The INSTITUTE of MARINE ENGINEERS
Transactions

1951, Vol. LXIII, No. 11

HIGHER STEAM CONDITIONS FOR SHIPS' MACHINERY

Problems in the Selection and Application of Cycle Components and High Temperature Materials

By M. L. IRELAND, JR., H. W. SEMAR and N. L. MOCHEL

Introduction

A discussion of this subject can be clarified by first defining what is meant by "higher" steam conditions.

It would be easy to say that one means higher than some specific currently accepted level of pressure and temperature such as 600 lb. per sq. in. and 850 deg. F. However, a more logical definition arises from the necessity for entry into a field of new problems and new concepts of design. Accordingly, the authors are referring to steam conditions which require a consideration of the effects of temperature upon the long term properties of the materials used in the construction of ships' machinery. No new types of problems are introduced by advances in pressure above say 600 lb. per sq. in., although existing problems may be so accentuated that new solutions are required.

Discussion will be confined to geared turbine drive since this is the most widely used type associated with advances in steam conditions and encompasses problems not encountered with electric drive. Security requirements limit the subject to a consideration of merchant vessel types and World War II Naval construction.

Naturally, the authors' viewpoint is based on conditions prevailing in the United States. Entirely different conclusions might be reached in other countries where the relations between costs of fuel, raw materials and labour are quite different. This situation has been ably discussed by Burkhardt(1).*

The major incentive for advances in steam conditions for commercial vessels is the prospect of a saving in fuel consumption. Certain classes of long haul deadweight carriers such as tankers and ore carriers can profit from a reduction in machinery weight as well as smaller bunker capacity but, as is shown in Appendix I, only minor weight savings, if any, are to be expected from increased steam conditions.

There have been some very interesting studies published on the continent(2) which indicate a reduction in the inherent costs of stationary power plants as steam conditions are increased. However, available information from published American and British sources(3), (4), (5), (6), (7), indicate that prices as distinct from inherent costs invariably become higher for both stationary and marine power plants. It seems evident that substantial weight savings must be effected if costs are not to increase, since the increased alloy content required for higher temperature materials is inherently more expensive.

It appears, therefore, that consideration of advances in steam conditions must require striking a balance between fuel savings and increased first costs. A study of this economic problem for merchant ships was published about three years ago.(4) It is now clear that the factual results of this study were destined to be somewhat conservative because the initial scope was too limited. The study was based on prices for only three vessels which inevitably penalized the higher steam conditions because of the development costs which had to be borne by a small number of units.

It is believed that quite different conclusions might be reached by concerns operating a large fleet of vessels, especially if long term replacements were considered. An excellent example of analysis from the viewpoint of fleet requirements appears in reference (8).

Whatever the combination of incentives, it is a fact that merchant ship steam conditions are currently advancing. It should also be recognized that these advances, if soundly based and ably prosecuted, will become widely adopted as development costs are mainly absorbed by the initial installations. Thus, the advanced steam conditions of today, if successful, are likely to become the standard of tomorrow and will be priced accordingly.

This result is well illustrated by present-day conditions in the United States where, for new construction of moderate to high power (9,000 s.h.p. per shaft and greater), few operators would seriously consider steam conditions less than 600 lb. per sq. in., 850 deg. F., whereas such conditions were exceptional before World War II. There are indications that a similar condition now applies to construction in the British Isles, though the combination most favoured appears to be 500 lb. per sq. in., 800 deg. F.

To the authors, the moral of all this seems to be that the strictly economic solution is likely always to be short sighted and that the wisest long term selection should go at least one step further. This conclusion is emphasized by the continued tendency for initial costs to increase as time goes on.

Pressure-Temperature Combinations

There are several factors, in addition to economics, which may influence the selection of a set of pressure-temperature conditions for a specific application. A number of these

*Figures in parentheses indicate references to be found on p. 249.

TABLE I

MARINE PRESSURE TEMPERATURE COMBINATIONS BASED ON ASA DIMENSIONAL STANDARDS AND INTERIM GUIDE RECOMMENDATIONS FOR FERRITIC PIPING MATERIALS

(a) *Facings other than ring joint* Steam temperature, deg. F.	Dimensional Standard	750	800	850	900	950	1,000	1,050
				Working Pressures				
C. steel	600 lb.	680	600					
C.-Mo. steel	600 lb.	720	680	640	*600*			
1-Cr. ½ Mo. / 1¼ Cr. ½ Mo.	600 lb.			640	*600*	530	420	290
2¼ Cr. 1 Mo.	*600 lb.			640	*600*	530	430	360
Carbon steel	900 lb.	1,020	900					
C. Mo. steel	900 lb.	1,080	1,020	960	*900*			
1-Cr. ½ Mo. / 1¼ Cr. ½ Mo.	900			960	*900*	795	630	435
2¼ Cr. 1 Mo.	*900			960	*900*	795	645	540
(b) *Ring joint facings (or welding end valves and fittings without flanges)*								
C. steel	600 lb.	850	730					
C.-Mo. steel	600 lb.	850	750	700	650			
1-Cr. ½ Mo. / 1¼ Cr. ½ Mo.	600 lb.	850	800	750	680	600	520	360
2¼ Cr. 1 Mo.	*600 lb.	850	800	750	680	*600*	540	450
C. steel	900 lb.	1,275	1,100					
C.-Mo. steel	900 lb.	1,275	1,125	1,050	975			
1-Cr. ½ Mo. / 1¼ Cr. ½ Mo.	900 lb.	1,275	1,200	1,125	1,020	*900*	780	540
2¼ Cr. 1 Mo.	*900 lb.	1,275	1,200	1,125	1,020	*900*	810	675

* Note that Interim Guide makes no specific provision for use of ferritic alloys of greater chromium content or for austenitic alloys.

are discussed in a very clear manner by Dr. T. W. F. Brown in a recent and excellent paper(9).

An important factor which must be considered in American practice is the generally adopted American Standards Association pressure-temperature ratings for American Standards Series valves and fittings. For marine installations, these are codified in the "Interim Guide"(10) published and adopted by the American Bureau of Shipping and largely included in the current regulations of the United States Coast Guard Bureau of Merchant Marine Inspection.

Owing to the American Standard Series basic pressure standards of 400, 600, 900 and 1,500 lb. per sq. in., the choice of steam pressures is likely to be limited to similar increments. Variations in the applications of these standards to different steam temperatures are available depending on the alloy content. Table I, based on the above reference(10), presents a condensation of the principal choices of current interest.

A recent version of the fuel savings which can be obtained with advances in steam conditions is shown in Fig. 1. This figure is a revision of earlier information(4) based on the considerations which follow. In the first place, the earlier curves were based on a certain standard level of turbine performance as a function of the power rating. Recent, and earlier published results, references (11), (12), (13), (14) and (15), show that this performance has been exceeded over the entire range as shown in Fig. 2. Use of this higher level of turbine performance produces more favourable results at the higher steam pressures and temperatures mainly due to its effect on the regenerative part of the cycle. Details of the method of obtaining these results are given in Appendix II.

Another factor not fully expounded in the earlier pre-

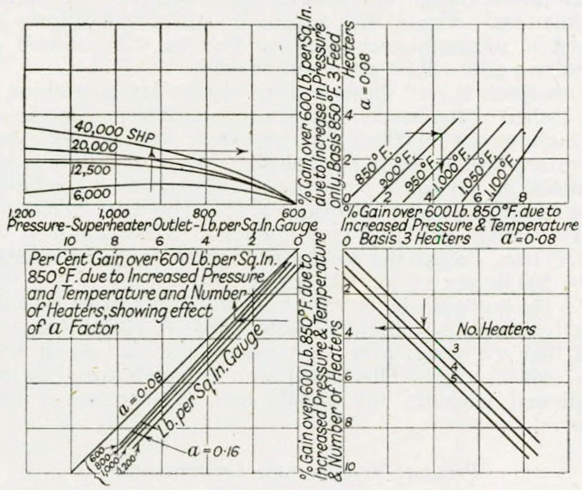


FIG. 1.—IMPROVEMENTS IN SHIP FUEL RATE DUE TO HIGHER PRESSURES AND TEMPERATURES

a = ALLOWANCE FOR AUXILIARIES AND HOTEL LOAD

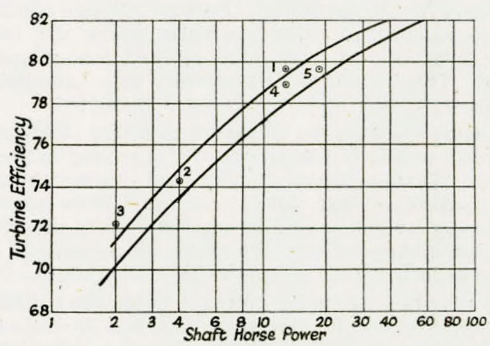


FIG. 2.—EFFICIENCY OF GEARED TURBINES FOR VARIOUS DESIGNED POWERS CORRECTED TO 590 LB. PER SQ. IN. ABS., 840 DEG. F., 28·5 IN. HG.

(1) ESSO SUPER TANKER REF. (11)
(2) GREAT LAKES FREIGHTER REF. (14)
(3) BULK FREIGHTER, WILLIAM A. IRVIN REF. (15)
(4) BETHLEHEM SUPER TANKER REF. (12)
(5) SUPER TANKER—ATLANTIC SEAMAN REF. (13)

sentation was the influence of variations in the ship's hotel load requirements. The new curves have been amplified to include this effect.

A practical consideration which has a bearing on the choice of steam pressure is the decided preference of some operators for the sectional header type of boiler. The principal advantages claimed are quicker plugging of damaged tubes and reduced time and cost of cleaning external surfaces. Selection of this type of boiler has in the past imposed a definite limitation on the working pressure of about 600 to 700 lb. per sq. in. due to the difficulty and cost of constructing headers for much higher pressures. It is understood that recent developments may increase the allowable pressure for sectional header boilers to about 850 lb. per sq. in.

This preference, in combination with a realization that the law of diminishing returns applies more stringently to increases in pressure than to increases in temperature, has resulted in some installations of moderate pressure which employ a quite advanced temperature [13], [16] and [17].

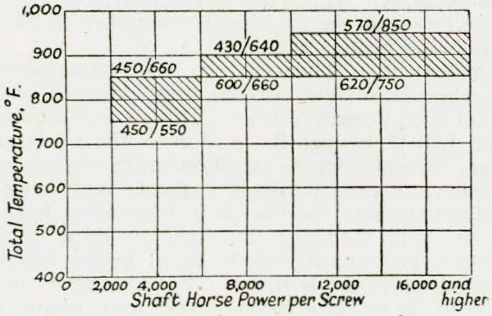


FIG. 3—SUGGESTED RANGE OF STEAM CONDITIONS FOR VARIOUS SHAFT HORSE POWERS. NUMBERS ON LINES ARE STEAM PRESSURE LB. PER SQ. IN. GAUGE

Such installations are well out of the range where moisture in the last few rows of low pressure turbine blading is a matter of concern. On the other hand, there are indications that the erosion problem for steam conditions resulting in 11 to 12 per cent moisture has been considerably reduced at least for moderate blade speeds by the provision of drains between blade rows[18].

The presentation of a recommended choice of steam conditions as a function of the power rating implies that all the foregoing considerations have been weighed and an economic balance obtained. Such a result can only apply to a specific set of conditions; however, a range may be indicated which would presumably include the majority of cases. Fig. 3 presents the authors' version of such a range in which account has been taken of increasing competition from Diesel propulsion at low and moderate powers per shaft. Previously published recommendations appear in references [4], [7], [9] and [19], and numerous other publications.

Selection of Principal Machinery Components

When a suitable temperature-pressure combination has been determined, the selection of cycle components can proceed along fairly definite lines based upon the following very simple principle: if an advance in steam conditions has been decided upon, then every reasonable effort should be made to realize the full advantages of such a step in the cycle arrangement and principal machinery component selection.

Inability to carry out this principle seems to be the reason why the then very advanced steam conditions employed in German merchant vessels between 1935 and 1939 did not result in outstanding fuel rate performances.

This point is illustrated by the figures given in Table II based on information [20] compiled in 1945.

Cycle arrangements will be discussed in the proper place but the first and paramount consideration is to obtain the best possible performance from the main propulsion turbines consistent with the highest standard of dependability. As previously indicated, usual American practice can be represented by a "standard" engine efficiency curve for one condition accompanied by a consistent set of correction factors for other pressures, temperatures and vacua[4].

Also, as previously stated, this so-called "standard" can be excelled if there is sufficient incentive. Still referring to American practice, some turbine manufacturers have established price increases applicable to specific percentage improvements over the "standard" performance.

In view of the above, the authors' suggestion is that turbine performances be studied over a range including the very highest that can be guaranteed. Then, having compared these results on a strictly economic basis including all factors that apply to the particular case, the final selection should lean toward an efficiency value exceeding the economic optimum. This recommendation follows from the same

TABLE II

SUMMARY OF GERMAN HIGH TEMPERATURE INSTALLATIONS—1935 AND 1939

Vessel	Gneisenau	Tannenberg	Pretoria	Eisenach (re-engd.)	Scharnhorst*	Potsdam*	Blohm and Voss † Hull 523
Type	C. and P.	C. and P.	C. and P.	Cargo	C. and P.	C. and P.	C. and P.
Year built	1935	1935	1937	1939	1935	1935	1940
Builder	Deschimag	Stettiner Oder-Werke	Blohm and Voss	Deschimag	Deschimag	Blohm and Voss	Blohm and Voss
Displacement, tons	24,250			9,680	24,250	23,600	36,000
S.h.p., total	26,000	12,500	13,000	4,500	26,000	26,000	44,000
No. shafts	2	2	2	1	2	2	2
R.p.m.	129	248	125	100	130	160	882 (throttle)
Superheater pressure, lb. per sq. in. gauge	735	882	1,135	735	735	1,320	877 (throttle)
Superheater temperature, deg. F.	878	896	896	842	877	896	
Fuel rate, lb. per s.h.p. hr.	0·638	0·574	0·68		0·638	0·67	

* Turbo-electric drive.
† Not completed.

considerations previously outlined regarding the initial selection of steam conditions.

With regard to boilers, it is possible at the present time to prescribe a limit of performance beyond which excessive maintenance is almost certain to develop. If the uptake temperature is below about 300 deg. F. corresponding to 88 per cent efficiency at the normal continuous rating, corrosion of gas heated type air heater elements is likely to be excessive. For satisfactory maintenance at this condition it is essential to provide air heater by-passes for reduced ratings and to maintain a rigorous schedule of soot blowing and cleaning of external surfaces.

The reference to air heaters as a matter of course may not be entirely satisfactory to some operators who have a decided preference for economizers. The important point to realize is that effective use of advances in steam conditions requires the fullest possible use of regenerative cycle features. However, this cannot be accomplished and at the same time obtain satisfactory boiler efficiency unless some form of air heater is employed.

For those who object to gas type air heaters there is available an alternate arrangement which employs bled steam for air preheating[12]. However, the thermodynamic capabilities of this system are distinctly limited for pressures exceeding about 900 lb. per sq. in. unless considerably higher air preheat temperatures are employed than is now customary with gas air heaters[21].

Another system which has been considered uses part of the boiler feedwater as an intermediate heat exchange medium, and while this arrangement is thermodynamically equal to direct air heating, its economic position has not as yet been determined by actual use.

Choice of Cycle Arrangements and Auxiliary Components

The general postulate of going one step beyond what appears at the moment to be the optimum choice from a strictly economic evaluation applies equally to the choice of cycle arrangement and to all its auxiliary components.

This consideration has apparently resulted in the use of a higher final feed temperature and more stages of feed heating than are indicated by economic studies. The point is illustrated by Fig. 4 wherein optimum and economic final feed temperatures are compared with current American practice. Considerable variation is to be noted between the results from different sources. The data from Salisbury[22] shows a particularly significant effect of type of heater arrangement upon the optimum final feed temperature.

It will be noted that the final feed temperature is directly related to the steam pressure and affected very little by steam temperature. This has been shown to be fundamentally correct [3], [22], [23]. The gains resulting from a larger number of feed heating stages are closely related to the total feed heating temperature range. It is not so generally recognized that extra feed heating stages produce a slight additional gain at higher steam temperatures irrespective of the steam pressure. This is due to a reduction in the inherent losses resulting from the use of highly superheated bled steam in the later feed heating stages.

This statement is illustrated by the values shown on Fig. 25 where a smaller correction is indicated when five heaters are used instead of four. This further improvement of about 1 B.Th.U. per lb. is due to the lower average superheat resulting from the use of one more extraction point.

One feature of present trends in the United States which merits elaboration is the use of a de-aerating feed heater capable of fully utilizing the main turbine cross-over pressure[11]. For approximately equal power distribution the cross-over pressure is higher for increased steam conditions. With a non-condensing type feed pump turbine drive, it becomes necessary to increase the exhaust pressure of these units above the usual 15 lb. per sq. in. gauge level. However, the increased effectiveness of the cross-over bleed point and the elimination of throttling often more than compensates for the increased feed pump turbine steam rate. Under some conditions overall fuel rate savings of about 3/4 per cent may be credited to this development with only small increases in cost of the affected equipment.

Following this same line of reasoning into the selection of steam conditions for auxiliary turbine drives led to a careful review of previous recommendations[4] for the use of desuperheated steam for such turbines. As a result of this review, it still appears that only moderate temperatures can be justified for non-condensing auxiliary turbine drives. While marginal improvements in the heat cycle can be obtained by using full boiler pressure and temperature steam for these turbines, the expense of such designs is not justified by the results.

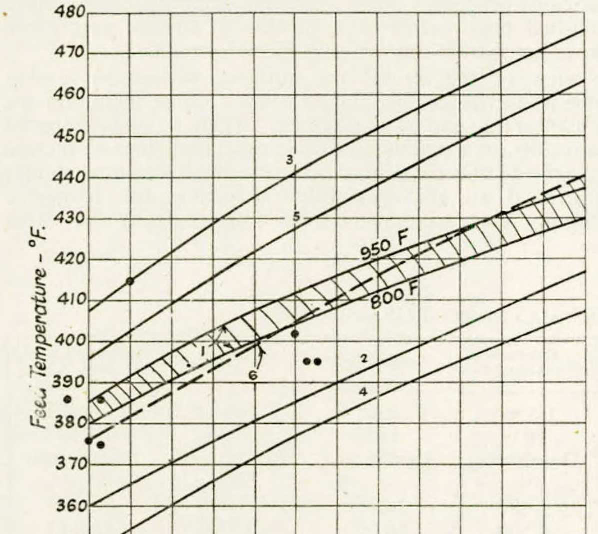


FIG. 4.—COMPARISON OF OPTIMUM AND ECONOMIC FEED TEMPERATURES WITH CURRENT PRACTICE

(1) OPTIMUM FOR 5 HEATERS 800/950 DEG. F. REF. (7)
(2) ECONOMIC FOR 5 HEATERS 800/950 DEG. F. REF. (7) ABOVE AT 50,000 KW. 29 INCH VACUUM
(3) OPTIMUM 5 CONTACT HEATERS REF. (22)
(4) OPTIMUM 5 FLASH HEATERS REF. (22)
(5) ECONOMIC 5 CONTACT HEATERS REF. (4)
(6) OPTIMUM 3 CONTACT HEATERS REF. (22)
● CURRENT U.S. PRACTICE—INSTALLATIONS AND PROPOSALS

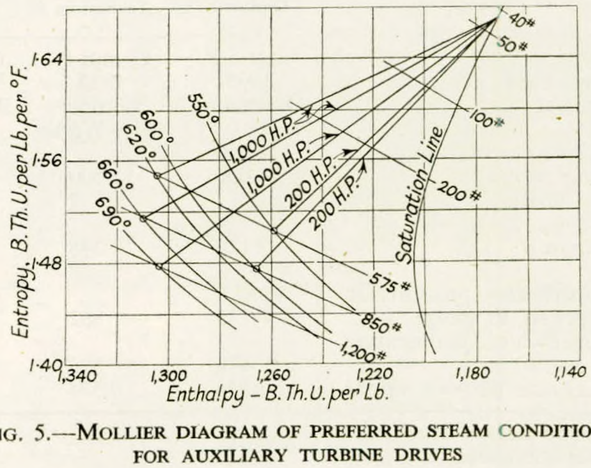


FIG. 5.—MOLLIER DIAGRAM OF PREFERRED STEAM CONDITIONS FOR AUXILIARY TURBINE DRIVES

A reasonable compromise would appear to be afforded by using full boiler pressure and reducing the inlet temperature by de-superheating to such an extent that the auxiliary turbine exhaust is near the saturation condition. Since this exhaust steam is normally used for feed water heating, inherent losses develop if it is superheated and at this point the gains from higher initial steam temperature begin to diminish. Fig. 5 has been prepared to illustrate this point and shows the inlet temperature which will produce saturation at 47 lb. per sq. in. absolute exhaust pressure for a 200 horse-power turbine and also for a 1,000 horse-power unit. From these results it would appear that 50 to 100 deg. F. initial superheat is all that can be justified for such units at the present time.

There is ample experience available with the de-superheating equipment needed to meet the above requirements and there appears to be no reason why it should not be used wherever its cost can be justified.

For auxiliary drive turbines whose power and load factor conditions are similar to auxiliary generator turbines, it may be desirable to consider the use of condensing units.

For such units and turbo-generator drives the maximum inlet steam temperature cannot be selected on the same basis as for non-condensing units, i.e. a condition of zero exhaust superheat, since no serious inherent losses are involved in the condensation of superheated steam. Therefore, no general basis can be offered for the selection of operating steam temperature for condensing type auxiliary turbines. Economic studies are required to suit the particular conditions and it may be noted that current practice appears to favour the use of full steam temperature for condensing type auxiliary turbines. However, the authors are doubtful that this can be justified for temperatures exceeding 900 deg. F.

There appears to be little basis for reducing the design pressure for auxiliary turbine units of either type below the main unit pressure unless this exceeds 900 lb. per sq. in. gauge. This is because there appears to be little cost saving when the necessary reducing valves are taken into account. Also the use of reducing valves in the steam supply line to a vital auxiliary may be considered to add an unwarranted hazard. On the other hand, de-superheating does not represent a direct thermodynamic loss and the necessary equipment has established its reliability in both land and marine power plant service.

It has been indicated that gains result from using a higher exhaust pressure for non-condensing turbines. It should be obvious that the converse holds as long as this exhaust is to be used for feed heating, since a lower exhaust pressure will reduce the amount of bleeding from the lowest main turbine extraction point where a greater amount of useful energy has been obtained before the steam is extracted from the turbine.

One very interesting development in feed pump drive for steam pressures exceeding about 600 lb. per sq. in. is the double compounded unit[11]. Two high speed pump impellers are operated in series but without physical connexion between the shafts. Each is driven by a separate turbine wheel and steam flows through these turbines in

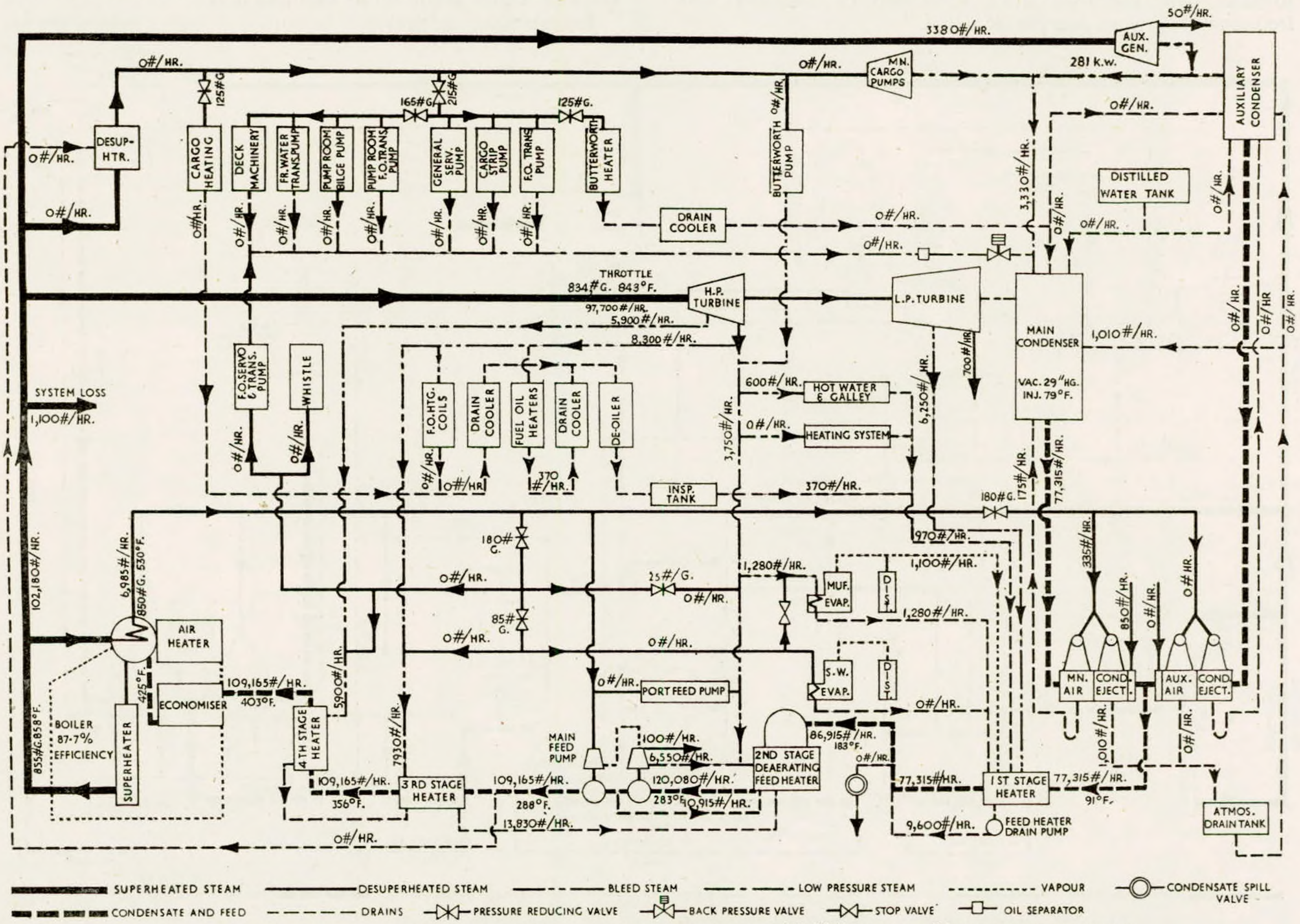


FIG. 6—HEAT CYCLE DIAGRAM—TRIAL PERFORMANCE OF 12,500 S.H.P. ESSO SUPERTANKERS
13,930 S.H.P. ; FUEL CONSUMPTION 0·498LB. PER S.H.P. HR.

series but again there is no physical connexion between the turbine rotor shafts. The high-pressure pump is driven by the low pressure turbine and the speed of this unit is adequately controlled by the hydraulic connexion to the other set. Two-stage turbine performance is obtained and the small size of the units results in a very compact arrangement. Experience to date with these units is understood to be highly satisfactory.

Two other practical alternatives remain to be considered for auxiliary unit drives and these are the use of motors or of direct attachment to the main units. Motor-driven feed pumps are widely used in land power plants but have been shown to be too costly to warrant marine use unless the specific conditions are such that no increase in generator capacity is required. The above condition would appear to be found in a tanker with motor driven cargo pumps or in a moderately powered vessel with large electric load demands in port. Attached feed pumps appear very unattractive because of manoeuvring conditions where momentary boiler feed requirements have no relation to main turbine output and speed.

Generators attached to the main reduction gear unit have been used by one owner for a number of years(24) and this practice has been continued on his most recent vessels.(13)

The obvious inducement of such installations is that electric power is generated at about the same efficiency as shaft horse-power. It is necessary, however, with geared turbine propulsion to consider the standby losses of an idling turbo-generator set which must be instantly available to take over the load upon reduction of attached unit frequency below an acceptable value.

Definite solutions to problems of the type discussed above can be obtained only for a specific set of conditions. Reference levels for such studies are offered on the cycle diagrams shown in Figs. 6 and 7. The first diagram presents an average of the official results obtained on the economy trials of three of a recent class of American tankers(11). The second diagram is generally based on the first and shows what might be expected in a larger powered twin-screw passenger and cargo liner using somewhat different steam conditions.

The heat cycle diagram, Fig. 7, for a twin-screw passenger liner illustrates another point concerning maximum utilization of thermal gains due to higher steam conditions. Bled steam is used as the primary source for supplying a number of the hotel service requirements. However, owing to the general use of non-ferrous materials in the ship's steam heating system, it is not possible to use bled steam directly for this purpose. Therefore, as many services as possible have been combined and supplied from a saturated steam generator supplied with bled steam.

The exhaust and condensate returned from services which are unlikely to result in contamination are returned to the circuit to best advantage.

Selection of Materials and Working Stresses for High Temperature

Basic Information

Before proceeding to a consideration of the specific problems occurring in superheaters, piping and turbine parts exposed to high temperature, it is necessary to summarize the basic problems common to all these items.

Returning to the previous definition of higher temperatures,

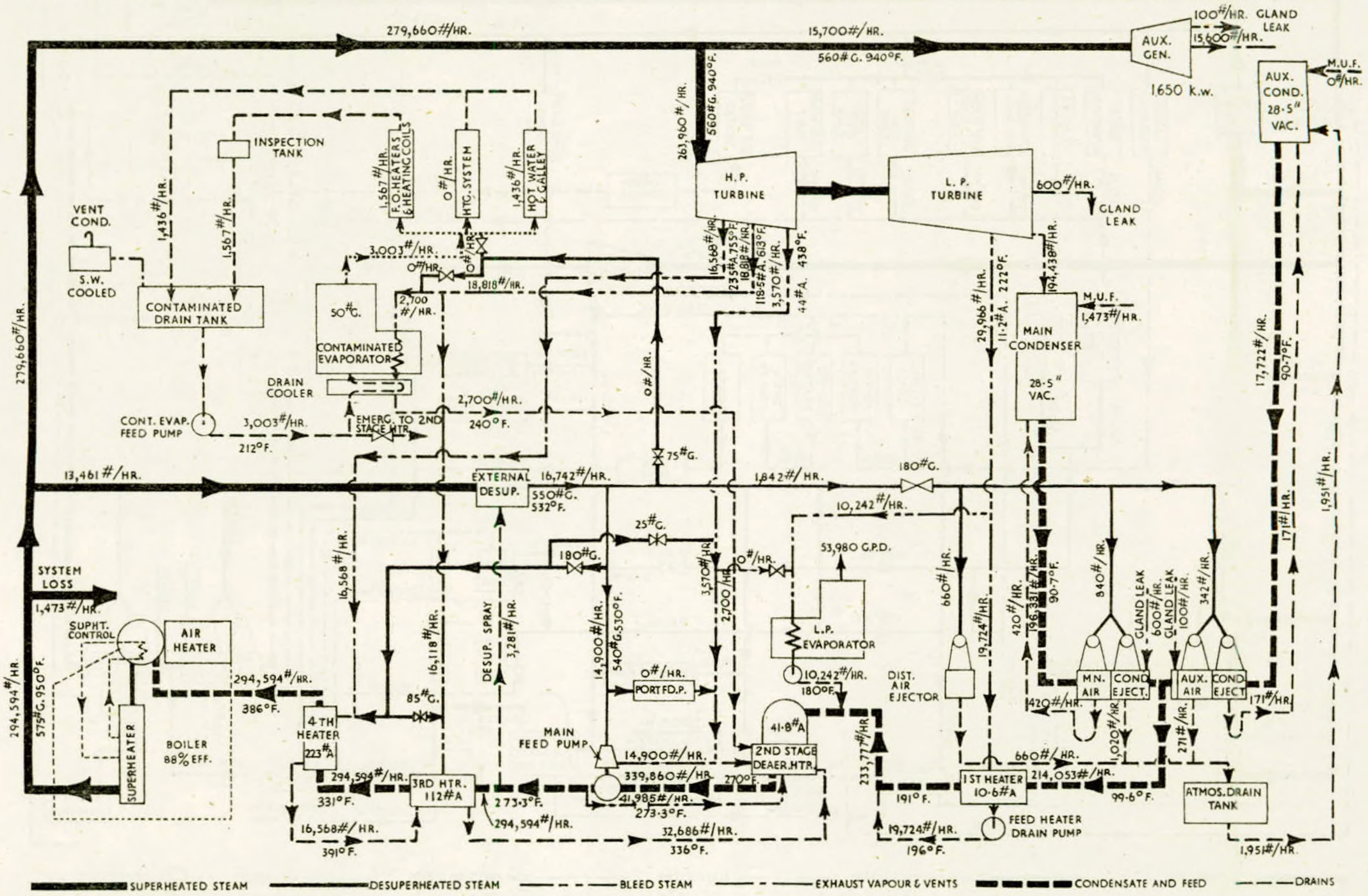


FIG. 7.—HEAT CYCLE DIAGRAM—PROPOSED ARRANGEMENT FOR 40,000 S.H.P. TWIN SCREW PASSENGER LINER
40,000 S.H.P; FUEL CONSUMPTION 0·511 LB. PER S.H.P. HR.

the most important new fact to be considered at these temperatures is the slow permanent deformation of materials under stress. This is the well-known phenomenon of "creep" which first assumes importance for ordinary materials like mild carbon steel at a temperature of about 800 deg. F. and for stresses above about 5,000 lb. per sq. in.

The literature on this subject is vast and most of it highly technical. For the present purpose, discussion will be limited mainly to descriptive statements following earlier sources [25], [26].

The paramount fact to realize about creep deformation is that, for a particular material, the process is a function of time, temperature and stress and that it proceeds at different rates according to the effects of all three quantities. This situation is represented in a qualitative way by the curves in Fig. 8 where total creep deformation is plotted as a function of time and the effects of various stresses and temperatures are shown.

Where long term performance is of paramount interest, the primary stage of creep shown in the figure can often be ignored and emphasis placed upon the secondary or constant creep rate stage. The tertiary stage which ultimately leads to rupture is also shown in the figure and, if this is of short duration before final rupture, it is obviously necessary to select design stresses such that this stage will not be reached during the useful life of the equipment.

A great deal of research effort has been spent and continues to be spent upon careful measurements of the second and third stage creep rates for various materials. It has been learned that several factors aside from alloy content influence the results obtained; heat treatment, method of forming and melting practice appear to be the most important of these. The effects of different testing procedures have been largely eliminated by standardization of these techniques[27].

A large amount of creep data has been obtained from tests usually lasting only a few thousand hours. For a wide range of materials and stresses, approximately constant creep rates are obtained in tests of such duration. It is also found that for moderate deformations a log-log plot of stress versus creep rate from such tests results in approximately straight lines as shown in Fig. 9.

The most commonly reported creep strength values are those sometimes described as the stress for 1 per cent creep

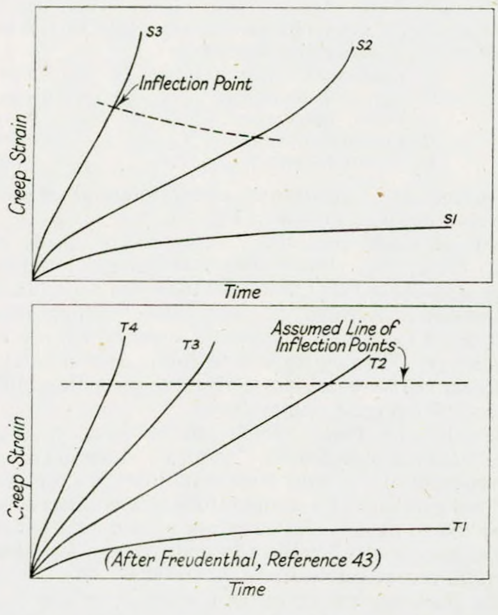


FIG. 8.—TOTAL CREEP VERSUS TIME ILLUSTRATING EFFECTS OF STRESS AND TEMPERATURE

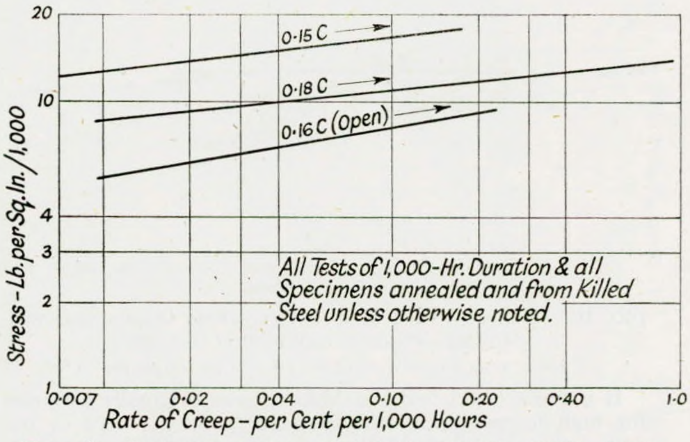


FIG. 9.—LOG-LOG PLOT OF STRESS VERSUS CREEP RATE. (FROM A.S.M.E.-A.S.T.M. 1938 CREEP COMPILATION REFERENCE 92)

in 10,000 or 100,000 hours, when what is actually referred to is the stress for 0.1 per cent or 0.01 per cent creep in 1,000 hours as determined from the above described log-log plot of constant creep rates determined in tests of a few thousand hours duration.

The results of creep tests carried out for many thousands of hours[28], [29], [30] and [31], indicate that the creep rates established in tests lasting a few thousand hours can be extended safely to longer periods only if the range of temperatures and stresses is such that the material is not undergoing structural transformation or surface deterioration.

The stress-time value at which fracture occurs at a particular temperature has assumed increasing importance as a basis for determining safe working stresses for long life requirements. This applies particularly in the case of structures which can, without difficulty, withstand creep deformations of one per cent or more.

The value of such tests also depends on the ability to safely extrapolate the results of data obtained in practical test periods to much longer times corresponding to the required life of the equipment. It was soon found that log-log plotting of fracture stress versus time resulted in straight line relationships such as those shown in Fig. 10[32]. As in the case of creep tests, these results can safely be extended to longer times only if there is assurance that intergranular or surface oxidation or structural transformation effects are unimportant[33]. Breaks in the stress rupture curves such as appear in this figure indicate that changes of this nature are taking place in the material under test.

Similar changes will be found in the slope of stress-time curves for a constant creep rate and, in addition, a marked decrease is observed in the amount of elongation at fracture. These indications all serve as a warning that the stress-temperature conditions under which they occur are probably unsuitable for long term operation. Discussion of the metallurgical changes associated with these effects will be given later with reference to specific materials.

Working Stresses and Factors of Safety

For a number of years the permissible hoop stresses in pressure vessels and piping representing U.S. practice have been set down in various portions of the A.S.M.E. Boiler Construction Code. These tables give the allowable stresses for various specification standard materials at various temperatures.

The relations between the A.S.M.E. 1949 Code stress and representative creep and rupture strength values for a few pressure vessel and piping materials of current interest are summarized in Table III. The strength properties shown in this table were taken from reference (34) and from other sources as noted in the table.

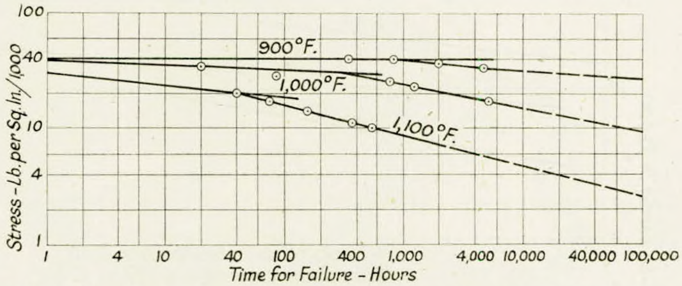


FIG. 10.—LOG-LOG PLOT OF RUPTURE STRESS VERSUS TIME FOR FAILURE—CARBON MOLYBDENUM STEEL

⊙ INDICATES INTERGRANULAR FRACTURE FROM REF. (45)

It is understood that the code stresses originally assigned for high temperatures corresponded to 80 per cent of the creep strength on the 0.01 per cent per 1,000 hours basis. Wartime provisions, which were later accorded regular status, raised the allowable stresses by 25 per cent thus making them equal to the above creep strength values[35].

A comparison of extrapolated 100,000 hour rupture strength values with the code stresses indicates that in most cases a "factor of safety" of at least two is available on a nominal stress basis.

It should be emphasized that the strength values in the table are averages and, as previously mentioned, wide variations may result from differences in melting practice, method of forming and heat treatment.

Certain of these A.S.M.E. Code stress values have recently been adopted by the American Bureau of Shipping and the United States Coast Guard Bureau of Marine Inspection mainly as the result of the recommendations of an Advisory Panel sponsored by the former agency[10].

It should be noted that lower temperature limitations were assigned to the various materials for marine use than appear in the above A.S.M.E. Code.

The effect of these temperature limitations is to require the use of carbon molybdenum alloy steel for temperatures exceeding 775 deg. F. and the addition of chromium as well as molybdenum for steam temperatures exceeding 875 deg. F. This temperature limitation for carbon molybdenum steel is somewhat more conservative than the recommendations of the Joint Committee on Graphitization sponsored by the American Central Station Power Industry[36]. Some disagreement with this limitation has been voiced as there are numerous instances of successful use of this alloy at temperatures exceeding 900 deg. F. including some marine applications.

On the other hand there are equally strong opinions in favour of lower temperature limits for both plain carbon and carbon molybdenum steel. In any event, the authors feel that for temperatures exceeding 875 deg. F. the extra cost of the alloys with added chromium content would appear to be well worth the more positive assurance of freedom from graphitization[37]. Other important considerations regarding safe temperature limits for these materials are discussed later.

Hoop Stress Formulae. The comparisons made above are between the primary circumferential tension or hoop stress in piping and pressure vessels and the results of tensile creep and rupture tests at the working temperature. They are subject to further qualification because the hoop stress varies throughout the pipe wall and various formulæ are used for its evaluation.

The most important working formulæ are, first the "common" or inside diameter formula, and, second Boardman's modified Lamé formula as now incorporated in the A.S.M.E. codes (with the addition of a corrosion allowance). A comparison of the values obtained from these two formulæ with Lamé's theoretically exact expressions

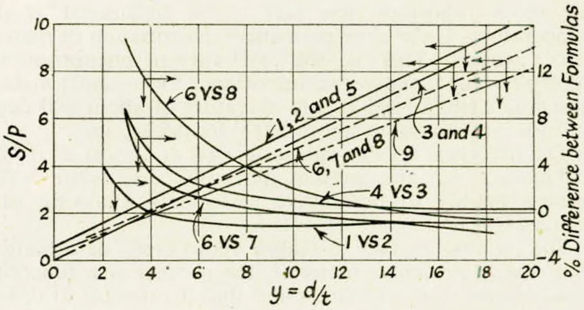


FIG. 11.—COMPARISON OF LOOP STRESS FORMULAE

(1) LAMÉ FORMULA FOR CIRCUMFERENTIAL STRESS AT INNER WALL

$$\frac{S}{P} = \frac{D^2 + d^2}{D^2 - d^2} = \frac{1}{2}\left(y + 1 + \frac{1}{y+1}\right)$$

(2) BOARDMAN APPROXIMATION TO LAMÉ FORMULA

$$\frac{S}{P} = \frac{D}{2t} - 0{\cdot}4 = \frac{1}{2}\left(y + 1 + \frac{1}{5}\right) = 3 + 0{\cdot}6$$

(3) SO-CALLED "COMMON" FORMULA FOR AVERAGE CIRCUMFERENTIAL STRESS IN PIPE

$$\frac{S}{P} = \frac{D}{2t} - 1 = \frac{y}{2}$$

(4) MEAN OF LAMÉ FORMULÆ FOR INNER AND OUTER WALL STRESS

$$\frac{S}{P} = \frac{1}{3}\left[\frac{(1 + 2/y)^{\frac{1}{3}}}{(1 + 2/y)^{\frac{1}{3}} - 1}\right] = 1{\cdot}02\left[\ \mathrm{I}\ \right], \text{approximately}$$

(5) SO-CALLED BAILEY-NADAI FORMULA FROM REF. (38)

$$\frac{S}{P} = \frac{D^2 + 3d^2}{D^2 - d^2} = \tfrac{1}{2}\left(y + \frac{1}{y+1}\right) = \mathrm{I} - 0{\cdot}5$$

(6) BAILEY'S 1935 FORMULA REF. (40)

$$\frac{S}{P} = 0{\cdot}2783\left[\frac{(1 + 2/y)^{\frac{1}{3}}}{(1 + 2/y)^{\frac{1}{3}} - 1}\right] \quad \begin{cases} m=2 \\ n=6 \end{cases}$$

(7) BELIAEV AND SINITSKI FORMULA FOR PLASTIC STATE OVER ENTIRE WALL REF. (43)

$$\frac{S}{P} = \frac{\sqrt{3}}{2}\left[\frac{1}{\log_e (1 + 2/y)}\right]$$

(8) NADAI'S 1937 FORMULA REF. (94)

$$\frac{S}{P} = \frac{1}{2\sqrt{3}}\left[\frac{1}{(1 + 2/y)^{\frac{1}{3}} - 1}\right] \quad (n = 6)$$

(9) RANKIN FORMULA BASED ON SHEARING RUPTURE STRESS REF. (39)

$$\frac{S}{P} = \frac{1}{0{\cdot}65 \times 4}(y + 1)$$

* THESE FORMULÆ ALSO PLOTTED AS PER CENT DIFFERENCE FROM OTHER CLOSELY RELATED FORMULÆ

S	CIRCUMFERENTIAL STRESS	LB. PER SQ. IN.
P	INTERNAL PRESSURE	LB. PER SQ. IN.
t	WALL THICKNESS	INCH
D	OUTSIDE DIAMETER	INCH
d	INSIDE DIAMETER	INCH

for average and maximum circumferential stress under elastic conditions is shown in Fig. 11.

It will be noted that the "common" or inside diameter formula (3) in Fig. 11 is a very good approximation to the average stress based on Lamé's solution, formula (4), whereas the Boardman formula (2) very closely approximates to Lamé's result (1) for the maximum stress which occurs at the inner surface. The agreement in both cases is so close that separate curves were not plotted but the percentage differences are shown for precise comparison.

The Advisory Panel which drafted the interim guide referred to above selected the "common" formula (3) as being most representative of long term conditions in a pipe subjected to internal pressure at a temperature where creep effects may be expected to modify the variation in circumferential stress. This conclusion is still subject to controversy, as indicated by three quite recent publications [35], [38] and [39].

In the past, several equivalent stress formulæ have been proposed which are based on various failure theories or limiting stress criteria which also take into account the axial

TABLE III

COMPARISON OF CREEP AND RUPTURE STRENGTH WITH ALLOWABLE HOOP STRESS VALUES IN 1949 A.S.M.E. BOILER CODE

(a) *Carbon and Carbon Molybdenum Steel*							
Reference Source for Principal Data							*Miller and Heger*[34]
Material	0·3 max. carbon steel type A 106 grade B						
Temperature, deg. F.	750	800	850	900	950	1,000	
Allowable hoop stress, lb. per sq. in.	10,400	9,100	7,400	5,600	3,800	2,000	
Stress for 0·01 per cent creep/1,000 hr., lb. per sq. in.		13,000	10,300	7,500	5,400	3,400	
Stress for 0·1 per cent creep/1,000 hr., lb. per sq. in.		19,000	14,200	10,800	7,200	5,000	
Stress for rupture in 100,000 hr., lb. per sq. in.[1]		16,600	13,200	10,000	6,800	4,600	
Stress for beginning of 3rd stage creep after 100,000 hr., lb. per sq. in.			12,200[2]			<3,400[3]	
Elongation at 100,000 hr. and above stress, per cent[5]			<1			<1	
Interim guide recommended limiting temperature, deg. F.	775						
Material	Carbon molybdenum steel type A 206 grade P1						
Temperature, deg. F.	750 and less	800	850	900	950	1,000	1,050
Allowable hoop stress, lb. per sq. in.	11,000	10,750	10,500	10,000	8,000	5,000	
Stress for 0·01 per cent creep/1,000 hr., lb. per sq. in.		20,700	16,600	12,600	9,200	6,500	4,000
Stress for 0·1 per cent creep/1,000 hr., lb. per sq. in.		29,000	24,000	19,500	15,000	10,500	6,500
Stress for rupture in 100,000 hr., lb. per sq. in.[1]			46,500	33,000	21,500	10,900	4,300
Stress for beginning of 3rd stage creep after 100,000 hr., lb. per sq. in.						6,500[4]	
Elongation at 100,000 hr. and above stress, per cent[5]						<1	
Interim guide recommended limiting temperature, deg. F.			875				
(b) *Low Chromium Alloy Steels*							
Material	1¼ chromium ½ molybdenum steel type A 158 grade P11						
Temperature, deg. F.	800	850	900	950	1,000	1,050	1,100
Allowable hoop stress, lb. per sq. in.	11,800	11,200	10,000	8,000	5,850	3,850	2,200
Stress for 0·01 per cent creep/1,000 hr., lb. per sq. in.	20,500	17,000	14,200	11,200	8,500	6,000	3,700
Stress for 0·1 per cent creep/1,000 hr., lb. per sq. in.	28,800	23,500	18,800	14,500	10,800	8,000	5,500
Stress for rupture in 100,000 hr., lb. per sq. in.[1]				(16,000)	11,700	8,500	5,700
Stress for beginning of 3rd stage creep after 100,000 hr., lb. per sq. in.					9,400[6]		
Elongation at 100,000 hr. and above stress, per cent[5]					<1		
Interim guide recommended limiting temperature, deg. F.						1,050	
Material	2¼ chromium-molybdenum steel type A 213 grade T22						
Temperature, deg. F.	800	850	900	950	1,000	1,050	1,100
Allowable hoop stress, lb. per sq. in.	11,800	11,200	10,000	8,250	6,250	4,800	3,700
Stress for 0·01 per cent creep/1,000 hr., lb. per sq. in.			22,000	12,000	7,200	5,500	4,500
Stress for 0·1 per cent creep/1,000 hr., lb. per sq. in.			40,500	19,000	13,000	9,400	7,000
Stress for rupture in 100,000 hr., lb. per sq. in.[1]				(18,000)	14,000	12,200	7,200
Interim guide recommended limiting temperature, deg. F.						1,050	
(c) *Higher Chromium and Austenitic Alloy Steels*							
Material	9 chromium molybdenum steel type A 213 grade T9						
Temperature, deg. F.	900	950	1,000	1,050	1,100		
Allowable hoop stress, lb. per sq. in.[7]	10,000	8,000	5,850	3,850	2,200		
Stress for ·01 per cent creep/1,000 hr., lb. per sq. in.	25,000	16,000	8,500	5,300	3,300		
Stress for 0·1 per cent creep/1,000 hr., lb. per sq. in.	33,400	19,000	12,000	8,700	4,800		
Stress for rupture in 100,000 hr., lb. per sq. in.[1]					11,500		
Material	18 chromium 18 nickel Columbium Stabilized type A 213 grade TP 347						
							6,000
Temperature, deg. F.	1,000	1,050	1,100	1,150	1,200		
Allowable hoop stress, lb. per sq. in.[7]	10,000	8,000	6,000	4,600	3,600		
Stress for 0·01 per cent creep/1,000 hr., lb. per sq. in.	(28,000)[8]		14,200	10,000	6,500		
Stress for 0·1 per cent creep/1,000 hr., lb. per sq. in.	26,000	21,000	16,800	13,500	10,000		
Stress for rupture in 100,000 hr., lb. per sq. in.[1]	(27,000)[8]	(19,500)[8]	14,400	10,400	6,700		

Notes:

(1) Extrapolated from data in reference (34)
(2) Rupture stress minus 1,000 lb. per sq. in.—from reference (93)
(3) Extrapolated from data in reference (28)
(4) Extrapolated from data in reference (44)
(5) Extrapolated as shown in figures 12a and 12b
(6) Extrapolated from data in reference (47)
(7) From A.S.M.E. Boiler Code 1949
(8) From reference (30)

and radial stresses resulting from internal pressure. A number of these formulæ are based on failure theories which have since been discredited even in the elastic range, and others are found to be inapplicable in the plastic range where creep effects are of importance. These latter have been disposed of rather effectively by Burrows and Buxton in reference (38).

However, the question is far from settled as may be seen from several proposed equivalent stress formulæ, (5) to (9) inclusive, shown in Fig. 11 which are all based on either plastic flow or rupture considerations.

It will be noted that the so-called Bailey-Nadai criterion, formula [5] from reference (38), indicates slightly higher stresses than the values obtained from the Lamé formula (1) for elastic inner wall stress. This result is at variance with the original conclusions, formula [6], reported by Bailey in his classic study presented in 1935 [40] and with other studies [41], [42], [93] made at about the same time. The discrepancy appears to be due to the use by Buxton and Burrows of shear stress at the inner wall as a limiting criterion instead of the inner wall tangential creep rate used by Bailey and others. Nadai [93], [94] still prefers to use the shear distortion energy criterion alone and evaluates this at the outer wall where it has the largest value under creep conditions. His results are also shown in Fig. 11 as formula [8] but numerically are not greatly different from Bailey's formula which is No. [6] of the figure.

The authors have found that a formula intermediate to the Bailey and Nadai results is obtained if the shear distortion energy criterion is applied to stress conditions at the inner wall of the pipe.

Another creep stress criterion, formula [7] of Fig. 11, taken from reference (43), page 430, is based on the assumption that creep rates eventually produce a state of completely plastic flow in the cylinder.

Rankin[39] proposes a criterion based on shearing rupture strength which is shown as formula [9] in the figure. A more complete discussion of these various stress criteria is given in Appendix IV.

In the normal range of thickness ratios (d/t exceeding 7) the last four criteria result in permissible wall thickness less than are obtained using the common or average stress formula [3]. For this range the common formula is thus seen to be conservative in relation to the more tenable of the current creep and rupture strength interpretations. A more rational selection of the most representative stress formula does not appear possible until data such as the tubular stress-rupture tests mentioned in reference (30) become available.

Metallurgical Significance of Stress Margins. The factors of safety or stress margins indicated by the data of Table III may appear quite startling in view of the customary room temperature factors of safety of four based on tensile strength and two or more based on yield strength. It must be realized, however, that a numerical factor of safety in relation to a rupture strength value for 100,000 hours of operation has no intrinsic significance whatever.

It might be better to consider the margin of safety expressed in relation to the expected service life, and on this basis the time ratio may be from 5 to 20 or more depending on the slope of the stress-rupture curve. This must be qualified by the extent to which the factor of safety based on calculated stresses must be considered as a factor of ignorance because of lack of precise knowledge of the actual stresses.

In order to visualize properly the probable margin of safety, it is necessary to consider the entire stress-creep to rupture history and also the type of failure of a particular material at its designed temperature. This point is illustrated by the two design charts shown in Fig. 12.

The upper curves, Fig. 12(a), show the behaviour of a normalized carbon molybdenum steel at 1,000 deg. F.[44]. It will be noted that stresses which permit longer times to

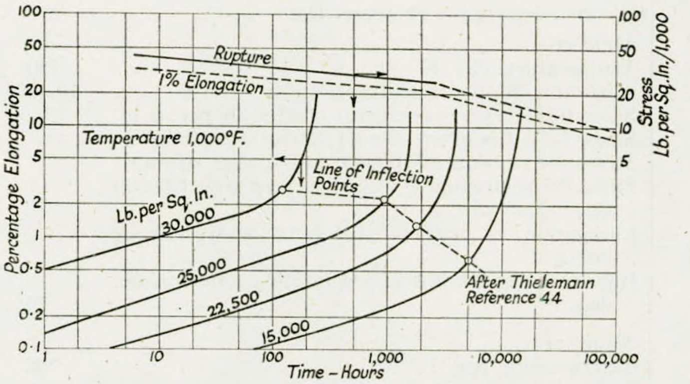


FIG. 12(*a*).—CREEP TO RUPTURE DESIGN CHART FOR CARBON ½ MOLYBDENUM STEEL AT 1,000 DEG. F.

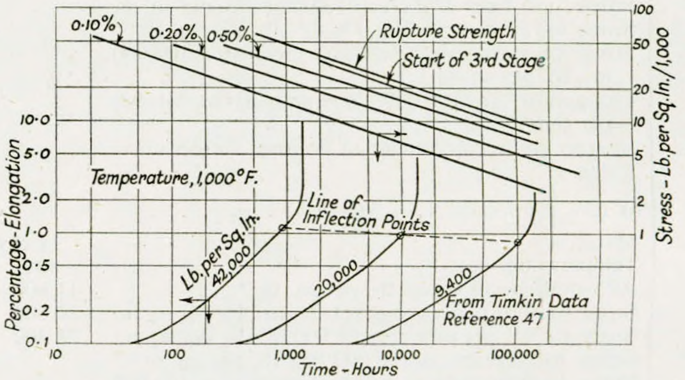


FIG. 12(*b*).—CREEP TO RUPTURE DESIGN CHART FOR 1¼ CR. ½ MO. STEEL AT 1,000 DEG. F.

fracture result in a transition to the accelerated stage of creep at increasingly lower values of total deformation.

As stated in the original article[44], this particular alloy had a relatively high manganese content which is believed to account for the comparatively high values of elongation at rupture. Other tests of this alloy with normal manganese content resulted in much lower elongations at fracture,[45] particularly in the longer time tests.

Examination of fractured specimens which failed with low ductility shows that failure was of the intergranular type associated with selective oxidation and crack formation at the surface. The occurrence of this type of failure is usually evidenced by a break in the stress-rupture curve [46] such as appears in Fig. 12(a).

Since oxidation is essentially a process which proceeds with time, its effects would be expected to be even more serious at lower stresses where the fracture times are longer.

The lower curve, Fig. 12(b), shows a similar stress-creep to rupture history for a 1¼ per cent chromium—½ per cent molybdenum alloy at 1,000 deg. F.[47]. In this case it will be noted that, although accelerated creep occurs at deformations of about 1 per cent, there is no marked tendency for the transition point value to drop off at longer times and lower stresses.

Examination of the fractured specimens showed that all failures were transcrystalline and accompanied by considerable grain distortion[48]. This results in greater total elongations at long fracture times than were observed in the former material. In fact, there appears to be no definite tendency for a reduction in the elongation at fracture of this material even for tests requiring 15,000 to 18,000 hours[49].

For a material which is to be used at a temperature which will result in ductile long time fracture characteristics of the type just described, it would appear that a rupture stress criterion might properly be based on the stresses existing in a fully plastic condition since such a state would necessarily be realized long before rupture could take place.

Since for a particular material the transition between the two types of behaviour described above is primarily a function of temperature, it is possible to restrict the use of any particular material to a temperature range where the latter characteristics are known to prevail. This criterion appears to be fulfilled within the above-mentioned temperature limits for carbon and carbon molybdenum steel.

It may be of interest to note at this point that the probable explanation for the marked improvement in high temperature performance due to a moderate chromium addition is that the formation of chromium carbide further inhibits the spheroidization process (50), (51), (52), which is known to continue in carbon molybdenum steels although at a slower rate than in plain carbon steel. Formation of such complex carbides in the grain boundaries appears to block or retard the progress of intergranular oxidation and cracking until appreciable amounts of crystalline slip, distortion and fragmentation have occurred.

Other Considerations. Hidden margins in the code tolerances for thinning of bends, corrosion allowances and manufacturing and procurement practices have recently been very ably disclosed(35), (53). It may be said that the recommendations of the interim guide have made a great step forward in eliminating some of these hidden margins from mandatory requirements.

A realistic allowance for corrosion would still appear to be desirable but, as will be shown later, it is doubtful that any margin need be allowed for thinning of bends if the bending procedure is carefully controlled. It does not seem possible to reduce manufacturing thickness tolerances for pipe and tubing without paying an excessive premium. Also, it is quite difficult at the present time in the United States to order tubing to exact rather than schedule thickness.

Additional piping and superheater tubing stresses of a more localized nature result from thermal expansion and from construction irregularities such as out of roundness in bends, abrupt changes of stiffness in valves and fittings, etc. Vibratory stresses may also result from localized resonance with propeller impulses or other sources of vibration. As will be shown later, all of these factors must be considered in high temperature applications and every effort made to minimize their effects.

Alloy Selection for Specific Applications

Superheaters. The highest metal temperature in any steam power plant system usually occurs in the final pass of the superheater. Figs. 13 and 14 have been prepared to illustrate the general relations between steam temperature, superheater location (i.e. gas temperature), rate of heat transfer and maximum metal temperature. In Fig. 13 the heat transfer rate per square foot of surface and also the steam flow per square foot of surface are related to the desired steam temperature and the superheater location as defined by the gas inlet temperature.

It should be emphasized that these curves are not intended to be used for design purposes. The many simplifying and restrictive assumptions necessary to prepare such curves are fully described in Appendix III.

The maximum metal temperature is shown in Fig. 14 in relation to steam temperature, gas inlet temperature and heat transfer rate. Again the results should be considered as illustrative only due to the necessary assumptions involved which are also described in Appendix III.

A study of these figures will show that maximum economy of surface is obtained by locating the superheater in a higher

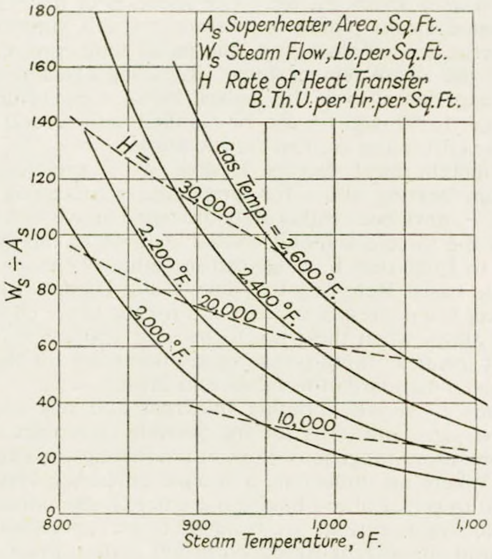


FIG. 13.—STEAM FLOW AND HEAT TRANSFER RATE PER SQ. FT. OF SUPERHEATER SURFACE AS INFLUENCED BY STEAM TEMPERATURE AND GAS INLET TEMPERATURE

gas temperature zone, but higher metal temperatures will be obtained which soon require the use of higher grade alloys.

Bailey in a very thorough study previously mentioned(40) indicates that thermal stresses due to the temperature gradient of heat transfer in a superheater tube may be unimportant but that the influence of temperature on the creep rate cannot be ignored. The American codes require that allowable stresses be based on the average metal temperature.

A compromise must be found between additional surface with attendant weight and space requirements and the increased cost and additional design and fabrication problems associated with more highly alloyed materials.

The U.S. central station power industry has now had over ten years' experience with steam plants operating at 900 to 950 deg. F. at the superheater outlet and many plants in the last few years have been put in operation with steam temperatures of 1,000 deg. F. to 1,050 deg. F. This experience, when applied to current and prospective marine installations with special weight and space requirements, indicates that above about 950 deg. F. steam temperature it is necessary to use austenitic materials in the final pass of the superheater. For lower temperatures ferritic alloys in the 2 to 9 per cent chromium range appear to be entirely satisfactory for most applications.

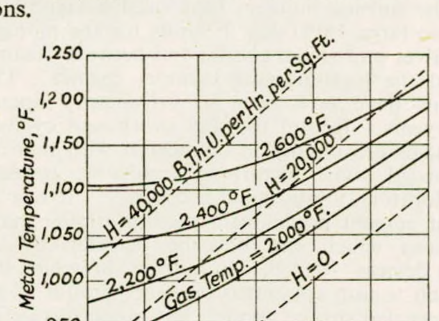


FIG. 14.—MAXIMUM METAL TEMPERATURE AS A FUNCTION OF STEAM TEMPERATURE, GAS INLET TEMPERATURE AND RATE OF HEAT TRANSFER

Superheated Steam Piping. The selection of materials for superheated steam piping is in some respects simpler than the superheater problem since the metal temperature is the same as the steam temperature. Referring again to power station experience, even the highest steam temperature now in use, i.e. 1,050 deg. F., can be handled with several of the moderate chromium content ferritic alloys.

The metallurgical factors leading to a preference for chromium bearing alloys for temperatures exceeding about 875 deg. F. have been rather fully discussed above. A choice between the three commonly available alloys of this type for the 900 to 1,050 deg. F. range can sometimes be made on an economic basis. Referring to Tables I and III it will be seen that lower hoop stresses are required for the lower chromium content alloys when these are to be used above 950 deg. F. and that lower working pressures are necessary for the same dimensional standards for valves and fittings.

A study of required piping thickness and resulting pipe and valve sizes and costs for the possible alloy selections in this temperature range may work out in favour of a particular alloy. Where no important economic difference appears, it is logical to prefer alloys having a greater chromium content. The following selections are believed to be representative of upper limits in current American power station practice:

Up to 775 deg. F.	Plain carbon steel
Over 775 deg. F. to 875 deg. F.	Carbon —½ Molybdenum steel
Over 875 deg. F. to 950 deg. F.	1 Chromium —½ Molybdenum steel
950 deg. F. to 1,000 deg. F.	1¼ Chromium—½ Molybdenum steel
Over 975 deg. F. to 1,050 deg. F.	2¼ Chromium—1 Molybdenum steel
1,050 deg. F. to 1,100 deg. F.	18-8 Columbium stabilized stainless steel

Referring to Table III it will be noted that a peculiar situation arises with regard to the higher chromium content ferritic type alloys. Despite their reported superior rupture strength properties, these alloys are not permitted higher working stresses.

Another alloy which appears to have unusual long term creep and rupture strength properties is a recent development [39], [52] of the molybdenum-vanadium steel investigated by Glen[54]. The new alloy contains 1 per cent chromium, 1 per cent molybdenum and ¼ per cent vanadium. Further details will be found in the references cited above.

Austenitic materials are now being used for main steam lines for land power plants for 1,050 deg. F. and higher temperatures because of improved creep and rupture stress properties which permit higher stresses and thinner walls. Already the turbine builders have used austenitic steel in a dozen or so large 1,050 deg. F. units for the piping between throttle valves and steam chests, and between steam or valve chests and the high-pressure cylinder casings. The earlier installations have now been in satisfactory operation for over two years. Recent thermal shock and cycling fatigue tests of austenitic piping, in connexion with marine piping, have recorded some disturbing results[55], particularly in some exploratory joints, and in castings. It has also been found that current grades of austenitic piping may contain serious flaws which can best be detected by ultrasonic inspection devices [56], [97]. In fairness, however, the utilization of such testing apparatus on other grades of alloy pipe has also revealed similar though less numerous defects.

Also, the oil refining and chemical industries have had a large amount of satisfactory experience with austenitic piping at high temperature. In summary it may be said that the use of austenitic steam piping is proceeding with the aid of special precautions with regard to welding and inspection, and with some reservations regarding the use of castings.

Combined Thermal and Pressure Stresses. The combined stress problem in steam piping is twofold. It is first necessary to determine the allowable expansion stresses which may be superimposed locally upon the primary hoop stress. The second step is to make sure that this stress is not exceeded in the design.

The interim guide permits a local combined stress equal to twice the allowable hoop stress for materials used at temperatures of 1,000 deg. F. or higher. This rule was restricted at lower temperatures to avoid exceeding the reported short time proportional limit of certain materials. A table of values is published in the guide which establishes a smooth transition between the "two thirds" rule which still is in effect up to 850 deg. F. and the factor of two permitted at 1,000 deg. F. and higher temperatures.

The use of such local combined stress limits may at first be considered hazardous since it apparently permits a factor of safety of one based on the 100,000 hour rupture strength. However, careful examination indicates that the basis is in fact quite sound.

In the first place, the maximum combined or equivalent stress is the value limited and this must include both internal pressure and thermal expansion effects. The results so obtained are usually localized point or line stresses. The thermal expansion components are subject to rapid alleviation due to creep effect, though they may recur in modified form when the pipe is cooled down and again brought up to temperature. Bailey[40] has indicated that creep effects are much more pronounced due to combined bending and hoop stress than for either stress acting alone.

Another feature which greatly reduces concern over thermal expansion stresses is the use of cold pull-up. The interim guide recommends that 100 per cent cold pull-up be used where this can be done without exceeding ⅔ of the yield stress in the cold condition. However, no more than ⅓ credit for cold pull-up may be used in the stress calculations. This is believed to be a wise provision since it is manifestly impossible to allow for the deflexion of anchorages or for relative movements of terminals due to the working of a ship in a seaway.

The best known and most widely used method of arriving at the maximum equivalent stress value is the Mises-Huber-Hencky distortion energy yield criterion (reference (43) page 256). This requirement simply states that the stress deviator or octahedral shearing stress for a multi-axial stress condition shall not exceed one half the permissible maximum combined stress. When this requirement is met, the maximum local shear distortion energy in the pipe has the same value as the shear distortion energy in a simple tension test.

The determination of thermal expansion stresses in steam piping has been the subject of many publications, and numerous ingenious methods have been devised to solve the flexure problems arising in complicated piping arrangements. A variety of these are described in references [57], [58], [59], [60], [61]. Space does not permit any discussion of these nor do the authors have anything new to add on the subject.

It may not be generally appreciated just how difficult and time consuming is the practical process of compromise between preliminary idealized technical piping layouts and the actual arrangement drawing with its multitudinous problems of clearances, anchorages and supports. The authors cannot recall a ship piping arrangement that did not require at least four complete sets of calculations for each pipe section before a satisfactory arrangement was achieved.

Local intensifications of thermal expansion stress also present a serious and continual problem. There has appeared quite recently a new and valuable contribution on the subject of stress intensification factors in elbows and bends[62].

One important case of stress intensification in pipe elbows appears to have escaped treatment in the literature except for one isolated note[63]. Examination of the equilibrium

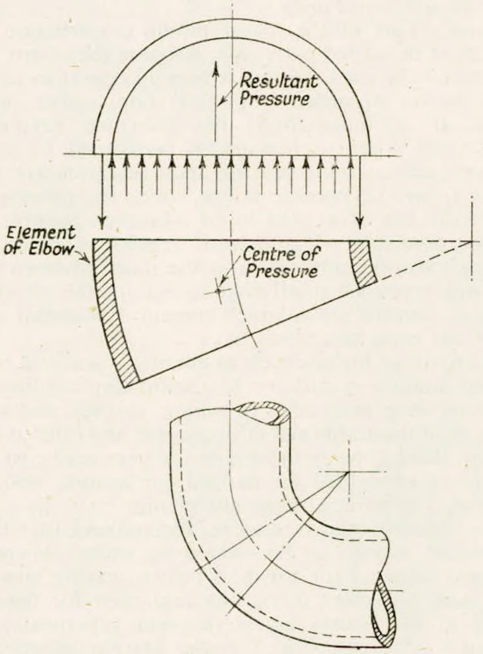


FIG. 15.—BENDING STRESS CONDITIONS IN A SHORT RADIUS ELBOW

conditions in the cross-section of an elbow shows that there is an increase of hoop stress on the inside of the bend which is of major proportions in a short radius elbow such as shown in Fig. 15.

Formulae for the stress conditions in a torus are given in a standard reference text(64). However, elementary analysis by requiring moment equilibrium conditions as shown in the figure yields hoop stress values which were found to be in good agreement with strain gauge measurements on an actual elbow similar to the one illustrated.

In this case the stress increase is general over the inner cross-section and extends for the full length of arc. The authors therefore conclude that such a stress must be considered in relation to limiting hoop stress values for straight pipe.

Further investigation indicates that this problem will arise mainly in forged turns where the method of forming restricts the natural tendency for thickening of the inner wall. In pipe bends of normal proportion which are formed around a mandrel the thickening effect of the bending process is approximately proportional to the increase of stress. This problem is avoided in cast fittings which are normally required to be appreciably thicker than the pipe wall(65).

A further consideration regarding pipe bends is that with proper control, the thinning which normally occurs on the outside of the bend will be of the same magnitude as the decrease of hoop stress. Therefore, the provision of extra design thickness to compensate for thinning appears to be an unnecessary precaution.

Thermal Shock. A great deal of attention has recently been focused on the effects of thermal shock on high temperature steam piping (53), (55). Except for its effect on flanged joints, the results of sudden quenching of superheated steam lines due to boiler water carry-over appear to have received some undue emphasis. Much has been made of the reduction in thermal stress which accompanies the use of thinner wall pipe. However, both theoretical analysis illustrated by Fig. 16 and actual temperature measurements(55) indicate that the benefit of thinner wall pipe applies only to sustained temperature gradients such as occur during starting of boilers

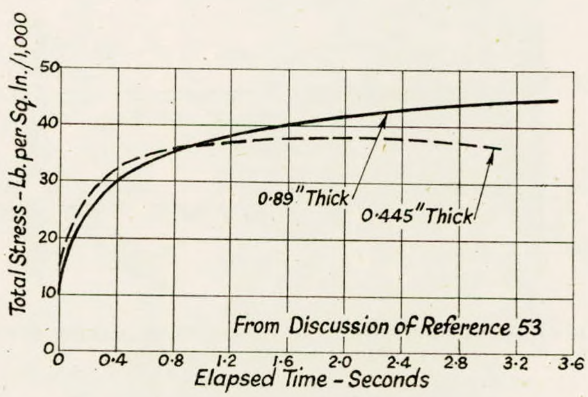


FIG. 16.—TOTAL STRESSES ON INNER SURFACE DUE TO THERMAL SHOCK AND PRESSURE

and warming up of steam lines or during steam temperature swings while operating under load.

Probably the most important lesson obtained from these studies and experiments is that thermal stresses approaching short term yield values may occur repeatedly in high temperature steam lines. It is therefore evident that high endurance strength and high short term fracture strength accompanied by considerable ductility are desirable properties in piping subjected to these conditions.

Vibratory Stresses. The influence of superimposed vibratory stresses in piping and related equipment has received relatively little attention. Some apparently conflicting results for gas turbine type alloys are presented in reference (66), Freudenthal also discusses these results in general terms (reference (43), page 385).

The use of a modified Gerber diagram to relate steady and alternating stresses at high temperature is suggested in references(66) and (35). Some data are presented in reference (67) from German sources which appear to justify the parabolic type of curve suggested in reference (35).

Unfortunately, information on the high temperature cyclic stress limit appears to be completely lacking for the ferritic alloys of current interest for piping. Information on damping capacity and the expected magnitude of exciting forces is also negligible. About the only information available is from personal experience regarding the general order of magnitude of shipboard steam piping vibration amplitudes under various conditions. When such amplitudes become excessive, common sense measures can usually be found which will reduce the amplitude to an acceptable level.

Main Turbines. Turbine casings are principally pressure vessels and materials and working stresses are selected on this basis. The casing also serves to maintain proper clearances between rotating and stationary elements and in this function is adversely affected by creep. However, the pressure-containing service is more critical and pressure stresses are based on a safe relationship with the creep-to-rupture strength of the material. With stresses so established, the creep rate is reduced to the point where adequate clearances can be maintained for the life of the machine.

High temperatures emphasize the need for material that is free of defects. For these reasons it would be desirable to make the casing element which is subject to inlet temperature of steel forgings. However, due to the complexity of shape, steel castings are more practical. It is necessary that these castings be examined by radiographic and magnetic particle examination, and careful re-examination is required after weld repair correction of all defects.

Rotor and blading materials usually operate at temperatures below which the creep-to-rupture strength or limiting total creep may determine the working stress. This is

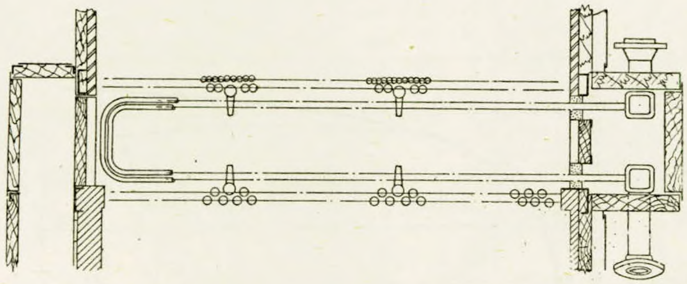


FIG. 17.—VERTICAL HEADER TYPE MULTIPASS MARINE SUPERHEATER WITH SCREENED HEADERS

because there is usually a considerable temperature drop through the first stage nozzles. In this case the working stress is based on the short term yield strength of the material at the operating temperature with the provision of an adequate margin of safety.

Initial blading and rotor stresses and temperatures may, however, be in the range where creep rates are appreciable but other considerations usually have a more important influence on design proportions. The most important of these is the vibratory stresses set up by steam impact in stages which are necessarily of the partial admission type. This problem received wide attention in the literature a few years ago and these taken with two more recent publications (68), (69) would appear to offer fairly complete information on the subject.

The standard low carbon 12 per cent chromium steel blading has so far proved entirely adequate for high temperature design work due to its excellent high temperature properties. There have been some developments in turbine rotor materials with primary emphasis on improved room temperature properties, and a high degree of physical and metallurgical stability. These materials which are complex nickel chromium molybdenum alloys are found to have very high creep strength properties.

Construction and Fabrication Problems

Superheaters

Returning to the superheater, the basic requirement for providing adequate surface is, as would be expected, a compromise between permissible gas and steam velocities based on pressure drop and metal temperature limitations, and a number of practical considerations. Recent experience has indicated the desirability of proper access for external cleaning and for detection of tube seat leakage. Fig. 17, showing the superheater arrangement proposed for some new cargo vessel construction, illustrates a vertical header design which meet these requirements.

Special problems arise where the steam temperature is high enough to require the use of austenitic tubing. Thermal stress problems as well as increased cost are urgent reasons for seeking to avoid the use of austenitic headers and a very satisfactory solution has been developed in stationary plant practice by employing ferritic "safe-ends" on austenitic superheater tubes.

The ferritic header and the safe-ends may be protected from radiation by a screen of insulation and refractory similar to that shown in Fig. 17. The tube seat may, therefore, be a normal expanded and seal welded design which presents no great problem as long as proper preheat and welding techniques are employed.

Welding of the ferritic safe-ends to the austenitic tubing is a rather special problem which has recently received a great deal of study (70), (71), (72). There is now ample experience to indicate that such joints can safely be made in superheater tubing.

Superheated Steam Piping

General. One of the major piping construction features which must be settled early in a design is the extent to which flanges are to be used. Some American operators insist upon flanged valves in order to permit shop, rather than ship, repairs. It is understood that overtime premium pay requirements when such work is performed by the ship's force have considerable bearing upon this situation.

Apart from such considerations, sound engineering practice dictates the use of welded joints wherever possible. Tools are now available which permit repairs in place even of gate valve seating surfaces, and the inconvenience of doing such work appears a small price to pay for the elimination of numerous flanged joints which remain a potential source of serious and even hazardous leaks.

Welding-in of fittings such as elbows, wyes, and headers is accepted practice in addition to aboard ship welding together of various shop assemblies of piping, fittings, and valves.

In spite of the additional radiographic and other inspections required, there appears to be a saving in overall cost from the substitution of welded for flanged connexions, and there is, of course, an appreciable weight saving.

The authors would, therefore, recommend that the entire superheated steam line be welded up except where flanged joints are required for top half turbine casing steam inlets. There does not seem to be any real need for flanged connexions at the boilers unless frequent superheater replacements are anticipated, and power station experience does not indicate that this is likely.

The development of necessary welding and inspection procedures even for the milder air hardening ferritic alloys is a major engineering task. Apart from the strictly welding technique features of optimum current and voltage requirements, actual handling of the electrode, slag removal, etc., there are many engineering problems involved in the choice of electrode compositions and coatings, amount of preheat and post-weld stress relief or heat treatment and in the mechanical design of the weld joints.

Each of the ferritic alloys will be found to have its own particular requirements in the above respects and, while this work may be new to shipyard organizations, there is available a large background of such experience in the firms which do pipe fabrication work for the power station industry. A number of special problems and features coming within the scope of the authors' experience are discussed below.

Structure and Heat Treatment of Welded Piping. The most desirable heat treatment from the standpoint of long term high temperature properties depends not only on the particular material and its melting and manufacturing history but also upon the various fabrication operations including bending and welding. The necessity for practical shop sequences must also be considered.

With regard to melting practice the avoidance of excessive amounts of aluminium in carbon and carbon molybdenum steels is recommended because the fine inherent grain resulting from aluminium additions is more susceptible to graphitization and also produces inferior long term creep and rupture properties. Steels with moderate chromium content are less sensitive to the effects of aluminium de-oxidation since chromium in amounts of 1 per cent or even less, according to some sources(37), appears to completely inhibit graphitization. There is some evidence, however, that aluminium in excess of $\frac{1}{2}$ lb. per ton of melt has an adverse effect on the long time creep strength of low chromium ferritic alloys(54).

In general it is preferable to order ferritic piping in the fully annealed condition to provide maximum initial ductility for cold bending. This treatment also removes most of the effects of cold finishing. The desirability of a recrystallization treatment following either cold or hot bending will be elaborated later. Further consideration of heat treatment must be related to a particular type of alloy, and the following discussion will be limited to two types, namely carbon molybdenum steel and the group of low chromium molybdenum alloys listed in the interim guide(10).

Carbon Molybdenum Steel. It has been rather well established that optimum long term creep and rupture properties for low to moderate carbon grades of this alloy are obtained from a grain coarsening treatment, i.e. normalizing. Tempering after normalizing or annealing instead of normalizing are undesirable in this respect because both processes involve appreciable time at temperatures which promote spheroidization and consequently a material loss of creep resistance(54).

The normalizing treatment may be applied after bending and completion of all sub-assembly welds except where pipe is to be joined to completed valves. Some objections have been raised to normalizing after welding of this material particularly if the welds were made with electrodes having an organic coating which will result in trapped hydrogen. Controversial points of this sort can be decided only by careful welding procedure and qualification tests for each particular case.

Obviously, it is not practicable to apply a furnace normalizing process to completed valves nor is it safe to apply a local normalizing temperature to the welded ends of such valves because of the high temperature differentials which may result.

In this case the next best treatment must be accepted which appears to be a 1,300 deg. F. stress relief anneal maintained for at least four hours. This treatment applied to carbon molybdenum steel which has previously been normalized appears to be effective in preventing graphitization(36). However, there is evidence of some loss in creep resistance and rupture strength, probably because of the spheroidization which takes place during this treatment.(54) Additional insurance against graphitization would appear to be the more important consideration.

Low Chromium Molybdenum Steels. The addition of larger amounts of chromium and molybdenum than the minimum necessary to prevent graphitization tends to produce inherently finer grain structures, even when little or no aluminium is used in the melting process. Also, the spheroidization process is strongly inhibited by chromium additions. As a result it would be expected that the long time high temperature properties of such alloys are less sensitive to heat treatment variables(54).

Normalizing is generally preferred to annealing mainly because of the higher short time rupture strength which provides greater protection from local stress concentration and thermal shock effects. Since the addition of chromium also produces an air hardening characteristic, normalizing must be followed by tempering above the usual stress relief range, and preheat is essential for all welding operations.

A further complication occurs at least in the upper chromium range under consideration where it has been found that normalizing lowers the rupture strength of the weld metal deposit(73).

As a result of these various factors, one recommended fabrication sequence for these alloys consists of a normalize and draw treatment after all bending operations whether cold or hot but before any welding is done. Welds are then stress relieved at about 1,350 deg. F. for one hour per inch or less depending on pipe size and thickness.

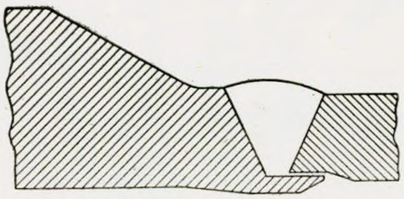


FIG. 18.—TYPICAL BUTT WELD DESIGN FOR PIPE TO CASTING CONNEXION

Welding—Design and Equipment. Two features of butt weld joint design may be mentioned, one is the use of chill rings of adequate thickness and the other is the application of common sense to the proportions of a joint between elements of considerably different thickness. Fig. 18 illustrates both features in a joint between straight pipe and a cast fitting. The chill ring is an integral part of the casting which has a smaller bore than the pipe. In addition, the end of the casting is tapered on the outside to provide a smooth transition which avoids discontinuity stresses and reduces the amount of weld metal to be deposited.

The necessity for preheating chromium alloy welds and for local stress relief of aboard ship welds requires extensive use of induction or resistance heating coils for this purpose. Even for shop work it is necessary to use this equipment for preheat and stress relief when welding pipe to finished valves in order to avoid distortion of valve seating surfaces.

In connexion with radiography it may be mentioned that where a large number of welds must be inspected, the use of cobalt 60 as a radiation source has been found to be fully as effective as radium and personnel hazards are limited to radiation exposure only while damage to a radium capsule presents the additional hazard of radon gas leakage.

The extensive use of cold pull-up in a practically all-welded piping assembly results in a certain number of ship welds which must be performed under restraint. Special fixtures must be used to hold the pulled-up connexions in alignment. However, the most important point is that it is essential to determine by actual test that the material and the fixture are capable of withstanding the expansion and shrinkage stresses of the welding operation.

Cast Fittings. Reference has been made to the use of cast fittings. The technical piping system designer would prefer easy bends to obtain the necessary flexibility but in practice close bends and wyes are inevitable. The choice between forged and cast fittings of ferritic material for this purpose should be mainly an economic one since either type can be expected to have long term high temperature properties comparable to the straight pipe(55).

The use of alloy castings requires careful radiographic and magnetic particle inspection to detect and eliminate defects, but once these have been corrected by repair welding followed by radiography a sound fitting is assured. The alternative for complex fittings such as headers and wyes is to weld together various forged sub-assemblies. The proportions and design of welds involved in such cases often make it evident that better results as well as reduced cost will be obtained by making the item a casting.

Bending Operations. The fabrication of shipboard piping assemblies, of course, involves a large number of bends. As would be expected, power station installations have less need for bends and where required they often occur in very heavy wall pipe due to the higher pressures employed. As a consequence, it is found that stationary piping fabricators generally do not cold form such bends and do not as a rule possess or use the large cold bending machines commonly found in shipyard piping shops.

This situation has resulted in some reluctance to use cold bending procedures for alloy piping for ship use. In one instance these objections were overcome by making destructive and long term high temperature tests of specimens from a sample cold bend of ferritic alloy piping made to a radius of five times the nominal pipe diameter.

Micro-examinations at the point of maximum elongation on the outside of the bend showed that the amount of grain structure orientation due to cold bending was surprisingly small and of random distribution. In fact, there was little difference from the micro-structure of the straight pipe which receives a certain amount of orientation during the final cold pass. A sub-critical stress relief treatment had no effect on the micro-structure as might be expected but all evidences of prior strain were completely removed by a normalize and draw treatment. Such treatment is also important following hot bending because of extensive coarsening at the hot working temperature.

One consideration which has been suggested with regard to cold bending is that the material should not be of a type which is subject to marked strain ageing since permanent strains of the order of 10 per cent occur during the bending process. Cold bending of ferritic piping has, therefore, in some cases been restricted to material of fine inherent grain size. However, there is ample satisfactory experience in cold bending of low aluminium content carbon-molybdenum pipe in the normalized condition which has a coarse grained structure.

It is also important to control the deformations resulting from bending operations. Thinning of the outer wall must be kept within reasonable limits but it is almost equally important to control the development of eccentricity.

With regard to thinning, it should be mentioned that ultrasonic equipment is now available which will measure the wall thickness in pipe bends with an accuracy of about 2 per cent. One of these devices is so simple to operate that it can be left in the hands of skilled shop mechanics once they are instructed in its use(74).

A different and more elaborate type of ultrasonic equipment has been used to examine piping for radial flaws which are not readily detected by other means(75). The authors' experience, which includes sampling inspection of straight ferritic pipe as received from the mill and similar pipe after bending, did not disclose any defects exceeding the 5 per cent tolerance limit mentioned in the reference.

When subjected to internal pressure, out of roundness introduces local bending stresses which add directly to the bursting stress. An analysis of this effect appears in an early American article(76), which also points out that bending stresses of opposite sign occur when an eccentric bend is subject to expansion stresses. However, the authors do not believe that any credit should be taken for this latter effect in view of the 100 per cent cold pull-up basis which is intended to eliminate expansion stresses at the working temperature.

From a practical standpoint it is not too difficult to limit out of roundness to 4 or 5 per cent using either the cold or hot bending technique. This tolerance may result in a superimposed bending stress at one section in the pipe nearly equal to the bursting stress. However, the resulting combined stress is of a localized nature and is in agreement with the maximum combined stress limit previously discussed.

One obvious disadvantage of the hot bending process is the additional loss of wall thickness due to oxidation and scaling in the 1,600 to 2,000 deg. F. forming temperature range. This may be as much as 0·03 inch or more depending on how long the pipe must be held at these temperatures. Another factor is that control of flats or excessive out of roundness is exceedingly difficult once such a tendency develops since these effects are due to local softening from unequal temperature conditions in the pipe while forming. Control of these features in the cold bending process is simply a matter of selecting and maintaining the proper dimensions of the forming dies and mandrels.

Flanged Joints. At the present writing it appears that any ship installation must have a certain minimum number of flanged piping joints because of the difficulties and expense encountered in placing the inlet nozzles in the bottom half casing of turbines requiring multiple nozzle valve control.

In this connexion, the bottom half casing piston valve arrangement described in reference (9) would hardly be acceptable in American practice because of the clearance flow losses.

Design of an efficient high temperature flanged joint is therefore a matter of considerable importance and some new work has recently been done in this direction(77), (78), (79) and (80). There is a long record of satisfactory marine experience with spiral wound asbestos filled steel gaskets for temperatures up to 900 deg. F. and some marine and power station experience at higher temperatures. Nevertheless, enough concern is felt over the behaviour of such gaskets under rapid temperature swings to have resulted in two major alternative developments.

The pressure sealed type of joint described in the above references appears to have excellent possibilities for land use. However, it is not attractive for ship piping due to the large amount of spring back required to open the joint and the difficulty of repairing damaged surfaces.

A modification of the pressure seal principle has been developed for valve bonnet joints which has undergone rigorous tests(55) and appears to have good possibilities for marine use.

The other development described in reference (79) employs a bellows type all-metallic gasket which utilizes the internal pressure to help maintain tightness. Like all bellows constructions it has the disadvantage of a welded connexion at a point of stress concentration. Fatigue tests to determine typical S-N curves for at least the number of cycles expected during the service life would appear to be required before such gaskets can be used with entire confidence.

Piping Supports and Anchorages. Hangers must be sufficiently closely spaced to maintain localized natural frequencies of piping system elements at values well above propeller blade tip frequency. If practicable, these frequencies should exceed twice the number of the blade tip impulses at maximum revolutions since second order resonances are commonly observed.

Solid type hangers are preferred for long horizontal runs of piping but spring hangers must be used wherever there are substantial changes in elevation which result in vertical expansion movements. The latter are arranged to support the weight of pipe and insulation in the cold condition but permit thermal expansion movement without excessive change in spring reaction. The natural frequency of spring supported systems is, of course, affected by the elastic constants of the springs which must be taken into account.

When excessive vibration is encountered this can be reduced to acceptable levels by the use of friction damping type supports which permit slow movement due to thermal expansion. Fig. 19 illustrates the details of this type of support.

Horizontal weight and inertia components of piping may be quite large when a vessel is rolling in a seaway. The forces are usually absorbed by a system of sway braces or links. Considerable ingenuity is often required in placing these

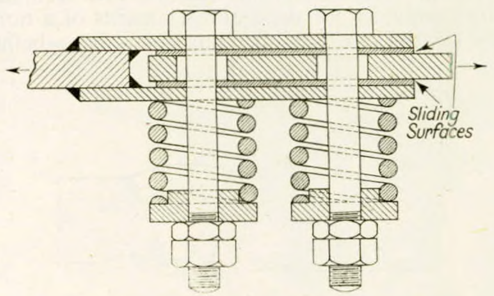


FIG. 19.—FRICTION DAMPING TYPE OF PIPE HANGER

links so that expansion movements of the piping closely follow the arc of travel of the piping end of the link.

Frequently the forces and moments imposed by piping attachments to turbines and boilers is of more serious consequence than the stress in the piping. It is necessary to inform the manufacturers of such equipment of the magnitude and direction of these forces and moments, and quite often additional piping flexibility must be provided to reduce the values to acceptable limits.

Safety Valve Escape Piping. The few steam line failures of recent record which resulted in casualties aside from battle damage experiences were mostly due to rupture of safety valve escape piping at the junction of the valve outlet, Fig. 20.

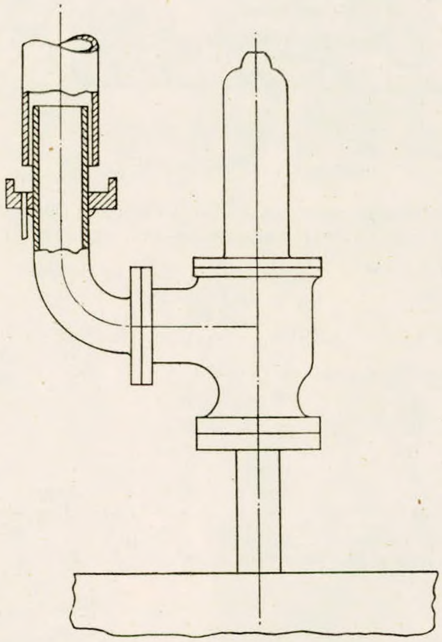


FIG. 20.—TYPICAL DESIGN OF OPEN END SAFETY VALVE ESCAPE PIPING CONNEXION

It is now well recognized that the entire vertical run of such piping must be supported and permitted to expand independently of the valve mounting.

Frequently this may be safely accomplished with an open end connexion as shown in the figure since advantage may be taken of the kinetic energy of the jet leaving the valve nozzle to provide an eductor effect. The published analytical procedures available to the authors[81] [82] are not sufficiently realistic for solving the flow and pressure drop relations involved in this case. Tests of small scale piping mock-ups using air instead of steam were found to give a satisfactory basis for design. The method of analysis involves the procedures described by Keenan[83] for compressible flow relations involving a shock front.

Main Turbines. Construction problems due to the use of higher steam temperatures in main turbines centre in those parts which are subject to full inlet temperature. These are the stationary parts, valves, steam passages and nozzles which are in contact with the inlet steam before it is expanded by the first stage nozzles to a lower temperature.

The design features to accommodate higher temperatures fall into two categories. The first process is simply the selection of suitable materials and dimensioning of usual constructions so that stresses are within allowable limits for the materials at their working temperatures. When this is inadequate, it becomes necessary to develop special constructions to reduce to safe limits the stresses which are set up by temperature differences and to design for lower stress concentrations at critical points to provide adequate margins above the rupture strength or creep limit at the working temperature.

Valve bodies for the turbine throttle and stop valves are examples which can be satisfactorily designed for higher steam conditions using only the first procedure described above.

Forgings have been selected for this service for several reasons. Thinner wall thicknesses are permissible which will minimize the temperature differences during heating. The use of forgings also avoids the extensive weld repair that is generally necessary with castings to obtain an end product that is free of defects. There is a secondary advantage in having a valve body of minimum weight in order to simplify its mounting to avoid vibration.

Fig. 27, showing a steam strainer and ahead and astern valve body for the propulsion machinery described in reference (13), is a typical example. It is a single-chamber pressure vessel and therefore is subject to one pressure and temperature. Heating and cooling are fairly uniform so that there will be no severe stresses due to temperature differentials. A material is selected which has adequate strength at its working temperature and the proportions are such that stresses are safe.

The problem of bolting is similar to that of the valve body. It involves selecting a material and proportioning the bolting to provide safe stresses. A choice is generally available between bolting of one alloy steel, or somewhat smaller size bolting of a higher alloy. In some cases, however, where maximum strength is required, the more expensive and special alloy requiring the use of critical materials must be used.

Bonnet and cover joints which are circular can be made with either metallic or spirally wound gaskets. Irregular openings, however, must be made metal-to-metal and must be designed and finished to the same high degree of care as the main turbine casing joint. The principles involved in the design of such joints have been well summarized in a brief paper by Campbell[84].

Valves and valve stem construction must be arranged with clearances that will permit seating of the valve even though the stem and seat are somewhat misaligned by temperature

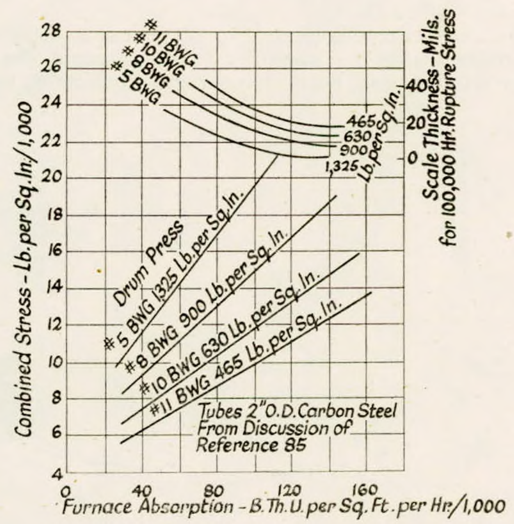


FIG. 21.—COMBINED THERMAL AND HOOP STRESSES AND EFFECT OF SCALE; ON FURNACE WATER WALL AND SCREENED TUBES

strains, and without the imposition of undue bending moments on the valve stems.

Valve stems and bushings, balance pistons and cylinders must be made with sufficient clearances to avoid binding and materials must be selected which retain their anti-galling characteristics at the operating temperature.

The most critical element of the main turbine for high temperature applications is the design of the inlet zone of the high pressure turbine casing. This zone is not only subject to maximum temperature and the limit which this places on allowable stresses, but is also subject to temperature differences during operation at partial loads when steam is admitted to only a fraction of the inlet nozzles.

In this respect, the high temperature marine turbine like its high temperature land counterpart must make use of special construction to avoid the stresses that would be set up by non-uniform heating in the conventional turbine.

For moderate temperatures, the multiple steam inlet passages to the first stage nozzles are cast integral with the cylinder as shown in Fig. 28 which illustrates the h.p. turbine for the tanker machinery described in reference (11). With this construction, the passage to the first group of inlet nozzles is open to the flow of steam before the other groups, causing the walls of this passage to heat at a greater rate than the surrounding walls.

The hotter walls are prevented from expanding by the surrounding cooler walls and expansion stresses are set up which are proportional to the temperature differences between the casing walls in this area. For inlet temperatures up to about 950 deg. F. these stresses can be safely sustained in arrangements of the type shown but it is desirable that attention be paid to retaining symmetry and circularity wherever possible in the design. With higher temperatures, the temperature differences are higher and the strength of the material is lower, both conditions increasing the severity of the resulting conditions.

One way to avoid severe thermal stress and distortion conditions is to design the inlet zone of the high pressure turbine in the manner illustrated in Fig. 29. This figure is a longitudinal section through the inlet end of a turbine installed in the Sewaren station of the Public Service Electric and Gas Company of New Jersey[95].

It will be noted that each inlet passage to the first stage nozzles is a single walled structure which is attached to the inner casing wall by a conical member so arranged that the nozzle chambers can expand without restraint. In this case there are six of these separate nozzle chambers, three in each half of the turbine casing.

The steam inlet pipe projects through the outer casing and is fitted inside the cylindrical inlet to each nozzle chamber. Leakage into the space between inner and outer casing walls is restricted by piston ring type joints between the inner end of the steam pipe and the nozzle chamber.

The inlet steam pipe is welded to a projecting nozzle of the outer casing to permit unrestrained expansion of the pipe.

With this arrangement the only parts subject to stress due to temperature differences are cylindrical, and the resulting

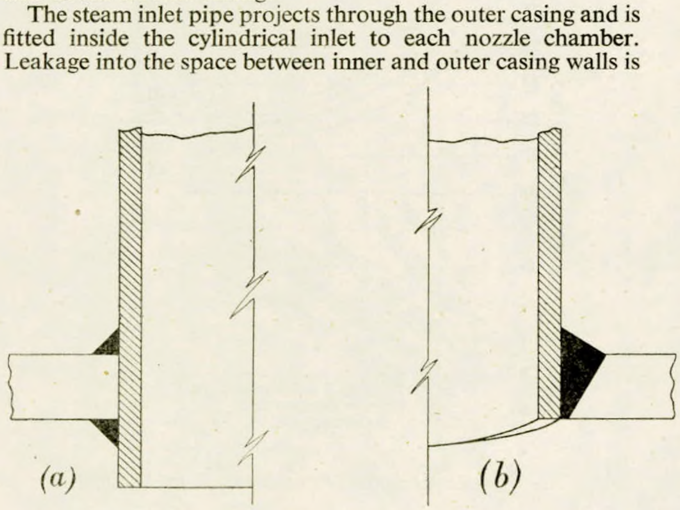


FIG. 22.—INTEGRALLY WELDED TUBE TO HEADER JOINT FOR HIGH PRESSURE BOILER (FROM A.S.M.E. CODE).

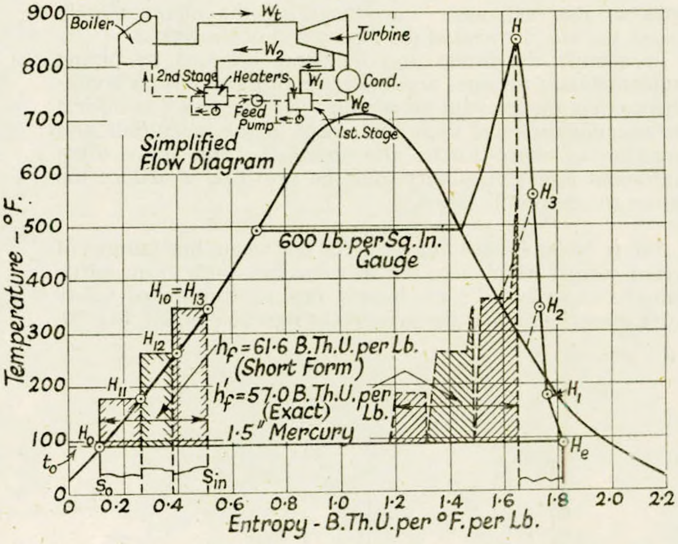


FIG. 23.—METHOD OF DETERMINING EXACT VALUES FOR ENERGY EXTRACTED FOR A FINITE NUMBER OF STAGES OF FEED HEATING

1ST HEATER BALANCE: $W_1\,(H_1 - H_{11}) = W_e\,(H_{11} - H_o)$

$$W_1 = W_e\left(\frac{H_{11} - H_o}{H_1 - H_{11}}\right)$$

2ND HEATER BALANCE: $W_2\,(H_2 - H_{12}) = (W_e + W_1)\,(H_{12} - H_{11})$

FRACTION OF EXHAUST FLOW EXTRACTED $= y_1 = \dfrac{W_1}{W_e} = \dfrac{H_{11} - H_{10}}{H_1 - H_{11}}$

$$y_2 = \frac{W_2}{W_e} = (1 + y_1)\,\frac{H_{12} - H_{11}}{H_2 - H_{12}}$$

$$y_\eta = \frac{W_\eta}{W_e}$$

$$= (1 + y_1 + y_2 + \cdots + y_{\eta\text{-}1})\,\left(\frac{H_{1\eta} - H_{1\eta\text{-}1}}{H_\eta - H_{1\eta}}\right)$$

RATIO: EXHAUST FLOW/THROTTLE FLOW $= \dfrac{W_e}{W_t} = R_e = \dfrac{W_e}{W_e + \Sigma W_\eta}$

$$= \frac{1}{1 + \Sigma y_\eta}$$

FRACTION OF THROTTLE FLOW EXTRACTED $= X_\eta = \dfrac{W_\eta}{W_t} = R_e\,y_\eta$

USEFUL ENERGY EXTRACTED FROM ANY STAGE PER LB EXHAUST FLOW

$$h_{e\eta} = y_\eta\,(H_\eta - H_e)$$

USEFUL ENERGY EXTRACTED FROM ANY STAGE PER LB THROTTLE FLOW

$$h_{t\eta} = R_e\,y_\eta\,(H_\eta - H_e)$$

THEREFORE

$$h_{t1} = \frac{W_1}{W_t}\,(H_1 - H_e) = R_e\,(H_1 - H_e)\left(\frac{H_{11} - H_0}{H_1 - H_{11}}\right)$$

$$h_{t2} = \frac{W_2}{W_t}\,(H_2 - H_e) = R_e\,(H_2 - H_e)\left(\frac{H_{12} - H_{11}}{H_2 - H_{12}}\right)(1 + y_1)$$

$$h_{t\eta} = \frac{W_\eta}{W_t}\,(H_\eta - H_e) = R_e\,(H_\eta - H_e)\left(\frac{H_{1\eta} - H_{1\eta\text{-}1}}{H_\eta - H_{1\eta}}\right)$$

$$(1 + y_1 + y_2 + \cdots + y_{\eta\text{-}1}) = X_\eta\,(H_\eta - H_e)$$

$h^1{}_f$ = EXACT VALUE FOR AVAILABLE ENERGY REMOVED FOR FEED HEATING (ON ADIABATIC BASIS) $= (\Sigma h_t)\,\dfrac{\sigma}{\eta_t}$

WHERE

$\sigma = 1$ — EXTERNAL LOSSES

η_t = OVERALL ENGINE EFFICIENCY = NET USED ENERGY/AVAILABLE ENERGY

h_f = SHORT FORM VALUE FOR AVAILABLE ENERGY REMOVED FOR FEED HEATING

$$= \left(1 + \frac{1}{\eta}\right)\left[(H_{1\eta} - H_0) - (460 + t_0)\,(S_{1\eta} - S_0)\right]$$

stresses can therefore be accurately determined and the dimensions proportioned accordingly.

For similar reasons the nozzle control valves which are normally attached to the turbine casing must be separated for temperatures above about 950 deg. F. For the unit shown in Fig. 29 the valve chest is mounted beside the cylinder and connected to the various nozzle inlets by pipes whcih are provided with sufficient flexibility to take care of their own expansion.

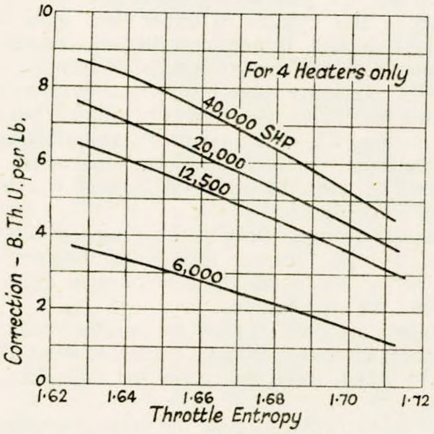


FIG. 24.—CORRECTION TO SHORT FORM VALUES FOR ENERGY EXTRACTED FOR FEED HEATING

FOR THREE HEATERS ADD 1 B.TH.U. PER LB. TO VALUE READ ON CURVE

FOR FIVE HEATERS SUBTRACT 1 B.TH.U. PER LB. FROM VALUE READ ON CURVE

EXACT ENERGY FOR EXTRACTION FEED HEATING $= h_f -$ CORRECTION

$$\text{WHERE } h_f = \left(1 + \frac{1}{\eta}\right)\left[(H_6 - H_0) - (460 + t_0)(S_6 - S_0)\right]$$

Austenitic materials were used for the inlet nozzle chambers and for the inlet steam pipes. There is also an austenitic section of the nozzle projections on the outer casing and an austenitic portion of the conical members supporting the nozzle chambers. Thus the only welds required between dissimilar materials are in the middle of these cylindrical and conical members.

By this arrangement the main turbine inner casing is subject only to the temperature of steam after expansion through the first stage nozzles. This lower temperature is the maximum to which the main turbine casing is subjected and simplifies its design.

The use of a by-pass to admit steam directly into the turbine casing for higher powers, which can readily be done at lower inlet steam temperatures, would present a difficult problem for a high inlet temperature casing design. By-passing would also subject the first rows of rotating blades to nearly full inlet temperature which would present an additional difficulty.

A critical effect of high steam temperature, as defined earlier in the paper, manifests itself in bolting. Bolts which are subject to full inlet temperature, such as those which attach the first stage nozzle blocks to the nozzle chamber, must be given special attention to avoid stress concentrations which may lead to failure by long term rupture. For this service, provisions must also be made for either accurate seating for the bolt heads or self-aligning washers to avoid the possibility of a bending moment being introduced into the body of the bolt while tightening. Tapered thread diameters, round bottomed threads and reduced body diameter are additional safeguards against stress concentrations. Tightening of the bolt must be controlled by measuring either the elongation of the bolt or the applied torque.

The remainder of the turbine in general follows conventional practice since the temperatures beyond the inlet zone fall within normal moderate temperature practice. However, the overall temperature level through the turbine is raised, which requires some innovations. Diaphragm packing seal rings operate at temperatures too high for the standard leaded-bronze construction and stainless steel fins must be used. Gland sealing piping may receive steam at temperatures above the limit for carbon-steel piping. The advantages of metallic labyrinth gland seal rings over carbon rings is amplified.

Astern operation presents an unusual problem for high temperature designs, particularly for large powers due to the increased windage losses generated in the ahead stages. The effect of higher temperatures is to aggravate the problem by raising the exhaust temperature level from which windage heating begins.

Enough is now known about the factors controlling reverse operation windage and thermal dissipation during astern operation to predict the astern "hot spot" temperatures and overall temperature differentials between rotor and casing with a satisfactory degree of accuracy. Proper clearances can then be assigned so that there is no likelihood of a rub during or after astern operation.

With the proper selection of materials, and the same care in the design and construction of the turbine, there is every reason for a high temperature turbine to maintain the same high standard of reliability as its moderate temperature predecessors.

Condensers. As was shown previously, the use of higher steam pressures and temperatures for modern propulsion machinery justifies and requires the use of three or four extraction type feed water heaters to obtain better cycle efficiency and therefore fuel rate. The extraction of steam for feed heating and the reduction of steam consumption for a given power output result in a considerable decrease of steam flow to the condenser, thereby making it possible to use a smaller condenser and obtain a saving in weight.

Higher steam pressures and temperatures do not impose any special design problems for a conventional surface condenser supported by the low pressure turbine or where the turbine supporting structure is built into the condenser shell above the top of the tubes. However, new problems arise in the case of installations where the turbine is mounted directly on top of the condenser with the support attached to the sides of the shell and there is considerable height between the turbine exhaust flange and the condenser supporting feet. This is because higher initial steam temperatures result in a large upward expansion of the condenser shell during astern operation when the exhaust temperature is relatively high due to the lower efficiency of the astern element. Adequate provision for differential expansion is obtained by arranging for additional flexibility in the condenser shell or by other means.

High Pressure Problems. As previously indicated, no new problems arise due to the use of higher pressures; however, a number of cases arise where designs that were entirely satisfactory at moderate pressures must be considerably modified for high pressure applications.

One obvious instance is the use of multiple pressure breakdown devices around moving shafts; this applies not only to steam turbine and feed pump rotors but also to high pressure valve stems.

The increase of saturation temperature with higher steam pressure has far-reaching effects on the boiler design. Economizer water outlet temperatures may be increased to correspond but, since feed water inlet temperatures usually increase also, the economizer absorption per pound of evaporation changes but little. A greater burden is, there-

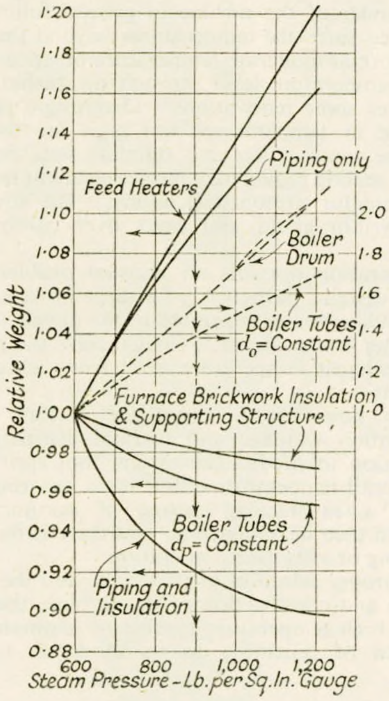


FIG. 25.—THEORETICAL EFFECT OF STEAM PRESSURE ON WEIGHT OF STEAM PIPING BOILERS, ETC.

fore, imposed on air heater recovery surface since the economizer gas exit temperature must be higher. The result is a tendency toward higher combustion air inlet temperature.

The necessity for increasing steam generating tube pitch to obtain adequate drum ligaments at higher pressures has a tendency to reduce the amount of steam generating surface which further increases the heat absorption duty of economizer and air heating surfaces.

Further, the density ratio of saturated vapour to liquid is less at higher pressures, approaching unity at the critical pressure. Therefore, it becomes increasingly important to provide adequate disengaging surface in the steam drum, and the moisture separating devices must be more efficient in action. Additional downcomers may also be required because the lowered density ratio reduces the circulation head of the steam generating tubes.

Another important effect on boiler design occurs in the waterwall and screen tubes of the steam generator; Fig. 21, reproduced from a recent publication(85), shows how the need for increased wall thickness aggravates the elastic state thermal stress conditions in such tubes. High steam temperatures are also a factor in this problem since the screen tube absorption rate is greater for higher furnace exit temperatures which have already been shown to have an important bearing on economical superheater proportions for high steam temperatures.

Another important high pressure effect is in relation to transient thermal stresses since the increased wall thickness required in drums, headers and piping aggravates the stresses resulting from sudden but sustained changes in water and steam temperatures.

Greater care must be exercised in starting of high pressure boilers to avoid excessive sustained temperature gradients in drum sheets, header walls and also in tube seat joints of economizers, waterwalls and superheaters. The recent considerable progress made in quick starting of high pressure land boilers is due mainly to a better understanding of the transient temperature conditions in these parts of a boiler based on careful measurement and analysis(86). The provision of drainable superheaters and reheaters has also assisted materially in reducing starting up time for such units.

The turbine starting problem is also accentuated in higher pressure units with more massive casing and diaphragm walls.

Finally, the increased mass of high pressure parts appears most prominently in the flanged joint problem. The necessity for thicker walls and heavier bolts combine to aggravate the delay in response of outer flange and bolt temperature to changes in steam and pipe wall temperature.

This effect is also evident in boiler tube seats where the need for heavier wall tubing and thicker headers has led to developments in the design and fabrication of expanded tube seats(87). Recently there has been a trend in very high pressure land boilers to use integrally welded header joints illustrated in Fig. 22 which result in an increased header ligament efficiency. In such cases the final connexion is a field butt weld between the tube and the stub which has been shop welded to the header(30).

Similar considerations apply to the design of tube seat and head joints in high pressure feed water heaters, and this appears to be the reason for the development and increasing use of return tube type heaters.

The problems involved in high temperature turbine casing and valve chest cover joints have been discussed above, and the extent to which these are magnified at higher pressures is apparent.

The effects of higher steam pressure upon control equipment will be discussed below.

Control Equipment and Operating Problems

A marked reduction in creep and rupture strength for the same increment of temperature at higher temperature levels is found even in the alloy materials selected for high temperature use. This is illustrated by the data in Table III and makes it evident that a positive form of superheat control is highly desirable for steam temperatures above about 850 deg. F.

It is obvious that a rising steam temperature characteristic with boiler output is very desirable since this provides a safer level for the temperature swings which are more likely to occur at light loads and during manœuvring. Unfortunately, this characteristic conflicts to some extent with economical superheater design for high temperatures since it is a result of decreased radiant absorption in the superheater. This situation further emphasizes the desirability of superheat control for high temperature boilers, to ease the operator's burden.

The simplest and most reliable control arrangement is accomplished with an auxiliary drum desuperheater which receives a controlled portion of the steam flow from an intermediate pass of the superheater. This desuperheated

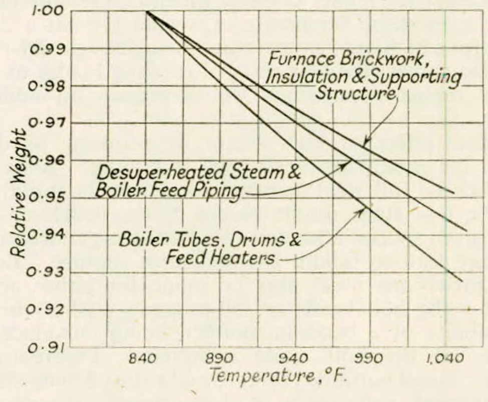


FIG. 26.—THEORETICAL EFFECT OF STEAM TEMPERATURE ON WEIGHT OF PIPING BOILERS, ETC.

steam is then returned to the inlet of the next superheater pass and the superheater outlet temperature controlled by the amount of steam permitted to flow through the desuperheater. The system has been in successful use for many years, and the original marine application is fully described in reference (24).

The use of a divided furnace arrangement for superheat

the bled steam feed-water heaters. This system was applied to the high pressure supertanker design previously mentioned[11], with notable results. In this case the de-aerating heater and the two high pressure heaters were all fitted with automatic live steam make-up valves of the air pilot actuated type controlled by the heater pressure. The setting of these valves was such that the minimum feed temperature at low

ow
out
nal
ing
am
not

am
tic

of
ing
ing
ect

int
ard
ge
on

as
an
ied

ver
yer
ers

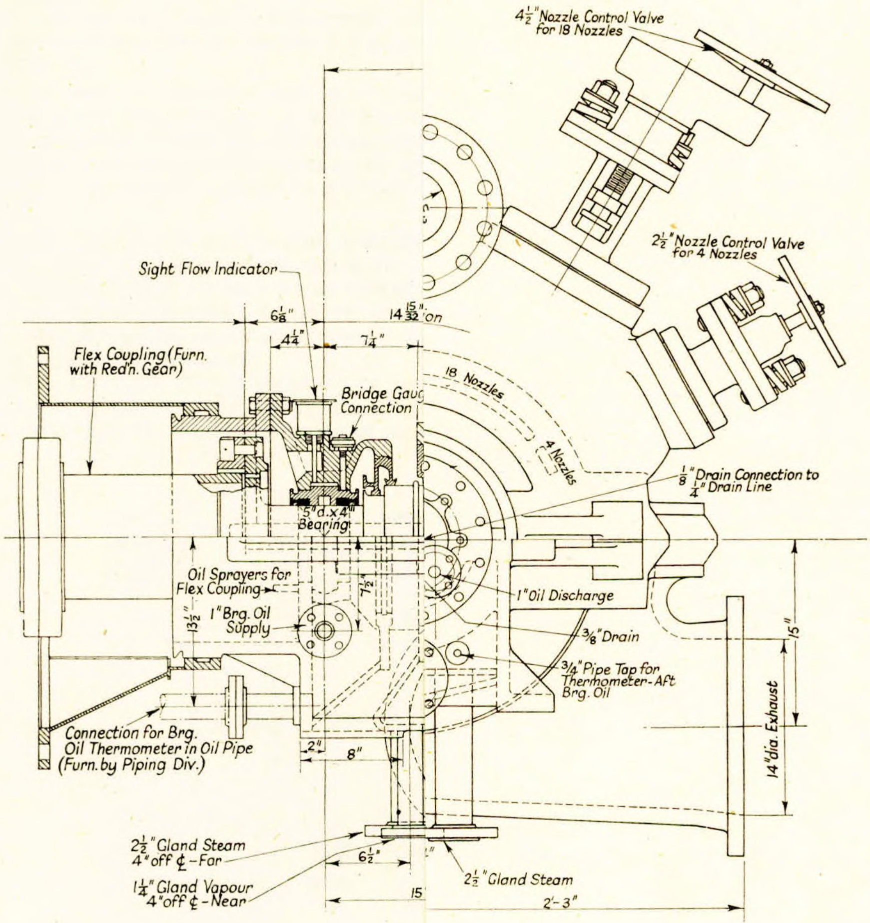


FIG. 27.—INLET STEAM STRAINER AND THROTTLE VALVE

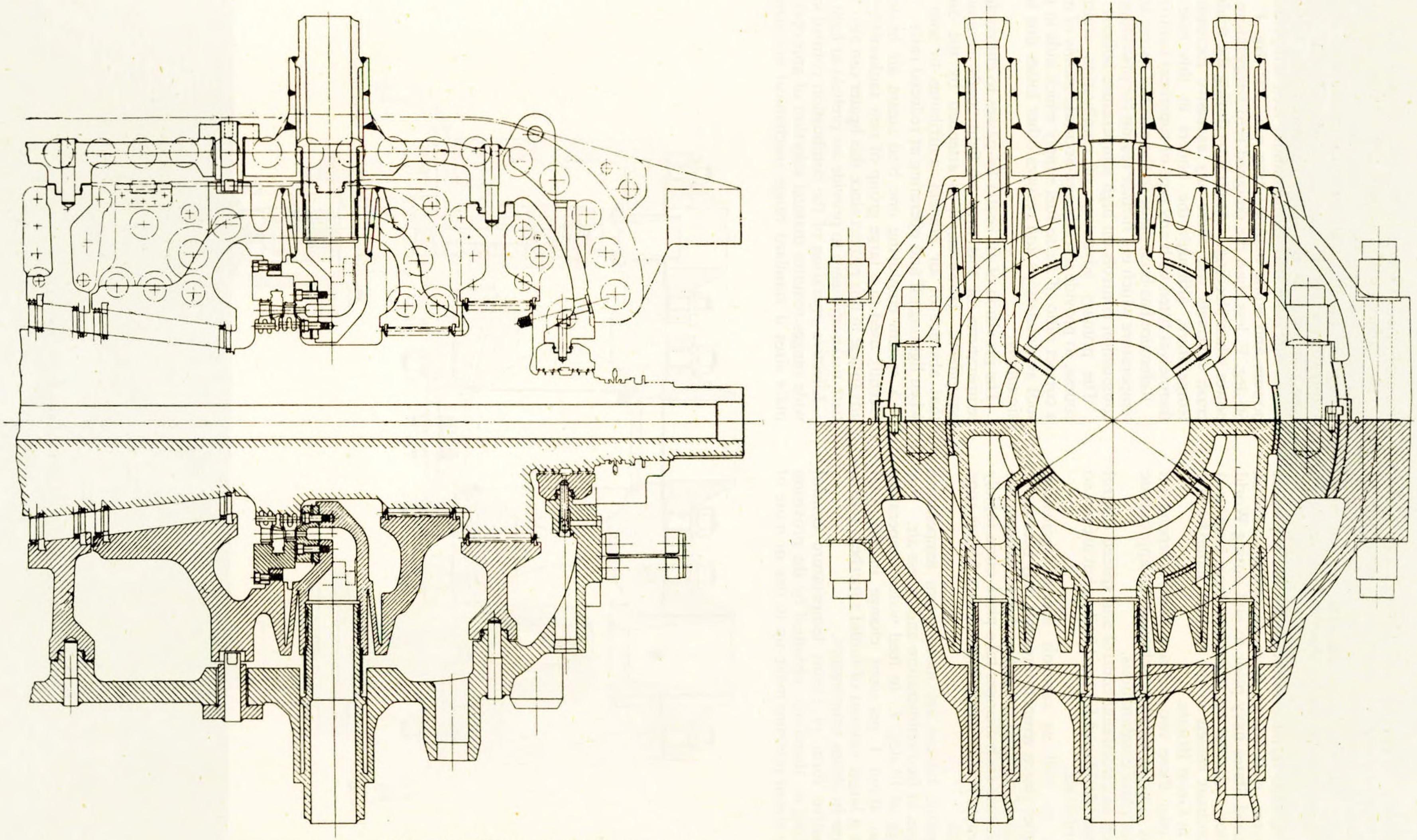


FIG. 29.—HIGH PRESSURE TURBINE NOZZLE AND CONTROL VALVE ASSEMBLY

HIGH TEMPERATURE CONSTRUCTION 1,500LB. PER SQ. IN. GAUGE, 1,050 DEG. F. TT DOUBLE WALL CASING WITH SEPARATE NOZZLE CHAMBERS AND INLET STEAM FEATURES OF 18-8 CHROME NICKEL ALLOY are used. Such a system cannot be considered fully automatic and its merit relative to a fully automatic system using either wide range steam atomizing burners or equally wide range return flow burners is still a highly controversial subject.

Considerable recent interest has been shown in steam atomizing type burners because of the increased seriousness of the superheater slagging problem at higher steam temperatures[88]. A great deal has appeared recently in the literature on this subject, and one item which appears to be of considerable significance is discussed briefly below.

Conclusive evidence has been presented in the case of stoker-fired land boilers of a reduction in superheater slagging due to introducing steam into the air supply below the fuel bed[89]. An explanation is offered which unfortunately does not seem to apply equally to pulverized fuel or oil-fired boilers. Briefly, the use of strongly humidified air causes the appearance of the well-known water gas reaction leading to the production of free hydrogen and carbon monoxide in the fuel bed, followed later by further combustion of these products. This first reaction is known to be endothermic and is therefore supposed to lower the combustion temperature above the fuel bed sufficiently so that vanadium compounds present in the fuel do not volatize and later deposit as slag on the superheater tubes.

Experience with pulverized fuel fired boilers indicates that the same reaction is not sustained because of the greatly reduced particle size and consequently greater speed of the combustion reaction. The reported reduction in superheater slagging due to use of steam atomizing fuel oil burners would appear to be due mainly to the resulting finer atomization and reduction in size of combustion particles.

The improvements in efficiency reported as a result of these humidification procedures are probably in the main due to increased average cleanliness of the various heat absorbing surfaces.

A further advantage claimed for steam atomizing burners is a reduction in soot deposits on economizer and air heater tubes primarily because such burners require less expert attention to maintain good combustion conditions at low rates of operation. The very wide range of burner capacity available when used with automatic combustion control would appear to reduce firing attention to a minimum. The amount of steam used is generally less than 1 per cent of the evaporation.

With further reference to feed system controls, the increased importance of feed pump losses in high pressure cycles emphasizes the advantages of constant differential pressure control of the main feed pump discharge pressure instead of constant pressure. This is particularly important for vessels operating over a wide range of powers but may be of significance even on a tanker or cargo vessel since appreciable throttling is required at normal power if the constant discharge pressure setting is based on the boiler drum safety valve condition.

The design of direct mechanical type pressure controls is greatly complicated by the requirement that the control shall be accurately governed by a comparatively small difference between two high pressures, i.e. feed pump discharge and steam drum pressure. The use of air pilot operated control for such purposes is therefore finding some favour despite its increased complexity.

Another control feature which is becoming standardized for high pressure designs is the provision of automatic make-up feed by a vacuum drag line which is controlled by the water level in the de-aerating heater. The tendency to increase the pressure in this heater to permit full utilization of the turbine crossover pressure emphasizes the importance of always having sufficient "net positive suction head" available at the inlet to the main feed pump which draws from the de-aerating heater.

Conclusion

The authors have attempted to discuss the major problems in the use of higher steam pressures and temperatures which are within the field of their experience. No doubt there are many others which the authors have overlooked and still others are due to appear in the future since the period of actual experience with high temperatures is comparatively short.

Further advances in power stations beyond the present top level of 2,500 lb. per sq. in. boiler pressure and beyond 1,050 deg. F. steam temperature associated with a somewhat lower pressure appear to be likely in the near future. It is understood that units are now being built for 1,100 deg. F. operating temperature.

Current U.S. marine practice is well illustrated by three supertanker designs now in operation to which reference has already been made[11], [12], [13]. These designs indicate widely varying tendencies but the majority of post-war installations are at the 600 lb. per sq. in., 850 deg. F. level. A brief description is also available of one C-3 prototype cargo ship now under construction employing 850 lb. per sq. in. gauge and 900 deg. F. at the superheater outlet[90]. There are, of course, other vessels under construction utilizing high steam conditions but these all come under national defence or other security classifications.

Regarding the near future, it appears probable that steam temperature for U.S. merchant marine construction will mainly be limited to about 850 deg. F. due to the present critical shortage of chromium and its need for use in more vital defence equipment. Looking beyond this limitation, which everyone hopes is transitory, an increase in steam temperature and pressure above 600 lb. per sq. in., 850 deg. F. is certainly to be expected.

Acknowledgements

The authors wish to express their appreciation for the very constructive suggestions obtained from Mr. W. F. Gibbs, Mr. O. R. Carpenter and Mr. Geo. W. Kessler during the course of preparing the paper. The authors are also indebted to Dr. A. Nadai for his valuable comments regarding hoop stress formulae.

Reference List

(1) "The Present Trend in Marine Engineering in the United States," by J. E. Burkhardt, Trans. S.N.A.M.E. 1938, Vol. 46, p. 34.

(2) "Considerations Governing the Choice of Live Steam Conditions in Condensing Turbo-Sets." *The Brown Boveri Review*, December, 1946.

(3) "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, Vol. L, January, 1934.

(4) "Economic Selection of Steam Conditions for Merchant Ships," by Douglas C. MacMillan, and Mark L. Ireland, Jr., Trans. S.N.A.M.E., 1948, Vol. 56, p. 149.

(5) "Power Plants for Postwar Vessels," by A. C. Rohn and W. E. Hammond, *Marine Engineering and Shipping Review*, Vol. 50, 1945.

(6) "A Modern Marine Power Plant," by A. W. Wood, *Marine Engineering and Shipping Review*, April, 1948.

(7) "Improvements in Thermal Efficiencies with High Steam Pressures and Temperatures in Non-Reheating Plant," by K. Baumann, Proc., I.Mech.E., 1946, Vol. 155, p. 125.

(8) "Factors Governing the Design of a Modern Tanker with Special Reference to Machinery," by W. Lynn Nelson, Trans., I.Mar.E. 1950, Vol. lxii, p. 177.

(9) "Geared Steam Turbines for Merchant Ship Propulsion" by T. W. F. BROWN, Transactions, North East Coast Institution of Engineers and Shipbuilders, 1949-50, Vol. 66, p. 315.

(10) "A.B.S. Interim Guide for the Installation of High Temperature Steam Piping on Ships," February 1950, published in *Marine Engineer and Naval Architect*, July 1950, p. 309.

(11) "Performance and Design of Propulsion Machinery for the 26,800 ton Esso Supertankers," by M. L. IRELAND, JR., M. D. WHEELER, and L. E. SPENCER, presented at the Society of Naval Architects and Marine Engineers Chesapeake Section meeting, 30th November, 1950.

(12) "The Design of a Class of 28,000 ton Tankers," by H. DELUCE and W. I. H. BUDD, Bethlehem Steel Co., Shipbuilding Division, presented at A.S.M.E. Annual Meeting, December 1949.

(13) "The Engineering of a 30,500 ton Super Tanker—The First Ship using One Thousand and Twenty Degree F. Steam," by DR. L. M. GOLDSMITH of the Atlantic Refining Co., presented at A.S.M.E. Annual Meeting, 30th November, 1950.

(14) "The Modern Great Lakes Freighter," by R. C. STANBROOK, *Marine Engineering and Shipping Review*, April 1947, p. 73.

(15) "New Great Lakes Bulk Freighters, *William A. Irvin* and *Governor Miller*," by E. B. WILLIAMS, *Marine Engineering and Shipping Review*, June 1938, p. 252.

(16) "Metropolitan-Vickers Marine Turbines," *Shipbuilding and Shipping Record*, 25th May, 1950, Vol. 75, p. 655.

(17) "The *Empire Orwell*," *Shipbuilding and Shipping Record*, 12th January, 1950, Vol. 75, p. 45.

(18) "Progress in Design and Performance of Modern Large Steam Turbines for Generator Drive," by G. B. WARREN, TRANS., A.S.M.E. 1941, p. 49.

(19) "High-Pressure Steam for Marine Propulsion," by R. H. TINGEY, TRANS., S.N.A.M.E., 1943, Vol. 51, p. 156.

(20) C.I.O.S. Report on Main Propulsion Steam Turbine Machinery of German Merchant Vessels, by MARK L. IRELAND, JR., U.S. Technical Industrial Intelligence Committee.

(21) "Economizers and Air Heaters in Marine Power Plants," by D. M. SCHOENFELD and E. P. WORTHEN, TRANS., S.N.A.M.E., 1949, Vol. 57, p. 573.

(22) "Steam Turbines and their Cycles," by J. KENNETH SALISBURY, John Wiley & Sons, 1950.

(23) "The Regenerative Cycle: An Efficiency Basis having Special Reference to the number of Feed Water Heating Stages," by H. S. HORSMAN, Proc. I Mech. E., 1941, Vol. 146, p. 5.

(24) "The High Pressure High Temperature Turbine Electric Steamship *J. W. Vandyke*," by L. S. GOLDSMITH, TRANS., A.S.M.E., 1939.

(25) "Problems Incidental to the Use of High Pressures and Temperatures in Marine Steam Power Plants," by C. RICHARD SODERBERG, Trans., S.N.A.M.E., 1939, p. 21.

(26) "Properties of Metals at Elevated Temperatures," by G. V. SMITH, *Mechanical Engineering*, October 1950, p. 799.

(27) A.S.T.M. Specification E22-41—Recommended Practice for Conducting Long-Time High-Temperature Tension Tests of Metallic Materials, A.S.T.M. Standards, 1949, Part I Ferrous Metals, p. 1281.

(28) "Influence of Time at 1000 F. on the Characteristics of Carbon Steel," by A. E. WHITE, C. L. CLARK and R. L. WILSON, PROC., A.S.T.M. 1936.

(29) "The Creep Curve and Stability of Steels at Constant Stress and Temperature," by S. H. WEAVER, TRANS., A.S.M.E. 1936, p. 745.

(30) "The Creep and Stress-Rupture Testing of Steam Boiler Materials," by J. B. ROMER and H. D. NEWELL, presented at A.S.M.E. Annual Meeting, 26th November to 1st December, 1950.

(31) "100,000 Hour Creep Test," by E. L. ROBINSON, *Mechanical Engineering*, March 1943, p. 166.

(32) "Fracture of Carbon Steels at Elevated Temperatures," by A. E. WHITE, C. L. CLARK and R. L. WILSON, TRANS., A.S.M. 1937, Vol. 25, p. 863.

(33) "Fracture of Steels at Elevated Temperature after Prolonged Loading," by R. H. THIELEMANN and E. R. PARKER, TRANS. A.I.M.M.E. 1939, Vol. 135, p. 559.

(34) Report on the Strength of Wrought Steels at Elevated Temperatures, by R. F. MILLER and J. J. HEGER, Special technical publication No. 100 of A.S.T.M., 1950.

(35) "Safety Margins and Stress Levels in High Temperature Equipment," by E. L. ROBINSON, TRANS., A.S.M.E. 1951, p. 89.

(36) Progress Report No. 5 of Joint E.E.I.-A.E.I.C. Sub-Committee on Graphitization of Piping, February, 1946.

(37) "Graphitization of Low Carbon Low Alloy Steel" An Appraisal of the literature by G. V. SMITH, *Welding Research Supplement*, published by Welding Research Council, June 1948.

(38) "Formula for Pipe Thickness," by W. J. BUXTON and W. R. BURROWS, presented at A.S.M.E. Annual Meeting, 26th November-1st December, 1950.

(39) "Design and Fabrication of Steam Piping," by A. W. RANKIN and R. W. CLARK, presented before Metropolitan Section of S.N.A.M.E., 20th February, 1951.

(40) "The Utilization of Creep-Test Data in Engineering Design," by R. W. BAILEY, PROC. I.M.E. 1935, p. 131-349.

(41) "Design of members subjected to creep at high temperatures," by JOSEPH MARIN, TRANS., A.S.M.E. 1936, p. A21.

(42) "The interpretation of creep tests for machine design," by C. R. SODERBERG, TRANS., A.S.M.E. 1937, p. 733.

(43) "The Inelastic Behavior of Engineering Materials and Structures," by ALFRED M. FREUDENTHAL, John Wiley, 1950.

(44) "Correlation of High Temperature Creep and Rupture Tests," by R. H. THIELEMANN, TRANS. A.S.M. 1941, Vol. 29, p. 355.

(45) "Some Effects of Composition and Heat Treatment on the High Temperature Rupture Properties of Ferrous Alloys," by R. H. THIELEMANN, PROC. A.S.T.M. 1940, p. 788.

(46) "The Rupture Strength of Steels at Elevated Temperatures," by A. E. WHITE, C. L. CLARK and R. L. WILSON, TRANS. A.S.M. 1938, Vol. 26, p. 52.

(47) Resume of High Temperature Investigations Conducted during 1946-47, published by the Timken Roller Bearing Co.

(48) "High Temperature Properties of Steels," by C. L. CLARK, published by Timken Roller Bearing Co., 1945.

(49) Private communication from Dr. C. L. CLARK, Timken Roller Bearing Co., Canton, Ohio.

(50) "The Creep Strength of 17 Low Alloy Steels at 1,000 deg. F.," by R. F. MILLER, W. G. BENZ aud W. E. UNVERZAGT, PROC., A.S.T.M., Vol. 40, 1940, p. 771.

(51) "Creep strength, stability of microstructure and oxidation resistance of Cr-Mo and 18 Cr—8 NI steels," by R. F. MILLER, W. G. BENZ and M. J. DAY, TRANS. A.S.M. 1944, Vol. 32, p. 381.

(52) "High temperature properties and characteristics of a ferritic steam-piping steel," by A. W. RANKIN and W. A. REICH, presented at A.S.M.E. Annual Meeting, 26th November—2nd December, 1950.

(53) "Thickness of Steam Piping for High-Pressures, High-Temperature Service," by H. C. E. MEYER, TRANS. S.N.A.M.E. 1949, Vol. 57, p. 379.
(54) "The Creep Properties of Molybdenum, Chromium-Molybdenum and Molybdenum-Vanadium Steels," by J. GLEN, Jl. Iron and Steel Inst., 1948, Vol. 158, No. 1, p. 37.
(55) "Thermal-shock and other comparison tests of austenitic and ferritic steels for main steam piping," by W. C. STEWART, TRANS. A.S.M.E. 1950, p. 1043.
(56) "Ultrasonic Flaw Detection in Pipes by Means of Shear Waves," by C. D. MORIARTY, presented at A.S.M.E. Fall Meeting, 19th-21st September, 1950.
(57) "Determination of the Expansion Forces in Piping by Model Test," by H. W. SEMAR, TRANS. A.S.M.E. 1939.
(58) "Piping Handbook," by SABIN CROCKER, McGraw-Hill Book Co., Inc.
(59) "End Reactions and Stresses in Three Dimensional Pipe Lines," by G. B. KARELITZ and J. H. MARCHANT, TRANS. A.S.M.E., June 1937.
(60) "Design of Piping Systems," by M. W. Kellogg Co., New York, N.Y., 1941.
(61) "Piping Flexibility and Stresses," by S. D. VINIERATOS and D. R. ZENO, Cornell Maritime Press, 1941.
(62) "Properties of Thin-Walled Curved Tubes of Short-Bend Radius," by T. E. PARDUE and IRWIN VIGNESS, TRANS. A.S.M.E. 1951, p. 77.
(63) Discussion of reference 53 by J. F. HARVEY, TRANS. S.N.A.M.E. 1949.
(64) "Strength of Materials," by S. TIMOSHENKO, 2nd Ed. 1941, Part II, p. 164.
(65) "American Standard Steel Pipe Flanges and Flanged Fittings" A.S.A.-B16e-1939 and Supplement No. 1 A.S.A.-B16e6-1949.
(66) "Problems in the mechanical design of gas turbines," by R. B. SMITH, TRANS. A.S.M.E. 1947, p. A101.
(67) Discussion by M. L. IRELAND, JR. of paper "Vibration of Steam Turbine Blading", by R. W. NOLAN, TRANS. S.N.A.M.E. 1949.
(68) "Vibration of Marine Turbine Blading," by R. W. NOLAN, TRANS. S.N.A.M.E. 1949, p. 518.
(69) "Mechanical Design and Testing of Long Steam Turbine Blading," by H. M. OWENS and W. E. TRUMPLER, JR., TRANS. A.S.M.E. 1950.
(70) "Cyclic Heating Test of Main Steam Piping Joints between Ferritic and Austenitic Steels, Sewaren Generating Station," by H. WEISBERG, TRANS. A.S.M.E., August 1949.
(71) "Some Considerations in the Joining of Dissimilar Metals for High Temperature, High Pressure Service," by O. R. CARPENTER, N. C. JESSEN, J. L. OBERG, R. D. WYLIE of the Babcock & Wilcox Co., presented at June meeting A.S.T.M., 1950.
(72) "Welds between dissimilar alloys in full size steam piping," by R. U. BLASER, F. EBERLE and J. T. TUCKER, JR., presented at annual meeting of A.S.T.M., 29th June, 1950.
(73) Private communication from O. R. CARPENTER, Babcock & Wilcox Co., Barberton, Ohio.
(74) "Portable Ultrasonic Thickness Gage," by NORMAL G. BRANSON, *Electronics*, Jan. 1948.
(75) Discussion by W. B. BUNN, the M. W. Kellogg Co., of "Ultrasonic Flaw Detection in Pipes by Means of Shear Waves," by C. D. MORIARTY, presented at A.S.M.E. Fall Meeting, 19th-21st September, 1950.
(76) "Bending of Curved Pipes," by SABIN CROCKER, *Heating, Piping and Air Conditioning*, May 1931.
(77) "Bending Moments and Leakage at Flanged Joints," by ROBERT G. BLICK, *Petroleum Refiner*, February, May and June 1950.
(78) "Effect of Internal Pressure on Stress and Strains in Bolted Flanged Connections," by D. B. WESSTROM, presented Nov.-Dec. Meeting A.S.M.E. 1950.
(79) "Joints for High-Pressure High-Temperature Piping," by I. H. CARLSON and W. S. BLACK, presented at A.S.M.E. Fall Meeting, 19th-21st September, 1950.
(80) "Performance Tests of Pressure-Seal Pipe Joints," by I. H. CARLSON and W. S. BLACK, Crane Co., *Valve World*, April-May 1947.
(81) "Sizing Vent Piping for Safety Valves," by MAX W. BENJAMIN, *Heating, Piping and Air Conditioning*, October 1941.
(82) "Design of Vent Piping for Safety Valves," by Paul L. GEIRINGER, *Heating, Piping and Air Conditioning*, March 1945.
(83) "Gas Tables, Thermodynamic Properties of Air, Products of Combustion and Component Gases and Compressible Flow Functions," by J. H. KEENAN and JOSEPH KAYE, John Wiley & Sons, Inc.
(84) "Practical Aspects of Turbine Cylinder Joints," by C. B. CAMPBELL, TRANS. A.S.M.E. 1938, p. A-49.
(85) "Procedures and Influencing Factors in the Design of Marine Boilers," by G. W. KESSLER, TRANS. S.N.A.M.E. 1948 (Discussion p. 230).
(86) "Latest Technique for Quick Starts on Large Turbines and Boilers," by J. C. FALKNER, D. W. NAPIER, C. W. KELLSTEDT, TRANS. A.S.M.E. 1950, p. 1111.
(87) "Practical Aspects of Making Expanded Joints," by C. A. MAXWELL, TRANS. A.S.M.E. 1943.
(88) "Formation and Removal of Slag from Superheater Tubes of Marine Boilers," by F. R. HOCK, J.A.S.N.E. November 1945, p. 508.
(89) "Fireside Deposits on Steam Generators Minimized through Humidification of Combustion Air," by P. MURPHY, JR., J. D. PIPER and C. R. SCHMANSKY, presented at A.S.M.E. Annual Meeting, 26th November-1st December, 1950.
(90) "Prototype Cargo Ship C3-S-DXI", *Marine Engineering and Shipping Review*, July 1949, p. 43.
(91) "The Effect of Carbide Spheroidization upon the Rupture Strength and Elongation of Carbon-Molybdenum Steel," by S. H. WEAVER, PROC. A.S.T.M., Vol. 46, 1946, p. 856.
(92) "Compilation of Available High Temperature Creep Characteristics of Metals and Alloys", A.S.M.E.-A.S.T.M. Joint Publication, 1938.
(93) *Journal of Applied Physics*, Vol. 8, No. 6, June 1937, pp. 418-432, article by A. NADAI on " Creep of Solids at Elevated Temperature."
(94) Private communication from A. NADAI, 23rd February, 1951.
(95) "The Design of Sewaren Generating Station and No. 1 Unit at Essex Station, Public Service Electric and Gas Company," by F. P. FAIRCHILD, TRANS. A.S.M.E. 1950, p. 247.
(96) "Marine Separately Fired Superheater Boilers," *Marine Age*, June 1944, p. 32.
(97) "Ultrasonics brings 100% Inspection to Power Piping," by W. B. BUNN and W. J. MITCHELL, *Power*, October 1950, p. 102.

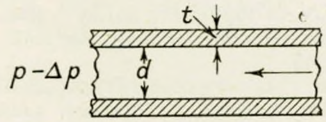
Appendix I

Influence of Steam Pressure and Temperature on Specific Weight of Machinery

Information on the variation of machinery weight with steam conditions was presented in Fig. 10 reference (4). This data was based on manufacturers' quotations and detailed estimates of piping weights. In all cases the specific weight per shaft horse-power was found to increase with higher pressure and decrease slightly with higher temperature.

There follows a theoretical analysis of the basic factors which determine the variation of weight with changes in steam conditions for the principal types of equipment affected by such changes.

(1) *Steam Piping—effect of increased pressure at constant temperature*



It will be assumed that the length of piping required is determined from arrangement considerations and is not influenced by the steam pressure. The following requirements must be met: (1) $pd/2t = \sigma$, constant, to satisfy strength requirements (2) $\Delta p/p = c$, a constant, i.e. pressure drop allowance is taken as a constant fraction of initial pressure.

For simplicity all constant quantities will be lumped together in a single symbol c in each expression which follows. The value of this constant is different for each equation but has no significance for the present purpose.

Now it is shown in Fig. 1 that the improvement in fuel rate due to higher pressures is quite small; however, there is a reduction in weight flow and a substantial decrease in specific volume. The change in weight flow is approximately proportional to the increase in available energy and thus may be determined from the approximate relation:

$$wi = c, \text{ a constant} \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where

$$i = c\, pv \left(1 - \rho^{\frac{k-1}{k}}\right) \quad . \quad . \quad . \quad (4)$$

using the following notation:

p = initial pressure, lb. per sq. in. abs.
v = initial sp. vol. cu. ft. per lb.
ρ = pressure ratio p_2/p_1
k = adiabatic expansion coefficient
w = weight flow lb. per hr.
W = total weight of equipment, lb.
A = inside area, sq. ft.
I = inside diameter, ft.
t = thickness, ft.
V = flow velocity, ft. per hr.
i = available energy, B.Th.U. per lb.
f = friction factor
l = length of pipe, ft.

The specific volume change is approximately proportional to inlet pressure, i.e.

$$pv = c, \text{ constant} \quad . \quad . \quad . \quad . \quad (5)$$

$$\Delta p = \frac{f\, V^2}{v}\frac{l}{d} = c\,\frac{w^2 vl,}{d^5} \quad . \quad . \quad . \quad (6)$$

since

$$V = c\,\frac{wv}{d^2}$$

Finally the piping weight is given by

$$W = c\,(d + t)\,t \quad . \quad . \quad . \quad (7)$$

with both d and t variable

Combining (2), (5) and (6)

$$\frac{\Delta p}{p} = c = \frac{w^2 vl}{pd^5} = \frac{w^2 l}{p^2 d^5} \text{ so } d = \left(\frac{w^2 l}{p^2}\right)^{\frac{1}{5}} c \quad . \quad . \quad (8)$$

Combining (3), (4) and (5)

$$wi = c \text{ or } w = c/\left(1 - \rho^{\frac{k-1}{k}}\right) \quad . \quad . \quad (9)$$

Also from (7) and (1)

$$W = c\left(d + \frac{pd}{2\sigma}\right)\left(\frac{pd}{2\sigma}\right) = cl\,\frac{pd^2}{2\sigma}\left(1 + \frac{p}{2\sigma}\right) \quad (10)$$

Then from (8) and (10) is obtained

$$W = \frac{clp}{2\sigma}\left(\frac{w^2 l}{p^2}\right)^{\frac{2}{5}}\left(1 + \frac{p}{2\sigma}\right) \quad . \quad . \quad . \quad (11)$$

or finally

$$W = cl^{\frac{7}{5}}\, p^{\frac{1}{5}}\left(1 + \frac{p}{2\sigma}\right)\Big/\left(1 - \rho^{\frac{k-1}{k}}\right)^{\frac{4}{5}} \quad . \quad (12)$$

Values of the relative weight $\frac{W_2}{W_1}$ obtained by substituting values of p in this expression are shown in Fig. 25 using reference values p_1 = 615 lb. per sq. in. abs., p_e = 0·75 lb. per sq. in. abs. It is apparent that the piping weight alone increases appreciably as the pressure is increased.

The weight of insulation has been neglected in the above analysis but experience indicates that it must be considered, as in many cases the insulation will be found to weigh nearly as much as the pipe. Thus if one writes:—

where W_p = weight of piping
W_I = weight of insulation
∂_s = steel density lb. per cu. in.
∂_I = insulation density lb. per cu. in.
t_I = insulation thickness, inch
(assumed constant = 4 inch)

$$\Sigma W = W_p + W_I \quad . \quad . \quad . \quad (13)$$

$$W_I = \pi\partial_I l t_I\,(d + 2t + t_I) \quad . \quad (14)$$

or combining (14) and (1)

$$W_I = \pi l \partial_I t_I\left[d\left(1 + \frac{p}{\sigma}\right) + t_I\right] \quad . \quad . \quad (15)$$

Thus (10) and (15)

$$\Sigma W = \pi\partial_s lpd^2\left(1 + \frac{p}{2\sigma}\right) + \left(\pi\partial_I\,\frac{t_I}{d}\right) ld^2\left(1 + \frac{p}{\sigma} + \frac{t_I}{d}\right) (16)$$

$$\Sigma W = \pi ld^2\left[\frac{\partial_s p}{2\sigma}\left(1 + \frac{p}{2\sigma}\right) + \left(\partial_I\,\frac{t_I}{d}\right)\left(1 + \frac{p}{\sigma} + \frac{t_I}{d}\right)\right] (17)$$

or finally

$$\Sigma W = \frac{\pi\partial_s l^{\frac{7}{5}}}{p^{\frac{4}{5}}\left(1 - \rho^{\frac{k-1}{k}}\right)^{\frac{4}{5}}} \times \left[\frac{p}{2\sigma}\left(1 + \frac{p}{2\sigma}\right) + \left(\frac{\partial_I}{\partial_s}\,\frac{t_I}{d}\right)\left(1 + \frac{p}{\sigma} + \frac{t_I}{d}\right)\right] . \quad (18)$$

In this case the three constants ∂_s, ∂_I and $\frac{t_I}{d}$ have relative significance. Numerical results were obtained by assuming $t_I/d = \frac{1}{2}$, ∂_s = 0·283, and ∂_I = 0·011. Values are shown on Fig. 26, and it is apparent that on the above basis the total weight of piping and insulation decreases with increased pressure.

(2) *Boiler pressure tubing—effect of increased pressure at constant temperature*

This case includes steam generating, superheater and economizer tubing in the boiler and differs from the steam piping only in that the tube surface is a function of the total heat to be transferred. Thus relations (1) to (7) of the former case still apply but in addition the proportions must satisfy the following:—

$$w\Delta h = Q \quad . \quad . \quad . \quad . \quad . \quad (19)$$

$$\frac{Q}{S} = U\Delta t = \text{constant, approximately} \quad . \quad . \quad (20)$$

$$S = cnl_1 d \quad . \quad . \quad . \quad . \quad (21)$$

where S = surface, sq. ft.
Q = total heat transferred, B.Th.U. per hour
U = heat transmission coefficient, B.Th.U. per hour per sq. ft. per deg. F.
Δh = unit heat transferred, B.Th.U. per lb.
Δt = overall mean temperature difference, deg. F.
n = number of tubes in parallel flow
l_1 = length of each tube, inch.

In practice d would probably remain constant, but to obtain the minimum weight increase it is necessary to assume that d may be varied as well as n and l_1.

The unit heat transferred Δh will be less due to a reduction in enthalpy with increased pressure both at saturation and at the superheater outlet. Also the inlet feed temperature should be increased to correspond to the higher pressure. A fairly close approximation to the overall effect for the combination of all three types of boiler surface is:—

$$\Delta h = c\,w \quad . \quad . \quad . \quad . \quad . \quad (22)$$

Thus

$$Q = c\,w^2 \quad . \quad . \quad . \quad . \quad . \quad (23)$$

and from (20) and (21)

$$\frac{w^2}{nl_1\,d} = \text{constant or } nl_1\,d = cw^2 \quad . \quad . \quad (24)$$

Returning to equation (6)

$$\frac{\Delta p}{p} = c = \frac{w^2 l_1}{n^2 p^2 d^5}\ So\ \frac{n^2 d^5}{l_1} = \left(\frac{w}{p}\right)^2 \quad . \quad . \quad (25)$$

Equations (24) and (25) define the requirements which must jointly be met by the three variables l_1, n and d. One more condition is needed to define them individually. If equation (24) is substituted in the weight formula (10), there results:—

$$W = \frac{cnl_1}{2\sigma}\,d^2 p\left(1 + \frac{p}{2\sigma}\right) = cw^2 dp\left(1 + \frac{p}{2\sigma}\right) \quad (26)$$

Examination of equation (26) indicates that a further condition which will result in minimum weight is

$$dp = \text{constant} \quad . \quad . \quad . \quad . \quad (27)$$

or

$$W = c\left(1 + \frac{p}{2\sigma}\right)\Big/\left(1 - \rho\,\frac{k-1}{k}\right)^2 \quad . \quad .(28)$$

Again the relative weight is obtained by substitution of numerical values. The results on Fig. 25 indicate a slight decrease in weight of boiler heat exchanger tubing with higher pressure.

However, it should be emphasized that if the tube outside diameter is required to remain constant, an increase in weight will result rather than a decrease. The weight formula for this case is

$$W = \left(\frac{c\,p}{\sigma + p}\right)\left[1 - \frac{p}{2\,(\sigma + p)}\right]\Big/\left(1 - \rho\frac{k-1}{k}\right)^2 \quad (29)$$

It will be noted that the heat transfer coefficient U was assumed constant in the above analysis. Certainly this is justified for boiler pressure heat exchange elements where the gas side coefficient is dominant.

(3) *Feed water heaters—effect of increased steam pressure*

In this case the water side coefficient is dominant and varies with $V^{0\cdot 8}/d^{0\cdot 2}$. Thus:—

$$U = c\,V^{0\cdot 8}/d^{0\cdot 2} \quad . \quad . \quad . \quad . \quad (30)$$

$$V = c\,\frac{wv}{nd^2} \quad . \quad . \quad . \quad . \quad (31)$$

Equations (19), (20) and (21) also apply except that the heat transfer coefficient U is no longer constant. Furthermore, there should be an increase in Δh and Δt to correspond to the increased feed temperature desired with higher steam pressure. The following assumptions appear to be a reasonable approximation:—

$$w\Delta h = \text{constant} \quad . \quad . \quad . \quad . \quad (32)$$

$$\Delta h = \Delta t \quad . \quad . \quad . \quad . \quad (33)$$

From which it follows that

$$Q = \text{constant} \quad . \quad . \quad . \quad . \quad (34)$$

and

$$U = c/S\Delta t = \frac{cw}{S} = \frac{cw}{nl_1 d} = \frac{cw}{nl_1} \quad . \quad . \quad (35)$$

since in practice d would remain constant.

Also from equations (30) and (31)

$$U = c\left(\frac{wv}{nd^2}\right)^{0\cdot 8}\frac{1}{d^{0\cdot 2}} = c\left(\frac{w}{n}\right)^{0\cdot 8} \quad . \quad . \quad (36)$$

since v is practically constant for water.

Thus combining (35) and (36)

$$c\left(\frac{w}{n}\right)^{0\cdot 8} = \left(\frac{w}{n}\right)\frac{1}{l_1}\text{ and } l_1 = c\left(\frac{w}{n}\right)^{0\cdot 2} \quad . \quad (37)$$

Now from equation (25)

$$\frac{\Delta p}{p} = c = \frac{w^2 l_1}{n^2 p^2 d^5} = \frac{w^2 l_1}{n^2 p^2} = \left(\frac{w}{n}\right)^{2\cdot 2}\frac{1}{p^2}\ so\ n = \frac{cw}{p^{\frac{10}{11}}} \quad (38)$$

From (37) and (38)

$$l_1 = c\left(\frac{wp^{\frac{10}{11}}}{w}\right)^{0\cdot 2} = c\,p^{\frac{2}{11}} \quad . \quad . \quad . \quad (39)$$

Thus in equation (26)

$$W = cnl_1 d^2 p\left(1 + \frac{p}{2\sigma}\right) = cnl_1 p\left(1 + \frac{p}{2\sigma}\right) =$$

$$c\ \frac{w}{p^{\frac{10}{11}}}\,p^{\frac{2}{11}}p\left(1 + \frac{p}{2\sigma}\right). \quad . \quad (40)$$

and finally

$$W = c\,p^{\frac{3}{11}}\left(1 + \frac{p}{2\sigma}\right)\Big/\left(1 - \rho\,\frac{k-1}{k}\right) \quad . \quad . \quad (41)$$

These results are also shown on Fig. 25, and it will be noted that the weight increases at about the same rate as the weight of steam piping only.

(4) *Boiler furnace, brickwork, insulation and supporting structure—effect of higher pressure or temperature*

Assuming the furnace to be a cube with walls of constant thickness, then the weight of brickwork, insulation casing and supporting structure will vary as the 2/3 power of the firing rate if no changes are made in furnace exit temperature or loading factors. The resulting decreases in relative weight, as shown on Figs. 25 and 26, are seen to be comparatively small.

(5) *Boiler drum*

Aside from the main propulsion turbines only this one important special case remains. Due to the reduction of liquid to steam density ratio with higher pressures, it may be assumed that the disengaging surface cannot be reduced for such pressures even though the weight flow of steam is reduced. If the drum diameter remains constant, the following simple relations result:—

(1) $pd/t = 2\sigma$, constant

$$d = \text{constant} \quad . \quad . \quad . \quad . \quad (42)$$

$$l = \text{constant} \quad . \quad . \quad . \quad . \quad (43)$$

$$(10)\quad W = c\frac{lpd^2}{2\sigma}\left(1 + \frac{p}{2\sigma}\right)$$

or

$$W = cp\left(1 + \frac{p}{2\sigma}\right) \quad . \quad . \quad . \quad (44)$$

Thus, as shown in Fig. 25, the boiler drum weight increases more rapidly with pressure than any other element in the power system.

(6) *Turbines—effect of increased pressure at constant temperature*

The weight variation of turbines with steam pressure is complicated by the introduction of rotor stress considerations, desirable ratios of blade height to diameter and other design factors. However, it is evident that the weight flow required must fall off at a lesser rate than the increase of available energy. Also the reduction in size due to increased specific density applies only to those parts of the turbine which operate at higher pressures than the prototype machine. Thus it is visualized that there may be a weight reduction in the moderate to low pressure part of the turbine due to reduced weight flow which is counterbalanced by increased weight in the initial stages. Therefore, little if any turbine weight saving is to be expected with increased pressure.

(7) *Effect of increased temperature at constant pressure*

The most important effects of higher steam temperature are the reduction in weight flow due to a substantial increase in available energy and the corresponding increase in specific volume of superheated steam. The available energy increases approximately with the square root of absolute temperature but specific volume increases as the first power of absolute temperature.

For various elements of the plant these opposing effects work out as follows:—

(a) Superheated steam piping.	No change unless stress is lowered.
(b) Desuperheated steam and boiler feed piping.	$W = c/(T)^{\frac{2}{5}}$
(c) Boiler tubes in steam generating and economizer sections (d_0 constant).	$W = c/T^{\frac{1}{2}}$
(d) Feed heaters and boiler drums (d constant in both).	$W = c/T^{\frac{1}{2}}$

Values are shown in Fig. 26.

Turbine weight increases due to increased available energy which requires an increase in the Parsons coefficient, Σd^2N^2, unless such increase can be obtained entirely by an increase in rotating speed N.

Appendix II

Basis for Fuel Rate Improvements for Higher Pressures and Temperatures

The numerical procedure involved in calculating fuel rate improvements for higher steam conditions is described in Appendices 1 and 2 of reference (4) and will not be repeated here.

The most important change in concept is that the short cut procedure for determining feed heating energy removed from the turbine was amplified by a set of correction curves to express the difference between short cut and exact energy extractions for feed heating.

Exact values were obtained by constructing estimated turbine internal condition lines for various initial steam conditions and powers, and obtaining the necessary enthalpy and available energy values from these condition lines at the various heater extraction points. Equal heater temperature rises were used and no losses were assigned outside the turbines and the use of the data assumes mixing or booster pump type heaters with zero terminal difference. Pumping losses were also ignored. A derivation of this method of computing the energy loss quantities is given in Fig. 23 in condensed form.

A comparison of exact values with short form data is shown in Fig. 24. Throttle entropy was selected as a parameter which most suitably expressed the variation in average superheat or moisture content at the various extraction points [23]. Further variations arise, as shown in Fig. 24, due to the changing slope of the turbine condition lines with power rating and due to changes in the number of feed heating stages desired.

A minor change was made from the original formulation of an allowance for auxiliaries and hotel load, the factor a shown on Fig. 1. The original data were reviewed and this factor standardized to provide equal percentage increments for higher pressures and temperatures. The expression selected for a is:—

$$a = a_0\left(1 + \frac{\Delta t}{2500}\right)\left(1 + \frac{\Delta p}{4000}\right)$$

where a_0 = base value at 600 lb. per sq. in. gauge, 850 deg. F. at superheater outlet

Δt = temperature increase over 850 deg. F.

Δp = pressure increase over 600 lb. per sq. in. gauge.

Representation of the results has been extended to show the separate effects of a greater range of independent variables. It was found that doubling the base load value a_0 had a secondary effect on the result which varied with both temperature and pressure. This is natural since the a factor works against the improvement in propulsion machinery performance. This effect is taken care of by the final family of curves in the lower left corner which are labelled $a_0 = 0{\cdot}16$.

A note on the construction of the turbine condition lines may be of interest. These must, of course, agree with the overall engine efficiencies shown in Fig. 2 corrected for pressure and superheat as indicated in Appendix 2 of reference (4), and with a further allowance for external losses. It was found that these lines could be calculated through from the initial conditions and arrive at the required end points on the basis of the following considerations:—

(1) Basic stage efficiency in superheat region increases slightly with power.

(2) Additional losses in the initial stages down to one half of inlet pressure are proportional to inlet volume flow.

(3) Additional losses in moisture region are proportional to average moisture content.

(4) Leaving loss is constant except for changes in exhaust moisture content.

(5) External losses (bearings, gears, glands, and astern turbine windage) in B.Th.U. per lb. are reduced at higher powers.

Appendix III

Superheater Characteristics as related to metal temperature

(1) *Heat transfer relations*

Notation

W_s Weight flow of superheated steam, lb. per hr.

Δh_s Heat added in superheater per lb. steam, B.Th.U. per lb.

Q_s Total heat absorbed in superheater, 1,000 B.Th.U. per hr.

W_g Gas weight flow past superheater, lb. per hr.

t_w Saturation temperature at superheater inlet pressure, deg. F.

t_s Superheater outlet steam temperature, deg. F.

t_3 Superheater inlet gas temperature, deg. F.

t_4 Superheater outlet gas temperature, deg. F.

Δh_g Heat absorbed by superheater per lb. gas, B.Th.U. per lb.

A_s Superheater outside tube surface, sq. ft.

A_g Mean gas flow area through superheater, sq. ft.

H Unit rate of total heat absorption in superheater, 1,000 B.Th.U. per sq. ft. per hr.

H_c Unit rate of convection heat absorption, 1,000 B.Th.U. per sq. ft. per hr.
H_r Unit rate of radiant heat absorption, 1,000 B.Th.U. per sq. ft. per hr.
Δ_m Corrected logarithmic temperature difference, deg. F.
C_{ps} Mean specific heat of superheated steam, B.Th.U. per lb. per deg. F.
C_{pg} Mean specific heat of gases, B.Th.U. per lb. per deg. F.
U_c Overall convective heat transfer coefficient, B.Th.U. per hr. per sq. ft. per deg. F.
U_g Gas side heat transfer coefficient, B.Th.U. per hr. per sq. ft. per deg. F.
U_t Tube heat transfer coefficient, B.Th.U. per hr. per sq. ft. per deg. F.
U_s Steam side heat transfer coefficient, B.Th.U. per hr. per sq. ft. per deg. F.
t_f Average gas film temperature, deg. F.
t_r Mean radiating gas temperature, deg. F.
t_t Outside tube temperature, deg. F.
p Absorption factor for radiation,

The following relations are fundamental:—

$$H = \frac{Q_s}{A_s} = \frac{W_s\,h_s}{1{,}000\,A_s} = \frac{C_{ps}W_s\,(t_s - t_w)}{1{,}000\,A_s} \quad . \quad (1)$$

$$H = \frac{W_g\,C_{pg}\,(t_3 - t_4)}{1{,}000\,A_s} \quad . \quad . \quad . \quad (2)$$

$$\Delta_m = C6\,\frac{(t_3 - t_s) + (t_4 - t_w)}{2} \quad . \quad (3)$$

Where $C6$ is a constant which includes ratio of logarithmic to arithmetic mean temperature difference and arrangement correction factor.

$$U = 1{,}000\,H_c/\Delta_m \quad . \quad . \quad . \quad . \quad (4)$$

$$\frac{1}{U} = \frac{1}{U_g} + \frac{1}{U_s} + \frac{1}{U_t} \quad . \quad . \quad . \quad . \quad (5)$$

$$H = H_c + H_r \quad . \quad . \quad . \quad . \quad . \quad (6)$$

$$H_r = 1{\cdot}72p\left[\left(\frac{t_r + 460}{1{,}000}\right)^4 - \left(\frac{t_t + 460}{1{,}000}\right)^4\right] \quad . \quad (7)$$

Also let $C_1 = W_g/W_s$ and assume that C_1 is constant for purposes of this study.

Then combining (1) and (2):—

$$t_3 - t_4 = \frac{C_{ps}}{C_1C_{pg}}\,(t_s - t_w)$$

The gas side heat transfer coefficient is dominant and may be expressed fairly accurately as

$$U_g = 0{\cdot}177\,C_2C_5\left(\frac{W_g}{A_g}\right)^{0{\cdot}55} \quad . \quad . \quad . \quad (8)$$

where

C_2 = correction for average gas film temperature which will be assumed constant.
C_5 = correction for tube size and arrangement factors which will also be assumed constant.

Since U_g is dominant, it may be said that
$U = C_3C_7U_g$

Where C_3 is a constant correction for the effect of the tube and steam side coefficients U_t and U_s, and C_7 is a similar constant representing the clean tube factor.

$$\text{Therefore,} \quad U = 0{\cdot}177\,C_2C_3C_5C_7\left(\frac{C_1W_s}{A_g}\right)^{0{\cdot}55} \quad . \quad . \quad (9)$$

Now $A_s = \pi\,n\,d\,l/12$
with n = number of tubes in superheater
d = tube outside diameter, inch.
l = tube length, feet with tube axis normal to gas flow
n = $144\,x\,y/p^2$
where p = tube pitch, assumed same in both directions, inch.
x = superheater bank depth in direction of gas flow, feet
y = superheater bank height, feet

So $A_s = \dfrac{12\pi d\,x\,yl}{p^2}$

Also

$$A_g = \left[y - \frac{d}{12}(12y/p)\right] l = yl\left(1 - \frac{d}{p}\right)$$

and $A_s = \dfrac{12\,\pi\,d\,x\,yl}{p^2\,yl\left(1 - \frac{d}{p}\right)} = \dfrac{12\pi\,d\,x}{p^2\left(1 - \frac{d}{p}\right)} = C_4$, assumed constant.

Therefore,

$$U = 0{\cdot}177\,C_2C_3C_5C_7\left(\frac{C_1C_4W_s}{A_s}\right)^{0{\cdot}55} \quad . \quad . \quad . \quad (10)$$

Now combining equations (1) (4) (6) (7) and (9) results in :

$$H_c = -\,H_r + H$$

And

$$U = 0{\cdot}177\,C_2C_3C_5C_7\left(C_1C_4\frac{W_s}{A_s}\right)^{0{\cdot}55} = \left\{-1{\cdot}72p\left[\left(\frac{t_r + 460}{1{,}000}\right)^4 - \left(\frac{t_t + 460}{1{,}000}\right)^4\right] + \frac{C_{ps}W_s\,(t_s - t_w)}{1{,}000\,A_s}\right\}\frac{1{,}000}{\Delta m} \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

Now $t_r = \dfrac{t_3 + t_4}{2}$ approximately and $\left(\dfrac{t_t + 460}{1{,}000}\right)^4$ is negligible in comparison.

So

$$0{\cdot}177\,C_2C_3C_5C_7\left(C_1C_4\,\frac{W_s}{A_s}\right)^{0{\cdot}55} = \left[C_{ps}\left(\frac{W_s}{A_s}\right)(t_s - t_w) - 1720p\left(\frac{t_3 + t_4 + 920}{2{,}000}\right)^4\right]\frac{1}{\Delta_m} \quad . \quad . \quad . \quad (12)$$

But

$$\Delta_m = C_6\left[\frac{t_3 + t_4 - (t_s + t_w)}{2}\right] = C_6\left[\frac{2t_3 - \frac{C_{ps}}{C_1C_{pg}}(t_s - t_w) - (t_s + t_w)}{2}\right] \quad (13)$$

So

$$0{\cdot}177\,C_2C_3C_5C_7\left(\frac{C_1C_4W_s}{A_s}\right)^{0{\cdot}55} = \frac{C_{ps}\left(\frac{W_s}{A_s}\right)(t_s - t_w) - 1{,}720p\left[\frac{2t_3 - \frac{C_{ps}}{C_1C_{pg}}(t_s - t_w) + 920}{2{,}000}\right]^4}{\frac{C_6}{2}\left[2t_3 - \frac{C_{ps}}{C_1C_{pg}}(t_s - t_w) - (t_s + t_w)\right]} \quad (14)$$

Now t_w may be assumed constant for the present purpose. Therefore, the above expression may be solved for $\dfrac{W_s}{A_s}$ in terms of the two other variables t_3 and t_s.

Resulting in:

$$\frac{W_s}{A_s}\underbrace{\left[C_{ps}\,(t_s - t_w)\right]}_{\text{A}} - \left(\frac{W_s}{A_s}\right)^{0{\cdot}55} \quad . \quad . \quad . \quad . \quad (15)$$

$$\times\ \underbrace{\frac{0{\cdot}177}{2}\,C_2C_3C_5C_6C_7\,(C_1C_4)^{0{\cdot}55}}_{\text{B}}$$

$$\times\ \underbrace{\left[2t_3 - (t_s - t_w)\left(\frac{C_{ps}}{C_1C_{pg}}\right) - (t_s + t_w)\right]}_{\text{C}}$$

$$= 1{,}720p\left[\frac{2t_3 + 920 - \frac{C_{ps}}{C_1C_{pg}}(t_s - t_w)}{2{,}000}\right]^4$$

or $\quad A\left(\dfrac{W_s}{A_s}\right) - BC\left(\dfrac{W_s}{A_s}\right)^{0{\cdot}55} = DT_r{}^4$

It is useful to review the meaning of the various terms in equation (15); the left hand group is $H - H_c$, i.e. the total heat absorbed minus the convection heat transfer which must, of course, equal the radiant heat transmission on the right hand side of equation (15). Other groupings have the significance indicated below:

$$T_r = t_r + 460 = \frac{2t_3 + 920 - \frac{C_{ps}}{C_1 C_{pg}}(t_s - t_w)}{2}$$

$$2\Delta_m = 2t_3 - (t_s - t_w)\frac{C_{ps}}{C_1 C_{pg}} - (t_s + t_w)$$

Numerical solution of equation (15) is accomplished by trial and error. The constants used in preparing the curves for Fig. 13 were as follows:

$t_w = 500$	$C_1 = 1{\cdot}1$	$C_s = 0{\cdot}85$
$C_{ps} = 0{\cdot}68$	$C_2 = 1{\cdot}15$	$C_6 = 0{\cdot}94$
$C_{pg} = 0{\cdot}31$	$C_3 = 0{\cdot}96$	$C_7 = 1{\cdot}0$
$p = 0{\cdot}07$	$C_4 = 50$	

(2) *Determination of Maximum Metal Temperature*

It is not practicable to make direct use of the values shown in Fig. 13 for estimating maximum metal temperatures for the following reasons:

(1) The overall heat transmission rates previously established are averages for the entire superheater; it is necessary to assume that local convection heat transfer coefficients will exceed these by about 20 per cent due to unequal inlet gas velocity and temperature distribution.

(2) The maximum metal temperature will be near the superheater outlet and in a zone where the gas temperature is high. For the present purpose it is assumed that the superheater outlet is located in the gas inlet zone, although this is not always the case in practice. This greatly increases the total radiant heat transmission above the average value.

(3) The maximum metal temperature depends to a considerable extent on the inside and tube wall coefficients U_s and U_t whose effects were small enough in the previous study to be lumped together as a constant correction factor, C_3.

(4) The tube metal temperature cannot be neglected in determining radiant absorption.

The curves in Fig. 14 were, therefore, prepared as follows:

(1) Values of the overall convection heat transfer coefficient, U_c, were obtained from the data used to prepare Fig. 13 at selected values of the gas inlet temperature t_3 and desired steam temperature t_s. The local overall coefficient U_c^1, at the superheater outlet was then established as $1{\cdot}20\ U_c$.

(2) An assumed value of 300 was used for the inside heat transfer coefficient U_s. The tube wall thickness was taken as 0·18 which with a thermal conductivity of 200 B.Th.U. per inch per hr. per sq. ft. per deg. F. results in a tube wall coefficient U_t of 1,100. This conductivity value was chosen to represent a low chromium molybdenum alloy at the average metal temperature and was obtained from reference (26).

(3) The following relations must hold:

(a) $H_c^1 = H^1 - H_r^1$

(b) $t_m = t_s + H^1\left(\frac{1}{U_s} + \frac{1}{U_t}\right)$

(c) $H_r^1 = 1{\cdot}72p\left[\left(\frac{t_3 + 460}{1{,}000}\right)^4 - \left(\frac{t_m + 460}{1{,}000}\right)^4\right]$

(d) $U_c^1 = \frac{H^1{}_c}{t_3 - t_m}$

(4) Numerical results were obtained by a trial and error procedure beginning with an assumed value of H^1 which is then used to calculate the metal surface temperature, t_m from equation (b). With this it is possible to compute the radiant heat absorption H_r^1 from equation (c). This value is then subtracted from the total transmission H^1 to obtain the convective heat transfer H_c^1 (equation (a)).

The value of U_c^1 is then calculated from equation (d) and compared with the desired value of $1{\cdot}2\ U_c$. This process is repeated until a value of H^1 is obtained which gives the desired result.

Appendix IV

Discussion of Equivalent Hoop Stress Formulæ for Piping

The application of creep data to design of piping has been studied by Bailey(40), Marin(41), Soderberg(42), Nadai(94), and others but most recently by Buxton and Burrows(38) and Rankin(39). A comparison of Bailey's and Nadai's work with the more recent proposals of Buxton and Burrows and of Rankin is given below in order to explain the major differences in viewpoint.

For convenience the following generally recognized notation will be used throughout:

Notation

- p = internal pressure, lb. per sq. in.
- r = radius, inch
- σ_r = radial stress, lb. per sq. in.
- σ_t = tangential stress, lb. per sq. in.
- σ_z = axial stress, lb. per sq. in.
- σ_{to} = stress in equivalent simple tension test, lb. per sq.in.
- d = inside diameter inch.
- t = wall thickness inch.
- $y = d/t$.
- v_r, v_t, v_z are the corresponding creep rates inch per inch per hr.
- Φ is the stress deviator or shear distortion energy function which is proportional to the octahedral shear stress.
- c is the creep rate constant which expresses proportionality between creep rate and the stress deviator.
- Subscripts 1 and 2 refer to inner and outer pipe walls, i.e., r_1 and r_2 are the inner and outer radii.
- τ = principal shear stress, lb. per sq. in.
- m = power exponent in the proportionality relating creep rate to the stress deviator.
- n = power exponent in the creep rate—stress relation.
- A = numerical constant in the creep rate—stress relation.

Bailey's Solution

Bailey develops general expressions for the three principal creep rates v_r, v_t and v_z in terms of the three principal stresses σ_r, σ_t, and σ_z on the basis that the creep rate is proportional to a power "m" of the stress deviator Φ or shear strain energy term and also proportional to a power "q" of the shear stresses on planes normal to the direction of creep.

It has been confirmed by careful measurements that the axial creep rate v_z is zero. In addition, it has also been established that there can be no appreciable volume change. This leads to the conclusion that v_t is proportional to the radius squared.

These conditions provide all the necessary information to establish the stress-strain relations in the pipe under creep conditions as follows:

$$v_r = c\,[(\sigma_r - \sigma_t)^q + (\sigma_r - \sigma_z)^q] \quad . \quad . \quad . \quad (1)$$

$$v_t = c\,[(\sigma_t - \sigma_z)^q + (\sigma_t - \sigma_r)^q] \quad . \quad . \quad . \quad (2)$$

$$v_z = c\,[(\sigma_z - \sigma_r)^q + (\sigma_z - \sigma_t)^q] \quad . \quad . \quad . \quad (3)$$

$$c = \Phi m = \left\{\frac{1}{2}[(\sigma_r - \sigma_t)^2 + (\sigma_z - \sigma_r)^2 + (\sigma_t - \sigma_z)^2]\right\}^m \quad (4)$$

$$v_z = 0 \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$v_t = \left(\frac{r_1}{r}\right)^2; v_{t\,1} = \left(\frac{r_2}{r}\right)^2 v_{t2} \quad . \quad . \quad . \quad . \quad (6)$$

for the case of simple tension, measured creep rates for a stable material may be expressed satisfactorily by the relation

$$V = A\,\sigma n \text{ and from the above} \quad . \quad . \quad . \quad (7)$$

$$n = 2m + q \text{ or } q = n - 2m \, . \quad . \quad . \quad . \quad (8)$$

It may be shown that equation (5) requires:

$$\sigma_z = \frac{\sigma_r + \sigma_t}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

The equilibrium condition for a small volume element is:

$$\sigma_t - \sigma_r = r\frac{d\sigma_r}{d_r}$$

Equation (10) can be expressed in terms of r, σ_r, and its derivative and then integrated. The constant of integration is obtained from the boundary conditions $\sigma_{r1} = -p$ and $\sigma_{r2} = O$. Finally the various stresses and creep rates may be expressed in terms of the pipe dimensions $r_1\,r_2$, the internal pressure p, and the creep constants A, m and n.

The important results for the present purposes are

$$v_{t1} = \frac{A}{2}\left(\frac{3}{4}\right)^m\left[1 + \left(\frac{1}{2}\right)^{n-2m}\right]\left(\frac{2}{n}\right)^n\left[\frac{p}{1-(r_1/r_2)^{2/n}}\right]^n \quad (11)$$

$$\tau_1 = \frac{\sigma_{t1} - \sigma_{r1}}{2} = \frac{1}{n}\left[\frac{p}{1 - (r_1/r_2)^{2/n}}\right] \quad . \quad . \quad . \quad (12)$$

For the case of simple tension the corresponding equations are

$$v_{to} = A\,\sigma_{to}n \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (13)$$

$$\tau_o = \frac{\sigma_{to}}{2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (14)$$

Bailey uses the inner bore tangential creep rate v_{t1} as a criterion and places this equal to the creep rate in a simple tension test with the result:

[6]* $$\frac{\sigma_{to}}{p} = \frac{2}{n}\left[\frac{3}{4}\right]^{\frac{m}{n}}\left[\frac{1 + \left(\frac{1}{2}\right)^{n-2m}}{2}\right]^{1/n}\left[\frac{1}{1 - (r_1/r_2)^{2/n}}\right] \quad (15)$$

Somewhat different results are obtained by Buxton-Burrows and by Nadai, and there are considerable differences in the interpretations, as will be shown.

Nadai's Solution

Nadai(94) obtains somewhat simpler expressions than Bailey, as he assumes that the creep rate equation (13) above applies directly to the octahedral shearing stresses and strains in the multi-axial stress condition.

Thus he arrives at a general expression for tangential stress:

$$\frac{\sigma_t}{p} = \frac{(r_1/r_2)^{2/n}}{1 - (r_1/r_2)^{2/n}}\left[1 - \frac{1 - 2/n}{(r/r_2)^{2/n}}\right] \quad . \quad . \quad . \quad (16)$$

Nadai states that this is to be evaluated at the *outer* wall of the pipe where the tangential stress is a maximum under creep conditions, resulting in:

[8] $$\frac{\sigma_{to}}{p} = \frac{\sqrt{3}}{n}\left[\frac{(r_1/r_2)^{2/n}}{1 - (r_1/r_2)^{2/n}}\right] = \quad . \quad . \quad . \quad . \quad . \quad (17)$$

$$\frac{1}{2\sqrt{3}}\left[\frac{1}{(1 + 2/y)^{1/3} - 1}\right] \text{ for } n = 6$$

Values of this equation have been indicated in Fig. 11 as a percentage difference from Bailey's equation [8].

Buxton and Burrows obtain stress equations which are the same as those used by Bailey and Nadai but for reasons which are not too clear to the authors use the shear stress at the inner bore τ_1, as a criterion and place this equal to the shear stress in the simple tension test resulting in:

[5] $$\frac{\sigma_{to}}{p} = \frac{2\tau_o}{p} = \frac{2}{n}\left[\frac{1}{1 - (r_1/r_2)^{2/n}}\right] \quad . \quad . \quad . \quad . \quad (18)$$

This last expression is seen to lack two terms which appear in the Bailey equation. Using the values $m = 2$ $n = 6$ recommended by Bailey it will be found that equation [6] equals about 0·85 times equation [5]. It will be seen that the Buxton-Burrows result also does not agree with Nadai's expression [8].

Rankin's Work

Rankin(39) makes some very interesting proposals with regard to a rupture strength criterion. He first evaluates the average shear stress which is:

$$\tau_{ay} = \frac{p}{2}\left(\frac{r_1}{t} + \frac{1}{2}\right) = \frac{p}{y}(y + 1) \text{ where } y = d/t \quad . \quad (19)$$

then on the basis of indirect shearing rupture tests it is concluded that the rupture stress in shear is 65 per cent of that obtained in a simple tension test, thus

$$0{\cdot}65\frac{\sigma_{to}}{p} = \frac{\tau_{ay}}{p} \text{ or} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (20)$$

[9] $$\frac{\sigma_{to}}{p} = \frac{1}{0{\cdot}65 \times 4}(y + 1) \quad . \quad . \quad . \quad . \quad (21)$$

As a more conservative shear rupture stress criterion Rankin also suggests the use of inner wall shear stress.

$$\tau_1 = \frac{p}{6}\left[\frac{1}{1 - \left(\frac{1}{1 + 2/y}\right)^{1/3}}\right] \quad . \quad . \quad (22)$$

then using a relation similar to (20) there results

$$\frac{\sigma_{to}}{p} = \frac{1}{0{\cdot}65 \times 6}\left[\frac{1}{1 - \left(\frac{1}{1 + 2/y}\right)^{1/}}\right] \quad . \quad (23)$$

This result is seen to be the same as dividing the so-called Bailey Nadai formula [5] by 1·3 or using 0·65 for the shear to tensile rupture ratio instead of one half.

Other Criteria

Buxton and Burrows also discuss the so-called Mises formula:

$$\frac{\sigma_{to}}{p} = \frac{\sqrt{3}}{4}\left(y + 3 + \frac{1}{y + 1}\right) \quad . \quad . \quad . \quad (24)$$

This formula expresses the condition for the beginning of plastic flow at the inner wall (Freudenthal reference (43) p. 432). This result is obtained by requiring that the stress deviator at the inner wall be equal to the corresponding tensile test stress limit. Use of this formula does not appear admissible since it is a criterion only for the *beginning* of plastic flow at the inner wall.

However, if the same criterion is applied to average wall stresses in the elastic state, the relation becomes:

$$\frac{\sigma_{to}}{p} = \frac{\sqrt{3}}{4}\left(y + 1 + \frac{1}{y + 1}\right) \quad . \quad . \quad (25)$$

which gives numerical results quite close to the Beliaev-Sinitski criterion [7] for a completely plastic state.

One further criterion suggested by the authors also arises from a consideration of the shear distortion energy in the plastic state. If this criterion is applied at the inner wall, there results:

$$\frac{\sigma_{to}}{p} = \frac{\sqrt{3}}{n}\left[\frac{1}{1 - (r_1/r_2)^{2/n}}\right] = \quad . \quad . \quad . \quad . \quad . \quad (26)$$

[5] $$\frac{1}{2\sqrt{3}}\left[\frac{1}{1 - \left(\frac{1}{1 + 2y}\right)^{1/3}}\right]$$

(for $n = 6$)

* Numbers in brackets [ ] refer to formulæ as numbered in Fig. 11.

This result is seen to be slightly more conservative than Bailey's 1935 formula [6] but, less conservative than Nadai's formula [8], since the factor $(r_1/r_2)^{2/n}$ has disappeared from the numerator. Since this expression is intermediate to the above formulæ, it might with some justification be described as the "Bailey-Nadai compromise" formula.

The remaining formulæ [1] [2] [3] [4] and [7] shown in Fig. 11 are generally self-explanatory or may be found in the references cited in the illustration.

This paper was read on Wednesday, 27*th June* 1951, *at the Central Hall, Westminster, London, S.W.*1, *at a meeting of the International Conference of Naval Architects and Marine Engineers, which was organized by the Institution of Naval Architects, Institute of Marine Engineers, Institution of Engineers and Shipbuilders in Scotland, and North-East Coast Institution of Engineers and Shipbuilders.*

DISCUSSION

Vice-Admiral(E) The Hon. D. C. Maxwell, C.B., C.B.E., said that it was most satisfactory to note that when naval architects and marine engineers held an international conference, it was truly international and leading men from other countries were prepared to put into the common pool the benefit of their experience and practice, to devote the time and effort that the preparation of a paper of this nature demanded, and to come over to this country to present it. If this free and whole-hearted co-operation existed in all fields of human intercourse, how much more smoothly might the world be running today !

The paper was one of very great merit. In it the authors had collected together a wide range of knowledge from many sources, to which they had added of their own great experience and they had assembled the whole in a manner which both provoked and stimulated discussion. It was an ideal paper for the members of the conference to get their teeth into, and he congratulated the authors upon it.

While the paper had been written from the aspect of merchant ship machinery, it might be of interest and perhaps of most value to the proceedings if he remarked on certain points from the Naval view of the same problem. It was necessary to appreciate that Naval machinery designs differed basically from those to meet mercantile requirements. One essential difference was that Naval machinery was utilized at full or high power for only a very small proportion of its life, and therefore they were prepared to sacrifice some full load efficiency to obtain optimum part load performance.

In the section dealing with control of equipment and operating problems, the authors stated that a steam temperature characteristic rising with boiler output was very desirable. In some Naval designs higher steam temperature at the lower powers was necessary in order to obtain the best possible efficiency at cruising speed; full use could thus be made of the fact that centrifugal stresses were much reduced below full power. A boiler characteristic of this type required control of steam temperature, and this allowed the temperature to be greatly reduced during manœuvring and whenever there were considerable and, more particularly, rapid changes in power.

Naval designs were largely controlled by considerations of space, for it was only by making every effort to minimize the space occupied by equipment that the best battle characteristics could be worked into the overall ship design. In the matter of turbines this had led to the use of high blade speeds with bypassed stages in the h.p. turbine and high leaving losses from the l.p. turbine. The authors had pointed out that with high temperature designs, bypassing of the initial stages of the h.p. turbine introduced difficulties in design. The alternatives involved additional space and weight and for Naval purposes this must be carefully balanced against the improved efficiency of the cycle.

Almost as important as the bulk of the machinery was the requirement for minimum weight of machinery plus fuel, and in some cases it paid them to design for 25 in. or 26 in. vacuum at full power instead of 28½ in. normal in the merchant service. The saving in weight of condenser and turbine allowed more fuel to be carried and increased the ship's radius of action, while at the lower powers at which they normally operated a high vacuum was readily obtained. Should they go to higher steam temperatures without greatly increasing the steam pressure, the l.p. turbine exhaust was likely to be considerably superheated; they had so far been unable to locate any positive evidence of the effect of this upon condenser design, and he would welcome any experience that the authors might have in this respect.

Little had been done in Naval engineering in the way of multi-stage bleed feed heating owing to the weight of the heaters and pipes involved, and to some extent to the complications that might arise when frequently changing power. Naval practice was to use non-condensing steam-driven auxiliaries, the exhaust from which was used for feed heating either in a pressure heater or de-aerator; this afforded the equivalent of single stage bleed feed heating, while preserving the essential lightness of the installation.

The use of higher steam temperatures posed a considerable problem in the layout of auxiliaries due to the complication of the steam and particularly of the exhaust pipes from those auxiliaries. The common solution in the merchant service was to use electric drive for all but the largest auxiliaries, but this as a complete solution was not so attractive to the Navy, because of the danger of interruption to electric supplies during battle.

One more difference between mercantile and Naval practice in design to which he wished to draw attention was that of cost. In merchant ship work, first costs, operating costs and maintenance costs were three guiding principles in deciding steam conditions. In a Naval design, they had, of course, to be certain that the taxpayer got value for money, value in this case being the military qualities of the ship, but they must also consider cost as being a yardstick of the effort required to produce, as manpower was the biggest single limitation of the country's effort in war. Not only must they reject a design which involved unduly complicated techniques, provided, of course, a less complicated design would serve the purpose, but they must also ensure that their designs in peace did not call for materials upon which they could not lay their hands in time of war. Stockpiling could not be relied upon as a full solution of a supply problem, and they might find that they had to limit themselves in steam conditions because of the danger of material shortages.

The paper had given all of them much food for thought, and he wished to express his appreciation and gratitude to the authors. He most certainly joined hands with them upon the desirability of going one stage beyond the economic optimum when designing advanced machinery. This was essential if they were to ensure in the future, as in the past, that marine engineers should always be capable of providing the best.

Mr. A. R. Gatewood (Member, I.Mar.E.) said that in reading the paper, he had noted that there were numerous references to the American Bureau's "Interim Guide" for making such installations, so it would seem proper at this time to point out that since formulating the Guide additional investigations had been completed which indicated that it should be altered in two places. Both of those investigations had been commented on in the paper.

The recommendation that the circularity of the pipe in way of bends be maintained within 2 per cent had been found to be somewhat restrictive from a practical point of view. Out of roundness resulted in a combined stress of a purely localized nature which would relieve itself very rapidly after the pipe was put into service. Combined stresses of the order of those tabulated in the Guide were felt to be safe. In some cases, however, this might result in an out of roundness tolerance of as much as 9 per cent, and since it was not felt desirable to encourage poor workmanship it was now recommended that the circularity of the pipe be maintained within 5 per cent, as this figure could easily be met.

The authors also explained why it was preferable to normalize low chrome-molybdenum steel pipes after bending. It had originally been thought that the pipes could be worked and welded and then subjected to the normalizing treatment. Subsequent investigations of that procedure, however, had indicated that normalizing lowered the rupture strength of the weld metal, so it was now recommended that after bending the chrome-molybdenum pipes should be normalized or annealed before welding but that the welds themselves should only be stress relieved even though this necessitated two heat treatments.

With regard to thermal shock from boiler water carry-over, the authors stated that they felt that this question might have received undue emphasis. It should be pointed out, however, that cracks had recently been experienced in several main superheated steam lines which were attributed to thermal shock, not from boiler water carry-over but from condensate carry-over from the combustion control line when the ship rolled. This was felt to be of particular importance since it had become routine practice to use combustion controls with those high temperature installations, and hundreds of successful installations had been made.

On the ship where the failures had occurred, there had been a relatively long, unlagged, horizontal run of the combustion control line, at the same level as the main steam line, before it dropped down to the combustion control board. With this arrangement the pipe could fill with condensate to the level of the horizontal lead and the condensate would periodically carry over into the steam line due to rolling of the ship. The good record of other installations indicated that there would be less chance of experiencing this type of thermal cracking if a generous vertical drop was provided in all dead-end lines at their point of connexion to superheated steam lines.

Dr. T. W. F. Brown (Member, I.Mar.E.) said he was sorry that in the paper the authors had not included any reference to reduction gearing in either the weight or materials sections, as it bore directly on the subject of the paper. It was evident that increased steam conditions would be accompanied by increased turbine speeds if high stage efficiencies were to be maintained, and since main shaft revolutions were tending if anything to become lower, the future trend was towards higher gear ratios. Thus, unless the rating of gears could also be increased, gearboxes would tend to become bulkier and heavier. The progress that was at present being made in gear-cutting technique, post-hobbing processes, and the use of stronger materials in order to increase gear ratings, was, it was felt, a necessary accompaniment to the use of higher steam conditions.

With reference to the choice of steam conditions, it was believed that for new construction in this country at the present time conditions of 650 per sq. in. gauge and 850 deg. F. would generally apply.

Did the figures given on the diagram in Fig. 3* represent pressure ranges? If so, it was difficult to see why the recommended pressures for the range 2,000 to 6,000 s.h.p. should be higher than for the range 6,000 to 10,000 s.h.p. It was felt that the authors' reference to increasing competition from Diesel propulsion at low and moderate powers did not completely represent the present position in this country, where the tendency was for single-screw turbine machinery to be used in preference to Diesel machinery for powers in the region of 7,000 s.h.p.

Difficulties were usually met when auxiliaries were driven by prime movers of relatively small powers in relation to advanced steam conditions. A possible arrangement would be to expand the feed-pump exhaust, together with bled steam from the main turbine, in a mixed-pressure turbine driving a generator, with live-steam supply for port use.

Regarding materials for high temperatures, the danger attending the extrapolation of creep curves was well known, and as such extrapolation, especially with low-alloy ferritic steels, must assume structural stability, the results obtained could be very wide of the mark. The data in Table III were very useful and admirably presented, and where comparison with British data was possible, had been found to be in reasonable agreement. It was unfortunate that the authors had been unable to include more information on the elongation at fracture for different stresses, temperatures and times. If more data of this sort were available, engineers would be in a position to check components after certain periods of operation and decide whether extension had been excessive for the known loads and how near they were to fracture.

In connexion with superheater materials, attention should be drawn to the very serious question of vanadium attack at temperatures in the region of 1,200 deg. F. The use of stabilized austenitic steel or other high-temperature alloys would not prevent this trouble, which at the moment seemed to impose a definite ceiling on permissible steam temperature.

On the question of materials for steam piping, development work in this country was being carried out on a ferritic steel of the CrMoWV type, but insufficient data were available at the moment regarding its long-term creep properties. MoV steel was favoured for the range 950 deg. F. to 1,050 deg. F., and the possibility of improving this steel by chromizing to prevent scaling was also receiving attention, with the object of obtaining better qualities at 1,050 deg. F. and above.

There was a definite trend in this country to use 18/13/1 CrNiCb steel in place of 18/8 Cb-stabilized in view of its much higher creep strength. The extent to which the CrMo steels had been developed in the U.S.A. had probably been influenced by the extensive occurrence of graphitization of the C-Mo steels that they had experienced. This was due to the American practice of deoxidizing with aluminium, additions of as much as 2 lb. per ton being quite common. British steelmakers rarely exceeded 2 to 6 oz. per ton, and failures due to graphitization were unknown. The addition of chromium was known to prevent graphitization.

The authors stressed the need for careful inspection of turbine castings intended for high pressures and temperatures and re-examination after repair of defects by welding. It should also be added that stress-relieving treatment was essential, especially after repair by welding.

The standard low-carbon 12 per cent chromium steel for blading, referred to on page 240, was equivalent to the stainless iron used in this country, and the authors' comment that it had proved entirely adequate for high-temperature designs supported our own experience.

On page 242 a criticism was made of a piston-valve type of nozzle-control valve described in reference [9] on the basis of clearance-flow losses, but he thought it should be pointed out that this criticism, if valid, would apply to a much greater extent to the turbine illustrated in Fig. 29, which had no fewer than six piston-ring-type joints whose function was to prevent leakage from nozzle boxes to wheelcase. This turbine was moreover, designed for three times the nozzle-box pressure of that shown in reference [9]. Incidentally the arrangement of Curtis wheel followed by end-tightened reaction blading shown in Fig. 29 would be considered undesirable in a marine turbine designed for those steam conditions. The arrangement illustrated in Fig. 28 was

*Since corrected.

much more suitable, and in fact was generally similar to modern British designs for merchant vessels.

Last year he had the privilege of seeing three turbine sets building at the works where two of the authors were employed, for steam conditions of 650 lb. per sq. in gauge and 1,020 deg. F., and in view of the repeated statements in the paper of the benefits to be obtained from the use of higher steam conditions the last paragraph of the paper seemed curiously restrained. After the powerful and, indeed, convincing advocacy of higher conditions one rather expected a more optimistic and confident conclusion!

Mr. L. Baker, D.S.C. (Member of Council, I.Mar.E.) said that in view of the short time now available for the discussion he would omit some of the things he had intended to say, and make a written contribution to the discussion.

He could not help saying that he thought that British turbine practice had been somewhat handicapped by the relatively great success achieved by Diesel propulsion. The majority of ships had been built in the power range in which the Diesel could compete very successfully with steam, and it was only since the war that powers had increased to a point at which steam could, if it would, become attractive.

It was hard in England to understand the American preference for header type boilers. Those boilers were inherently more liable to defect than the normal 2-drum type, and the ease with which tubes could be plugged was matched by the greater frequency with which they needed to be plugged! Moreover, there could be no disputing the greater cost of routine inspection and cleaning, nor could there be any question of the greater weight for a given output with corresponding loss of dead weight capacity—a loss, it should be noted, that existed for the life of the ship.

So far as the limiting gas temperature was concerned, the risk of dewpoint corrosion, of course, was dependent upon the fuel. From the limited information available, the fuels from American sources had higher sulphur contents than the majority from the Middle and Far East. The experience with gas temperatures from 240 to 260 deg. F. from the Middle East and Borneo fuels had been entirely satisfactory, as no corrosion in air heaters or economizers had been experienced. Attention, of course, was paid to gas bypasses and to hand cleaning at quarterly intervals. Combustion equipment also affected the risk of corrosion greatly.

In this country there was as yet little experience of the risk of thermal shock with advanced steam conditions. Apart from the need for service fluctuations, temperature variations must occur when lighting up and shutting down, and were usually involved in the manœuvring operations. In addition, salts or moisture might be carried with the steam from the drum into the superheater and thence in some cases onwards. The direct solution of solids in steam became appreciable at pressures in excess of 1,200 lb. per sq. in., but there was some evidence at lower pressures that any solids present in moist steam leaving the drum might be deposited differentially in superheater, steam line or turbines. They had experienced the deposition of "gooseberries" of pure sodium phosphate in the manœuvring valve of a Victory ship operating at 450 deg. per sq. in. 750 deg. F. With this in view, they had paid attention to the development of detection devices to safeguard the plant from the deleterious effects, and in their new steamers they were installing protection against either solid or water content of the steam leaving the drum. In this way they hoped to avoid all risks of corrosion conditions in addition to pure stress variations.

Finally, he was in entire agreement with the authors when they suggested that the correct procedure was to take one step beyond the economic optimum. Ships in Great Britain were built for at least twenty-five years' useful life, and frequently for longer. Unless the machinery were thoroughly up to date and preferably in advance of the standard product of its generation, it was difficult, if not impossible, to ensure a reasonable income from the ship in its later life when it was faced with competition from newer fleets. In general, he believed that in this country they set undue emphasis on reliability without setting against it the price they were paying—in other words, fear of the unknown had tended to make them unduly conservative. On the other hand, he would like to quote a remark that had been made to him in America. He had been prodding an American about the way in which the American had been following up what appeared to be a completely wasted effort, and the American said, "Well, even a turtle must stick its neck out sometimes." As engineers, they had to be careful that they did not stick their necks out too far too often.

Capt. R. A. Smyth, U.S.C.G., commenting on the reference on page 230 of the paper, to varying preferences by vessel operators for either economizers or air heaters, said that some improvement might be made in the design of both. The successful operation of marine boilers at higher temperatures with maximum efficiency would depend upon maintaining those heat exchangers in the plant system. It was to be hoped that experience with past failures in this respect would lead to improved design and arrangement for proper routine maintenance.

On page 234, it would appear that the position of the authors of the "Interim Guide" regarding the limitations of carbon steel to 775 deg. F. and carbon-molybdenum steel to 875 deg. F. was now vindicated by recent reports of a major industry in the United States indicating serious graphitization in service units operating above the ranges mentioned here.

It was not believed that immediate future designs would contemplate pressures in the 600 lb. range. It would appear that, as the operating results of recent installations were known more generally than at present, the trend would be towards 850 lb. per sq. in—850 deg. F. during the period in which chrome alloys were restricted for reasons given by the authors.

Mr. H. N. Pemberton (Member of Council, I.Mar.E.) noted that the authors pointed out that the major incentive for advances in steam conditions for commercial vessels was the prospect of saving in fuel consumption. Several ships had been built in the United Kingdom in recent years with turbines operating at around 600 lb. per sq. in. steam pressure and 800-850 deg. F.; with provision for steam reheating, fuel consumption for propulsion in the region of ½lb. fuel per s.h.p. per hour could be obtained.

These were not particularly complicated installations, and in general the steam piping had been plain carbon steel, a few pipes in ½ per cent moly-steel being used in certain cases. Those ships were regarded as being modern, and the owners had no reason to be dissatisfied with their performance. Moreover, they were proving to be reliable.

One wondered, therefore, when studying the paper and pondering on all the multitude of problems—materials, stresses, design, fabrication, etc.—to which the authors referred, how much further one might expect to go. The shipowner had little interest in pioneering advances in steam conditions, beyond those which had been proved to be thoroughly reliable and which permitted him to burn the cheapest form of fuel oil at a reasonable rate of consumption.

His own view was that there would be a period of consolidation at figures approximating to those which he had just mentioned. He had no doubt, however, that there would be further advances in marine plant, if only because, in the long run, marine practice was bound to follow land practice, and in any case it would become increasingly necessary to compete with the gas turbine for high powers in the years to come. In the circumstances, whether they liked it or not, they had to face the problems associated with the more advanced steam conditions of temperatures of over 1,000 deg. F. referred to in the paper.

In the short time at his disposal, he wished to make only one further comment. He thought that British marine engineers could admit that they used mild steel for steam pipes in a somewhat extravagant way. Permissible working stresses were lower than they need be. But this unnecessary weight of material was not of primary importance, and indeed had probably been partly responsible for the fact that steam pipe failures in ships were extremely rare. With the higher steam conditions now contemplated, they could not afford such extravagance in material, not only because of the higher cost of steel alloys required, but also because thermal conditions demanded that the pipe be as thin as possible, consistent with strength and safety.

He had been looking at the stress figures set out in Table III. This referred to the allowable stresses recommended in the "Interim Guide" issued by the American Bureau. He noted that these were based on values for rupture strength extrapolated from the results of 1,000-hour creep tests. The authors had themselves referred to the danger of extrapolated values which took no account of deterioration in material exposed to high temperature and stress over periods much longer than 1,000 hours.

With this in mind, he would like to know on what type of creep test these recommended stresses were based. Dr. R. W. Bailey, one of the foremost British authorities on creep testing, had on more than one occasion pointed out that 1,000-hour creep tests at the working temperature, using stress to accelerate creep strain brought into account only to an insignificant extent the influence of thermal action on the material, whereas his own method of testing at the working stress and using temperature as the creep accelerating factor brought this thermal action more adequately into account.

He did not dispute the stress figures given in Table III, but had no doubt they pointed the way in which one should go. Nevertheless, it was important to keep clearly in mind the type of creep test data from which these extrapolated values had been derived, and before accepting them unreservedly he would like to be satisfied that the 1,000-hour creep tests on which they were based conformed to the principles expounded by Dr. Bailey; if not, some confirmatory tests were desirable.

Mr. A. W. Davis, B.Sc. (Member, I.Mar.E.) said that in the realm of marine propulsion the gas turbine was acting very much as a catalyst in the development of the steam turbine; in fact the upper practical limit of steam temperature presently foreseen was defined by practical experience with the gas turbine. Many of the difficulties to be overcome were, of course, entirely different and the association of very high pressures with high temperatures tended to be peculiar to the steam turbine; the combination of limit conditions in pressure and temperature could, however, be avoided by the adoption of reheating and advantage be thereby taken of other desirable characteristics which became of greater significance with higher steam conditions. The authors had made no reference to the reheat cycle and it would be very interesting to have some remarks from them on the matter.

With regard to high pressure steam piping, marine work was somewhat handicapped in this country as compared with the United States by the heavier scantlings that were required, possibly to allow for a degree of wastage that would be associated with the effects of saturated rather than superheated steam, and this made it all the more difficult to provide for expansion in arranging pipe lines. In this connexion the authors had commented on the wisdom of allowing no more than a one-third cold pull up in settling the stresses. He would like to have the authors' views as to the adoption in practice of a greater cold pull up—say, two-thirds—when high temperatures were involved, having regard to the effect of creep in relieving strain in the high temperature condition.

CORRESPONDENCE

Mr. L. Baker wrote that, in addition to his remarks in the verbal discussion, he wished to raise a number of other points. He had already referred to the American preference for the header type boiler and indicated his preference for the single furnace D type boiler. For mercantile service there was no need for superheat control over a wide range of boiler outputs; if superheat control was necessary at all it was required in two ways, firstly to get full service output at either high or low temperature and secondly to get a fine control of temperature at full output and high temperature to ensure that the controlled limits could be, say +0 deg. F./−25 deg. F. These conditions could easily be met by a single furnace boiler with damper control with advantages in weight saved and simplicity of operation. In fact by using remote control for the dampers it was easy to put the control of steam temperature where it should be, i.e., under the personal control of the *turbine* operator.

Two design points were of interest. The experience with 240-250 deg. F. final gas temperature leaving the vertical tube air heater had been quite satisfactory. Care, of course, was taken to use soot blowers daily, to use the bypasses when at lower power and lighting up and to clean the gas side by hand at quarterly intervals. In this connexion British experience with American multi-nozzle soot blowers had not in general been as satisfactory as with the conventional single nozzle type, and several Victory ships had been re-equipped with British sootblowers.

With regard to main turbines, two main points arose. Firstly, more information would be welcome regarding the American practice of quoting the extra cost to be paid for specific improvements in performance. No such system was operated elsewhere so far as he was aware and it was one that had much to commend it, provided that testing facilities of the required degree of accuracy were available. This would be virtually impossible in a ship.

Secondly, the analyses carried out in this country by shipowners seemed to indicate a lower optimum pressure than that obtained from Fig. 1. Investigation suggested that this was due to reduced losses in the h.p. end of the main turbines of U.S. design. Fig. 28 did not indicate so marked a difference in practice and this would seem to warrant more detailed investigation.

Finally, some information on relative costs might be of value. A recent enquiry for a 8,000 s.h.p. steam turbine gave £20,000 extra capital cost for the change from 475 lb. per sq. in. 850 deg. F. to 600 lb. per sq. in. 950 deg. F. for a saving of at least £4,000 per annum. In other words, when designing for a minimum ship life of twenty-five years, the gains were adequate to cover any increased maintenance cost that prudence, in the absence of practical experience, suggested might arise.

Mr. Evers Burtner referred to Fig. 2, in which the authors showed curves of geared turbine engine efficiency versus shaft horse power, together with spot values for five recent installations constructed in the U.S.A. It should be noted that two of the five turbine sets whose engine efficiencies were plotted were designed and constructed by shipyards which had maintained technical and shop staffs capable of such work. Fig. 2 showed that the performance of the shipyard units compared very favourably with marine turbines constructed by prominent builders of stationary turbo generator sets.

Although some American marine engineers felt that shipyards should avoid propulsion turbine manufacture, from personal experience he knew it encouraged research and development by the shipyard technical staffs and the shops. This, furthermore, was useful in improving the morale of these two departments.

Early in the paper, when discussing pressure temperature combinations, the authors indicated that the header form of boiler was limited to a maximum of from 600 to 850 lb. per sq. in. pressure due to difficulty and cost of construction of the sectional headers for higher pressures. Perhaps a greater and very definite disadvantage of the sectional header boiler for higher pressures was the fact that its circulation was inherently less satisfactory than the two drum type. All water leaving, and also water and steam entering the drum, flowed through the down comer nipples at top of the front headers and through the circulation tubes rolled into the back of the drum. Increase in steam pressure and the corresponding decrease in the difference in density of steam and water made the provision of adequate circulation in natural circulation boilers more difficult. The basic design of the boiler for high pressures should not discourage circulation. However, one must realize that many cross drum sectional header boilers were used for relatively high pressures in stationary service.

Mr. H. J. Chase, in offering the following notes referring to turbines, was aware that consideration of the economic and performance characteristics of the turbines involved but one aspect of the whole problem. Differences noted in experience with turbines did not detract from the writer's respect for the excellent work of the authors.

It was suggested that Fig. 1 showed too little gain in heat rate in going to pressures higher than 600 lb. per sq. in. In particular, for turbines of 12,500 h.p. and larger, it was believed that the gain in going from 600 lb. per sq. in. to 1,200 lb. per sq. in. should be nearly double that shown by Fig. 1. The authors mentioned that the level of turbine efficiency had improved, and offered Fig. 2 as reference. Fig. 1, however, did not appear to take full advantage of the improvement available at higher pressure on relatively high ratings. Extending this thought to Fig. 3, the writer would expect that for units "16,000 h.p. and higher" a definite advantage would be realized at pressures up to 1,200 lb. per sq. in. and possibly also at temperatures up to 1,000 deg. F.

The authors noted that in the field of power station turbines, construction of a turbine for 1,100 deg. F. was underway. The writer's company was engaged in design and construction of a turbine for 1,100 deg. F. and 2,300 lb. per sq. in. Some details of construction of large units for power station use were described in a paper presented in London by Edwin E. Parker at the Fourth World Power Conference, July, 1950. In particular, illustrations of double shell construction and valve arrangements similar to the authors' Fig. 30 were included.

While central station turbine practice could not be compared directly with marine practice, fuel economy and many cost factors had much in common in the two fields. It was interesting to note that the writer's company had seen the following trends in very large turbines delivered to owners in the past five years:

Turbines designed for steam pressure above 850 lb. per sq. in—from 10 per cent of total to 95 per cent of total.

Turbines designed for steam temperature above 950 deg. F. —from 7 per cent of total to 52 per cent of total.

This indicated that the advantages of higher pressures and temperatures had been realized, and it pointed to engineering developments in all fields which had justified owners in these advances.

The authors noted that high inlet temperature presented a difficult problem in bypassing stages for higher power. It was pointed out that an increase in pressure might eliminate the need for bypassing by reducing the size of the inlet connexions and the first stage nozzles. Alternatively, an internal bypass, such as from the first stage to the third, could accommodate the higher power, and at the same time reduce the actual temperature on the first rows of rotating blades.

The authors pointed out that special attention must be given to astern operating conditions when high steam temperatures were employed. The writer agreed that this situation should be examined in detail, not only with respect to rotor clearances, but also with respect to the temperatures reached by the first ahead stage during astern running. It was the writer's experience, however, that units for large powers were no more difficult to handle than units of moderate rating.

The writer shared with the authors the expectation that the near future would bring about installations to operate at pressures above 600 lb. per sq. in. and at temperatures above 850 deg. F., but only where the rating of the units was relatively large.

Mr. I. D. Eby wrote that while it was the desire of the merchant ship operator to obtain better fuel performance, lighter machinery and boilers per horse power and savings in space, it was also his concern that these gains be not achieved at the expense of greatly increased maintenance costs or at a sacrifice of reliability. With the cost per day to operate a merchant ship at its present high level, an appreciable saving in fuel would be needed to compensate for even one day lost time while undergoing repair. It should furthermore be considered that with higher temperatures and pressure, came the requirement of special materials frequently unobtainable which further increased the maintenance problem. With regard to operation, higher steam conditions demanded more skill on the part of the engineers, which factor must also be balanced against economies achieved.

In this paper the authors had gone to considerable trouble to ascertain the results obtained by operators of land installations and by marine operators in other countries. They had also realized that with higher steam conditions came added operating problems but that with care in choice of materials and in layout of plant satisfactory results would obtain. It was believed that the greatest problem in adapting higher conditions to merchant ships would not be in selecting proper boilers, turbines and major auxiliaries but in tying these units together in such manner that the normal merchant crew would be able effectively to manœuvre the vessels and to carry out normal voyage maintenance. It was noted in the paper that investigation had been made of automatic controls, combustion controls, safety devices, piping layouts and simplicity of equipment. The effectiveness of this work would largely determine whether the plant would be a success or failure so far as normal operation was concerned.

It was believed that the second consideration should be whether major overhaul of the vessel could be performed in the normal shipyard. The use of special materials and alloys would require in many cases improved techniques and facilities and the merchant operator would not wish to be limited to an extremely small number of shipyards for their overhauls or to have the time which elapsed during overhaul unduly extended.

The following specific comments referred to the particular parts of the paper indicated:

Page 229—The sectional type boiler not only lent itself to easier cleaning and reduced time for repairs, but was also conducive to keeping soot tight. If practically the same economy were obtainable and if this type could now be used to develop higher pressures, it would be preferable for merchant ship operation.

Page 230—High boiler efficiency should not be obtained at the sacrifice of other parts of the ship. An uptake temperature of less than 300 deg. F. would almost certainly result in a dirty ship unless special means were taken to eliminate soot either by blowers or soot arresters. It was possible that the added cost of these items would offset the increased fuel economy.

Pages 230-231—Operation had shown that with the non-condensing type feed pump no less overall economy resulted if its exhaust were used to operate the distilling plant and far fewer maintenance troubles occurred in its use.

Page 232—Generators attached to the main reduction gear unit would not be desired; however, if these were used, reliability could be obtained through the use of a standby Diesel generator with an automatic air starting valve, functioning when voltage dropped below acceptable value.

Page 234—The increased cost of chromium alloys would not be a deterrent to their use in the steam cycle but the difficulty of obtaining replacements during overhauls should be considered.

Page 238—Austenitic piping, if subject to failure from thermal shock, should be very carefully investigated before it was adapted to marine use since fluctuations of temperature were often severe during manœuvring.

Page 246—Superheat control should be made as simple as possible and adequate means provided which would enable the average merchant crew safely to keep superheat within proper level. Separately fired superheaters would not be desired. It was his company's opinion that the automatic live steam pressure made up to one or more of the bled steam feedwater heaters would be a preferable method of control.

Page 247—Combustion control systems had proved reliable and economical and should be included on all modern high pressure vessels; their installation would be especially desirable in cases where excess air was of importance in maintaining proper superheat control.

Page 249—Any method which would reduce slagging of superheater surfaces or the deposit of soot on air heater and economizer tubes would be highly desirable but no conclusive evidence has been presented which would indicate that steam atomization is the answer.

Page 240—The vertical header type superheater shown in Fig. 17 would greatly lessen the difficulty of cleaning superheater tubes and detecting leaks. When designing a superheater, however, more attention should be given to ease of replacement of superheater tubes and to the proper location and required number of soot blowers.

There was strong objection to the elimination of flanges on main steam line valves for not only was it difficult to repair these in place but it was often necessary to replace valves entirely, a process which would be inconvenient.

It would be desirable to weld up the superheater steam line entirely except for valve and turbine casing flanges but extreme care should be used in the piping layout since it was frequently necessary to remove the superheater steam line in order to open turbine casings and move large machinery parts.

Mr. C. W. Hasek referred particularly to superheater design and slagging.

The amount of superheater surface to be installed (see page 237) was a compromise of several items which must be considered to obtain an economical and reliable design. The steam velocities should be high enough to maintain reasonable tube temperatures but not so high that the resulting pressure drops gave excessive design pressures. The heat input to the superheater surface was dependent upon the diameter and pitch of superheater tubes, flow relationship between gas and steam, screen tube pitch which governed radiant heat leakage from the furnace, burner location, etc. Header and nozzle sizes, location of handhole fittings, type of seal welding, etc., must be considered from the shops' viewpoint.

The problem was then one of correlating the effects of these items so that a practical design was obtained which would meet the ship's steam temperature requirements.

The superheater designs for various amounts of superheat would be affected in different degrees by the above items and it was, therefore, difficult to establish simplified data which would apply over a fair range of steam total temperatures.

Appendix III and Figs. 13 and 14 were, he believed, generally correct for the assumptions considered in the analysis. However, certain assumed values would change with changes in steam temperature. For instance, low chrome moly materials could not be used for the higher metal temperatures of Fig. 14. The tube temperatures would change quite perceptibly for resulting changes in conductivity, tube thickness and steam side heat transfer rate.

As noted under item (2) on page 256 (Appendix III), radiant heat leakage from the furnace was not included in the analysis. This leakage could be a considerable amount, as high as 25 per cent of the maximum total absorption per sq. ft. and resulted in a considerable increase in tube temperature. Thus it was usually preferable to have a parallel flow relationship between gas and steam in the high temperature passes of the superheater.

On page 241, reference was made to the reduction in superheater slagging obtained on coal fired boiler units by introducing steam into the air supply below the fuel bed. It was further suggested that the "water gas reaction" theory applied to this reduction in slagging might also apply to steam atomization of oil.

The writer understood that the reduction in superheater slagging had been noticed on coal fired underfeed stoker and travelling grate boiler units, and that the steam was injected in the back half of the grates. The following was a possible theory regarding this operation:

(*a*) As the coal first entered the fuel bed the heat drove off the moisture and volatiles. This resulted in the low grate temperature which had been noted in several installations.

(*b*) As the coal travelled toward the rear of the grates the coal was reduced to coke, carbon plus ash. This burned at high temperature, indicated in part by the sudden increase in grate temperature at this location.

(*c*) The constituents of the ash sublimate at the high temperatures, condensed in the cooler zones of the furnace and screen, and formed slag (on the superheater surfaces).

(*d*) The addition of water or steam below the grates reduced the fuel bed temperature by 200-300 deg. F. Water gas was formed (H_2O+CH_2+CO) and the constituents combined before entering the boiler bank.

(*e*) The reduction in fuel bed temperature reduced sublimation of the ash such that less slagging material was carried into the furnace and slagging of boiler surfaces was reduced.

(*f*) Briefly, the injection of steam below the fuel bed kept the ash in the fuel bed and, therefore, it could not deposit as slag on the boiler surfaces.

If this theory were correct, it did not appear to be the correct analysis for the apparent reduction in superheater slagging obtained with his company's Y-jet steam atomizer on oil fired marine boilers, because the oil ash was not kept out of the furnace. On stationary practice, the amount of moisture in the coal did not have an appreciable effect on the slagging of pulverized coal boiler units which, on the basis of the above theory, was because the ash was not kept out of the furnace.

Incidentally, the British had been obtaining a similar reduction in slagging by recirculating flue gas through stoker grates.

Mr. G. W. Kessler was heartily in agreement with the general views of the authors and their interpretations of the advantages and disadvantages of past and present practices. Therefore, his remarks were more in the nature of a re-emphasis and expansion of a few of the various subjects discussed.

First, as cycle efficiencies were increased by the use of higher steam pressures, higher steam temperatures, and more regenerative feed heating, it became increasingly necessary to use air heaters if boiler efficiency were to be maintained at, or to exceed, the present high levels. However, this did not mean economizers would be eliminated; instead we might find that in many instances conditions would dictate the use of both economizers and air heaters.

Because of their low fluid temperatures, economizers were certainly far better heat absorbers than the last sections of high pressure boilers and, further, their use would prevent the necessity of operating air heaters with exceedingly high exit air temperatures. In addition, a better distribution of the resistances to gas and air flows could probably be obtained, with a good possibility of reduced overall air pressure requirements when using both economizers and air heaters.

The authors mentioned air heater operation with 300 deg. F. uptake gas temperatures. This usually was quite satisfactory, but if fuel oils with higher sulphur contents were used in the future, it would be necessary to consider higher exit gas temperatures (and consequently lower efficiencies), air bypassing, or air recirculation at normal ratings. Small steam air heaters also might be used to preheat the air a few degrees before it entered the main air heater. This would entail a small loss in efficiency. It must be noted that in boilers designed for operation at 300 deg. F. exit gas temperatures at normal rating, the exit gas temperatures would be well below this figure during port and low load services.

The authors discussed in considerable detail the various formulæ used to establish hoop stresses. The American Society of Mechanical Engineers had before it for review and adoption a new formula for the determination of hoop stress.

The proposed formula,

$$t = \frac{D}{2}\frac{(P/s)}{(P/s+1)} + C$$

where t = thickness, inch
D = outside diameter, inch
P = pressure, lb. per sq. in.
s = stress, lb. per sq. in.
C = corrosion factor = $0{\cdot}0156D$

was presented for review and comment in the May 1951 issue of "Mechanical Engineering." It was expected that the new formula would be adopted by July 1951.

The American Bureau of Shipping's Interim Guide, of December 1949, recommended the use of the formula —

$$t = \frac{PD}{2s + XP}$$

where X is an adjustment factor—equal to 2·0 for temperatures of 95 deg. F. and higher
for establishing hoop stresses in piping.

It was noted that, if the corrosion factor in the proposed A.S.M.E. formula were eliminated, and when X equalled 2·0 in the Interim Guide formula, both formulæ might be expressed in the form —

$$t = \frac{Pd}{2s}$$

where d = inside diameter, inch.

Thus, in the near future there should be a better correlation between formulæ used by different agencies.

Superheater design, performance, and maintenance were all important in boiler design and the authors covered this difficult subject in an excellent manner. Too few of us recognized the necessity for careful study of the disposition of superheater heating surface, and the fact that high grade alloys could often be eliminated, or their requirements reduced, by variations in the resistance to steam flow and inlet gas temperatures.

The boiler designer was faced with a difficult problem as steam temperatures increased. The trend toward steam temperatures of 950-1,050 deg. F. required superheaters which must absorb a very large portion of the total heat absorbed in the entire unit. Thus, the natural tendency would be to move the superheater closer toward the furnace. This resulted, however, in higher inlet gas temperatures, higher tube metal temperatures, and much greater possibilities of slagging. Increasing the pitch of the tubes would help to minimize slagging but would require still more surface.

If, in the future, the size of the superheater and the possibility of slagging were cause for concern, the use of gas tempering might very well be the best solution. With such a method of operation a certain amount of the products of combustion could be taken from the zone behind the boiler or the zone between the economizer and the air heater and carried back to the furnace where it would be intimately mixed with the furnace gases just prior to their entrance into the boiler water screen. This procedure would result in a twofold advantage—first, the gas temperature entering the superheater would be lowered appreciably, thus minimizing the possibility of slagging, and secondly, the increased gas flow would increase convection heat transfer rate and consequently superheater size might be reduced.

This method of operation also led to another advantage, that of superheat control. This control could be obtained by reducing the amount of gas used for tempering as the rating was decreased. This allowed the furnace exit gas temperature to increase (but not over the tempered gas temperature established for normal rating) and consequently higher steam temperatures could be maintained at the lower boiler ratings.

The authors discussed the use of ferritic and austenitic alloys, safe ending, and tube to header welding in superheaters. In this connexion an important item should be noted. His company had had occasion to fabricate all austenitic superheaters, that is, austenitic tubing and headers. During service they had noted a rather strange and peculiar trait of the austenitic materials. The welded joints gave excellent service but the expanded joints, used in the inlet and low temperature intermediate superheater headers, leaked after operation. These joints were rerolled and tested tight under hydrostatic pressure; however, they leaked after subsequent service and again were rerolled. This sequence of rerolling, leakage and rerolling was repeated several times and always it was noted that in every case the same size expander had to be used. In other words, the material, after service, always came back to its original size regardless of the amount of expanding. As a result of this experience, it was felt that joints between austenitic materials should be welded even though they operated at low temperatures.

There was still considerable objection to the use of steam atomizing oil burners, although they performed in excellent fashion and facilitated wide range automatic operation. This objection, of course, stemmed from the loss of fresh water which might amount to one-half to one per cent of the total steam output. In view of this situation, considerable attention had been given to the development of wide range return flow mechanical pressure oil atomizers and the results had been most promising.

Tests of variable supply, variable return, and variable differential pressure return flow burners had shown that very good conditions could be obtained over a wide range of operation with a minimum amount of return oil. Further, sprayer plate wear and maintenance had been decreased appreciably.

Mr. J. V. C. Malcolmson commented on the selection of boiler components for higher steam conditions; it was noted that for steam pressures above 900 lb. per sq. in. the use of steam air heaters was unfavourable because the bled steam requirements of these cycles were large and it was questionable whether or not there would be adequate steam for air heating purposes. In order to realize the best boiler efficiency, the use of economizer and air heater was indicated; however the corrosion of gas air heater tubes as a result of low steaming rates and low stack temperatures made necessary the bypassing of the heater, with a resultant loss in efficiency. In order to realize the use of a gas air heater with lower gas temperatures, he asked whether any further consideration had been given to the use of the regenerative type of air heater which utilized rotating corrugated metal plates which were alternately heated in the exhaust gas and cooled in the combustion air.

The problem of slag formations on superheater tubes had given origin to the use of steam atomizing fuel oil burners in vessels of the Texas Company. In addition, in the interest of proper atomization, there had been installed aboard one of the vessels using steam atomization, a viscosity control instrument which automatically maintained a desired value of viscosity for the fuel to the burners, regardless of the viscosity of the bunkers which were received. A sample of the fuel leaving the fuel oil heater was supplied to the control instrument, which in turn measured and controlled the viscosity through the use of an air actuated control valve governing the admission of steam to the heaters. Instruments of this type had been used successfully in industrial process applications and although their use in the marine field was in an embryonic state, it was felt that the marine application of this type of control might have merit and be worthy of consideration in the future.

Mr. W. Hamilton Martin (Member, I.N.A.) wrote that in reference to the authors' remarks on materials for steam and superheating tubes and turbine blades which would have to withstand higher steam conditions at temperatures from 1,050 to 1,200 deg. F., it would seem that no high temperature alloy steel was yet available effectively and continuously to resist such destructive agents as vanadium pentoxide, sodium vanadate or other compounds released in steam which would eventually destroy parts made from the alloys in present use. Austenitic steels were being used and nimonic steels were coming into wider use as the nearest solution for high steam conditions. Nimonic steels were being used in jet aero-engines and for gas turbine parts and accessories which were exposed to intensive chemical corrosion at high mechanical stresses and excessive temperatures by impingement of hot combustion gases carrying destructive incandescent particles; periodical cleaning, resurfacing and contouring was therefore required, thereby avoiding costly renewals and saving time taken in overhauls.

A process of electro-polishing had recently been developed and perfected in this country by a British electro-chemist, by which such parts were successfully cleaned, resurfaced, polished and rebuilt to original overall dimensions. In its present stage of development, parts made from materials such as stainless steels, stainless iron, high duty nickel-chrome alloys and the like could be descaled, foreign deposits removed from entire surfaces, corrosion and pitting removed and a new surface built, polished and brightened, the original sectional dimensions restored and put into service again in little time, avoiding costly renewal.

It would appear that for these higher steam conditions, in which nimonic steels and the like were used, this electro-polishing process ought to find similar useful applications for steam tubes, superheater tubes (final passes especially), boiler wall tubes, and possibly economizer or air heater tubes. For turbines, blading and nozzle plates, and throttle valves, subjected to these higher steam conditions, such parts might be likewise cleaned, brightened, polished, resurfaced, contours restored and put into service again without their having to be renewed. A tube surface could be cleaned and polished, both externally and internally, to an appreciable length and minimum diameter. The maintenance of marine and land reliability and freedom from untimely breakdown, should be turbo installation at these higher steam conditions, and their reliability and freedom from untimely breakdown, should be improved by the use of this process. In time tube surfaces, nozzle plates, blades, etc., and many parts exposed to excessive mechanical, chemical and high temperature effects in steam practice, cracking and distillation, and chemical plant, heat exchanges, atomic, etc., equipment may have their surfaces cleaned, polished and brightened before being built in to extend their first service period and reduce chances of premature shutdowns.

By this process electro-machining and cutting could also be done, which should be especially useful in cases where parts or surfacing are inaccessible to ordinary machining methods.

Mr. A. W. Rankin wished to discuss several particulars of high-temperature design in which his opinions were not in complete accord with those of the authors, but these would not lessen to any considerable degree the writer's compliments to the authors on a paper of considerable practical value.

The authors mentioned that a large amount of creep data "has been obtained from tests lasting only a few thousand hours." This was obviously true, but cognizance should be taken of the large number of American creep tests which had been carried out for many thousands of hours. Specifically, the writer's company had run one set of four bars in constant-load creep tests out to a full 100,000 hr. (31); another set of two bars was run out for 50,000 hr. Sixty-seven tests of more than 20,000-hr. duration had been completed, while at the present time there were forty-eight bars in constant-load creep tests aimed at 25,000-hr. Of the latter series, a number of bars had already been in test for more than 20,000 hr. In addition, other American firms had published data showing tests of well over 15,000 hr. duration. In the writer's organization, the short-time creep test of a few thousand hours duration was used primarily as a sorting tool, but when a steel was to be used in production design it was subjected to a creep test of much longer duration.

In the writer's opinion, the phenomenon of creep strength was somewhat secondary to that of rupture strength, and in many parts of turbine design creep strength was essentially of minor importance, and the major emphasis was placed on rupture strength. The most obvious point at which this was true was in the design of bucket dovetail hooks. The overall length of these was so small that creep could not seriously affect turbine clearances, while the desirability of alleviating stress concentrations made it worth while to welcome a certain amount of plasticity in order to provide against the likelihood of a rupture failure. It should also be emphasized that this concern was not an academic one, since sufficient rupture failures of various turbine and boiler parts had shown that long-time rupture was a factor which must be given due respect. It should be remembered that this type of failure might occur after a prolonged period of apparently successful operation, during which period the designer might have repeated that particular design in a number of other installations. In particular, it was self-evident that no turbine installation could be analysed so accurately that the complete stresses were known; there would be residual stresses of various types and stress concentrations at various locations, and with some of the stiffer alloys which must be used for the higher operating temperatures these unexpected stresses could easily encroach on the long-time rupture strength and jeopardize the long-time reliable operation.

With reference to the knees which occurred in the curves of Fig. 10, the authors stated that these served as a warning that the particular stress-temperature conditions under which they occurred were probably unsuitable for long term operation. Such knees in the log-log plot occurred in the majority of steels which were employed for long-time high-temperature operation, and did not render the steels unsuitable for service. The tests which were used to set the allowable design stresses must be carried out far enough to get appreciably past this knee so that the extrapolated value had adequate validity, but the major effect of the change in the curve in making the design problem more difficult was that it resulted in a lower value of allowable stress.

With respect to the effect of alloys and heat treatment on high-temperature strength, the writer was not in agreement that moderate additions of chromium resulted in any marked increase in high-temperature long-time strength, at least of the normalized and tempered steels of the more usual alloy combinations. Tests made, for instance, on 1 per cent mo. steel and 1 per cent cr.-1 per cent mo. steel had shown no marked difference in the high-temperature strength of the chromium-containing steel. The addition of the chromium would increase the graphitization resistance (particularly of an aluminium-killed steel), increase the oxidation resistance, and improve the heat-treating characteristics, but would increase the high-temperature strength only moderately. Of course, in other alloys, different results might be obtained; for instance, a 1 per cent cr.-1 per cent mo.-½ per cent V. alloy had better high-temperature properties than a similar chromium-free composition.

Cognizance must also be taken of the relative strength of a stress-relieved weld in a normalized and tempered structure. By normalizing and tempering, the moderate alloys could be given their best high-temperature strength, but where the weld, of a composition corresponding to the base material, could be given only a post-weld stress relief and temper, the weld would often be considerably weaker than the base material. This should be included in the design calculations so that the weldment could be given depth enough for adequate strength. In this respect the tests of the writer's company indicated that normalizing a weld, of the moderate alloy combinations, would raise the high-temperature strength rather than lower it as stated by the authors.

The authors had done a commendable job in presenting a further analysis of the effect of the different theories in developing a satisfactory formula to study the stress distribution in a pipe wall during creep. As stated in a previous publication [39], the writer proposed a formula which gave the diametral creep on the outer surface, and could see no particular reason for concern over creep at the inner surface except in so far as the (larger) inner-surface creep might result in rupture failure. The writer's average-shear expression given by the authors' formula (21) was developed while studying the possibility of rupture, and was used as described in reference [39] only to show that if rupture were controlled by shear stress, the common formula (or the writer's variant thereof) was sufficiently conservative. Until actual long-time rupture data on tubular specimens were available, it would be difficult to evaluate the degree of safety of the various formulæ, and reliance must be placed on engineering judgment, as was so often necessary in the design of complex high-temperature equipment.

The writer had given the above comments so as to present some of the results of the long experience of his company in building both land and marine turbines. Economic factors were driving both land and marine units to higher pressures and temperatures—some marine units were already operating at the 1,000 deg. F. level, while the writer's company was now engaged in the construction of land turbines for 1,100 deg. F. operation—and these steps to the higher operating conditions required the painstaking care of the engineering, manufacturing, installation, and operating organizations.

Mr. E. Kemper Sullivan considered that the prophecy of the authors that the steam conditions of the future would be considerably higher than those of today was sound. We had not yet reached the limitations imposed by materials and manufacturing techniques available today, and when that point was reached it seemed fairly certain that these limitations would have been removed to higher levels. As for going one step further than was economically justified, it seemed there was more advantage to all parties if the improvements be made in increments, as has been the general policy in the past.

In advanced steam conditions there was more to consider than the striking of a balance between fuel savings and increased first cost. Such factors as operating cost, maintenance and repair cost and reliability were also important; and today the question of scarcity of materials must be given careful consideration. The use of critical materials, such as chromium, nickel and fuel oil must be evaluated from the point of view of availability as well as cost.

As an example of the cost of a steam power plant as a function of steam conditions, economic studies made for the "Mariner Class" fast cargo ship now building for the Maritime Administration resulted in the following comparative costs:—

Steam Conditions at throttle	Annual increased fuel cost	Annual decreased fixed cost	Net increased annual cost
850 lb. per sq. in.—890 deg. F.	$0	$0	$0
600 lb. per sq. in.—840 deg. F.	$9,500	$7,960	$1,540
585 lb. per sq. in.—890 deg. F.	$6,400	$6,400	$0

Although the gain obtainable by using the higher temperature may seem small, it was real, and would be greatly amplified in the near future when the cost of development could be spread out, as pointed out by the authors. For the above mentioned "Mariner Class," however, the gain from higher temperature was not believed to be warranted when all factors were considered.

Of course, going to higher steam conditions without making use of all the advantages of those conditions in the selection of the steam cycle and machinery components seemed wasteful. All practical means of fully utilizing those benefits should be provided when economically justified. As an example of this thinking, a prototype cargo ship now building for the Maritime Administration employed 850 lb. per sq. in. gauge—890 deg. F. steam. The cycle for this ship included four stages of feedwater heating, high efficiency main and auxiliary turbines, and boilers fitted with both economizers and gas air heaters. Throughout the cycle attempts were made to utilize the energy available.

With regard to air heaters, it was believed that for most cases the use of steam air heaters should be limited to steam pressures of about 600 lb. per sq. in. gauge or less. Regenerative type gas air heaters offered promise of providing a means safely to obtain boiler efficiencies in excess of 88 per cent. There were three marine installations now being made in America, and it would be interesting to follow the performance record.

In discussing steam pipe stresses, the authors referred to one hundred per cent cold pull-up, with one-third credit allowed in the pipe stress calculations. The writer believed that a more equitable procedure for all conditions of service was to apply fifty per cent cold pull-up and to give no credit allowance in the stress calculations.

The need for superheat controls at elevated temperatures was pretty much agreed. The authors described a method of using an auxiliary desuperheater. As an alternative means, injection of feedwater into the coils of the superheater header offered promise. This method had the added advantage of protecting the superheater tubes during temperature surges.

AUTHORS' REPLY

Mr. Ireland, replying, said that Admiral Maxwell had brought out some very interesting distinctions between the design aspect for Naval machinery and that for commercial machinery. Practically all the points that Admiral Maxwell had raised were being given serious consideration in U.S. naval work also. He was sorry that he could not give a specific answer to the question about the influence of superheated exhaust on condenser performance, because there were two sets of results available at the present time, neither of them complete and both contradictory. His own results indicated that so long as the conditions were such that the tube could not actually become dry, if condensation did take place on the tube, the rate of heat transfer would not suffer. However, if it was possible for the tube to become completely dry so that it was acting as a de-superheater, then the heat transfer rate would suffer.

The question of availability of materials in wartime mentioned by Admiral Maxwell was quite important and was receiving attention currently in the U.S.A. Iron was the basic material needed of course, while nickel, molybdenum and chromium were the alloying elements required for high temperature designs.

One other point of interest in connexion with Admiral Maxwell's remarks was that, according to the latest information, it was possible to obtain a maximum steam temperature at cruising speeds with a gradual falling off at higher ratings without using temperature control for this purpose. This did not alter the views stated in the body of the paper regarding the desirability of control to prevent large swings in steam temperature.

He had particularly appreciated Mr. Gatewood's remarks, and he would like to add that when the story about condensation producing thermal shock had become known, there had been a very immediate and very thorough check of all such connexions on work that was then in hand.

Dr. Brown had asked for some information on gearing. He agreed that some remarks should be made on that subject, but for the moment he would only say briefly that the recent advances made in the United States in regard to gear cutting had been standardized for commercial work to the point where all leading manufacturers were now willing to offer designs based on K factors of 100 for the high speed reduction and 75 for the low speed reduction. The use of hardened pinions and shaving or lapping processes was inherent, of course, in any such advances.

It was interesting to note that, according to Dr. Brown, steam conditions in both the United Kingdom and the United States were now at the same level. He would like to explain that the temperature limitation to 875 deg. F. in the United States, which Capt. Smyth had referred to, was due to the chromium shortage and the need of it for more vital equipment than commercial processes. Once that need had been surmounted he hoped that they would be able to proceed. He felt that the marine industry had the "know-how" for 950 deg. F. as a routine commercial design, and that they could go further for special cases, when needed.

He was glad that Dr. Brown had called attention to one remark in the paper which, he agreed, was somewhat provocative. In the Sewaren unit, shown in Fig. 29 of the paper, there were piston ring connexions at the inlet. He would say, however, that there appeared to be this difference, that the ones in the unit shown in the paper moved wholly due to thermal expansion, whereas he had the impression that the valve in the British units referred to must travel through quite a distance in order to expose successive nozzles and also must be operated by a manual or other mechanism.

The question raised by Dr. Brown with reference to Fig. 3 disclosed that an error had crept into the labelling, which it was hoped could be corrected in the published text. The figures representing steam pressure on the 850 deg. F. line should be interposed as between the 2,000 to 6,000 s.p.h. range and the 6,000 to 10,000 s.h.p. range.

The authors certainly agreed with Dr. Brown concerning the desirability of having more information on the elongation at fracture of stress rupture specimens. One of the authors had strongly advocated this for many years, and it was now a growing practice in the U.S.A. to record such data and include it on creep-rupture design charts.

As mentioned by Dr. Brown, it was imperative to stress relieve turbine castings, particularly after repair welding, and this was normal practice in the U.S.A. for all types of castings used in high-pressure high-temperature steam plants.

The other points mentioned by Dr. Brown relative to steam cycle arrangements, selection of high temperature materials and the problem of vanadium attack were appreciated, as they were very timely and added to the value of the paper.

He had been interested to find that Mr. Baker was a proponent of the 2-drum type boiler. Speaking as a shipbuilder, he maintained a completely open mind on that subject, and would work entirely to the owner's preferences in either case.

In reply to Mr. Baker's further remarks, the authors were very much interested in his statements regarding satisfactory experience with air heaters and economizers at 250 deg. F. exhaust gas temperature. According to the latest information from U.S. sources, corrosion of these elements and of breechings and smoke pipes had now become a serious problem even with 320 deg. F. exhaust temperature.

With regard to the Victory ship soot blowers, it had now become quite common practice to use 250 lb. air pressure instead of 125 lb. pressure on these vessels. Also, there had been some design improvements since these early installations of the air puff type soot blowers.

Mr. Baker's question with regard to pricing of improvements in steam rate performance of propulsion turbines should, of course, be referred to the manufacturers.

With reference to the possibility that losses in the h.p. end of the main turbines could be reduced, Mr. Baker was correct in assuming that this was a matter of detailed investigation, some phases of which were briefly mentioned in reference (11).

As to costs, the authors found that Mr. Baker's figures were in close agreement with those given in reference (4) of the paper when these were corrected to current price levels, provided the vessel was assumed to operate at a high load factor. This was perhaps surprising since the gross figures obtained by Mr. Baker would be expected to be considerably lower than those quoted currently in the United States.

Mr. Baker's remarks on the detection of carryover products in the steam line were particularly interesting, and it might be mentioned that a similar device based on electrical conductivity measurement was understood to be used on the vessels described by reference (13).

Capt. Smyth's remarks concerning economizers and air heaters lent emphasis to what had been said previously about the current seriousness of the corrosion problem.

Mr. Pemberton had raised the question of the extrapolation of creep and rupture tests. There was a brief mention of this in the paper, coupled with a few references to long-time tests. Generally, the matter could be summed up by saying that for materials which had metallurgical stability within that temperature and stress range where they maintained stability, it appeared to be perfectly safe to extrapolate these results. This had been confirmed by tests lasting 10,000, 15,000 and 20,000 hours on a number of materials and in a very few tests lasting as long as 100,000 hours. There was a reference to such tests in the paper. This information had been amplified by Mr. Rankin in his contribution to the discussion.

The fuel rate mentioned by Mr. Pemberton of ½lb. per s.h.p. hr. had already been attained without reheating in the vessels described by reference(11) with steam conditions of 850 lb., 850 deg. F.

Mr. W. Hamilton Martin's suggestions relative to repair of wasted surfaces by electro-plating were of great interest. The extent to which such a process was economical would appear to depend upon the ease with which the affected parts could be removed from service and also upon their actual size. It was a little difficult to visualize electro-plating equipment capable of economically resurfacing heat exchanger tubes twenty or more feet in length.

Mr. Davis had referred to re-heating. Re-heating had first been introduced in the United States when a temperature level had been established, and he thought that the position should be the same and that as long as increases in temperature were possible, it was well to adhere to that line; when a level was reached which appeared to be necessary for some years to come, it was time to reconsider the use of re-heating. Mr. Davis had referred to the use of cold pull. The authors' recommendation was 100 per cent physical cold pull. The limitation of one-third was the amount of credit to be taken for such cold pull in making the calculations.

As already mentioned, Mr. Rankin had added some valuable information concerning his company's long-time creep and rupture testing programme as well as some very interesting statements on the application of such information to the design of turbine parts. The authors certainly agreed that long-term creep and rupture strength data were both important and that one or the other might dominate a particular application.

Mr. Rankin had taken the authors to task for indicating that it was undesirable to use high temperature materials for conditions beyond the knee in the stress-rupture curve. The authors agreed that the original statement was perhaps capable of misinterpretation but still felt that the occurrence of such a knee in combination with a marked reduction in ductility at fracture, as illustrated in Fig. 12(a), should be taken as a warning to exercise caution in the use of a particular material under these conditions. Furthermore, they felt that it was wise to be assured of about one per cent creep elongation before the start of third-stage creep which would ultimately lead to failure.

The authors did not believe that Mr. Rankin intended any disagreement with the data shown in Table III, which generally indicated substantial improvements in creep and rupture strength for the alloys with higher chromium and molybdenum contents. The authors would agree that for higher temperatures and longer test durations, the improvement in surface oxidation resistance was probably the controlling factor in producing such results.

Mr. Rankin's point concerning the relative strength of the weld following various postweld treatments was an important one. It was noted in the paper that a normalized condition was preferred for welds in carbon-molybdenum pipe but this was not found to be satisfactory for weld metal containing about two per cent of chromium. Mr. Rankin's experience, presumably, related to alloys of lower chromium content where he found an improvement in high temperature strength of welds due to normalizing. Differences in electrode carbon content and ductility and the use of synthetic electrode construction might well account for the different results noted by Mr. Rankin. In general, the finer grain structure resulting from normalizing would be expected to lower the high temperature strength.

With regard to the problem of hoop stress formulations for the high temperature range, the authors were pleased to note that the lack of tubular stress rupture data mentioned by Mr. Rankin would be filled by a publication to be presented at the annual meeting of the American Society of Mechanical Engineers in December 1951.

Mr. Chase had made some valuable comments with regard to current trends in U.S. central station practice and on the bypassing and astern heating problems. His statements on the improvements to be expected from the use of higher pressures were understood to represent a change in viewpoint which had taken place since the publication of reference (4). It would be recalled that the material in this reference was used to prepare Fig. I of the present paper, and it might be mentioned that when reference (4) was presented, Mr. Alan Keller, an associate of Mr. Chase, commented to the effect that he was in agreement with the turbine performance information contained therein.

Upon investigation, the authors had learned that this change in viewpoint was based on the actual performance of recently installed high pressure, high temperature central station units which exceeded their anticipated performance. Until there was more general agreement on this point, the authors would prefer to let Fig. 1 stand as representative of results that could be readily obtained.

Some of Mr. Burtner's comments relative to turbine manufacturers in the United States touched upon a matter wherein the authors were naturally at variance among themselves. It need only be said that such questions continued to be handled on an amicable basis.

With regard to Mr. Burtner's remarks concerning the inherent circulation characteristics of header and two-drum type boilers, the authors believed that the boiler manufacturers took the attitude that entirely satisfactory circulation characteristics could be obtained in either type of boiler.

Mr. Eby had presented some very important considerations from the operator's viewpoint, and all of his suggestions needed to be carefully considered in any new installations. The difficulty of obtaining special replacement materials in outlying ports must be overcome in the main by designing high temperature parts for even greater reliability than was now obtained in the same parts working at lower temperatures. Much progress was being made along this line, such as improvements in design of superheater tube supports, valve internals and the like. Superheater tube renewals and similar replacements should not exceed what was now considered to be normal, and these could be deferred until arrival at the home port where the normal provision of spares should prevent undue delay.

Mr. Malcolmson had raised an interesting point relative to the use of regenerative rotary type air heaters. It was understood that experience with these in recent British vessels had generally been satisfactory. Mr. Sullivan had mentioned that there were three vessels under construction in the United States employing this type of air heater. The U.S. marine industry would await with interest the results of the initial installation.

The use of steam atomizing burners and a fuel oil viscosity monitor mentioned by Mr. Malcolmson were both of considerable interest, and the authors looked forward to hearing more about these developments as time went on.

The cost figures presented by Mr. Sullivan would appear to indicate a ten-year pay out time for the increased cost of 585 lb.—890 deg. F. steam conditions as compared to 600 lb.—840 deg. F., which was not greatly different from similar information appearing in reference (4) if a high load factor was assumed. As indicated in the paper, such determinations must be made for the particular ship and route under consideration before a decision could be made. Just how large a step might seem wise was also an individual matter involving, among other items, assurance to all parties concerned that the necessary technical "know-how" and background information was at hand.

Mr. Sullivan's comments with reference to the importance of maintenance and repair items, the need for maintaining efficiency throughout the plant cycle and other matters were well taken. The authors would only add a word of caution with regard to the injection of feedwater into the superheater intermediate header for temperature control. From what they had seen of the large-scale spray type desuperheaters used in central station plants, a well proportioned venturi sleeve with ample length for mixing was necessary to prevent impingement of water particles on the pipe walls. Such a device could best be installed in a section of straight pipe external to the superheater header.

Mr. Kessler had contributed some valuable remarks on the natural relationship between air heater and economizer surface, a point that was also mentioned by Mr. Malcolmson. The reference to use of a small steam air heater to raise the dewpoint at the cold end of the gas air heater was particularly interesting, as one of the authors had been trying to interest the boiler manufacturers in this scheme for several years.

The new A.S.M.E. hoop-stress formula was shown by Mr. Kessler to be quite similar in form to the Interim Guide expression. It should be noted that a joint task force had recently been appointed under the auspices of the American Standards Association to review the entire field of hoop-stress formulae, with the intention of arriving at a single formulation which would be acceptable to all the agencies concerned.

The prospects of gas recirculation mentioned by Mr. Kessler had undoubted advantages with respect to superheater design and control but would require auxiliary hot gas fans which introduced sizable weight and space requirements as well as maintenance problems.

With reference to the superheater tube seat problem described by Mr. Kessler, the authors had recently learned that there was a strong trend to the use of external strength welds similar to those shown in Fig. 22.

Mr. Hasek had contributed an interesting statement of the numerous and complex factors entering into superheater design and had clarified the current theories regarding the relative effectiveness of steam injection in stoker fired versus pulverized fuel and oil firing.

INSTITUTE ACTIVITIES

Autumn Golf Meeting 1951

It is hoped that the competition held between the Institute and the Institution of Naval Architects at the Royal Automobile Club's Golf Club at Woodcote Park on Thursday, 4th October, will be the first of many. If the success of future occasions is as great as that of the first, there should be no doubt that the competition will become an annual event.

With this particularly in view, members are reminded that notices were sent out with the September issue of the TRANSACTIONS concerning the proposed formation of a Golfing Society. It is hoped that many members will enrol on the register of this Society.

The morning competition consisted of a match between fifteen nominated members each of the Institute and the Institution of Naval Architects, and prizes were awarded also for the best stroke score, virtually two competitions being played off together. The match was won by the Institute by the margin of nine to three, with three halved. In the stroke competition the first prize, a Smith's alarm clock in a pigskin case, was won by Mr. J. G. Edmiston, with a net score of 72. The second prize, six silver tea spoons, went to Mr. A. McDougall with a net score of 73, and the third prize, a tankard, went to Mr. W. Sampson with a net score of 74.

In the afternoon four-ball bogey greensome competition, which had been arranged by ballot for all entrants, Messrs. E. F. J. Baugh and J. H. F. Edmiston came in first with 2 up, and each received a set of heat-resisting table mats; Messrs. R. K. Craig and W. Smith came in second with 2 down and received Ronson Queen Anne table lighters, while Messrs. J. Irwin and W. Ridley were third with 4 down and received Bayard alarm clocks.

In the absence of Mr. A. Robertson (Convener of Social Events) through illness, Mr. J. Turnbull (Chairman of Council) assisted by Mr. R. K. Craig, presented the prizes. Mr. Turnbull mentioned that this was the first occasion of a joint meeting with the Institution of Naval Architects; he was sure all would agree that it had been most successful and, in hoping for its repetition, he received general support.

Votes of thanks to the Royal Automobile Club Committee for the excellent arrangements which had been made for the meeting, and to the Social Events Committee for their work in conducting the competitions, were proposed by Messrs. Craig and Baugh respectively. Mr. Turnbull moved a very hearty vote of thanks to the prize donors—Messrs. R. K. Craig, T. A. Crompton, C. P. Harrison, H. R. Humphreys, J. A. Rhynas, A. Robertson, W. Sampson, A. Walker and the Institution of Naval Architects, and the proceedings terminated.

Sydney Local Section

A meeting of the Sydney Local Section was held at Science House, Gloucester Street, Sydney, on Friday, 5th October, 1951, at 8 p.m. Forty-six members and guests were present and the Chair was taken by Mr. W. G. C. Butcher.

Professor A. H. Willis, Associate Professor of Mechanical Engineering, N.S.W. University of Technology, gave a lecture entitled "Stressed Rubber". The lecture was well illustrated by lantern slides and conveyed a great deal of information in the modern use of rubber in engineering. Afterwards, an excellent discussion took place, to which Captain(E) Bull, Messrs. Findlay, Butcher, Short, Wardell, Buls, Lees, Flaherty and McLachlan contributed.

A vote of thanks to Professor Willis was proposed by Mr. H. P. Weymouth and seconded by Mr. D. N. Findlay, and carried by acclamation.

Junior Section

Birkenhead Technical College

A Junior lecture was given on Wednesday, 10th October 1951, at the Birkenhead Technical College, Conway Street, Birkenhead. Mr. A. G. Arnold (Member) took the Chair and represented the Institute, while the college was represented by Mr. Oulsnam.

112 students, apprentices, members and visitors were present to hear Lieut. Com'r(E) A. P. Monk, D.S.C., R.N.(ret.) (Member) deliver a lecture entitled "The Construction of Marine Watertube Boilers". This large attendance was particularly gratifying as Commander Monk had given this lecture on Merseyside only twelve months previously.

A large number of questions were asked and answered and a hearty vote of thanks was accorded to the author for his interesting paper.

Acton Technical College

On the 11th October 1951, Messrs. M. W. T. Rees, B.Sc., and G. J. Tuke, B.Sc., gave a lecture at Acton Technical College entitled "Electric Propulsion of Ships". The Principal of the College, Mr. R. W. MacAdam, B.Sc., M.I.Mech.E., was present, also Mr. Williams of the Mechanical Engineering Department, and the Institute was represented by Mr. C. P. Harrison (Member of Council).

About 150 students attended the meeting and their interest and appreciation were evident from the vigorous questioning of the authors which followed the lecture. The meeting was generally considered to have been most successful.

Southern Junior Branch I.N.A. and I.Mar.E.

The Annual General Meeting of the Southern Junior Branch was held on Friday, 26th October 1951, at 7.15 p.m., at the Polygon Hotel, Southampton. Forty-eight members were present. In the absence through illness of Dr. Dorey, Mr. R. W. L. Gawn, O.B.E., R.C.N.C., Vice-President, presided over the meeting, and Mr. W. J. Ferguson, Chief Executive to Lloyd's Register of Shipping, read Dr. Dorey's Presidential Address, "Science and the Marine Engineer". A motion was passed to the effect that the Secretary should write to Dr. Dorey expressing regret at his illness and wishing him a speedy recovery, and to thank him for his services to the Branch. A message from Dr. Dorey, praising the work of the Council, was read by Mr. Ferguson.

In presenting the Council's report the Secretary, Mr. T. W. Paradise, said that it was five years ago, on 25th October 1946, that the Branch held its first general meeting; there were then only sixty-three names on the mailing list compared with over 200 today. The programme for the session was as follows:—

22nd Nov. (Thurs.) 7.30 p.m. "Shallow Draught Vessels", by A. R. Mitchell, M.B.E., M.C., at the Municipal College, Portsmouth.

17th Dec. (Mon.) 7.30 p.m. Film evening at University College, Southampton, when the following films would be shown: "The Making and Shaping of Steel"; "We've come a long way"; "A Harbour Goes to France".

In January, a junior member of the Branch, John Leathard, B.Sc., Ph.D., would read a paper on modern propeller theory at Portsmouth.

The February meeting, to be held in Southampton, would be for the lecture by Mr. T. Clark (Member, I.Mar.E.) on the work of Trinity House, "Ships that Serve Ships", and a film on the subject would be shown.

The details of the March and April meetings were not yet settled; in March there would be another film evening at Portsmouth, and it was hoped that at the April meeting at Southampton a paper would be presented by a junior member of the Admiralty Experimental Works at Haslar.

In May, Mr. J. Calderwood, M.Sc. (Vice-President, I.Mar.E.) had promised to read his paper, "The Combustion Turbine", at Portsmouth.

A visit to the new oil refinery at Fawley, Hampshire, was being arranged to take place early in 1952. Later in the year a visit to the works of Saunders Roe, Ltd., at Cowes, Isle of Wight, was being arranged. A visit to one of the "Queens" in Southampton Docks was also under consideration.

The Treasurer, in his report, said that the Council intended to recommend the publication of the TRANSACTIONS in book form.

Messrs. W. J. Ayers, J. R. Cousins, J. Leathard, B.Sc., Ph.D., B. W. Ramsey, R.C.N.C., J. E. Tope, T. W. Paradise and M. Varvill were elected to the Council for 1951-52.

A motion which was passed empowering the Council to elect Honorary Members was necessary as a result of the loss of the services of experienced Members of Council on account of the age limit of thirty. When the Council had considered the question of age limit, the view had been unanimously expressed that provided the interests of junior members, for whom the Branch was formed, were fully considered, the time had come to explore the possibility of developing the organization into the Southern Branch of the two parent bodies, to embrace all grades of membership. Whilst meetings were well attended by junior members, there was always a core of senior members which remained continually in support, being sometimes in the majority. This interest by senior members augured well for the progress of the Branch and it would be a natural step forward, as well as being an acceptance of the real position, if the Branch were to expand to include those members. The Council did not for the moment wish to put this suggestion forward as a resolution but, in coming to a decision, would welcome any comments sent in to the Secretary. If the Council considered this measure to be both desirable and practicable, their proposals would be submitted to a general meeting of the Branch.

OBITUARY

W. HINCHCLIFFE (Member 5212) began his engineering career with J. Samuel White and Co., Ltd., Cowes, in 1898; after gaining sea experience with the Shaw, Savill and Albion Co., Ltd., he joined the drawing office staff of the Wallsend Slipway and Engineering Co., Ltd. In 1911 he was appointed technical assistant to the superintendent engineer of the Ellerman Lines, in 1915 he was assistant superintendent engineer of the Ellerman Hall Line, and in 1924 he was appointed to the position of chief superintendent engineer. From then until his retirement he was responsible for some eighty ships in the company's fleet; he also supervised the hulls and propelling machinery for the whole of the fleets of the Ellerman Group. Mr. Hinchcliffe retired in 1947 and lived at Cowes in the Isle of Wight until his death in October 1951.

He was a member of the Institution of Mechanical Engineers and the Institution of Naval Architects and was elected a Member of the Institute in 1924. In 1933 he was elected chairman of the Liverpool Marine Engineers' and Naval Architects' Guild and took a prominent part in the incorporation of the Liverpool Engineering Society, of which he was president in 1939. He was a director of C. and H. Crichton (1921) Ltd., and of Amos and Smith, Ltd., of Hull; in addition, he was managing director of the Ellerman Engine Works in Bootle and a consultant for the Wilson Line.

G. A. PULLEN (Member 1927) was born in 1880. He attended Birkbeck College, London, and was a pupil at the Royal Shipbuilding and Engineering Co., Ltd., in Flushing, Holland, from 1895 to 1898. Until October 1899 he was an improver, fitter and erector with Maudsley, Son and Field, Ltd., of Lambeth. At this time he entered the business carried on by his father, a machinery merchant having a large connexion with the principal Dutch shipbuilders and engineers. The merchanting side of the business did not greatly interest either him or his brother and soon afterwards they formed the firm of Frederick A. Pullen and Company, which specialized in pumping machinery, a business they carried on together until April 1950 when the firm was made into a limited company under the style of Frederick A. Pullen and Co., Ltd., of which they were both directors.

Mr. Pullen was elected a Member of the Institution of Heating and Ventilating Engineers in 1906 and for many years he served on their council; he was elected to Membership of the Institute in 1907.