p.m. when a paper entitled "Marine Machinery Breakdowns" by J. H. Milton (Member of Council) was presented by the author to an audience of seventy members.

A vote of thanks to the author was proposed by Mr. R. H. Rees, O.B.E. (Member) and seconded by Mr. H. S. W. Jones (Associate Member). A vote of thanks to the Chairman was proposed by Mr. R. G. Turnbull (Member).

Minutes of the Proceedings of the Student Meeting held at the Institute on Monday, 15th January 1962

A meeting of the Student Section was held at the Memorial Building, 76 Mark Lane, London, E.C.3, on Monday, 15th January 1962, at 6.30 p.m.

A lecture on "Motorship Auxiliary Machinery" was given by Mr. H. Taylor (Member).

46 Members and visitors were present.

The lecture was followed by an interesting and lively question period.

Mr. G. F. Gatward (Associate Member) was in the chair, and the vote of thanks to the author by the Chairman, was carried by acclamation.

The meeting ended at 8.20 p.m.

OBITUARY

Alfred Cecil Hardy

An appreciation by J. Calderwood, M.Sc. (Vice-President)

To many who did not know him personally Cecil Hardy was looked upon only as a technical journalist who wrote largely in a semi-technical vein. He was often subject to criticism for the nature of some of his articles but his critics did not appreciate the object of the bulk of his writing.

Much that he wrote was published in papers read largely by non-technical or semi-technical readers and he had the ability to write on subjects of a technical nature in language which made them clear to laymen; to a large extent he made it the object of his writings to keep the lay world informed of what was happening in shipbuilding and, further, to publicize new methods, new ideas and new types of equipment.

On one occasion in his early days a friend referred to him as the Edgar Wallace of technical journalism. At the time he was very hurt by this remark but it was, in fact, in the nature of a compliment in that he wrote in a style that made it very difficult not to read to the end of any article once you had picked it up. One of the best compliments to the value of his writing was paid by the manager of a small marine engine works in the north of Spain. I was near it on holiday and walked round this very interesting though very small works. Afterwards the manager asked me if I knew Hardy. On my saying that I did he produced one of his books. He then told me that to them and to other people who were out of the main run of shipbuilding and marine engineering Hardy's books and articles were of the greatest possible value as they kept them in touch with what was happening in the shipbuilding world as a whole.

It is fair to say that the publicity which he gave to new ideas and the energy with which he backed them has greatly helped to introduce many of them into shipbuilding and marine engineering and his sound technical sense is shown by the fact that few of the ideas that he backed at an early stage failed to be developed successfully later.

Cecil Hardy had a vivid personality and the ability to mix on terms of good friendship with men of all ages. Few men

worked harder than he and he was always willing and ready to help others, particularly younger men in the profession. His loss will be felt deeply by all of his friends and by many others who knew him only by his writing.

Commander A. C. Hardy, B.Sc. (Associate Member) an Associate Member of Council of this Institute, died at Littlehampton on 17th December 1961.

Born in 1898, he was trained as a naval architect at Palmer's Shipbuilding and Iron Company, Jarrow, and graduated B.Sc. at Durham University. He had shipbuilding experience in various shipyards in the United Kingdom and the United States; consulting experience with Messrs. Camps and Co., naval architects; he was chairman and consultant of Hardy, Tobin and Co. Ltd. in the City of London.

A prolific writer on all aspects of the maritime industries, in 1923 Commander Hardy was assistant editor (technical) of "The Marine Engineer and Naval Architect", later becoming editor of "The Motorship" New York and for over thirty years was a regular contributor to "Lloyds List".

During World War II he was a lecturer on shipping, City of London College, and a member, Royal Corps of Naval Constructors; secretary, Shipping Security Co-ordination Committee, Admiralty, 1942; deputy command constructor officer, British Naval Commander-in-Chief, Germany; personal technical assistant to Chief of Combined Operations at the First Quebec Conference in 1943; in the same year supervising the conversion of cargo ships to assault carriers; and senior British technical member, Tripartite Naval Commission, Berlin, 1945. He was also Honorary Consultant to the Fisheries Division, Food and Agriculture Organization of the United Nations.

An Associate Member of the Institute since 1923; an associate member of the Council almost continuously since 1937, he was also a member and, from 1953-56, member of council of the Royal Institution of Naval Architects; a member, North East Coast Institution of Engineers and Shipbuilders; a member, Institute of Petroleum; a fellow, Royal Geographical

Society; a liveryman, Worshipful Company of Shipwrights; a freeman of the City of London; and a founder member and subsequently co-ordinator of the International Cargo Handling Co-ordination Association.

A. C. Hardy played a long and invaluable part in the affairs of the Institute and besides being several times elected to Council, served as chairman of the Marine Engineers National Memorial Appeal Committee, the Library Committee and the Publicity and Advertising Committee. From 1936 he served as a member of the Papers and Transactions Committee and also on the Special Conference (1962) Committee.

David Frank Cormack (Member 7992) died on 3rd November 1961 aged 65 years. Having served his apprenticeship with Messrs. Morrell, Mills and Co. of Manchester, he went to sea between 1920-27 as fifth and then third engineer with the British Tanker Company, the Eastern Telegraph Company and Elders and Fyffes Ltd. respectively. In 1928 he became engineer and boiler inspector with the Manchester Steam Users' Association, a position he held for a year. In 1930 he joined Messrs. Bowring and Co. where he served in various capacities, eventually as senior second engineer. In 1936 Mr. Cormack came ashore and took up an appointment with the Aeronautical Inspection Directorate, with whom he remained until 1947. In that year he joined the staff of W. J. Fraser and Co. Ltd. of Romford, Essex, a firm of chemical engineers, where he acted as their chief inspector. He continued in this appointment until his retirement on 30th November 1960.

Mr. Cormack held a First Class Board of Trade Steam Certificate with Motor Endorsement and had been elected to full Membership of the Institute on 4th November 1935.

George Hallewell (Member 8340) was born on 28th January 1897. Indentured to the Campbell Gas Engine Co. between 1914-18, Mr. Hallewell joined the Royal Air Force in the First World War in charge of the power station at West Fenton aerodrome. In 1919 he rejoined the Campbell Gas Engine Co. as a draughtsman and the following year went as chief draughtsman to H. Widdop and Co. Ltd., Keighley. In 1925 he returned for two years to the Campbell company as engine designer and followed this with a second period with Widdop's, again as chief draughtsman. In 1933 Mr. Hallewell joined Tangyes Ltd. as chief engineer in the Diesel department.

From 1935 until his death Mr. Hallewell has served with Blackstone and Co. Ltd., Stamford, where his first appointment was as chief draughtsman. At Blackstone's he later became responsible for development projects and was elected to the board some five years ago. In his capacity as chief engineer he was responsible for the development and introduction of Blackstone engines now in worldwide use. He had invented a flexible coupling and nodal damper which has proved of value not only in marine installations but in industrial power houses too.

Mr. Hallewell was elected a Member of this Institute on 7th December 1936, and was also a member of the Institution of Mechanical Engineers. He died suddenly on 3rd January 1962, and is survived by a widow, a son and a daughter.

John Joseph Joyce (Member 22738) was born on 19th March 1909. After leaving school he attended Greenock Technical School where he took the O.N.C. mechanical engineering course while serving an apprenticeship with John G. Kincaid and Co. Ltd. of Greenock. After completing his training as an apprentice, in 1929 Mr. Joyce joined Alfred Holt and Co. Ltd. (Blue Funnel Line) for the next seven years in a seagoing capacity. While serving with the Blue Funnel Line he gained experience with Burmeister and Wain, Doxford and Werkspoor Diesel engines, and also with steam turbine and reciprocating engines. It was in this period, also, that he studied at Watts Memorial School, Greenock and obtained his Second and First Class Board of Trade Motor Certificates together with a First Class Steam Endorsement.

In June 1937 Mr. Joyce entered the service of the Aeronautical Inspection Directorate in the employ of the Air Ministry/Ministry of Supply and was for five years an inspector resident at contractors' works engaged on the manufacture and repair of aircraft for the R.N. and R.A.F. The war years were spent at the headquarters of the A.I.D., where Mr. Joyce was employed in the technical direction of field inspection staff. In September 1946 he was appointed to the staff of the Directorate of Aircraft Production, a directorate responsible for all aspects of military aircraft production, on which he served in various grades until his promotion, in 1950, to that of principal production officer (Engineer I) in which capacity he was serving at the time of his death. At the Directorate of Aircraft Production, Mr. Joyce was responsible for ensuring that a group of aircraft contractors under his departmental care was efficient and capable of producing aircraft in the most economical manner to the required programme. His duties also included the study of production aspects of new designs.

Mr. Joyce became a full Member of the Institute on 17th October 1960 and was also an Associate Fellow of the Royal Aeronautical Society. He died on 23rd August 1961 and leaves a widow, two sons and a daughter.

James Edward Smith (Member 18615), former managing director of the North Eastern Marine Engineering Co. Ltd., died on 7th January 1962 at the age of 47 years. His apprenticeship was served with the company of which he was later to become head, between 1930-35 during which time he attended Sunderland Technical College, finally gaining the Higher National Certificate in Mechanical Engineering. Immediately after completing his apprenticeship, Mr. Smith became an engineering draughtsman and estimator with North Eastern Marine at Sunderland but in 1937 he moved to Wallsend as a marine engineering estimator. In 1940 he became assistant to the engine works manager and three years later was appointed boiler works manager, to which in 1945, was added the further duties of engine shop manager. Three years later Mr. Smith was appointed works manager of Richardson, Westgarth and Co. Ltd., members, like North Eastern Marine, of the Richardsons, Westgarth Group. In 1951 he was put in charge of the marine department and a year afterwards was made works director. In 1956 he reached the position of managing director of North Eastern Marine Engineering Co. Ltd. and only severed his 29-year-old link with the group to join Hawker Siddeley Industries as managing director of the National Gas and Oil Engine Co. Ltd. at Ashton-under-Lyme, a firm which has now been merged with another company of which Mr. Smith was a director, Mirrlees, Bickerton and Day Ltd. In October 1961 he was appointed managing director of the Brush Electrical Engineering Co. Ltd., but was unfortunately prevented by ill-health from taking up this position.

Mr. Smith was at one time a council member of the North East Coast Institution of Engineers and Shipbuilders. He also served on the executive committee of the Engineering Employers' Association in the North East and Manchester, and was elected a Member of this Institute on 13th March 1957.

George Alfred Williams (Associate Member 10724) was born on 23rd June 1913 of British nationality. He served his apprenticeship at H.M. Dockyard, Devonport between 1928 and 1933, having become a Whitworth prizeman in 1932. After completing his apprenticeship, he acted as an engine fitter at Devonport for the following year and then transferred to second class draughtsman until 1935, when he was posted to Admiralty in Whitehall. From 1939 until 1946 he served as a first class draughtsman and was elected an Associate Member of the Institute on 19th March 1946. Mr. Williams was first elected to the Committee of the West of England Section in May 1956, and this was repeated in January 1958. He lived for some time at Widcombe Hall, Bath. The date of his death is unknown.