FEED HEATING FOR HIGH THERMAL EFFICIENCY

ECONOMIES OF 25,000-KW. POWER STATIONS USING SINGLE- AND MULTIPLE-STAGE CONDENSER HEATERS, WITH AND WITHOUT ECONOMIZERS. DETERMINED TO DEMONSTRATE THE EFFECT OF VARYING THE FEEDWATER TEMPERATURE

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The author investigates the problem of determining the correct feedwater temperatures for conditions of high thermal efficiency. He applies his methods to a 25,000-kw. power station, investigating the conditions when using single- and multiple-stage condenser heaters, both with and without economizers. In appendices are given all the necessary data for the solution of the problems and the formulas by which the results are obtained.

The paper is illustrated with curves obtained from the solution of the problem, various conditions being considered and the results compared. The paper considers the problem from the viewpoint of thermal efficiency only and does not include other factors such as investment, operating cost, etc.

For power plants using single- or multiple-stage feedwater heaters of the condenser type, the temperature of the boiler feedwater as it leaves the heaters should not be less than 150 deg. fahr. when using economizers and, with the main unit operating on 330-lb. per sq. in. steam pressure, 200 deg. fahr. superheat, and 29 in. vacuum, probably not more than 260 deg. fahr. for two-stage heating when not using economizers. Three- and four-stage heating, however, permit improving thermal efficiencies up to temperatures of 300 to 340 deg. fahr., corresponding to pressures of 72 to 120 lb. per sq. in. absolute in the heater. The maximum of this range was established by consideration of fuel charges only. The lower limit of 150 deg. fahr. was chosen to avoid mechanical difficulties met when sending colder water to economizers, although this temperature, as will be seen, is less than the lowest temperature justified by purely thermal analyses of the 25,000-kw. plant used as a basis for the present studies.

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2 That there exists for any fixed set of conditions a definite feedwater temperature at which the efficiency of power generation is a maximum is most readily demonstrated by consideration of a theoretically perfect generating plant using perfect condenser heaters. Figs. 3 and 5 show that in this case, assuming that the quantity of condensate heated remains fixed, for both single- and double-stage heating, the weight of the exhaust steam required to heat the condensate is practically directly proportional to the rise in the temperature of the feedwater. If an open type of feedwater heater were used, maintaining a constant back pressure on the units supplying the exhaust steam, the load carried by these units also would

![Graph showing theoretical work done by each pound of steam used for heating water.](image)

**Fig. 1 Theoretical Work Done by Each Pound of Steam Used for Heating Water**

(The expansion is adiabatic from the boiler pressure to that corresponding to the vapor tension of the feedwater. Initial steam pressure, 325 lb.; superheat, 200 deg. fahr.)

be approximately directly proportional to the rise in the temperature of the feedwater. However, since the feedwater heaters are considered as being perfect condensers, the pressure within them, and for the purposes of the theoretical analyses, the back pressure on the auxiliary turbine or at the extraction point on the main unit supplying the heating steam will be that corresponding to the vapor tension of the feedwater, and so will increase as the temperature of the feedwater increases. This increase in pressure in turn will increase the water rates of the steam units exhausting to the heaters, or, from another viewpoint, will decrease the work capable of being done by a pound of steam used for heating the feedwater. This is shown by the curve of Fig. 1.
3 Consequent upon increasing feedwater temperatures, therefore, we have a reduction in the amount of work obtainable from

![Graph](image)

**Fig. 2 Theoretical Heat Available for Raising the Temperature of the Boiler Feedwater**

(Curve 1 is determined by subtracting from the heat content of the exhaust steam the heat content of the feedwater. The heat content of the exhaust steam, curve 2, in turn, is determined by subtracting from the initial heat content of the steam the heat equivalent of the work done by it expanding adiabatically to the pressure in the feedwater heater. Initial steam pressure, 325 lb. superheat, 200 deg. fahr.)

![Graph](image)

**Fig. 3 Weight of Exhaust Steam Required**

(The upper curve shows for single-stage heating the theoretical weight of exhaust steam required to heat the boiler feed from 79 deg. fahr., corresponding to a vacuum of 29 in. in the condenser of the main unit. The lower curve shows that required when the condensate temperature is 92 deg., corresponding to a vacuum of 28.5 in. Initial steam pressure, 325 lb.; superheat, 200 deg. fahr.)

each pound of steam used for heating the feedwater and an increase in the total amount of steam required for heating purposes. The relation between these offsetting effects is such that the work done
by the steam used for heating the feedwater increases as the temperature is increased to a certain point, after which it decreases. This is also shown by curves of Figs. 4 and 5 for both single- and double-stage heating. Any increase in feedwater temperature beyond that for which maximum work obtains continues to augment the demand for exhaust steam, but the capacity of a given quantity of the steam to do work is so reduced that the result is diminution in the total power generated by it.

**EFFECT OF MULTIPLE-STAGE HEATING**

4 A conception of what happens may be had by reference to the temperature-entropy diagrams, Figs. 6 and 7, given for both

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**Fig. 4 Work Done by Exhaust Steam**

(Curve 1 shows for single-stage heating that the theoretical maximum work obtained from steam subsequently used for heating boiler feedwater is obtained with a final feedwater temperature of approximately 240 deg. fahr. when the initial temperature of the feedwater is 79 deg. fahr. Curve 2 is based on a condensate temperature of 92 deg. and shows that in this case the maximum work is obtained with the final feedwater temperature at 260 deg. Initial steam pressure, 325 lb.; superheat, 200 deg. fahr. The peaks of the curves establish the feedwater temperatures for maximum theoretical thermal efficiency.)

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single- and double-stage heating, assuming for simplicity that saturated instead of superheated steam is used. For convenience in illustrating the effect of varying the feedwater temperature, the areas representing the work done by the house turbine were placed above the water line of the temperature-entropy diagrams of the main unit. They should be viewed as distinct from these diagrams, however, though the scales used in either case are the same. It is evident that as the temperature of the feedwater \( T_f \) is increased
from $T_e$ along $AB$ the area $abcd$, which represents the work done by the house turbine for single-stage heating, at some point becomes a maximum. This is also true for the sum of the corresponding areas for double-stage heating. The temperatures at which these maxima occur are those for the best theoretical efficiency. Lowering either the initial or back pressure lowers the feedwater temperature for maximum efficiency, while increasing either of these pressures raises this temperature.

5 As the number of stages of feedwater heating increases, the work derived from the steam used for heating the feedwater increases

![Diagram](image-url)

**Fig. 5 Theoretical Weight of Exhaust Steam to Heat Feedwater and Total Work Done by This Steam**

(With two-stage heating the theoretical weight of exhaust steam required to heat the feedwater follows the curve marked $A$, while the total work done by this steam is given by curve marked $E$. Maximum work is derived from the exhaust steam at a feedwater temperature of approximately 300 deg., which theoretically, therefore, is the temperature for best thermal efficiency. Initial steam pressure, 325 lb.; superheat, 200 deg. fahr.; vacuum, 29 in.)

and the temperature of the feedwater, as established for maximum theoretical efficiency, approaches that of the initial steam. Using an infinite number of stages, the temperature of the feedwater for best efficiency is equal to that of the initial steam and the efficiency of the theoretical power-generating cycle is that of Carnot's cycle. This is shown on the temperature-entropy diagram for infinite-stage heating.

**Description of Assumed Power Station**

6 Illustrative of what is involved in the practical problem of determining the most efficient feedwater temperatures for power
stations, the effect of varying this temperature was determined for several assumed stations of 25,000 kw. capacity, using various methods of heating the feedwater. To illustrate the influence of factors such as the water rate of the main unit and the slope of the Willans line of this unit, the internal Rankine-cycle efficiency of the bled steam and the Rankine-cycle efficiencies of the house turbine, two cases—designated Case 1 and Case 2, respectively—

![Temperature-Entropy Diagram](https://example.com/fig6)

**Fig. 6 Temperature-Entropy Diagram for Single-Stage Heating Using Dry Saturated Steam**

(Large area ABCD represents the work done by the steam passing through the main unit. The shaded area 2, above the water line, represents the work done, before entering the heater, by the steam used for heating the feedwater. The shaded area 3, below the water line, represents the heat added to the feedwater and is equivalent to the area under line ad. As indicated by areas 1 and 4 of the diagram at the bottom of the illustration, the area representing the work done by the steam used for heating the feedwater becomes rather small when the feedwater temperature deviates largely from that for best efficiency. The letters $T_f$, $T_h$, and $T_l$ indicate the feedwater temperature for different positions of the area showing the work done by the steam used for heating the feedwater. $T_g$ is the temperature of the initial steam and $T_c$ is the temperature of the condensate.)

were worked out for each arrangement of feedwater heating. The Rankine-cycle efficiency used in Case 1 for the bled steam based on the steam pressure on the turbine side of the throttle was approximately 67 per cent, the slope of the Willans line was 12 lb. per kw-hr., and the water rate of the main unit when carrying the total gross station load was 10.6 lb. per kw-hr. For Case 2 the
corresponding values were 80 per cent for the Rankine-cycle efficiency of the bled steam, 9 lb. per kw-hr. for the slope of the Willans line, and 10.26 lb. per kw-hr. for the water rate of the main unit.

7 The Rankine-cycle efficiencies of the house turbines are given in Figs. 11 and 14. The efficiencies for the 650-kw. house turbines used in Case 1 were taken from curves similar in shape to those shown in Fig. 11, but were otherwise practically the same as those used for Case 2. At low back pressure the Rankine-cycle-efficiency curve of commercial house turbines is likely to deviate from the curves shown, improving rather rapidly in the vicinity of 26 in. vacuum and falling off beyond 28 in. The lines for Case 2, however,
represent sufficiently accurately for the present purpose the efficiencies that may be expected of the better designs of house turbines, while the shape of the curve for Case 1 probably approximates what may be expected of house turbines designed primarily for high back pressure. An average value of 80 per cent was used for the Rankine-cycle efficiency of the bled steam on the main unit, though this will vary 2 or 3 per cent either way, depending on the conditions of bleeding and the design of the turbine.

All auxiliaries in these stations were considered as being motor-driven during normal operation, dual exciters and spare steam-driven boiler-feed pumps being provided to assure continuity of operation in case of disruption of the electric service. As these steam units were for use only in emergencies they did not influence the various heat balances worked out. Surface condensers were used on the main units and the condensate before going to the feedwater heaters was passed as cooling water through the evaporator system supplying boiler-feed make-up water so long as this operated on exhaust steam. When the temperature of this exhaust steam was too low to evaporate the water efficiently and the use of live steam became necessary, the condensate was returned directly to the feedwater heater and the water used for the condenser of the evaporator was taken from the boiler feedwater previously heated.

![Temperature-Entropy Diagram Using an Infinite Number of Stages](image)

(The shaded area is the total work done by the steam used for heating the feedwater. The total work cycle, \( \text{AECD} \), is seen to be a rectangle. The efficiency of this cycle is equivalent to Carnot's cycle.)
in the economizers or in the feedwater heaters. With this arrangement the use of live steam on the evaporators did not materially affect the heat balance, and the pressure of the exhaust steam used for heating the feedwater was determined by the pressure within the feedwater heater rather than the requirements of the evaporator system. A schematic arrangement of these plants is shown in Fig. 9. Economizers are shown, but heat balances as well were worked out for plants not using economizers, in which case the boiler feedwater was delivered directly to the boiler.

![Diagram of Station Layout](image)

**Fig. 9 Diagram of Station Layout**
(Schematic diagram of station layout for three-stage feedwater heating, showing distribution of steam and the various sources of water and steam losses. The condensate from the main unit is circulated through the condenser of the make-up water evaporator system before going to the feedwater heater.)

9 Radiation losses from the exhaust-steam piping and losses from low-pressure traps were considered as being independent of the feedwater temperature, although additions to these losses and to the necessary boiler-feed make-up water were made to compensate for increasing leakages from the low-pressure system with increasing pressure in that system. The extent of these additions is illustrated by the dotted heat-consumption curves of case 2, and by the curve of make-up water. The quantities of steam lost from the low-pressure systems in any case can only be roughly estimated, although they should increase with the pressure in the heaters and therefore tend to lower the temperature for best econ-
omy. In this study the losses were considered as occurring by intermittent leakages, so that under usual operation the full amount of steam bled or supplied by the house turbines would be available for heating the feedwater. Radiation losses from the low-pressure system were taken as equivalent to 2 per cent of the heat in the low-pressure steam. The steam pressure at the boiler was taken at 350 lb. per sq. in. and that at the throttle of the turbine as 330 lb. per sq. in., the superheat in either case being taken as 200 deg. fahr. The terminal difference, that is, the difference between the temperature of saturated vapor at the pressure in the heater and the temperature of the leaving boiler feedwater, was taken as 5 deg. fahr. The auxiliary power requirements were taken as 1300 kw., making the total load on the main unit when carrying the entire station 26,300 kw., at which load its steam consumption was 279,000 lb. per hr. for Case 1 and 270,000 lb. per hr. for Case 2. For every 100 kw. decrease in load on the main unit the steam consumed by it was reduced 1200 lb. per hr. for Case 1 and 900 lb. per hr. for Case 2. High-pressure drips were 1000 lb. per hr. and equivalent in heat content to 390,000 B.t.u. Condensate losses were 1000 lb. per hr. The heat content of the exhaust or bled steam was determined by the following formula:

\[
\text{B.t.u. per lb. of exhaust or bled steam} = \frac{HW - 3415(L + K)}{W}
\]
where —

\[ H = \text{heat content of initial steam, B.t.u.} \]
\[ W = \text{weight of steam passing through house turbine or bled from the main unit, lb.} \]
\[ L = \text{load on house turbine or carried by bled steam, kw.} \]
\[ K = \text{factor for radiation, friction and generator losses expressed in kilowatts. For the house turbine, } K \text{ was approximately 10 per cent of the full-load capacity of the house turbine used. For bled steam, the no-load losses were considered as being carried by the steam passing on to the condenser and } K \text{ was equal only to the radiation losses and did not exceed 1 per cent of the power generated by the bled steam.} \]

\[ \text{Fig. 11 Rankine-Cycle Efficiencies for Case 1} \]

(Estimated Rankine-cycle efficiencies of 1500-1700-kw. house turbine generators as used for determining the heat balance for Case 1. The efficiencies used for the 650-kw. house turbine for Case 1 were approximately the same as those of Case 2, though the shape of its curve is similar to the ones given here. Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.)

When bleeding the main unit the drop in pressure in the bleeder piping varied with the amount of steam bled, and this was taken into consideration. The boilers were operated between 175 and 200 per cent of rating when economizers were not used, and under this condition had an efficiency of 78 per cent. When economizers were used the operating capacity was increased to 225 per cent of rating and the efficiency reduced to 75 per cent, not including the economizers. The power taken by the induced-draft fans was not included in the auxiliary load, and to obtain the true heat-consumption rate for the stations using economizers the equivalent heat consumption of these fans will have to be added to the rates that are given.
(Steam pressure, 330 lb. gage.; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

**Curve 1.** B.t.u. per kw-hr. using a 1600-kw. house turbine and not bleeding the main unit. Economizers were not used. The feedwater temperature for best efficiency is approximately 210 deg. fahr.

**Curve 2.** B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit. Economizers were not used. The feedwater temperature for best economy is slightly higher than for Curve 1.

**Curve 3.** B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 21,875 sq. ft. area and bleeding the main unit. The economizer surface is approximately equivalent to 50 per cent of the boiler area. The feedwater temperature for best economy is approximately 190 deg. fahr. The heat consumption of the induced draft fans, equivalent to the power consumed by them, is not included.

**Curve 4.** B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 49,450 sq. ft. of surface and bleeding the main unit. Though the economizer area is double that for Curve 3, the feedwater temperature for best economy is reduced only 10 to 15 deg. The power taken by the induced-draft fans was not included with the auxiliary power when determining the heat-consumption curves.

**Curve 5.** Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit. If low-pressure leakage losses are ignored the temperature for best economy when not using economizers would be determined by the peak of this curve, which occurs at 240 deg. fahr.

**Curve 6.** Load developed by the 1600-kw. house turbine supplying exhaust steam to the feedwater heater. A comparison of the temperature at which the peak of Curve 6 occurs with that of Curve 5 indicates the influence that decreasing Rankine-cycle efficiencies with increasing
back pressures on the house turbine have on the feedwater temperature for best economy. If low-pressure steam leakages are ignored the feedwater temperature for best economy for Curve 6 would be 220 deg., or 20 deg. less than for Curve 5. By reference to Curves 1 and 2, showing the B.t.u. rates per kw-hr., it is seen that the real influence of the decreasing Rankine-cycle efficiencies with increasing back pressures is small.

FIG. 13 DOUBLE-STAGE HEATING — CASE 1
(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

Curve 1. B.t.u. per kw-hr. using a 1400-kw. house turbine and bleeding the main unit to obtain steam for the second stage, not using economizers. The feedwater temperature for best economy is approximately 280 deg. fahr.

Curve 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit at two points, not using economizers. The feedwater temperature for best economy is slightly higher than that shown by Curve 1, though, due to the flatness of the curves, it may be considered the same. It is noticeable that as the number of stages used for heating the feedwater increases, the curve of heat consumption per kw-hr. flattens out.

Curve 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 18,180 sq. ft. area and bleeding the main unit at two points. The economizer surface is approximately equivalent to 45 per cent of the boiler area. The feedwater temperature for best efficiency is between 210 deg. and 250 deg. fahr. The power taken by the induced-draft fans was not included in the auxiliary load.

Curve 4 (dotted). Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit at two points. If low-pressure losses are ignored the temperature for best economy when not using economizers would be at the peak of this curve, or 300 deg. fahr.

Curve 5. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 47,000 sq. ft. area and bleeding the main unit at two points. The feedwater temperature for best economy is between 180 and 220 deg. fahr., or 35 deg. less than shown by Curve 3.
Heat balances for two arrangements using single-stage heating and no economizers were worked out. In one arrangement the auxiliary power was obtained from a house turbine of 650 kw. capacity, or one-half the total auxiliary load. The remainder of the auxiliary load was carried by the main unit. Steam, in addition to that supplied by the house turbine, in this case was bled from the main unit for heating the feedwater. The other arrangement used a house turbine with its point of best economy at 1600 kw. for Case 1 and 1700 kw. for Case 2, and was considered as being so designed that the house turbine could deliver power to the main bus. With this arrangement no means for bleeding the main unit were provided, all steam for heating feedwater being obtained from the house turbine. As indicated by the curves 1 and 2 of Figs. 12 and 15, the first arrangement is thermally the more economical and also requires for best efficiency a slightly higher feedwater temperature than the second arrangement. It is interesting to note that whereas the theoretical feed temperature for best efficiency is in...
Curve 1. B.t.u. per kw-hr. using a 1700-kw. house turbine and not bleeding the main unit. Economizers were not used. The feedwater temperature for best economy is approximately 190 deg. fahr. This is 20 deg. lower than the best feedwater temperature for the corresponding conditions of Case 1, though, due to the flatness of the curves over this range, the proper temperature for either case may be considered as approximately the same. The difference in temperature indicated is due to the difference in the slopes of the Rankine-cycle efficiency curves for the house turbine.

Curve 2. B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit, not using economizers. The feedwater temperature is slightly higher than for Curve 1.

Curve 3. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 21,875 sq. ft. area, and bleeding the main unit. The feedwater temperature for best economy is between 160 and 180 deg. fahr. This is lower than the corresponding temperature for Case 1, due largely to the difference between the slopes of the Willans lines of the main units. The power consumed by the induced-draft fans was not included in the auxiliary power.

Curve 4. B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 49,450 sq. ft. of surface, and bleeding the main unit. The power consumed by the induced-draft fan was not included in the auxiliary power.

Curve 5. Load developed by the steam used for heating the feedwater using a 650-kw. house turbine and bleeding the main unit.

Curve 6. Load developed by the 1700-kw. house turbine supplying exhaust steam to the feedwater heater.

Curves 7, 8 and 9 (dotted) are the same as 2, 3 and 4 respectively except that the low-pressure steam losses are not included.
the neighborhood of 250 deg. fahr., the actual temperature is closer to 200 deg. fahr. This is due to various factors, among them be-

**Fig. 16 Double-Stage Heating — Case 2**

(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 29 in.)

**Curve 1.** B.t.u. per kw-hr. using a 1500-kw. house turbine and bleeding the main unit to obtain steam for the second stage, not using economizers. The feedwater temperature for best economy is approximately the same as for the same conditions of Case 1.

**Curve 2.** B.t.u. per kw-hr. using a 650-kw. house turbine and bleeding the main unit at two points, not using economizers. The feed temperature for best economy tends to be slightly higher than for Curve 1, though the difference is not appreciable. Curve 3 is the same except that losses due to leakage from the low-pressure steam piping are not included.

**Curve 4.** B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 18,180 sq. ft. area and bleeding the main unit at two points. The feedwater temperature for best economy is between 200 and 240 deg. fahr., or slightly lower than for Case 1. The power taken by the induced-draft fans was not included in the auxiliary power load. Curve 5 is the same but does not include the low-pressure steam losses.

**Curve 6.** B.t.u. per kw-hr. using a 650-kw. house turbine, economizer of 47,000 sq. ft. area and bleeding the main unit. The feedwater temperature for best economy is between 160 and 200 deg. fahr. This is lower than for Case 1, due to difference in the slope of the Willans lines of the main units. Curve 7 is the same but does not include the low-pressure steam losses.

**Curve 8.** Load developed by the steam used for heating the feedwater when using a 650-kw. house turbine and bleeding the main unit at two points.

**Curve 9.** Load developed by the steam used for heating the feedwater when using a 1500-kw. house turbine and bleeding the main unit. Fig. 18 shows the load carried by the house turbine.

...ing the decreasing Rankine-cycle efficiency of house turbines when operating at successively higher back pressures.
HEAT BALANCE FOR DOUBLE-STAGE HEATING

Similarly heat balances for two arrangements using double-stage heating and no economizers were worked out on the basis that the heating effect was divided equally between the two stages. In one of these arrangements the first-stage heaters derived the steam for heating the feedwater from a house turbine with its point of best economy at 1400 kw. for Case 1 and 1500 kw. for Case 2, only the second-stage heater deriving steam by bleeding the main unit. The other arrangement used a house turbine with its point of best economy at 650 kw., additional steam required by the first-stage heater being obtained by bleeding the main unit. The feed-

Fig. 17 Four-Stage Heating — Case 2
(Steam pressure, 330 lb. gage; superheat, 200 deg. fahr.; vacuum on the main unit, 20 in.)

Curve 1 shows the heat consumption per kw-hr. for four-stage heating when not using economizers. Curve 2 is the same except that losses due to leakage from the low-pressure steam piping are not included. The difference between the two curves indicates the effect that the assumptions made as to these low-pressure steam losses have on the slope of the heat-consumption curves. Curves 3 and 5 are heat-consumption curves when using economizers of 18,180 and 47,000 sq. ft., respectively, and curves 4 and 6 are the same but do not include losses due to low-pressure steam leakage. Curve 7 shows the load carried by the steam used for heating the feedwater. Steam was derived from a 650-kw. house turbine and by bleeding the main unit. The temperatures for best economy are approximately 250 to 350 deg. when economizers are not used, 210 to 240 for the 18,180-sq.-ft. economizer arrangement and 190 to 220 deg. when using the 47,000-sq.-ft. economizer.
FEED HEATING FOR HIGH THERMAL EFFICIENCY

water temperatures for best efficiency as indicated by curves 1 and 2 of Figs. 13 and 16 are seen to be approximately the same in either case, and about 25 deg. below that indicated by theoretical considerations alone. Both curves are rather flat over a range of 50 deg. in the vicinity of the point of best efficiency. The more efficient of the two arrangements, as with single-stage heating, is that one using the smaller house turbine, or the one bleeding the largest amount of steam from the main unit. This latter arrangement, with a feedwater temperature of 275 deg. fahr., showed a possible saving of approximately 320 B.t.u. per kw-hr. as compared with single-stage heating. The maximum efficiency using single-stage heating, however, was obtained with a feedwater temperature of approximately 200 deg. fahr. Comparing single-stage with double-stage heating on the basis that a temperature of 210 deg. fahr. was not to be exceeded, the difference between double-stage and single-stage heating is about 225 B.t.u. per kw-hr.

EFFECT OF ECONOMIZERS

12 Economizers in connection with single- and double-stage heating were applied to those stations which used a 650-kw. house turbine and bled the main unit, this arrangement being the more economical. The economizers were assumed as having a heat-recovery factor of 85 per cent and a heat-transfer rate of 5 B.t.u. per hour per deg. mean temperature difference between the flue gases and the water. Two sizes of economizers were applied to each
station to illustrate the effect of varying the size of the economizers. The boilers, as previously stated, were assumed to operate at approximately 225 per cent of their rated capacity, and their efficiency in this case was taken as 75 per cent, or 3 per cent less than that used when the stations had no economizers. The temperature of the flue gases entering the economizers in all cases was taken as 580 deg. fahr., which meant that for the type of boilers selected the percentage of rated capacity developed did not vary with the temperature of the feedwater. This, of course, required that the total

operating capacity be slightly decreased as the temperature of the feedwater entering the boiler was increased. The weight of the flue gases per pound of coal burned was taken as 19 lb. and independently of the feedwater temperature, but, inasmuch as the weight of coal burned per pound of steam generated varied with the feedwater temperature, the weight of gas per pound of steam generated also varied.

13 Analyses of heat balances as affected by feedwater tem-
temperatures did not involve considerations of boiler-room efficiency for those stations not using economizers. When economizers were used, however, the temperature of the flue gases leaving the economizer, for given equipment and operating conditions, was determined by the temperature of the water entering the economizer.

Increasing the temperature of the water entering the economizer simultaneously increased the temperature of the flue gases so that the combined efficiency of the economizers and boilers was decreased. However, as the temperature of the feedwater leaving the heaters was increased above that of the condensate, the efficiency of converting steam to power, as previously demonstrated, increased,
thereby opposing the consequent decrease in efficiency of steam
generation. The relative rates at which the efficiency of steam
generation decreased and that of power generation increased deter­
mined the temperature of the feedwater for best efficiency, these
rates being equal for the condition of best efficiency. The rate at
which the efficiency of steam generation decreases with increase in

![Graph](image-url)

**Fig. 21** Ratio of Weight of Flue Gases through Economizer to Weight of Water Flowing, and Rise in Temperature of Water

(The upper set of curves shows the ratio of the weight of flue gases passing through the
economizers to the weight of water flowing. The data are based on 19 lb. of gas per pound of
coal burned. The lower curves show the rise in the temperature of the water passing through
the economizers for various entering water temperatures.)

feedwater temperatures depends on the relative area of the econo­
mizers and the boilers. The larger the economizer relative to the
boiler, the more rapid is the rate at which this efficiency falls off.
In consequence of this, the feedwater temperatures for best efficiency
were found to be lower for the larger economizers than for the
smaller ones, though the differences were not of great moment.
14 The temperature for best economy when economizers are used, as given by the heat-consumption curves 3 and 4 of Figs. 12 and 15 for single-stage heating, lies between 175 and 190 deg. fahr. for Case 1 and 165 and 175 deg. fahr. for Case 2. With double-stage heating, Figs. 13 and 16, the corresponding temperatures are 225 and 175 deg. fahr. for the smaller and larger economizer surfaces, respectively, of Case 2, and 230 and 200 deg. fahr. for Case 1. The temperatures for Case 1 are lower than those for Case 2, due to the difference in the slopes of the Willans lines used. For each kilowatt-hour developed by the steam used for heating the feedwater in Case 1, the steam condensed in the main unit's condenser was reduced 12 lb., while for Case 2 this figure was 9 lb. The benefit derived from carrying load on steam used for heating the feedwater was therefore relatively less for Case 2 than for Case 1, while the economizer and boiler efficiency remained the same. The differences in temperature are small, however, and indicate that with fair

![Graph showing heat consumption per kw-hr. for various stages of feedwater heating - Case 2](image)

(The curve illustrates that as the number of stages increases above two the value of the last stage decreases rather rapidly. The heat-consumption rates given by the curve do not include the losses due to the steam leakage from the low-pressure steam piping and are for the arrangements which do not use economizers.)
accuracy the desirable feedwater temperature may be considered as a range which is largely independent of the characteristics of the steam equipment used. The economizers used for double-stage heating were slightly smaller than those used for single-stage heating, commercial considerations indicating that this was justified. The difference in the sizes of the economizers did not, however, materially affect the feedwater temperature for best efficiency. The rate-of-heat-consumption curves are rather flat over a considerable range in proximity to the temperature for best economy, and increasing the area of the economizers even to the extent of doubling them need not require large changes in the temperature of the feedwater. The various data used in connection with the economizers are given in curves, Figs. 19, 20 and 21.

COMPARISON OF RESULTS WITH AND WITHOUT ECONOMIZERS

15 A comparison of feedwater temperatures for best economy as indicated by Case 1 and Case 2, respectively, shows that when economizers are not used ordinary variations in the Rankine-cycle efficiency of the bled steam, the efficiency curve for the house turbine, and the slope of the Willans line of the main unit have no considerable effect. When economizers are used, the temperatures are reduced by decreasing the slope of the Willans line of the main unit, but, as previously stated, the influence is not large. It is evident that for the purpose of establishing the proper feedwater temperature for best economy minute accuracy in the determination of the various turbine efficiencies is not required. The overall efficiency of the entire station is influenced, of course, by these efficiencies, but the best feedwater temperature changes only slightly with them. For this study a constant efficiency for bled steam was used. However, the house-turbine efficiencies were considered as being a function of the feedwater temperature, and a comparison of the temperatures for best economy obtained when using a house turbine alone with those obtained when using a smaller house turbine together with bleeding the main unit illustrates the influence of the variation in the Rankine-cycle efficiency. The difference in the temperatures that were obtained is between 10 and 15 deg. fahr., but as the curves are flat this is of no great consequence.
MULTIPLE-STAGE HEATING

16 By similar methods of computation, heat balances for four-stage heaters with and without the use of economizers were determined and the results are presented by the curves of Fig. 17. The average Rankine-cycle efficiency for the bled steam in this case was taken at 79 per cent instead of 80 per cent, as the average efficiency over four stages would probably be somewhat less than that for one stage. A curve, Fig. 22, is also given showing the heat consumption per kilowatt-hour for the various methods of heating the feedwater worked out for Case 2, using the 650-kw. house turbine. As the number of heating effects increase, the value of the last effect decreases, which is to be expected. It is interesting to observe also that as the number of effects increase, the heat-consumption curves for the various stations flatten out in proximity to the temperature for best economy and that it would therefore seem undesirable to go beyond a certain feedwater temperature regardless of the number of stages. When economizers of 21,850 sq. ft. are used the gains in efficiency referred to a basis of no heating for single-, double-, triple-, and quadruple-stage heating are 1.86, 2.88, 3.35, and 3.64 per cent, respectively. A proper comparison of two-, three- and four-stage heaters would require a study of investment and operating charges, and the curves are given only as an indication of what may be expected from the viewpoint of the thermal efficiency when the number of heating effects are increased. As stated above, with increasing number of effects the value of the last heating decreases, and in determining the proper number of stages consideration must be given to the investment and also to the operating problems encountered.

THE FORMULAS USED

17 The various formulas and data used in making this study are given in the following pages. In general, when economizers are not used the temperature for best economy occurs in the neighborhood of that which gives a maximum for the power generated by the steam used for heating the feedwater. When economizers are used the temperature for best economy is given rather closely by determining the temperature which makes —

\[
\frac{\Delta L}{\Delta t} = \frac{W' \left( \frac{1}{W' + \frac{W'b}{aT + 0.475GHi + 0.5}} \right)}{n \left( H_S - t_i + 32 \right) - h}
\]
where \( \frac{\Delta L}{\Delta t} \) = rate of increase of load carried by steam used for heating the feedwater with increase of feedwater temperature

\( W' \) = weight of water passing through economizer per hour

\( G \) = weight of gases per pound of coal

\( i \) = percentage of recoverable heat

\( b \) = heat value of coal in B.t.u. per lb.

\( n \) = slope of Willans line of main unit

\( H_s \) = heat content of steam in initial state

\( a \) = total area of economizer in sq. ft.

\( T \) = rate of heat transmission in B.t.u. per hr. per sq. ft. per deg. mean temperature difference

\( h \) = B.t.u. required by the house turbine to produce 1 kw-hr.

\( t_1 \) = temperature of condensate from main unit

\( t \) = temperature of feedwater entering economizer

\( H \) = total over all heat requirements of station when feedwater is at temperature \( t \), i.e., B.t.u. in coal fired.

\( \Delta L/\Delta t \) may be found by determining the increments in \( L \) for definite increments in \( t \) and dividing the increments in \( L \) by the chosen increments of \( t \), using for single-stage heating the formula

\[
L = \frac{r BS' (t - t_1)}{3415 C + r B n (t - t_1)}
\]

where \( r \) = Rankine-cycle efficiency of house turbines

\( B \) = adiabatic heat drop from initial steam condition to the temperature of the feedwater

\( S' \) = weight of steam consumed by the main unit when carrying the total load including that of all auxiliaries

\( C \) = heat available per lb. of heating steam for raising temperature of feedwater.

The value of

\[
W' \left( \frac{1}{\frac{W'}{W' b} + \frac{a T}{0.475 G H_1} + 0.5} \right) \frac{n (H_s - t_1 + 32) - h}{n (H_s - t_1 + 32) - h}
\]

does not vary greatly with the feedwater temperature, so that one determination suffices. As the value of this expression decreases, the feedwater temperature for the best economy increases. It is of value, therefore, in judging the influence of various factors, though for determining the proper feedwater temperature the formulas in the Appendix should be used.
AIR ECONOMIZERS

18 If air for the boilers is preheated by means of air economizers and no feedwater economizers are used, such preheating does not affect the feedwater temperature for best economy. However, the air may be preheated by exhaust steam similarly to heating feedwater. Also air economizers and exhaust-steam air heating may be used with or without the coincident use of water economizers, and with single- or multiple-stage feedwater heating. If air is heated by exhaust steam alone, the problem of determining its temperature for maximum efficiency is not different from that for determining the temperature of feedwater heated only by exhaust steam. This, however, is not the case if both air and water economizers are used simultaneously.

19 Air and water economizers, if used simultaneously, may be placed either in parallel or in series. If placed in parallel the effect of the air economizer is to reduce the amount of gases available for heating the feedwater, and so alter the operating characteristics of the water economizers so far as these are determined by the ratio of the waste gases to the water heated. If the air heater is placed in series with the water economizer and in the coldest part of the flue gases, the leaving temperature of these gases is no longer determined by the temperature of the boiler feedwater. For such an arrangement it would appear desirable to heat the boiler feedwater to that feedwater temperature giving the maximum efficiency for the conversion of steam energy to electrical power. The feedwater would then be further heated in a water economizer while the temperature of the flue gases leaving the water economizer could be reduced to a desirable point, justified, of course, by the efficiency of the air economizers and their cost. On this basis the temperature of the feedwater entering the economizers would correspond to that previously determined when no economizers were used. The air economizers in this case, of course, would have to reduce the temperature of the flue gases below that established for the use of water economizers alone.

CONCLUSIONS

20 So many factors enter into the determination of the proper feedwater temperature of a plant that figures determined for one station should not be directly applied to another. The tem-
perature should be lower for plants using single-stage heating than for those using multiple-stage heating, and also should be lower for plants using economizers than for plants not using economizers. Efficiencies of auxiliary apparatus as well as of the main generating units have their influence. If the feedwater temperature be raised much above 212 deg. fahr., factors incident to the use of pressures well above atmosphere in the auxiliary exhaust piping come into play. The proper feedwater temperature is dependent somewhat upon the initial steam pressure and temperature and on the temperature of the condensate, increasing as these increase. It is quite impossible, therefore, to determine, except within wide limits, feedwater temperatures applicable to all plants, and the purpose has been rather to indicate in a broad way the effect of feedwater temperatures on power-plant efficiency, leaving out the matter of costs, and to give some basis for estimating the sacrifice in fuel made to assure practicable operation of the feedwater-heating system as laid out for a particular station.
APPENDIX NO. 1

DATA

21 The following data were used as bases for the heat-balance study of this paper:

Net station load.................................................. 25,000 kw.
Load on auxiliary bus.......................................... 1,300 kw.
Gross station load............................................... 26,300 kw.
Steam consumption of the main unit when carrying the gross station load:
Case 1: 279,000 lb. per hr.
Case 2: 270,000 lb. per hr.

High-pressure drips............................................. 1000 lb. per hr.
Condensate losses................................................ 1000 lb. per hr.
High-pressure steam losses................................... 3000 lb. per hr.
Radiation losses from low-pressure steam:
2 per cent of total heat in steam used for heating the feedwater
Pressure of steam at throttle................................. 330 lb. per sq. in. gage
Boiler pressure.................................................. $350 lb. per sq. in. gage
Superheat.......................................................... 200 deg. fahr.
Heat content of boiler steam................................. 1326 B.t.u. per lb.
Heat content of steam at throttle........................... 1324 B.t.u. per lb.
Vacuum on main unit........................................... 20 in. Hg.
Temperature of condensate.................................. 75 deg. fahr.
Temperature of make-up water entering evaporator....... 60 deg. fahr.
Slope of Willans line of main unit......................... Case 1: 12 lb. per kw-hr.
Case 2: 9 lb. per kw-hr.

Heat content of the high-pressure drips recovered........ 390 B.t.u. per lb.
Radiation, friction and generator losses of the house turbine in kw.:
650-kw. turbine, 65 kw.
1500-kw. turbine, 135 kw.

Radiation losses from bled steam on passing through the main unit:
1 per cent of load developed by the bled steam

Internal Rankine-cycle efficiency of bled steam based on
steam pressure after throttle............................... Case 1: 67 per cent
Case 2: 80 per cent

Boiler efficiency when not using economizers............. 78 per cent
Boiler efficiency when using economizers but not including
the economizer efficiency.................................... 75 per cent
Coefficient of heat transmission through economizers.... 5 B.t.u. per sq. ft.
per deg. mean temperature difference between flue gases and water
Cost of economizers per sq. ft. of surface................. $4.00
Cost of boilers per sq. ft. of surface...................... $4.50
Annual charges against economizers which vary as their size:
20 per cent of their cost

Annual charges against boilers................................ 15 per cent of their cost
Use factor of economizers..................................... 50 per cent and 100 per cent
Use factor of boilers.......................................... 85 per cent
Temperature of gases entering economizers......................... 580 deg. fahr.
Specific heat of flue gases.................................................. 0.2375
Percentage of recoverable heat recovered by economizers........ 85 per cent
Flue gases per lb. of coal.................................................. 19 lb.
Heating value of coal.......................................................... 13,500 B.t.u. per lb.
Cost of fuel............................................................................. $5.00 per ton
Ratio of the load developed by the boilers to the full-load rating
of the boilers when using economizers..................................... 2.25

APPENDIX NO. 2

FORMULAS

I—FORMULA FOR DETERMINING THE LOAD DEVELOPED BY THE STEAM USED IN A
SINGLE-STAGE HEATER WHEN THE STEAM IS OBTAINED FROM A HOUSE TURBINE
AND BY BLEEDING THE MAIN UNIT

22 Nomenclature:

\( M \) = make-up water in lb. per hr.
\( C \) = condensate from the main unit, after deducting losses, in lb. per hr.
\( S_B \) = steam bled from the main unit in lb. per hr.
\( S_H \) = steam derived from the house turbine in lb. per hr.
\( D \) = high-pressure drips in lb. per hr.
\( R \) = radiation losses in per cent of the heat in \( S_B \) and \( S_H \)
\( L_H \) = load developed by the house turbine in kw.
\( L_B \) = load developed by the steam bled from the main unit in kw.
\( W_H \) = water rate of the house turbine in lb. of steam per kw-hr.
\( W_B \) = pounds of steam bled per kw-hr. developed by the bled steam
\( C_1 \) = steam supplied to the main unit when carrying the full station
load, including that of all auxiliaries when no steam is bled
\( a \) = condensate losses
\( C_1 - a \) = condensate to be heated when the main unit carries the entire
station load and no steam is bled
\( T \) = temperature of the feedwater
\( L \) = total load developed by the steam subsequently used for heating
the feedwater
\( h_m \) = heat content of make-up water in B.t.u. per lb.
\( h_c \) = heat content of condensate in B.t.u. per lb.
\( H \) = heat content of high-pressure steam in B.t.u. per lb.
\( h_d \) = heat content of high-pressure drips in B.t.u. per lb.
\( n \) = slope of Willans line of main unit in the region of the load carried
by it
\( I_H \) = radiation, friction, and generator losses of the house turbine in kw.
\( I_B \) = radiation losses of the bled steam passing through the main unit
in per cent of the load developed by the bled steam.
Since the heat content of the mixture leaving the heater is equivalent to the heat content of the fluids entering the heater,

\[
(S_B + M + C + S_H + D)(T - 32) = h_m M + h_c C + (1 - R)[HSH - (L_H + l_H)3415] + (1 - R)[HSH - L_B(1 + l_B)3415] + Dh_d
\]

\[
C = C_1 - a - nL
\]

\[
S_H = L_H W_H
\]

\[
L_B = L - L_H \text{ and}
\]

\[
S_B = (L - L_H) W_B
\]

from which —

\[
L = \frac{h_m M + h_c(C_1 - a) + (1 - R)H_L W_H - (1 - R)(L_H + l_H)3415 - (1 - R)H_L W_B + (T - 32)(W_B - n) + h_c n - (1 + l_B)3415 L_H (1 - R) + Dh_d - (T - 32)(M + C_1 - a + L_H W_H + D - L_H W_B)}{(1 - R)H_L W_B + (1 - R)(1 + l_B)3415}
\]

**II — FORMULA FOR DETERMINING THE LOAD CARRIED BY THE STEAM USED IN A SINGLE-STAGE HEATER WHEN THE STEAM IS DERIVED ONLY FROM A HOUSE TURBINE**

**24 Nomenclature:** same as for Formula I, except for the following:

\[
S_1 = \text{steam consumption of the house turbine at full load}
\]

\[
S_2 = \text{steam consumption of the house turbine at half-load}
\]

\[
K = \text{full-load rating of the house turbine}
\]

\[
S_H = \frac{2(S_1 - S_2)L}{K} + 2S_2 - S_1
\]

\[
W_1 = \text{full-load water rate in lb. of steam per hr.}
\]

\[
W_2 = \text{half-load water rate in lb. of steam per hr.}
\]

\[
S_H = L(2W_1 - W_2) + K(W_2 - W_1)
\]

Since the heat content of the mixture leaving the heater equals that of the fluids entering the heater,

\[
[(M + C_1 - a - nL + L(2W_1 - W_2) + K(W_2 - W_1) + D)](T - 32)
\]

\[
= h_m M + h_c C + (1 - R)H_S - (1 - R)(L + l_H)3415 + Dh_d
\]

from which —

\[
L = \frac{(T - 32) [(M + C_1 - a + K(W_2 - W_1) + D)] - h_m M - h_c(C_1 - a) - (T - 32) n - (T - 32)(2W_1 - W_2) + h_c n + (1 - R)K(W_2 - W_1) + (1 - R)l_H 3415 - Dh_d}{(1 - R)H(2W_1 - W_2) - 3415(1 - R)}
\]

**III — FORMULA USED FOR DETERMINING THE LOAD DEVELOPED BY STEAM USED IN A DOUBLE-STAGE HEATER, THE FIRST STAGE DRAWING STEAM FROM A HOUSE TURBINE AND THE SECOND STAGE OBTAINING STEAM BY BLEEDING THE MAIN UNIT**

26 Since the heat given up by the steam equals the increased heat content of the water heated in the heater, using the previous nomenclature, and using \(T_1\) and \(T_2\) to designate the temperatures of the feedwater entering the first- and second-stage heaters, respectively, and \(T_3\) for the temperature leaving the second-stage heater.

\[
(1 - R)[H_L W_H - (L_H + l_H)3415] - L_H W_H(T_2 - 32)
\]

\[
= (M + C_1 - a - nL + D)(T_2 - T_3)
\]
for the first-stage, and —

\[
(1 - R) [HLBWB - LB (1 + lb) 3415] - LBWB (T_s - 32)
= (M + C_1 - a - nL + D + LHWH) (T_s - T_2)
\]

for the second-stage heater.

\[
WH = (2W_1 - W_2) + \frac{K}{LH} (W_2 - W_1)
\]

from which —

\[
LH = \frac{(M + C_1 - a - nL + D) (T_s - T_1) + (1 - R) 3415lH -}{(2W_1 - W_2) [H (1 - R) - T_s + 32] -}
\]

\[
\frac{K (W_2 - W_1) [H (1 - R) - T_s + 32]}{(1 - R) 3415}
\]

\[
LB = \frac{(M + C_1 - a - nL + D + LHWH) (T_s - T_2)}{(1 - R) [HWB - (1 + lb) 3415] - WB(T_s - 32)}
\]

\[
L = LH + \frac{[M + C_1 - a + D + K (W_s - W_s) + (2W_1 - W_2 - n) LH]}{(1 - R) [HWB - (1 + lb) 3415] - WB (T_s - 32) + n (T_s - T_2)}
\]

Substituting the value of \( L \) in the equation for \( LH \) permits solving for \( LH \).

IV — Formula for determining the load developed by steam used in heating when the steam is derived from a 650-kw. house turbine and by bleeding the main unit at two, three or four points accordingly as two-, three- or four-stage heaters are used.

27 The notations used are the same as previously used except for the following:

\( L'B, L''B, L'''B \) and \( L''''B \) designate the loads developed by the bled steam used in the first-, second-, third- and fourth-stage heaters, respectively.

\( W'B, W''B, W'''B \) and \( W''''B \) designate the water rates for that part of the load carried by steam bled for the first-, second-, third- and fourth-stage heaters, respectively.

28 Since the heat given up by the steam entering the heater equals the increased heat content of the water heated,

\[
(1 - R) [HLHW - (LH + lb) 3415] - LHWH (T_s - 32)
+ (1 - R) [HL'WB'B - L'B (1 + lb) 3415] - L'BWB' (T_s - T_2)
= (M + C_1 - a - nL + D) (T_s - T_1)
\]

for the first-stage heater, from which —

\[
LB' = \frac{(T_s - T_1) (M + C_1 - a - nL + D) - LHWH [(1 - R) H - T_s + 32]}{WB' [(1 - R) H - T_s + 32] -}
\]

\[
\frac{(1 - R) 3415 (LH + lb) 3415}{(1 - R) (1 + lb) 3415}
\]

\[
LB'' = \frac{(M + C_1 - a - nL + D + LHWH + LB'WB') (T_s - T_3)}{(1 - R) [HWB'' - (1 + lb) 3415] - WB'' (T_s - 32)}
\]

\[
LB''' = \frac{(M + C_1 - a - nL + D + LHWH + LB'WB' + LB''WB'') (T_s - T_3)}{(1 - R) [HWB''' - (1 + lb) 3415] - WB''' (T_s - 32)}
\]
\[ L_{B'''} = \left( \frac{M + C_1 - a - nL + D + L_b W + L_B' W_B' + L_B'' W_B'' + L_B''' W_B'''}{1 - R} \right) \left( H_{W_B''''} - (1 + l_B)3415 \right) - \frac{T_3 - T_4}{W_B'''} (T_3 - 32) \]

\[ H_{H} = (1 - R) \left( H - T_3 + 32 - \frac{1 - R}{W_B} \left( 1 + \frac{l_H}{L_H} \right) \right) 3415 \]

\[ H_{B'} = (1 - R) \left( H - T_3 + 32 - \frac{1 - R}{W_B'} \left( 1 + l_B \right) \right) 3415 \]

\[ H_{B''} = (1 - R) \left( H - T_3 + 32 - \frac{1 - R}{W_B''} \left( 1 + l_B \right) \right) 3415 \]

\[ H_{B'''} = (1 - R) \left( H - T_3 + 32 - \frac{1 - R}{W_B''''} \left( 1 + l_B \right) \right) 3415 \]

\[ H_{H}, H_{B'}, H_{B''}, H_{B'''} \text{ and } H_{B''''} \text{ represent the heat content, not including that of the liquid, in the exhaust steam from the house turbine and in the steam bled for the first, second, third and fourth stage heaters respectively.} \]

Let \[ \left( \frac{T_3 - T_2}{H_B''''} + \frac{T_3 - T_2}{H_B'''} + 1 \right) \left( \frac{T_3 - T_2}{H_B'''} + 1 \right) \left( \frac{T_3 - T_2}{H_B''''} + 1 \right) - A \]

Then \[ L = L_B' \left[ W_B' \left( 1 + \frac{H_B'}{T_2 - T_1} \right) A + 1 \right] + L_H \left[ W_H \left( 1 + \frac{H_H}{T_2 - T_1} \right) A + 1 \right] - L_H W_H H_H \]

\[ L_B' = \frac{(T_2 - T_1) \left[ M + C_1 - a - nL + C_H \left( 1 + \frac{H_B'}{T_2 - T_1} \right) + 1 \right] + D}{W_B' H_B' + n \left[ A W_B' \left( 1 + \frac{H_B'}{T_2 - T_1} \right) + 1 \right]} (T_2 - T_1) \]

\[ L_B'' = \frac{T_3 - T_2}{H_B''''} W_B'' \left[ L_B' W_B' \left( 1 + \frac{H_B'}{T_2 - T_1} \right) + L_H W_H \left( 1 + \frac{H_H}{T_2 - T_1} \right) \right] \]

\[ L_B''' = L_B' W_B' \left( 1 + \frac{H_B'''}{T_3 - T_2} \right) \left( \frac{T_4 - T_3}{W_B''''} \right) \]

\[ L_B''' = L_B''' W_B'' \left( 1 + \frac{H_B''''}{T_4 - T_3} \right) \left( \frac{T_4 - T_3}{W_B''''} \right) \]

The load on the house turbine \( L_H \) is determined by the capacity of the house turbine which in the case considered was 650 kw.

**V — Formula for determining the steam passing through to the condenser of the main unit**

\[ C_1 - nL = \text{steam condensed} \]

**VI — Formula used to determine the combined boiler and economizer efficiencies when using economizers, and the size of the economizer for best commercial efficiency**

**Nomenclature:**

- \( g \) = weight of flue gas per lb. of coal
- \( S \) = sq. ft. of economizer surface per lb. of water
$H_S$ = heat content of steam as delivered by the boilers

$T_2$ = temperature of the water leaving the economizers

$T_1$ = temperature of the water entering the economizers

$r$ = charges against economizers in percent of their initial cost (These charges are those which vary with the size of the economizer)

$C$ = cost of economizers in dollars per sq. ft.

$r'$ = fixed charges against boilers in percent of their initial cost

$a$ = use factor for economizers

$b$ = use factor for boilers

$T_{o1}$ = initial temperature of the flue gases

$T_{w1}$ = initial temperature of the water

$D = T_{o1} - T_{w1}$

$T = coefficient of heat transmission through economizers$

$G = pounds of flue gas per lb. of water$

$R = rise in temperature of the feedwater$

$e = boiler efficiency not including economizer$

$F = cost of fuel in dollars per ton$

$L = ratio of load developed by boilers to the full-load rating of the boilers$

$C_B = cost of boilers in dollars per sq. ft.$

$B = heat value of coal in B.t.u. per lb.$

$i = percentage of recoverable heat.

30 The areas of the economizers used were determined by the following formula:

$$S = \frac{K \sqrt{D} - \frac{1}{T}}{0.475 G + 0.5}, \quad K = \sqrt{\frac{a}{TrC} \left(\frac{4.375 F}{Be} + \frac{10 C_B r'}{33400 Lb}\right)}$$

$S$ represents the area of economizer surface per pound of water flowing through the economizer. The area of the economizer so determined gives the best commercial efficiency, assuming that the cost of the economizers, and the variable charges against them which vary with their size, may be expressed as a fixed ratio of their area.

31 The increased efficiency of steam generation due to the addition of economizers was determined by the following formula:

$$e = \frac{100 g R}{GB}$$

in which —

$R = \text{rise in temperature of water passing through the economizer}$

$$D = \frac{1}{S T} + \frac{1}{0.475 G t + 0.5}$$

$$G = \frac{g}{Be} \left( H_S + 32 - T_1 - \frac{D}{\frac{1}{ST} + \frac{0.475 G t + 0.5}} \right)$$

This equation is readily solved by first approximating $G$. 
DISCUSSION

G. G. Bell. Mr. Helander has presented a very interesting paper on the effect of feedwater heating on plant economy. He has made a very complete study of the subject, and shows the theoretical relation between plants equipped with large and small house turbines, both with high boilers without economizers and medium-sized boilers with economizers.

In the spring of 1922, two more 30,000-kw. units were purchased for the Windsor station, the boiler drum pressure was raised from 250 to 350 lb., and on account of this higher pressure, steel-tube economizers were substituted for the cast-iron economizer which had been installed with the first 16 boilers in the plant.

The troubles which other plants had had with corrosion in steel-tube economizers necessitated the installation of deaerating apparatus. Manufacturers of deaerating apparatus claimed that air separation was easier at temperatures above 160 deg. fahr. Experience with a closed system of feedwater heating which prevented enrichment indicated that an average oxygen content in the feedwater of 1/4 c.c. per liter could be maintained, provided the feedwater temperature was held at 210 deg. fahr. Investigations of an existing plant equipped with economizers had indicated some advantages in increasing the feedwater temperature to 210 deg. fahr. by utilizing the exhaust steam from a house turbine of sufficient size to supply the auxiliaries with power; although subsequent investigations have demonstrated that probably there would be a slightly higher saving if the study had been made for a temperature of 190 instead of 210 deg.

In the new addition it was decided to heat the condensate by steam bled from the main unit instead of exhaust steam from the house turbine, and it was thought that the best temperature at which the feedwater should enter the economizers should not be less than 210 deg. on account of the more efficient use of the steam in the main unit. To check this a study was made, as a result of which it was decided to heat the feedwater to the highest temperature possible by extracting steam from the thirteenth stage.

Upon the preparation of Mr. Helander's paper, these studies were revised. Additional data on the turbine was obtained so as to give complete information for bleeding all stages from the eighth

1 West Penn Power Co., Pittsburgh, Pa.
to the fifteenth inclusive, and the effect on the economy of the station of single-, double- and triple-stage bleeding within these limits was studied.

Data on steam extracted from the 30,000-kw. G. E. Curtis turbine at a load of 28,000 kw., steam conditions at throttle 300 lb. and 200 deg. and 1 in. back pressure, are shown on Fig. 23.

Curve No. 1 shows the absolute pressure at the various stages.

**Fig. 23  Data on Steam Extracted from 30,000-kw. G. E. Curtis Turbine**

Steam condition 300 lb. 200 deg. 1 in. load on turbine 28,000 kw.

No. 1 Absolute pressure at various stages
No. 2 Deg. superheat in bled steam
No. 3 Per cent moisture in bled steam
No. 4 Average B.t.u. per lb. of bled steam
No. 5 B.t.u. available for heating feedwater to maximum practical temperature by bled steam
No. 6 No. of lb. of live steam added per 10,000 lb. of steam bled.

Curve No. 2 shows the degree of superheat in the steam extracted from the various stages.

Curve No. 3 shows the percentage of moisture in the steam extracted from the various stages.

Curve No. 4 shows the average B.t.u. in each pound of steam bled.

Curve No. 5 shows the number of B.t.u. available for heating the feedwater to the maximum possible temperature with steam extracted from any stage.

Curve No. 6 shows the number of pounds of live steam that
must be added at the throttle in order to permit of the extraction of 10,000 pounds of steam at various bleeding points.

In determining the temperature to which the feedwater could be heated by each of the various stages, it was assumed that there was a loss between the turbine and the heater of 1 pound when steam was extracted from the fifteenth stage, of 1½ pounds when steam was extracted from the fourteenth stage, and 2 pounds when steam was extracted from the thirteenth or any higher stage; and that in all cases the maximum temperature to which the feedwater could be heated was 10 deg. lower than the temperature corresponding to the pressure of the steam at the heater.

The boilers purchased for Windsor are 14 tubes high and 42 tubes wide, Babcock & Wilcox cross-drum boilers, the boilers being equipped with a slag screen and Babcock & Wilcox inclined baffle. The front headers are set 21 feet above the floor, the drum center being 35 ft. 3 in. Four boilers are provided per unit, although it was assumed that when the turbine was operating at its point of best efficiency, which is 28,000-kw., three boilers would supply steam for the unit, the output of each boiler being about 100,000 lb. when feedwater is supplied at a temperature of 210 deg.

The extra boiler capacity would be utilized in reducing the rating on the boilers in the older section of the plant which are not as efficient or as liberally stokered as the newer boilers.

Fig. 24 shows the efficiency of the new boiler and steel-tube economizer at various ratings.
DISCUSSION

Curve No. 1 shows the combined efficiency of boiler and economizer.

Curve No. 2 shows the efficiency of the boiler alone.

Curve No. 3 shows the boiler efficiency corrected for the effect of change in capacity on the efficiency of the economizer.

Curve No. 4 is that portion of the preceding curve which is used, amplified so as to permit of reading any slight variation in the relative efficiency of the boiler when operating at slight differences in output.

These figures are based upon 12 per cent CO₂. It is necessary in order to compare the results of bleeding from various stages to work to a fraction of a per cent. All computations were checked by the comptometer.

Fig. 25 gives similar data for a 20-tube high boiler without economizers.

Curve No. 1 gives the efficiency of the boiler at various ratings.

Curve No. 2 is that portion of the preceding curve which is used, amplified so as to permit of reading any slight variation in the relative efficiency of the boiler when operating at slight differences in output.

In connection with the 14-tube high boiler, 8341-sq. ft. steel economizers are installed. The following tabulation gives the rise in the economizer when operating at a constant output of 100,000 lb. of steam per hour with feedwater temperatures of 135, 170, 210 and 250 deg. The figures are derived from guarantees based

---

**Fig. 25 Efficiency of 20 High, Cross Drum B. & W. Boiler**

No economizer. Steam pressure at drum 325 lb. Temperature at superheater outlet 225 deg. Feedwater temperature 210 deg.
on 210-deg. feedwater and are proportional to the arithmetical mean of the temperature differences.

<table>
<thead>
<tr>
<th>Feedwater temperatures, deg. fahr</th>
</tr>
</thead>
<tbody>
<tr>
<td>135</td>
</tr>
<tr>
<td>Gas temperature entering economizer, deg. fahr</td>
</tr>
<tr>
<td>Gas temperature out of economizer, deg. fahr</td>
</tr>
<tr>
<td>Drop in economizers, deg. fahr</td>
</tr>
<tr>
<td>Average gas temperature, deg. fahr</td>
</tr>
<tr>
<td>Water inlet to economizer, deg. fahr</td>
</tr>
<tr>
<td>Estimated rise in economizer, deg. fahr</td>
</tr>
<tr>
<td>Temperature water leaving economizer, deg. fahr</td>
</tr>
<tr>
<td>Average water temperature, deg. fahr</td>
</tr>
<tr>
<td>Average thermal difference in economizer, deg. fahr</td>
</tr>
<tr>
<td>Rise in economizers, deg. fahr</td>
</tr>
</tbody>
</table>

These results are plotted on Fig. 26, which also shows a comparison of single-extraction heating and heating the condensate by exhaust steam from a house turbine if the Windsor plant is equipped with a 14-tube high boiler and 8341-sq. ft. economizer.

Curve No. 1 shows the heat consumption of the plant if the
turbine is arranged for single-stage bleeding. These results are arrived at by calculating the heat requirements with bleeding at various stages from the tenth to the fifteenth, and drawing a curve through the points thus obtained. The curve represents the results which might be obtained if the turbine were designed with an infinite number of stages and could be bled at any one of them. As a matter of practice these results can only be obtained when steam is bled from any one of the six stages coming within the limit of this curve, and the feedwater is heated as hot as possible by the steam extracted from the turbine.

Curve No. 2 shows the effect of bleeding from the thirteenth stage, and throttling the amount of steam bled so as to get a varying temperature. This is practically a straight line. For the purpose of this study the condensate was assumed to be heated first approximately 20 deg., by the heat contained in the steam escaping from the steam seal. Twenty degrees is also the amount the condensate would be heated if before passing into the first bleeder heater it first cooled the generator air and then absorbed the heat in the bearing and transformer oil for the unit.

On curves Nos. 1 and 2 it was assumed that the boiler feed pump was motor-driven. If in place of a motor-driven boiler feed pump a steam-driven pump is used, Curve No. 3 would represent the heat requirements of the station. When sufficient steam is bled from the main unit to heat the feedwater to the maximum obtainable from the thirteenth stage, the heat consumption in the plant equipped with a motor-driven pump is approximately 1 per cent less than in a plant equipped with steam-turbine-driven pump. This is when the exhaust steam from the steam-driven boiler feed pump is discharged into the same heater as the steam bled from the main unit. When using a steam-driven boiler feed pump the minimum temperature is increased from 97.5 to 127.5 deg fahr.

Curve No. 4 shows the results if steam from the boiler feed pump is discharged to a separate heater and used to heat the feedwater sufficiently above the temperature of the feedwater leaving the extraction heater to get the necessary reëvaporation to permit the deaërator to function satisfactorily. Where the pressure will permit, the exhaust steam from the turbine glands is discharged into the same condenser as the exhaust steam from the boiler feed pump. The curve indicates that approximately the same results can be obtained by using the turbine-driven pump as the motor-driven pump, provided that separate heaters are used
and the feedwater is heated approximately 25 deg. above that used with a motor-driven boiler feed pump.

Curve No. 5 shows the results which would be obtained if the feedwater were heated by exhaust steam from a house turbine. Guarantees were obtained on house turbines designed for three different back pressures, and in figuring this curve the steam consumption used in each case would apply only if a special house turbine were designed for the operating conditions under consideration. This curve indicates that there is very little difference in the economy of using a house turbine in connection with the 14-tube high boiler and 60 per cent economizer, with feedwater temperatures between 170 and 190 deg. Assuming that this house turbine had been bought for a back pressure corresponding to 180 deg., if the feedwater temperature were then varied by increasing or decreasing the output from the house turbine, the heat requirements of the plant over the range would be higher than shown in No. 5, and would only coincide with Curve No. 5 when using a feedwater temperature of 180 deg., the point for which this particular turbine was designed.

Curve No. 6 shows the amount of power which can be obtained from the house turbines when heating the feedwater to temperatures shown in Curve No. 5.

Curve No. 7 shows the results which would be obtained if the feedwater were heated by exhaust steam from efficient high-speed geared turbines driving the auxiliaries. In this case one design of turbine is used and operated with various back pressures, so that full advantage is not taken of the increase in vacuum which it is possible to get with the lower feedwater temperature. The water rate for various feedwater temperatures is as follows:

<table>
<thead>
<tr>
<th>Final feedwater temperature, deg. fahr.</th>
<th>Back pressure on turbine exhaust, lb. per sq. in.</th>
<th>Water rate, lb. per brake hp-hr.</th>
</tr>
</thead>
<tbody>
<tr>
<td>135</td>
<td>4.53</td>
<td>16.5</td>
</tr>
<tr>
<td>170</td>
<td>7.99</td>
<td>17.0</td>
</tr>
<tr>
<td>210</td>
<td>16.13</td>
<td>21.0</td>
</tr>
</tbody>
</table>

The curve indicates that within the range of the temperatures under consideration there is practically no difference in the heat requirements of the plant, although if the small geared sets were so designed as to show an increase in economy with a decrease in
back pressure, it would pay to lower the feedwater temperature to the lowest point that was practicable and still prevent the sweating of the economizer tubes.

Curve No. 8 shows the amount of power generated by the geared turbine-driven sets. The reason that Curves Nos. 5 and 7 approach each other at the lower temperatures is that the house turbine is of larger capacity, and at low feedwater temperatures is operated at part loads, whereas in Curve No. 7 it is assumed that only enough of the geared driven units are driven to give the required temperature when each turbine is carrying the maximum load.

Fig. 27 shows a comparison of results on the Windsor plant if the turbine were arranged for single-, double- or triple-stage extraction heating if the plant were equipped with 14-tube high boilers and 8341-sq. ft. economizers.

Curve No. 1 shows the heat consumption of a plant arranged for single-stage bleeding at any of the temperatures within the limits of the curve. This curve is the same as Curve No. 1 of Fig. 26 and shows best results at a feedwater temperature of 225 deg. fahr. with a very slight increase in heat requirements by increasing or decreasing the feedwater temperature 25 or 30 deg.

Curve No. 2 shows the heat consumption of a plant arranged for bleeding from the fourteenth and a higher stage.

Curve No. 3 shows the heat consumption when bleeding from the thirteenth stage and a higher stage.

![Fig. 27](image-url)
Curve No. 2 shows that minimum heat requirements are obtained by a combination of the fourteenth and twelfth or eleventh stages at a temperature of 225 or 251 deg. fahr.

Curve No. 4 shows that the best results with triple-stage heating are obtained with a combination of the fourteenth, twelfth and tenth stages, the best results being obtained at a temperature of 275 deg. fahr.

Curve No. 5 shows the heat consumption with triple-stage heating using the fourteenth, eleventh and a higher stage. This combination is not as efficient as that shown in Curve No. 4.

These studies are all made for a plant operated with a motor-driven boiler feed pump. The study indicates that there is very little difference between bleeding the eleventh, twelfth or the thirteenth stage with single-stage heating; and that for double-stage heating a combination of the fourteenth and twelfth or eleventh stages gives the best result, the heat requirements per net kw-hr. being about 16,750 as against 16,940 B.t.u. for single-stage heating. In triple-stage heating a combination of the fourteenth and twelfth and the tenth stages gives the best results, the heat requirements for triple-stage bleeding being about 16,660 B.t.u. as compared with 16,750 B.t.u. for the double-stage heating, and 16,940 B.t.u. for single-stage heating.

While there are a number of points to consider in obtaining plant heat requirements, the work can be reduced to a comparatively simple form; and with a set of curves as are given in Fig. 23 giving information for various stages and temperatures, etc., a point can be determined every twenty minutes by using a slide rule. However, a slide rule is not accurate enough to give smooth curves.

In order to illustrate the method used in making these computations, the formulas used are given below:

\[ A = \text{the number of pounds of steam required by the main turbine to produce 28,000 kw. per hour when no steam is extracted from the turbine} \]

\[ X = \text{the number of thousands of pounds of steam bled from the main unit to heat the feedwater in the first-stage heater} \]

\[ Y = \text{the number of thousands of pounds of steam bled from the main unit to heat the feedwater in the second-stage heater} \]

\[ Z = \text{the number of thousands of pounds of steam bled from} \]
the main unit to heat the feedwater in the third-stage heater

\( r_x \) = the number of pounds of steam which have to be added per thousand pounds extracted from the \( x \) stage

\( T_c \) = the temperature in deg. fahr. of the condensate leaving the main condenser

\( T_x \) = the temperature of the condensate leaving a heater supplied with steam from the \( x \) stage

\( B \) = lb. of steam entering condenser

\( C \) = B.t.u. by test in the amount of steam required for sealing the high-pressure gland of the main unit. This is a constant, and is 5,500,000 B.t.u. above the liquid temperature of 78 deg. Where it is desired to use the number of B.t.u. in the gland steam above 32 deg., a figure of 5,700,000 is used

\( D \) = total number of pounds of live steam to be supplied to turbine

\( E \) = net kw-hr. put out by the plant

\( H_x \) = average total heat in the steam bled from the main unit at the \( x \) stage

\( F \) = equivalent number of pounds of steam that each boiler would evaporate if supplied with feedwater entering the 20-high boiler or the economizer in case boiler is equipped with economizer at 210 deg. fahr.

\( H_s \) = total heat in the steam leaving the superheater

\( h_x \) = total heat in the condensate above 32 deg. after being heated by the steam bled from the main turbine at the \( x \) stage

\( R \) = rise in the economizer after condensate has been heated by steam extracted from the one or more stages of the main unit

\( R_{210} \) = rise in the economizer when condensate enters economizer at a temperature of 210 deg.

\( B_e \) = boiler efficiency

\( P_e \) = ratio between results which we expect to get in ordinary operation as compared with the results which might be expected from the manufacturer’s guarantees. This ratio includes the plant losses from condensation, steam leaks, soot-blower loss, radiation from steam pipes, etc.

\( G \) = total steam output of boilers in operation supplying
each unit. In this study each boiler was assumed to have an efficiency of 75 per cent when the boiler and economizer together were supplying 100,000 lb. of steam per hour or 78 per cent when steam was supplied by a 20-high boiler operating at the same output.

Total live steam to turbine = \( A + Xr_x + Yr_y + Zr_z \)

Amount of steam to be bled by first-stage heater

\[
X = \frac{(A - X - Y - Z + Xr_x + Yr_y + Zr_z)}{H_x - h_x} (T_x - T_c) - C
\]

Amount of steam to be bled by second-stage heater

\[
Y = \frac{(A - Y - Z + Xr_x + Yr_y + Zr_z)}{H_y - h_y} (T_y - T_x)
\]

Amount of steam to be bled by third-stage heater

\[
Z = \frac{(A - Z + Xr_x + Yr_y + Zr_z)}{H_z - h_z} (T_z - T_y)
\]

To find the relation between \( X \) and \( Y \) by changing the form of the equations, we get

\[
\frac{X'(H_x - h_x) + C}{T_x - T_c} = A - X - Y - Z - + Xr_x + Yr_y + Zr_z
\]

\[
\frac{Y (H_y - h_y)}{T_y - T_x} = A - Y - Z + Xr_x + Yr_y + Zr_z
\]

Subtract

\[
Y = \left[ X + \frac{X (H_x - h_x) + C}{T_z - T_c} \right] \frac{T_y - T_z}{H_y - h_y}
\]

All the quantities in the above equation are known except \( X \) and \( Y \) so that a definite relation can be established between them.

Similarly by combining the expressions for \( Y \) and \( Z \) the relation between them will be found to be

\[
Z = Y\left[ 1 + \frac{(H_y - h_y)}{(T_y - T_x)} \right] \frac{(T_z - T_y)}{(H_z - h_z)}
\]

Substituting the known values for a combination of the fourteenth, eleventh and eighth stages we find that these equations when solved give simple answers.
DISCUSSION

\[ Z = 0.771Y \]
\[ Y = 1.110X + 5915 \]
\[ Z = 0.771Y = 0.855X + 4550 \]

Substituting these values of \( Y \) and \( Z \) in the equation for \( X \) permits us to find \( X \) and as the relation of \( X \) to \( Y \) and \( Z \) are known we can then readily find the amount of steam bled at each stage and multiplying it by the ratio of the amount of live steam that has to be added per 1000 lb. bled we can find readily the total steam supplied to the turbine, as follows:

\[ D = A + Xr_z + Yr_y + Zr_z \]

The steam passing to the condenser is

\[ B = (D - X - Y - Z) \]

There are two variables in considering the boiler rating; one is the number of pounds of steam to be evaporated, and the other is the number of B.t.u. to be added, which varies on account of varying feedwater temperature. When the economizer was supplied with 210-deg. feedwater, each of the three 14-high boilers and economizer was assumed to be producing 100,000 lb. of steam or a total of 300,000 lb., the efficiency of the boilers alone were taken as 75 per cent (see Fig. 24) and the boiler efficiency under all other conditions was determined from this by correcting for changes in efficiency due to changed output and taking into account the effect of variation in feed temperature.

If \( F \) = the output of each boiler in pounds of steam per hour referred to 210-deg. feed temperature as a basis,

\[ F = 100000 \times \frac{(H_s - h_s - R_s)}{H_s - h_{210} - R_{210}} \times \frac{D}{300,000} \]

Fig. 24 gives the efficiency of the 14-tube high boiler.

The net B.t.u. per kw-hr. output of the plant is obtained from the formula:

\[ D \times \frac{(H_s - h_s - R_s)}{E} \times \frac{1}{B_e} \times \frac{1}{P_e} \]

where \( E \) is equal to the net kw-hr. put out by the plant.

Where the plant was equipped with economizers and induced-draft fan, and a motor-driven boiler feed pump was used, the plant auxiliary power requirements were assumed to be 1600 kw. and \( E \) was equal to 28,000 — 1600 or 26,400 kw-hr. Where a tur-
bine-driven boiler feed pump was used, the plant auxiliary power requirements were assumed to be 1400 kw. and $E$ was equal to 28,000 — 1400 or 26,600 kw-hr.

In case the high boiler is used without the economizer, the rise in the economizer is neglected in the last two equations, and the boiler efficiency is taken from Fig. 25 instead of Fig. 24.

Where the high boiler is used with natural draft, the boiler feed pump requirements are reduced from 200 to 175 kw., and

![Graph](image)

**Fig. 28** Effect of Single-, Double- and Triple-Stage Extraction Heating on Feedwater Temperature and Station Economy

North extension of Windsor power station. 30,000-kw, G. E. turbines. 28,000-kw. load. Steam at throttle 300 lb., 200 deg., 1 in. 20 high, cross drum B. & W. boilers. No economizer, natural draft.

the auxiliary power is reduced 150 kw., on account of the omission of the induced-draft fan.

In case the steam-turbine-driven pumps are operated, Item C is increased by the amount of heat in the exhaust steam. In arriving at the B.t.u. in the exhaust steam allowance was made for variations in back-pressure by increasing the water rates of this unit \( \frac{1}{2} \) per cent per pound increase of back pressure.

A great many of the above factors are constant, and only a few change so that when one is familiar with the method the points on the curves can be obtained at the rate of two or three per hour.

Fig. 28 is a study of single-, double- and triple-stage heating
for the Windsor turbines, in combination with a 20-tube high boiler having an efficiency of about 78 per cent at the point at which it is operated.

Curve No. 1 shows the results obtained with single-stage bleeding.

Curves Nos. 2 and 3 show the results obtained with double-stage heating. There is apparently very little difference whether

![Fig. 29 Comparison of Windsor and Helander Study of Effect of Various Stage Bleeding on Power Station Economy](image)

the thirteenth or fourteenth stage is used as the first stage.

Curve No. 5 shows the result obtained with triple-stage heating, the best results being obtained by a combination of the fourteenth, eleventh and eighth stages. It is possible that a higher stage might be slightly more efficient, but the data for the higher stages were not available.

Fig. 29 is a comparison of the results obtained by Mr. Helander for various stage bleeding, with the results of the Windsor study. Mr. Helander limited his study to four stages, but the
curve is projected so as to show the approximate results for five stages. The Windsor study was made for three stages, but is extended so as to indicate the approximate results for four stages.

Fig. 30 shows the temperature at which the best results were obtained for single-, double- and triple-stage bleeding in the Windsor study and single-, double- and quadruple-stage bleeding in the Helander study.

These studies clearly indicate the advantage of bleeding the main unit with or without economizers, there being a gain of 1.10 per cent of double-stage over single-stage heating and approximately one half that amount in addition if triple-stage is used in place of double-stage heating, that is, for a 14-tube high boiler equipped with economizer; whereas for a 20-tube high boiler there is a gain of 1.64 per cent in double-stage heating over single-stage, and an additional 1.00 per cent if triple-stage heating is used in place of double-stage.

Regarding reliability, while the heater condensers will complicate the condensate piping and increase the pumping head, with the possible exception of the effect of breakage of extraction heater tubes, it is difficult to see how they will affect the reliability of the plant or complicate the operating problems. The breakage of a condenser tube can be taken care of either by installing check valves

![Fig. 30 Comparison of Windsor and Helander Studies Showing the Temperature at which Best Results are Obtained for Various Stage Bleeding](image)
between the heater and the main unit or in the lower-stage heaters by putting in drip lines of large enough capacity to take care of possible leakage. Condensate that leaks through a broken tube in this way is returned to the condenser or condensate system, and is not lost. Gate valves should be installed between the heater and the main unit so that the heater can be disconnected from the unit if desired.

Regarding the capacity of condensing equipment, as the bleeding of the main unit reduces the amount of heat passing to the main condenser, some reduction in its size is permissible. Fig. 31 shows the pounds of vapor and condensed steam entering the main condenser per hour for various bleeding combinations. This study indicates that for single-stage bleeding the condenser need only be 93 per cent of that required if no steam is bled from the main unit. For double-stage heating this ratio becomes 91 per cent and for triple-stage heating 90 per cent.

Fig. 26 indicates that heating the feedwater by exhaust steam from the house turbine is much less efficient than by bleeding the main unit, there being an advantage of approximately 1.85 per cent in favor of bleeding the main unit. This has led to a change in the type of house turbine installed. The tendency seems to be to carry only as much load on the house turbine as is necessary.
for the sake of reliability, paralleling the house turbine with the main unit and carrying all the load on the main unit, the switch being so arranged that in case of a heavy overload on the system, the house turbine with certain auxiliaries will pull away from the main unit and the house turbine will carry these auxiliaries at a slightly lower frequency until such time as the load can be again picked up by the main unit. The latest proposition is to carry no load on the house turbine but have it running so as to be able to pick up the necessary auxiliaries in case of trouble to the main unit. With this latter arrangement it is proposed to run a small pipe from the exhaust end of the house turbine to the condenser of the main unit, a check valve being placed in the main exhaust pipe from the house turbine, maintaining in this way a rarefied medium for the rotor to spin in, so that it will not overheat when running idle. The vacuum required in the exhaust end of the house turbine to prevent overheating varies with the design of the house turbine. It is only with the most efficient types of turbine that there is any danger that the vacuum which it is possible to maintain in the exhaust end of the house turbine will not be high enough during the warm summer months. The losses of such a stand-by house turbine when running in a high vacuum are very small.

It is possible to heat the condensate about 13 degrees by using the condensate to cool the air in a closed generator cooling system; and a rise of 7 degrees more may be obtained by absorbing the heat in the transformer and turbine oil. The use of condensate in these cooling coils will keep them clean; but there is some slight complication in regard to the operation of such a system during the warm summer months, or in case of dropping of load by the main unit. This latter is principally important in case the transformers and turbine are not paralleled as a unit on the high side of the transformers but paralleled with the other units on the low side of the transformers. These transformers will stand an interruption in the cooling water supply for several minutes without injurious effect. In studying such a system and comparing the reduction in heat requirements, consideration must be given to the fact that by absorbing this waste heat the amount of steam which can be bled from the main unit is reduced. While there is a possible reduction in the heat requirements of the plant of 1 2/3 per cent by absorbing the waste heat, if instead, additional steam is bled from the thirteenth stage of the main unit the heat requirements will be about 136 B.t.u. per net kw-hr. higher than if the
DISCUSSION

condensate temperature is raised by the waste heat in the generator air and transformer and turbine oil or the net gain in station economy of absorbing this waste heat is about eight-tenths of one per cent.

T. E. Keating. In view of the present agitation for improved efficiency in central stations, Mr. Helander's paper is quite timely. It is somewhat surprising to find that in the application of economizers, with 350-lb. boiler pressure and stage heating that the most efficient feed temperature is close to 200 deg., as it has been a commonly accepted belief that the efficient use of economizers demanded a somewhat lower temperature. If the desirability of this higher value can be proven, it will eliminate to a great degree various problems such as tube sweating, and if open heaters or surge tanks are used, the deaeration of water, which becomes increasingly difficult and expensive with lowered feed temperatures.

The use of multi-stage heaters is receiving considerable study in central-station engineering and the application involves many practical problems, such as the size and design of heaters, the heat transfer in event of superheated steam being bled at high pressures, the size of piping with low pressures, the use of traps and pumps, and provision for avoiding flooding the turbine. The problems of station layout seem to limit the practical application to four or five stages as Mr. Orrok states, and Mr. Helander's curve, Fig. 22, further indicates that with 350-lb. pressure the thermal gain is relatively small beyond three stages.

When a house turbine is used in combination with main unit bleeding, there is sometimes a question as to whether this auxiliary unit should be operated with approximately atmospheric exhaust pressure or at some pressure below atmospheric. Inasmuch as the efficiency of a commercial type house turbine at atmospheric exhaust is relatively poor, the greater thermal economy should occur with a house turbine expanding to a fair vacuum. However, if the last stage of heating is done in the main unit, the feed temperature will vary with the station load, while if done with a house turbine, the auxiliary load on the house unit may be adjusted so as to maintain a constant feed temperature. This same result could be obtained by use of a house turbine expanding to low vacuum, and in addition fitted with a bleeder valve for extracting steam at constant pressure. There is probably very little change in overall station efficiency due to the variable feed temper-
ature with changing load, and inasmuch as there seems to be a tendency to supply more rugged electrical equipment and more protective apparatus to insure continuity of energy supply to auxiliary feeders, the use of a house turbine as secondary source of power supply will probably become less frequent. Therefore, the house turbine should not be a controlling factor in the determination of the thermal efficiency of a station.

Mr. Helander's paper does not touch on the use of condensate for recovering the heat from the air used for cooling the main generator, presumably because he felt the thermal gain was not sufficient to warrant its use in a purely heat study. While the use of a closed cooler in connection with air recirculation is of great value, both in reducing the fire hazard and maintaining cleanliness of the generator, it is doubtful if the cost of the cooler and piping system to utilize condensate for this purpose can be commercially justified on a thermal basis when compared with obtaining the same amount of feed heating by bleeding the main unit.

F. H. Rosencrants. Mr. Helander's paper comes at a very opportune time in view of the fact that many new power plants under consideration propose incorporating feedwater heating systems designed along the lines discussed. The variety of schemes with reference to the number of stages of heating and the final temperature of feedwater proposed are indicative of the lack of harmony of thought among engineers on this subject. Such deductions as follow in this discussion are predicated on the figures as they stand in the paper with no attempt at revision or checking.

Mr. Helander points out that with an infinite number of stages of bleeding the cycle of power generation may be transformed from the Rankine cycle to the Carnot cycle, and the question arises, "What would be the corresponding ultimate saving?" Taking the author's assumed conditions of 330 lb. gage pressure and 200 deg. fahr. superheat at the throttle and a back pressure at the turbine exhaust of 1 in. Hg. absolute, the Rankine cycle efficiency figures out 35.35 per cent; the Carnot cycle (as modified by superheat) for the same conditions gives an efficiency of 40 per cent. The difference of 4.65 points in efficiency represents a saving of 13.15 per cent.

While the above efficiencies are those of the ideal cycles, it is probably fair to assume that each of the cycles may be approached in practice with equal perfection, and that, therefore, the actual
ultimate saving in practice will not be far from that calculated for the perfect cycles.

If we refer to Fig. 22 showing the B.t.u. per kw-hr. with various stages of feedwater heating, we will observe that one stage of heating gives a saving of approximately 3.8 per cent, 2 stages 5.8 per cent, 3 stages 6.8 per cent, 4 stages 7.3 per cent, and 5 stages 7.6 per cent. As Mr. Helander has already pointed out, the gain from multiplying stages rapidly diminishes with each successive stage added. From the above figures it would seem that three or perhaps four stages is about the limit for practical application, and that in view of the complications of equipment, piping, etc., it is doubtful if heating stages beyond three can be justified.

In all cases, the results in the paper are tied up with the use of house turbines. It would be interesting to draw comparisons of this type of installation with an installation in which all feedwater heating was accomplished with steam bled from the main unit. There is no question that the latter type of installation would be the more efficient, the difference between the two being a portion of the price we pay for the added reliability of auxiliary drive brought about by the use of the house turbine. We say “a portion of the price we pay,” since in addition to the loss of efficiency the house turbine set also complicates the piping lay-out and station operation.

A point of particular interest brought out in Mr. Helander’s paper is the effect of the efficiency of stage bleeding for feedwater heating on the application of economizers. In the past, there has been a tendency on the part of many engineers and manufacturers of equipment to ignore the less efficient transformation of power in the turbine room incident to the return of condensate to the economizer inlet cold. This in effect is to ignore the superiority in efficiency of the Carnot cycle over that of the Rankine cycle. These same engineers insist the gain of boiler and economizer efficiency as a result of this cold water as compared with hot water to be a net gain, and refuse to admit of an offset against this gain due to a reduction of efficiency in the steam cycle. With this idea in view, many engineers have insisted that for maximum economy the water should be returned to the economizers at the coldest possible temperature and that the practical limit was that temperature at which sweating of the economizers tubes, with consequent corrosion, begins to give trouble. Even with inefficient auxiliary drive turbines this was not true and with the much more
efficient use of heating steam as accomplished by stage bleeding of the main unit, it is more emphatically untrue. Summarizing from that portion of Mr. Helander's paper dealing with "Case 2," the following most economical temperatures with and without economizers are shown:

<table>
<thead>
<tr>
<th>Heating stages</th>
<th>Most economical feed temperatures</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Without economizers</td>
<td>With economizers</td>
</tr>
<tr>
<td>1</td>
<td>190</td>
<td>185</td>
</tr>
<tr>
<td>2</td>
<td>270</td>
<td>220</td>
</tr>
<tr>
<td>4</td>
<td>330</td>
<td>240</td>
</tr>
</tbody>
</table>

It will be observed that for multiple-stage bleeding the entering water temperature at the economizer is so high as to give but a small temperature difference between water and flue gas. An idea may be gained of the effectiveness of the economizers (50 per cent of boiler surface) by further summarizing from "Case 2" of the paper the B.t.u. per kw-hr. with and without economizers (both at the most economical feed temperature) and calculating the saving. The figures follow:

<table>
<thead>
<tr>
<th>Heating stages</th>
<th>Heating B.t.u. per kw-hr.</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Without economizers</td>
<td>With economizers</td>
</tr>
<tr>
<td>1</td>
<td>17370</td>
<td>16700</td>
</tr>
<tr>
<td>2</td>
<td>17050</td>
<td>16680</td>
</tr>
<tr>
<td>4</td>
<td>16850</td>
<td>16600</td>
</tr>
</tbody>
</table>

\[
\frac{17370 - 16700}{17370} = 3.8 - 0.50 = 3.30 \text{ per cent}
\]

\[
\frac{17050 - 16680}{17050} = 2.2 - 0.50 = 1.70 \text{ per cent}
\]

\[
\frac{16850 - 16600}{16850} = 1.5 - 0.50 = 1.00 \text{ per cent}
\]

0.50 has been deducted from the calculated gross gain as the allowance for the power requirements of the induced draft fan. In stating these figures attention should be called to the fact that a
boiler efficiency of 78 per cent without economizers and 75 per cent with economizers has been assumed. Accepting the above figures as representative of what may be actually expected, it gives us cause to question the practicability of combining economizers with multiple-stage bleeding and in view of the fact that the multiple-stage bleeder heater system, plus additional boiler capacity is likely to be less expensive than the economizer installation, it seems proper to ask the question, "Has an economizer a place in the central station of modern design?"

If we may for the moment assume the economizer eliminated, the field is left open for the application of air pre-heaters. There is, however, at the present time little data available on the cost of installation or on the operation of such equipment, though there are a number of installations for which air pre-heaters are being considered and if adopted will furnish interesting material for future papers.

An added advantage of the closed-heater system incident to the application of heating in stages with bleeder steam not touched upon in the paper is that the condensate at all points after leaving the condenser is under pressure and therefore the opportunity for the condensate to absorb air, which will later result in the corrosion of piping, economizers, boilers, etc., is eliminated. If we may assume a temperature of condensate in the economizers to be within two or three degrees of vacuum temperature, a figure which is being reached in condensers of modern design, the condensate upon leaving the condenser should show very small air content. It only remains to deaerate the make-up supply before admitting it to the system to obtain feedwater to the boiler plant in practically an air-free condition.

Oscar F. Junggren. The design of power stations is, of course, a compromise like many other things, but it is well to take stock every once in a while to see what is the ideal condition we wish to obtain. In reality, the ideal condition is to produce power most economically and this will always be done in the main unit. Whether the power is consumed by auxiliaries or sold, the most economical arrangement is to produce the power in the main unit and to heat the feedwater by bleeding. It has been felt that not all auxiliaries can be motor driven, but steam-driven auxiliaries should be reduced to a minimum. By bleeding the turbine we have the advantage of reducing the congestion in the low-pressure end
FEED HEATING FOR HIGH THERMAL EFFICIENCY

of the turbine. The modern tendency is toward motor-driven auxiliaries, permitting the highest efficiency in the production of power.

FRANCIS HODGKINSON. Stage heating of a main unit for the purpose of heating boiler feed is a reversion to ancient practice. It seems curious that we have been so slow to adopt it in modern steam turbine power house practice.

I think all turbine manufacturers realize the saving which will be the result of this practice, and later design large turbines built by the Westinghouse Electric & Mfg. Co. are provided with means for such heating. Four connections are generally provided on the turbine, where, at the point of best steam consumption, the temperature will be 140 and 200 deg. fahr., and 50 and 120 lb. per sq. in. absolute, these points being subject to slight adjustment, if necessary.

I am surprised that the author laid so much stress in his paper on the application of the so-called house turbine, for less economy is to be secured by its use, as compared with stage bleeding of the main unit. The house turbine was employed as a substitute for separate steam driven auxiliaries and with the idea that, inasmuch as this house service system would be independent of the rest of the plant, reliability of the auxiliary service would be secured. However, it is entirely possible to secure at least as high a degree of reliability in other ways, as, for instance, operating the auxiliaries from the main bus bars through a motor generator set, which motor generator set has a simple turbine connected to it which would run idly, having its governor adjusted so that it will come into operation whenever there is a fall in frequency. It has been suggested that this turbine exhausts to the atmosphere when operating under steam through a non-return free exhaust valve, there being a small connection to the main condenser, so that when the turbine is running idly, it will be under the same vacuum as the main turbine for the purpose of reducing the windage loss.

Another alternative would be an additional generator driven by the main turbine solely for the operation of the auxiliaries.

I shall be surprised if house turbines are employed very much in the stations of the future.

NEVIN E. FUNK. Theoretically, stage bleeding is probably the most efficient method of heating feedwater, but the writer is not quite sure but that it lends itself to more complication than is found
in other means of obtaining exhaust heat. Admittedly the main unit is most efficient and the heat obtained from it for feedwater heating is not thrown away in the circulating water and is obtained with a better water rate than is possible with auxiliaries. However, when things go wrong in a power plant, it is well not to have everything so connected that all operation must cease. Simplicity makes for efficiency. Many schemes which look feasible on paper will not work in practice because of the human element which is always involved.

C. M. Hardin. Everyone concerned with steam power generation realizes that reliability is of first importance and must be obtained even at the sacrifice of plant efficiency and other requisite factors.

Without discounting reliability, it is, however, vitally important that high efficiency be obtained. To obtain maximum plant efficiency every integral part of the station equipment must be of such correlative design and construction as will individually produce and maintain high efficiency.

Stage bleeding from the main prime mover is without question the method that does give highest feedwater heating efficiency as compared with using the house turbine or other steam driven auxiliaries. If stage bleeding should even in the slightest degree tend toward unreliability as regards continuity of service, the problem is to perfect the equipment used and allied with the scheme.

Advancement in any art can only be procured by development and by strengthening all links in the chain. Therefore, rather than advancement, which is vitally necessary, do not stick to ease and conservatism by adhering to old and less economical practices.

The Author. The value of recovering low temperature heat, such as can be obtained from the ventilating air used on generators, is dependent principally on the temperature to which the feedwater is heated in the first-stage heater and the efficiency of the method of heating the feedwater. Except where other factors than that of fuel economy influence the layout, investments, such as at present appear necessary to recover the equivalent of the generator losses, would seem to be of speculative value. The absorption of heat by the feedwater before it enters the first-stage heater requires that the steam bled from the main unit be reduced

1 Ross Heater & Mfg. Co., Inc., 2 Rector St., New York, N. Y.
by an amount equivalent in heat value to the heat absorbed, and this largely offsets the gain from the recovered heat. The curve of Fig. 32 shows in per cent of the recovered heat the reduction in the heat consumed by the generating station obtained by recovering low-temperature heat. With single-stage heating and a feedwater temperature of 150 deg., the heat regained is seen to be about \( \frac{1}{3} \)

![Fig. 32 Factor to Determine Value of Recovering Low-Temperature Heat with Single-Stage Heating](image)

When low-temperature heat is recovered, as is done when the ventilating air of the main generator is cooled by condensate, the reduction in the heat consumed an hour by the generating room may be approximated by multiplying the quantity of heat recovered by the heat recovery factor shown by the above curve. For accurate analyses, exact data based on the bleeding characteristics of the turbine and the method of heating the feedwater should be used to determine this factor.

of the heat absorbed by the feedwater. If the generator losses are taken as 800 kw. for a 30,000-kw. generator, operating at 25,000-kw. load, the heat regained per hour is 270 kw-hr. and this, with a generating room thermal efficiency of 25 per cent, is equivalent to 68 kw. at the switchboard. If the fuel cost of power is \( \frac{1}{2} \) cents per kw-hr. the gain in money will be 34 cents per hr. or $2,980 a year.
of 8,760 hr. This latter figure assumes a use factor of 100 per cent and a load factor of 100 per cent. Inasmuch as the combined use and load factor probably will not exceed 50 per cent, the net return in money value by recovering the generator losses will probably not exceed $1500 a year, which with 15 per cent fixed charges warrants an investment of not over $10,000. With multiple-stage heating the saving will be less.

The comments regarding the elimination of the house turbine indicate a healthy desire for both simplicity and economy of operation. Justification for the use of house turbines, in so far as these have been justified, lies in the substantial insurance they provide against disruption of the auxiliary power supply, and this insurance has been knowingly purchased with a sacrifice in economy. Even with large and interconnected stations, if the auxiliary power is derived directly from the main bus, faults on the system which cause the frequency to drop may entail a complete shut-down of the station due to the consequent slowing down of the condenser circulating pump. Some source of independent power supply is therefore required for the essential auxiliaries and particularly for the circulating pumps on the condenser of the main unit.

A house turbine of sufficient capacity to supply power to the essential auxiliaries is an altogether adequate safeguard against complete disruption of service, and as such the reduction in the station economy due to its no-load losses are not of great concern. For a 650-kw. house turbine, the no-load losses would be approximately equivalent to 65 kw. which would seem a small expenditure for insurance in a station developing 25,000 kw. The suggested method of floating a house turbine on the line without load but available for emergency service would probably mean the use of units with at least an equal, if not greater, no-load loss. The stand-by house turbine probably would be of sufficient size to carry the entire auxiliary load or for the case covered by the paper, 1300 kw. If, due to special design, the no-load losses are made half of those of an ordinary house turbine of equal capacity, the no-load losses would be 65 kw. or the same as that of a 650-kw. house turbine. If, however, a motor generator set is used in conjunction with the stand-by turbine, the combined losses of the turbine and motor generator set would be approximately 165 kw. or 100 kw. more than those of the 650-kw. house turbine. The comparison between these methods should be based, however, not upon the no-load losses but rather upon the relative steam con-
sumptions, in the one case running a house turbine under load and in the other case running it without load.

On the basis of such a comparison the method of driving the house turbine without load would no doubt show an improvement in economy over that obtainable when the house turbine is loaded, provided a motor generator set is not used. If a motor generator set is used in connection with the stand-by turbine it is quite likely that the steam consumption will be approximately the same as that required when using a 650-kw. house turbine. The efficiency obtainable in favor of the stand-by turbine not loaded is due to more efficient utilization of steam in the main unit than in a house turbine. From the point of view of capital charges, however, the house turbine should be credited with being a power unit, equal in money value to the cost of its equivalent in main unit capacity.

Stations with four or more main units might advantageously use house generators coupled directly to the main unit shafts, interconnecting these generators. These auxiliary house generators would be preferably of sufficient capacity to carry the essential auxiliaries such as the circulating pump, and, in addition, to provide power for driving the circulating pump of a unit about to be started or having an auxiliary generator out of service. The other auxiliaries could then be carried by main unit power. Some reactance to compensate for phase shifts between the auxiliary generators would probably be required, and, as the generators will have slower speeds than house turbine generators, investment charges would have to be included in a proper comparison between the two methods of supplying auxiliary power. This arrangement from the point of view of fuel economy would no doubt have advantages over the house-turbine layout. When the station has but two or three units in operation, the auxiliary power bus will no doubt be interconnected with the main bus for starting-up purposes and to provide an additional source of power to replace that of a non-operative auxiliary generator. Otherwise heavy reactances between the auxiliary generator and oversized generators would be necessary. Interconnection with the main bus, however, is a recognized hazard.

Probably the simplest and, in some measures at least, the most economical method of eliminating the house turbine is to provide dual driven circulating pumps with the turbine running idle but available for immediate service should the auxiliary power supply be disrupted. A 300-kw. turbine would approximately meet the
power requirements of the average circulating pump of the con-
denser of a 25,000- or 30,000-kw. unit. This 300-kw. turbine would
have small no-load losses, probably not exceeding 15-kw., and
would give adequate assurance that the circulating pump would
not be out of service. Objection to the arrangement would lie in
the unsatisfactory piping layout that would probably result.

In the end it appears likely that engineering judgment will
determine the method of providing reliable auxiliary power, and
though it appears that the house turbine, as it now stands, will be
modified it is likely that, for some engineers at least, this modifi-
cation will take the form of reduced size rather than complete
elimination. The main consideration is not economy but reliability
of service. In any case, the effect on the heat balance and on the
feedwater temperature of choosing between the various methods
available for providing insurance against disrupted auxiliary power
will not be sufficient to warrant changing the feedwater tempera-
ture as already determined. Mr. Bell has very completely worked
out heat balances involving all the various methods of driving
the auxiliaries and from his results an estimate of the advantages
of the various methods may be obtained. It should be observed,
however, that Mr. Bell’s analysis is based on a house turbine cap-
able of carrying the entire auxiliary load.

As Mr. Rosencrants points out, the improvements in economy
obtainable by using economizers is reduced by stage heating.
Whether economizers can be justified or not depends largely on
the relative cost of boiler surface and economizer surface. Recent
quotations indicate that economizer surface costs less per square
foot than boiler surface and if this relationship holds when build-
ing and other construction charges are included, it is altogether
possible that small gains in commercial economy may be had by
using economizers. The value of one per cent improvement in
economy using economizers is small but would be worth obtaining
if the investment charges are not increased beyond that justifiable.
This investment would be determined by the price of coal and so
would vary with the locality.

Mr. Bell’s discussion has the added value of being based on
data obtained from manufacturers’ guarantees and from the lay-
out of an operating station. Mr. Bell, after determining the heat
consumption rate of his station from manufacturers’ guarantees,
divides this by 95 per cent to determine his station B.t.u. rate and
this largely accounts for the differences in the heat required to
produce a kw-hr. as determined by the author and Mr. Bell. The author took into account leakages and make-up losses but these were not equivalent to the use of a factor such as 95 per cent for correcting guarantees to operating conditions. The temperature of the feedwater entering the first-stage heater used by Mr. Bell was 97 per cent while that used by the writer was 75 per cent. Radiation and leakage losses in the author's study were considered as increasing with the temperature of the feedwater and this would tend to establish a lower feedwater temperature than the method used by Mr. Bell. Taking these matters into account the agreement between the feedwater temperatures determined by Mr. Bell and the author is close. The discrepancy between the temperature obtained by Mr. Bell and the author appears to be about 20 per cent more at the single-stage point than at the other points. By reference to the curves of Fig. 33 which show the summary of the

**Fig. 33 Heat Consumption per kw-hr. Neglecting Leakages from Low Pressure Steam Piping**

The full line curves 1, 2 and 3 are for single-, double-, and quadruple-stage heating respectively, without economizers. Curves 4, 5 and 6 are the same except that economizers of 21,830 sq. ft. area, equivalent to approximately 50 per cent of the boiler area, were used.

The temperature of the feedwater for best efficiency is seen to increase with the number of stages.

The curves are rather flat in the vicinity of best efficiency so that ordinary deviations from the best feedwater temperature will not affect materially the economy.

For comparative purposes the same economizer area was used for curves 5 and 6 as for curve 4, though commercial considerations would warrant reducing the economizer area for double- and quadruple-stage heating to 18,150 sq. ft.
author's data on the same basis as that prepared by Mr. Bell, except for the initial temperature of the feedwater and the method of accounting for radiation and leakage losses, it is evident that this difference is not of considerable concern due to the flatness of the heat consumption curves. For this set of curves, leakage losses from the low-pressure steam piping were ignored and the same size of economizer was used throughout.