a check of the heat rate that is within the accuracy of the heat-balance diagram, Fig. 2.

The result also may be checked by either Equation [33] or Equation [35], which are different forms of an approximation. Applying Equation [35] to find η_i and the heat rate

$$\left.\begin{aligned} \eta_i &= 1 - [(1 + 6.38325)(0.0011721 + 0.0019323) \\ &\qquad - 6.38325 \times 0.014188] \\ &= 0.9894925 \\ \text{hr} &= 7538.803 \end{aligned}\right\} \quad \ldots [59]$$

This is an excellent check of the result found in Equation [58].

Summary

It is seen that the theoretical gain due to addition of reheat without interconnections may be found by using Equation [44]. The fraction of this gain that is realized when interconnections exist may then be found by using Equation [32] or [35]. By multiplying the two, the actual gain is found. It is seen that η_i is very nearly unity and that when it is applied to G_{rt} the actual gain G_r is found with excellent accuracy.

These results suggest a method of closely approximating the heat rate of a reheat cycle. Given the heat rate of a low-pressure high-temperature cycle, the theoretical gain resulting from superposition of a reheat portion may be found assuming contact heaters above the reheat point and no interconnections, using the relationship of Equations [17] and [18]. Since in a typical case it was seen (see Equation [56]) that $\eta_i \approx 0.99$, this factor may then be applied to the theoretical gain to obtain a close approximation of the actual gain. The latter may then be applied to the low-pressure cycle heat rate to obtain the reheat heat rate.

Additional Loss in Low-Pressure System

The foregoing analysis of the loss due to interconnections dealt entirely with degradation of energy through its incomplete utilization in the high-pressure system and subsequent delivery at lower than ideal availability to the low-pressure system; that is, the faults which have been found are primarily those of the high-pressure system.

An additional loss is suffered in the low-pressure system because of the low availability of the energy that is supplied to it by the interconnections. The magnitude of this loss will now be defined in terms of the ratio β of the thermal efficiency in the low-pressure system of the degraded energy to that of the prime flow of this system, that is

$$\beta = \frac{\text{thermal efficiency at which } \alpha Q_i \text{ is utilized}}{\text{thermal efficiency at which } Q \text{ is utilized}} \quad \ldots . [60]$$

If the heat rate of the low-pressure system without interconnections is r_{no}, it may be shown that with interconnections it becomes

$$r_n = \frac{r_{no}}{1 - \alpha \dfrac{Q_i}{Q}(1 - \beta)} \quad \ldots\ldots\ldots\ldots\ldots [61]$$

This represents an additional loss, not implicit in the previous analysis which pertained to the high-pressure system alone.

If we wish to find the total loss due to interconnections we must add the two. Neglecting higher order terms it may be shown that the total loss, including both components is given by

$$\frac{(\Delta r_r)_i}{r_r} = \left(\frac{G_{rt}}{1 - G_{rt}}\right)(1 - \eta_i) + \alpha \frac{Q_i}{Q}(1 - \beta) \quad \ldots [62]$$

In most instances the value of β will be between 0.5 and 0.8, and must be evaluated by detailed study of the low-pressure system. Assuming it to be 0.8 for illustrative purposes and evaluating Equation [62] from the previous data

$$\left.\begin{aligned} \frac{(\Delta r_r)_i}{r_r} &= \frac{0.1894126}{0.8105874}(1 - 0.989735) \\ &\qquad + 0.011721(1 - 0.8) \\ &= 0.0023986 + 0.0023442 \\ &= 0.474 \text{ per cent} \end{aligned}\right\} \quad \ldots [63]$$

Acknowledgments

The author wishes to thank the Bailey Meter Company for which this work originally was undertaken for its permission to publish this initial step in the solution of the complex problem of performance monitoring. In addition, the kindness of Mr. David Nabow of the Duke Power Company is acknowledged for permission to reproduce a heat balance for the addition to the Allen plant.

Discussion

V. F. Estcourt.[8] The author's clear and simple presentation of a rather difficult subject has been accomplished without detracting in any way from a painstaking and rigorous analysis of the factors involved. From a purely theoretical point of view he has presented a new and valuable method for the evaluation of the components in a reheat cycle. In a sense, the treatment may be regarded as a building-block approach and the manner of utilizing the ratio of heat rejected to heat supplied (q/Q) represents a useful simplification of concept.

It is pointed out that the widely used method of analyzing the gain due to reheat is to determine "the difference in heat rate between a system in which the high-pressure turbine is exhausted directly into a low-pressure turbine, and a system in which the exhaust steam of the high-pressure turbine is returned to the boiler for reheating to higher temperature before admission to the low-pressure turbine." The author presents a radically different concept wherein "the reheat cycle is considered to consist of a low-pressure, high-temperature, regenerative cycle system upon which is superimposed a high-pressure, high-temperature turbine, a reheater, and additional feedwater heaters." The advantages claimed for this method are:

(*a*) The low-pressure turbine remains unchanged both physically and thermodynamically as one goes from nonreheat to reheat.

(*b*) Since the thermodynamic mechanism by which the heat rate at the low-pressure turbine is achieved is amenable to analysis by existing methods, this new approach requires that new methods of analysis be applied only to the high-pressure system.

As pointed out by the author, the gains under this new approach are obviously much greater because of the larger number of components in the system which are credited to the addition of reheat.

Certainly, the method described is an ingenious and valuable tool in the light of strictly theoretical considerations, and it adds a great deal in providing a better understanding of the thermodynamic components of both the reheat and nonreheat cycles. However, when viewed in terms of what we actually desire to know in selecting a reheat or nonreheat cycle, and later in attempting to compare the actual gains with the theoretical gains, it is doubtful whether we are as interested in evaluating the benefits relating

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to the superposition of a high-pressure turbine and reheater upon a low-pressure turbine as might have been the case some years ago. The practical application of this concept, although observed by the author to have been common one or two decades ago as applying to the superposition of a high-pressure turbine on an existing low-pressure plant, is of limited current interest for well-known reasons. Economic studies today center more frequently around the choice of a high pressure-low pressure turbine combination with or without reheat, and therefore the evaluation of these two cycles in relation to first cost and fuel rate is usually the point of particular interest.

It is noted that the author states his paper represents the "initial step in the solution of the complex problem of performance monitoring" which is being undertaken in conjunction with the Bailey Meter Company. Since we have had considerable to do with bringing the two principals together in this effort we have more than an average interest in the ultimate outcome of this undertaking. Although this initial effort by the author should stand on its own feet as an outstanding contribution in the field of thermodynamic analysis, we believe that it is appropriate here to consider also the significance of the author's methods in their possible application to the practical field of performance monitoring.

When attempts are made to determine by test methods whether the cycle is performing according to calculated values, we are confronted with a major problem. By using the most up-to-date methods and test instrumentation, the best attainable accuracy is slightly better than ± 0.5 per cent. On a unit having a gross heat rate of 8000 Btu per kwhr this is of the order of ± 40 Btu per kwhr. This is a relatively large figure when one is considering the effects on cycle performance of variations in equipment or cycle arrangement. Actually, with a large reheat machine having a capacity of 150–200 mw, this represents an annual fuel cost of approximately $20,000 per year. The equivalent in capital outlay is roughly $175,000.

The concept of performance monitoring must include not only the measurement of changes in heat rate but also the determination of the actual over-all performance of the turbine cycle. If the problems were confined to changes in heat rate resulting from deterioration in the performance of the various components, the difficulty in measuring small values is minimized to some extent because some of the errors in measurement may be fixed quantities which will cancel out. However, we are also greatly interested in proving by actual test whether the performance levels of various cycle components in relation to over-all heat rate have been realized and thereby have justified the capital outlay in each instance. A simple illustration will suffice.

In presenting examples of the application of the equations derived by the author, both the accuracy and simplicity of such applications are fully demonstrated. As an example, in applying Equation [55], the heat rate of the complete reheat cycle without interconnections is compared with the actual heat rate, and it is shown that the latter is about 19 Btu per kwhr poorer because of these interconnections and the use of a drain-cooler heater instead of a contact heater. This loss is of the order of 0.25 per cent of the actual heat rate. An attempt to measure such small numbers by actual test is out of the question. In our theoretical study we can substitute a drain cooler for a contact heater, calculate the gain or loss to the cycle, and evaluate the economic justification for the one or the other. However, having made the decision we cannot afford to construct the plant by both methods in order to determine the actual difference in performance. A test of over-all cycle performance is our only tool, and the best accuracy we can obtain is ± 40 Btu per kwhr when we are actually looking for 19 Btu per kwhr.

The problem is further aggravated by the fact that the actual performance of the various pieces of equipment in the cycle also will vary from the original calculations. Seldom do any components perform exactly as designed because it is not practical to construct the equivalent hardware of the various cycle components to match the theoretical computations exactly. Furthermore, each group of hardware such as heat exchangers, pumps, turbine, and so forth, is bought on the basis of individual performance guarantees. The basis of these performance guarantees probably will not match exactly the values assumed in the heat-cycle computations. Even though performance may be better than guarantee in certain components, this may serve to aggravate the problem of measuring the over-all cycle heat rate. As an example, the turbine is designed to meet certain extraction conditions which are rarely duplicated in actual service in terms of the small heat values in which we are interested. Our only recourse is to resort to a multitude of cycle corrections in order to determine whether the guarantee has been met. These cycle corrections merely remove us one or more steps further away from the measurement of actual performance as found. The use of the main turbine to drive the boiler feed pump adds still another headache to the problem of cycle evaluation by testing.

Another factor which is rapidly becoming a major stumbling block in attempts to reconcile actual performance with calculated performance is the rapid deterioration in turbine heat rate which occurs as a result of relatively minor deposits on the blades. The cycle gain in raising the throttle steam pressure from 1450 to 1800 psi is approximately 1.8 per cent. Based upon a number of turbines tested, the results indicate that half of this (roughly 70 Btu per kwhr) may be lost within the first month owing to slight films of deposits of the order of 2 or 3 mils.[9]

Although theoretical calculations of the contribution of each component to the cycle heat rate are entirely valid for the assumptions made, the difficulty in determining how much is realized in actual practice lies in the large number of variables which inevitably do not conform to these assumptions and the relatively crude techniques available for testing. It has been shown already that errors which are introduced in attempts to simplify the problem of testing, even though small, frequently represent a sizable annual fuel cost. On the other hand, the author's demonstration of the extreme accuracy of his theoretical analysis is not only appropriate but also substantiates some important conclusions in his paper.

With further reference to the possible use of the methods set forth in the paper for practical monitoring purposes, it also should be pointed out that it is desirable to analyze the actual performance of the heater chain as a component separate from the main turbine or condenser. Such a breakdown is to be preferred for a number of practical reasons when monitoring the deterioration of the various items of cycle hardware and in programing corrective measures to restore the cycle to its maximum performance. The author's consideration of the low-pressure turbine with its extraction heaters as one component and the high-pressure turbine with the reheater and additional feedwater heaters as another component does not appear to lend itself readily to this approach. We are confident that the author is aware of these difficulties, at least to some extent, and it is hoped that these comments will serve to place additional emphasis upon their importance as well as to establish in some degree a larger perspective of the general problem.

H. H. Gorrie.[10] In recent years great interest has developed in the continuous monitoring of the performance of boilers and

[9] "Observed Effects of Deposits on Steam Turbine Efficiency," by J. Angelo and K. C. Cotton, ASME Paper No. 57—A-116.

[10] Vice-President, Bailey Meter Company, Cleveland, Ohio. Mem. ASME.

turbine cycles, because it presents one of the few remaining economic opportunities for improving performance, year in and year out.

The new approach to reheat-cycle analysis developed by the author provides a means for "sectionalizing" the heat cycle for the purpose of continuous-performance monitoring. This work is particularly timely. The general acceptance of single boiler-single turbine heat cycles makes continuous-performance monitoring practical, and the present and projected size of generating units makes it an economic necessity to know the relation between day-to-day operating and design efficiency.

A potential advantage of the author's concept as compared to conventional fault-monitoring procedures lies in the manner in which operating information may be displayed. It is practical to apply analog-computer equipment that will display continuously the over-all cycle heat rate. Using the author's concept it may be possible to segregate and display continuously the performance of the high and low-pressure sections of the turbine cycle in a quantitative manner, for example Btu per kwhr.

By displaying these data as a deviation from test operation, the operating personnel is made aware of deterioration in either the reheat or low-pressure section of the cycle and can take immediate action.

General acceptance of the concept of performance display can only come from a demonstration of its accuracy as an index of performance under all conditions of abnormality or deterioration in efficiency of prime mover or auxiliary equipment. This requires further work which currently is in process, and which will be accelerated greatly by the practical application of continuous-performance-monitoring equipment.

The author's disposition of certain "nuisance" factors such as drains, leakages, and so forth, represents one method, but certainly is not the only method of accounting for these items. Fortunately, however, these items come under the heading of "bookkeeping" and seem to be of relatively small importance when it is considered that the total effect of all such factors on the heat cycle usually is about 1 per cent or less, and the utility of the index is not affected as long as the same method is employed at all times.

In the practical application of performance-monitoring equipment to present-day heat cycles, these nuisance factors must be "prescheduled" since they are numerous and difficult to measure. Since the disposition of these items in the cycle affect the heat rate to a minor degree, this gives rise to the thought that cycle designers might well consider the disposal of leakages and drains in such a manner as to simplify and improve the accuracy of continuous performance-monitoring techniques.

This thought is prompted by the philosophy that a slight decrease in design efficiency can be tolerated if means can be devised to narrow the gap between the design heat rate and that generally achieved in operation. A continuous display of performance in the major sections of the heat cycle is the first requirement in obtaining this goal.

The author is to be congratulated for his contribution and it is hoped that his work may stimulate others to make a critical analysis of power-plant heat cycles since a thorough knowledge of the heat cycle with emphasis on the problems of continuous measurement and computation is a prerequisite to any thought of automatic optimization of generating-unit operation.

D. Nabow.[11] The author has again emphasized the value of sound analysis as a tool which the power-plant designer can use in his continuing effort to build the economically efficient power plant. His method is especially useful because it calculates fractional percentage losses from the cycle parameter with an accuracy which is within acceptable limits of the losses themselves. This method permits study in depth of many phases of the cycle. Thus for the 275/295-mw units for our Allen Plant units Nos. 3 and 4, a total of 22 different arrangements of drain coolers and drain pumps have been studied for the seven heaters, and the losses for each arrangement were determined in reference to the "ideal cycle" consisting of contact heaters. The heat-balance diagram shown in the author's Fig. 2 was arrangement No. 13 with a loss of 0.15 per cent. The poorest arrangement, consisting of seven heaters without drain coolers or drain pumps, showed a heat rate loss of 0.81 per cent. The final evaluated and adopted arrangement No. 17 with six drain coolers and one heater drain pump on the No. 5 heater (HP heater No. 1) showed a loss of only 0.06 per cent.

Other cycle losses which have been evaluated include:

1 Loss in heat rate resulting from each degree of terminal difference at each heater and due to each 1 per cent pressure drop in each extraction line.

2 Loss in heat rate as a function of drain-cooler effectiveness.

3 Distribution loss resulting from distortion from "equal rise" distribution.

Within the subcritical range of operation we are approaching a condition of narrowing horizons of performance gains which, in the past, were made possible by use of larger size units and higher pressures and temperatures. Larger size of units and rising fuel costs make it necessary to attack every possible source of losses. The author has given us an effective weapon for the attack on the losses in the feedwater-heating cycle.

M. J. Steinberg.[12] This paper extends the analysis of power-plant cycles to include the reheat cycle. To this extent the paper is a most valuable addition to the available literature and gives promise of becoming an important tool for the calculation of cycle thermal performance and the evaluation of the effect of cycle design changes on the over-all performance of reheat cycles.

The author has introduced a new concept in method of approach by analysis of the effect of superimposing a reheat section upon the nonreheat condensing section of the cycle. A review of the mathematical analysis discloses no errors in fundamental theory. Such simplifying assumptions as have been made are considered reasonable in light of acceptable precision. The fact that the author's methods result in heat rates, as shown by Equations [58] and [59] of the paper, which vary by less than 0.5 Btu per kwhr from the value derived by conventional methods of computation, indicated in Fig. 2, is sufficient evidence of the effectiveness of the author's method.

It is noted with interest, however, that the author states in his introduction that the method which he developed for analysis of the nonreheat cycle has not hitherto been applied and is not directly applicable to a reheat cycle. With respect to this statement, we have been privileged to supervise a thesis[13] by two students who did succeed in extending the author's method of analysis to include the reheat cycle. The material in the thesis is too voluminous for inclusion herein, but it may be of interest to describe generally how this was accomplished. First it should be noted that the author's method of application to nonreheat cycles involves a three-step calculation:

1 The theoretical maximum gain due to regenerative feed

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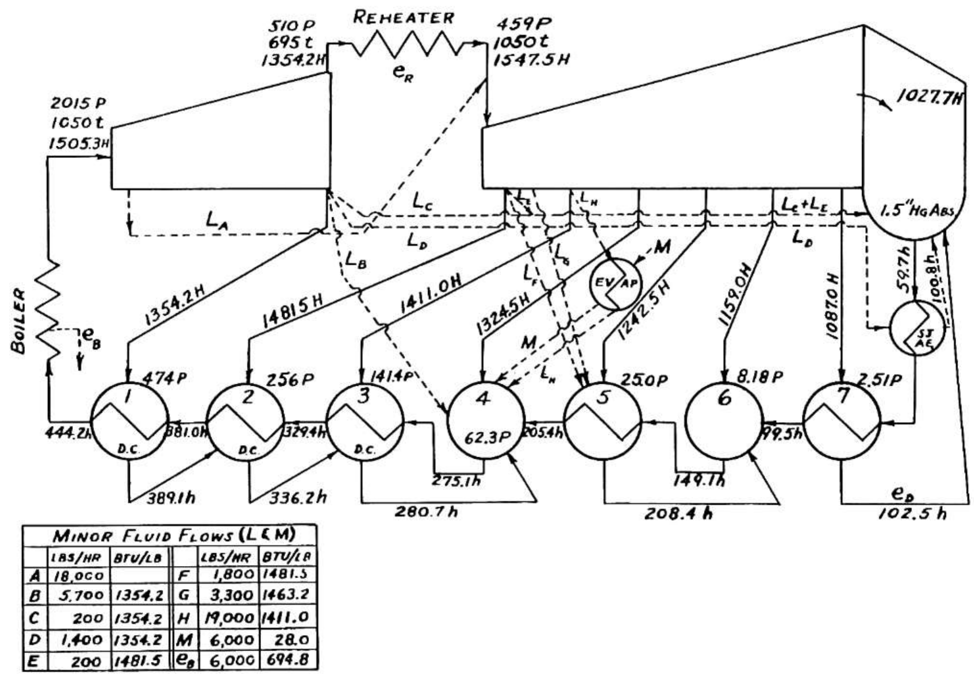
[12] Division Engineer, Consolidated Edison Company of N. Y., Inc. Mem. ASME.

[13] "A Method of Analysis for Steam Turbine Cycles," by S. M. Brodsky and E. A. Cotty, a thesis submitted in partial fulfillment of the requirements for the degree of MME at the Polytechnic Institute of Brooklyn, June, 1955.

Table 1 Chart calculations of a 3F-1C-1F-1C-1F cycle for numerical example. TFs = 1,335,000 lb per hr; 2015 psig; 1050 F/1050 F reheat; 1.5 in. Hg abs.

Heater number	1	2	3	4	5	6	7	
Type	F	F	F	C	F	C	F	
Group	I				II		III	
TF for group	$1 - \frac{y_4}{C_4} - \frac{F_3y_3}{C_4} - \frac{F_3F_2y_2}{C_4} - \frac{F_3F_2F_1y_1}{C_4}$				$1 - \frac{y_6}{C_6} - \frac{F_5y_5}{C_6}$		$1 - y_7$	Summary
R	63.2	51.6	54.3	69.7	56.3	49.6	39.8	
T	965.1	1145.3	1130.3	1049.4	1034.1	1009.9	984.5	
y	0.065485	0.045054	0.048040	0.066419	0.054443	0.049114	0.040427	
D	52.9	55.5	5.6	...	59.3	...	...	
z	0.046189	0.049102	0.0053364	...	0.058719	...	...	
F	0.953811	0.950898	0.9946636	...	0.941281	...	...	
C	...	...	...	1.066419	...	1.049114	...	
a	0.84595	0.88692	0.93272	0.93772	0.89722	0.95319	1	
ay	0.055397	0.039959	0.044808	0.062282	0.048847	0.046815	0.040427	
Σ_{ay}	...	...	...	0.202446	...	0.095662	0.040427	
TF_{group}	...	...	...	0.797554	...	0.904338	0.959573	
$(TF_I)(TF_{II})$	...	...	...	...	...	0.721258	...	
TF_x	...	...	...	...	...	...	0.692100	923,954 lb/hr
Basic e_D	...	...	...	...	...	...	0.029158	38,926 lb/hr
Basic $e_R = 1 - y_1$	...	...	...	...	...	...	0.934515	1,247,578 lb/hr



	LBS/HR	BTU/LB		LBS/HR	BTU/LB
A	18,000		F	1,800	1481.5
B	5,700	1354.2	G	3,300	1463.2
C	200	1354.2	H	19,000	1411.0
D	1,400	1354.2	M	6,000	28.0
E	200	1481.5	e_B	6,000	694.8

Fig. 5 Heat-balance diagram of a 3F-1C-1F-1C-1F cycle for example

heating, referred to a nonextraction cycle, is calculated assuming the use of an infinite number of contact heaters to heat the feedwater to saturation temperature.

2 The fractional part of the theoretical maximum gain is calculated by assuming an "ideal" cycle consisting of a finite number of contact heaters or equivalent, with equal spacing.

3 The third step involves the approximation of losses due to the departure of the actual cycle from the ideal cycle.

The approach used in the afore-mentioned thesis requires a two-step calculation:

1 The calculation of the heat rate for a "basic" cycle. This cycle is the actual cycle with the assumption that no fluid leave or enter the cycle, and the absence of steam leakages and reheater bypass.

2 Correction of the basic cycle heat rate for such items as steam leakage, introduction of make-up, pump work, generator air coolers, and so forth.

Basically, the thesis outlines a method for determination of flows which, together with appropriate enthalpy values, permits application of Equation [3] of the paper. The flows are determined by the methods developed by the author with some modifications in the interest of simplification. The calculation of the flow through the reheater is rather simple and involves no complicated formulas.

A sample calculation of the thesis method (Fig. 5 and Tables 1 and 2 of this discussion) applied to the reheat cycle of an actual installation is included herein. A variation of 4.2 Btu per kwhr, equivalent to 0.056 per cent, was obtained. Additional refinements could reduce this variation although we consider this to be unnecessary since the results are within acceptable engineering precision without such refinements.

The availability of another method of approach does not detract from the value of the paper, and we offer our congratulations to the author for an excellent and original approach to the problem.

Table 2 Leakage and make-up adjustments of a 3F-1C-1F-1C-1F cycle for numerical example

Leakage flows descrip	L_A byps rhtr	L_B to htr 4 byps rht	L_C to cond byps rht	L_D to SJAE byps rht	L_E to cond	L_F to htr 5	L_G to htr 5	L_H to htr 4 via evap	M to htr 4 via evap	e_B boiler blwdwn
L(lb/hr)	18000	5700	200	1400	200	1800	3300	19000	6000	6000
L(lb/lb T.F.)	...	0.0042697	...	0.0010487	...	0.0013483	0.0024719	0.014232	0.0044944	...
H_L	...	1354.2	1354.2	1354.2	1481.5	1481.5	1463.2	1411	28	694.8
h	...	275.1	59.7	100.8	59.7	208.4	208.4	275.1	275.1	28
T_L	...	1079.1	1294.5	1253.4	1421.8	1273.1	1254.8	1135.9	−247.1	666.8
T	...	1049.4	...	984.5	...	1034.1	1034.1	1049.4	1049.4	...
y_L	...	1.02830	...	1.27313	...	1.23112	1.21342	1.08243	−0.23547	...
$y_L - 1$	...	0.02830	...	0.27313	...	0.23112	0.21342	0.08243	−1.23547	...
$L(y_L - 1)$	...	0.00012083	...	0.00028643	...	0.00031162	0.00052755	0.0011731	−0.0055527	...
$aL(y_L - 1)$	...	0.00011330	...	0.00028643	...	0.00027959	0.00047333	0.0011000	−0.0052069	...

Line	Item	Group I	Group II	Group III	Totals	Summary
1	$\Sigma\, aL(y_L - 1)$	−0.0039936	0.00075292	0.00028643		...
2	$\Sigma\, AL(y_L - 1)$ group	−0.0034656	0.00072248	0.00028643	−0.0024567	...
3	Basic $\text{TF}_x(1 + M)$	...	...	...	0.69521	...
4	Lines 2 + 3	...	...	...	0.692753	924,825 lb/hr
5	Bypass turb. exh. (L_C and L_E)	...	...	...	...	400 lb/hr
6	Corrected TF_x (line 4 − line 5)	...	...	...	...	924,425 lb/hr

A. O. White.[14] This paper adds another important contribution to the many that the author has made to the analysis of steam power plants and their cycles. As such, it warrants careful study to insure a thorough understanding of the principles involved and application to investigations of reheat cycles.

The writer is pleased to see that the author has finally developed a strictly analytical method of handling reheat cycles, since this is a problem that has bothered the author since he published his paper, "The Steam Turbine Regenerative Cycle—An Analytical Approach" (footnote 2*b* of the paper). The method presented, while entirely consistent with his previous approach to thermodynamic cycles, is certainly a novel one, but displays that insight which he has previously displayed, and which is necessary to the solution of such problems. While the point of view is almost the exact opposite of the conventional one, it is a perfectly valid approach, and probably the only one possible for the task the author set himself. We would be well advised to become thoroughly familiar with it.

Anyone who takes the trouble can follow the reasoning and the method quite readily, particularly if he is familiar with the previous basic analysis. The writer believes the division of the problems into the determination of an "ideal gain" and then the "realization factor," η_i, based on the actual cycle arrangement is useful, both in simplifying the analysis and providing an insight into the effect of various parameters on a steam cycle.

Those who follow the example will agree that the method is simple to apply and the accuracy completely adequate. It is to be hoped that this analytical approach achieves general adoption in the industry as its use will save untold man-hours of tedious calculation in evaluating various cycles and cycle changes.

It is hoped the author will be able to publish at some later date further information on his work on performance monitoring.

Author's Closure

Mr. Estcourt's characterization of the method presented in the paper as a "building-block method" is appropriate. Work done since presentation of the present paper further breaks the plant down into smaller building blocks. This work is nearing completion and will be published at a later date.

The method of the paper is directed primarily at analysis of power plant cycles, for the purpose of performance monitoring, rather than at the selection of a type of cycle; naturally, however, it is applicable to both. The term "superposition" is a figure of speech, not related to physical apparatus, but rather to the process of thermodynamic analysis.

Mr. Estcourt's good offices in bringing the principals together in the current activity are well recognized and appreciated by both. Provided the work that has been undertaken ultimately is successful in yielding a satisfactory performance-monitoring system, he will have performed a service of inestimable value to the utility industry, in the author's judgment. His continued interest and encouragement are much appreciated.

The author agrees that one cannot test for a heat rate deviation as small as 19 Btu, and moreover that one cannot afford to undertake a complete installation to find such a magnitude by test, even if it were possible. This represents a situation where one must rely completely on analysis which, after all, is the essence of good engineering. The theory by which such changes in heat rate may be calculated is well established; the calculation of differences of such small magnitude represents the calculation of a *slope* of the heat-rate curve, rather than the absolute level of the heat rate itself. It is widely agreed that differences calculated by analytical methods are sufficiently accurate to permit the predicating of a decision on the installation of equipment upon them. Such differences are accepted on the same basis that "correction factors" are accepted, since these also represent the calculation of slope, rather than absolute magnitude.

The extension of the author's work that currently is under way is predicated on the monitoring of individual components of the complete power plant system. Since each of these components can be monitored with considerable accuracy, the entire plant can be monitored with commensurate accuracy, since the total deviation in plant heat rate obviously is equal to the sum of the individual component deviations, subject only to the root mean square value of the error in the individual quantities. When a single comprehensive test, such as the acceptance test, is used as a reference level, the method provides a means for determining changes in plant heat rate with great accuracy.

The extension of the present paper into monitoring of the components nicely takes care of the problem of which Mr. Estcourt is apparently so acutely aware, namely, that the heater

[14] Gas Turbine Department, General Electric Company, Schenectady, N. Y. Mem. ASME.

cycle does not normally operate in accordance with the cycle assumptions used for guarantees. This is a widely recognized problem. The current work, sponsored by the Bailey Meter Company, is nearing completion and will be published after it has been confirmed by numerous examples and test cases. It also provides a ready means for checking the deterioration of turbine performance that is mentioned by Mr. Estcourt.

Mr. Estcourt's recognition of the ". . . large number of variables. . ." and the complexity of the cycle is noted by the author with great satisfaction. That the literally hundreds of pressures, temperatures, enthalpies, flows, leakages, mechanical and electrical losses, etc., can be monitored by an ABC system is too much to expect. However, by using the "building-block" approach and certain over-all criteria for each component of the system, a considerable reduction in cost of equipment will be accomplished.

To allay Mr. Estcourt's concern, expressed in the last paragraph of his discussion, it is reiterated that the present paper is only the "initial step." The original plan was to present a paper on performance monitoring, but difficulties in presenting the material lucidly developed, and it finally was concluded that the present paper was necessary as a preamble to the performance monitoring paper.

The second step has included from the start precisely the approach mentioned, that is, the separation of performance of the heater system from the turbine and condenser. It was, in fact, because of the obvious need for such a breakdown that the present paper was written—to first break the plant down into two large pieces. The present effort further breaks these large pieces down into smaller ones.

The author is in general agreement with Mr. Gorrie's statements. While it is true that the nuisance items come under the heading of "bookkeeping," nevertheless the utilities can afford to do much bookkeeping for 1 per cent in efficiency at present prices. All efficiency studies and evaluations are essentially bookkeeping, anyway.

The author has discussed with Mr. Gorrie the matter of disposition of leakages in such a manner as to improve the accuracy of performance monitoring. Such an approach is highly desirable if *no loss* is incurred. On the other hand, if such rearrangement of the secondary flows of the system requires that we accept a "built-in" loss that is present for the life of the plant, it will be necessary to demonstrate in each specific instance that the possibility of better monitoring will more than offset the built-in loss. That is, the author's plea is for the best possible *design* efficiency consistent with the economics, without compromise for convenience except in those instances where a profit can be demonstrated. High *design* efficiency is a sine qua non of an efficient plant—monitoring can do no more than assist in maintaining a high level of efficiency—it cannot raise the level.

The author wishes to pay public tribute to Mr. Nabow for his keen appreciation of the value of careful and precise analysis and evaluation of power plant cycles, as demonstrated over the author's many years of association with him. His primary criterion, aside from the practicability from an operating standpoint, has always been economic evaluation of the many alternatives that invariably are available to the designer. The author undertook the work described in his discussion with recognition that decisions depended upon the results—a most satisfying environment that calls forth one's best efforts.

Mr. Steinberg's support of the author's work over at least seventeen years deserves, and is hereby tendered, the author's deepest gratitude. He has not only taught a series of graduate courses from the author's book, but he has diligently encouraged promising students to extend the work into new areas, of which his discussion is one example. Mr. Steinberg's devotion to such work represents one means for combating the current complete apathy on the part of mechanical engineering students in the field of steam power plants. The author feels that it is incumbent on the utilities, universities, and trade associations to overcome this apathy if we are to avoid finding ourselves without adequately trained people in the not too distant future.

Mr. Steinberg's continued interest and valuable contributions in the author's current effort to extend the principles of his earlier work to cover performance monitoring are solicited.

The author is appreciative of Mr. A. O. White's kind words, particularly because, as his former associate, he regards him as his most competent critic and the only one with sufficient tenacity to wade through the manuscript of footnote 2(*a*) of this paper.

The absence of a solution by the author of the reheat cycle analysis problem has resulted from his diversion by less important matters in the intervening years, and the lack of an incentive as challenging as the performance monitoring problem. With this as an objective, the present work will be extended into areas that will require the present paper as a foundation for the additional analysis.