for feedwater-heating purposes without taking a loss relative to the extraction of wet steam. This results from the fact that the feedwater temperature leaving a heater usually is determined by the pressure of the extracted steam without regard to its superheat. Even when a feedwater heater is designed to use the superheat in extracted steam to decrease the heater terminal-temperature difference, the temperature of the feedwater leaving the heater is many degrees below the temperature of the extracted steam.

Fig. 3 indicates the portion of the loss in the feedwater-heating cycle that is caused by superheat in the extracted steam. This loss was evaluated by calculating the extra "entropy of superheat" that would have been extracted if the heat of superheat were available at saturation temperature. Fig. 3 shows that the loss from this source amounts to about 0.2 per cent at the 338 F feedwater temperature leaving the No 4 heater.

To get an over-all idea of the feedwater-heating loss, the Fig. 3 curves cover the entire feedwater-temperature range from hot well to boiler saturation. To calculate the point at the 489 F boiler saturation temperature, a fictitious fifth stage of feedwater heating was assumed which delivers saturated water to the boiler at 614.7 psia pressure, heated by steam extracted from the turbine throttle. For this imaginary condition, the boiler heat output was kept unchanged from the actual 119,864,127 Btu per hr value by assuming a boiler mass flow of 126,685 lb per hr which is converted by the boiler from saturated water at 614.7 psia pressure to superheated steam at 614.7 psia pressure 825 F temperature. Of this 126,685 lb per hr boiler flow, 18,846 lb per hr is then extracted from the turbine throttle to the imaginary heater, leaving 107,839 lb per hr to flow to the turbine first stage, the same as before. With this imaginary feedwater heater in the picture, the entire feedwater-heating system from hot well to boiler saturation temperature extracts 4244.5 Btu/deg F/hr less entropy from the turbine than it adds to the feedwater. This 4244.5 Btu/deg F/hr value represents the saving which could be made if a perfect feedwater-heating system were available which delivered saturated water to the boiler.

Reheat Effects

The entropy-balance diagram is strictly correct for showing where cycle losses actually take place; however, it gives slightly conservative values when used to evaluate the gains which would be made if losses were eliminated or reduced.

Again, as an example, consider the No. 2 heater extraction pipe. Suppose that the pressure drop in this pipe were reduced and at the same time the extraction point in the turbine moved downstage slightly so as to maintain the same heater pressure. The state line enthalpy of the extracted steam would then be reduced slightly so that more extraction steam flow would be required to get the necessary heat for the feedwater heater. This increased extraction flow necessarily would cause less mass flow through the turbine low-pressure stages and hence less entropy flow to the condenser.

In evaluating the increased output that would result if the No. 2 heater extraction-pipe pressure drop were eliminated, it has been assumed that the decrease of entropy flow to the turbine exhaust would exactly equal the entropy increase in the extraction pipe. This is true except for one thing, namely, turbine stages are not 100 per cent efficient, and any reduction in the flow through a stage will also reduce the stage loss. Thus the entropy flow to condenser will always be decreased slightly more than any increase in the amount of entropy extracted.

At the turbine flange extracting to the No. 2 heater, the specific entropy is 1.7344 Btu/deg F/lb. At the turbine exhaust it is 1.8171 Btu/deg F/lb. Thus the "reheat factor" that must be applied to entropy extraction at the No. 2 heater opening is approximately 1.8171/1.7344 or 1.048. The word "approximately" is used in the foregoing expression because an entropy of liquid must be subtracted from both numbers, which will make the true reheat factor more complicated than the simple quotient shown.

If a turbine were 100 per cent efficient relative to the Rankine cycle and had an isentropic expansion line, all reheat factors would be 1.00. Since all large turbines have Rankine-cycle efficiencies between 80 and 90 per cent, the reheat factors can never get appreciably above 1.00. About 1.10 to 1.15 is the highest reheat factor that will be encountered in any normal installation; thus considering it as unity will cause no great error in perspective for a given study.

Simplified Calculation Methods

It can be shown that for a throttling process

$$ds = -0.1850 \frac{pv}{T} \, d \log_e p \quad [3]$$

Since the value of pv/T generally changes very little along a turbine-expansion line, it may be considered as approximately a constant.

Using 0.585 as an approximate value of pv/T, Equation [3] becomes

$$\Delta s = +0.108 \log_e \frac{p_1}{p_2} \quad [3a]$$

in which p_1/p_2 is the pressure ratio of a throttling process, and p_1 is the higher pressure.

Table 2 shows the actual increase of specific entropy associated with the 10 per cent pressure drop in each of the four extraction pipes of Figs. 1 and 2, together with the values calculated by Equation [3a]. Table 2 shows that Equation [3a] is quite accurate as a means of estimating entropy increases caused by throttling.

Simple procedures also can be worked up for estimating quickly the entropy increases associated with heater terminal-temperature differences, and the like, without reference to a steam table.

Conclusions

The entropy-balance method is a fundamentally sound and easily applied method of isolating and evaluating the different losses in the steam cycle of a condensing power plant. Although a sizable amount of calculation work is required to make a complete analysis of a given installation, any one question may be answered quite quickly after a heat balance is available. Preliminary estimates of many losses also may be made prior to a heat-balance calculation by making judicious assumptions as to the mass flows involved.

The entropy-balance diagram is not a substitute for other methods of analysis. It is another way of looking at a problem which should be used when it is helpful and not used when other methods seem more straightforward to apply.

Discussion

R. E. Hansen.[4] The evaluation of cycle losses by calculation of entropy generation is convenient, particularly when losses under study are too small to show up in a heat-balance computation.

The author states on the second page of the paper that the entropy rejection (a better word than "loss") of the theoretical

[4] Engineer, Elliott Company, Jeannette, Pa. Mem. ASME.

cycle is equal to the entropy received in the boiler. He does not define his theoretical cycle, but if the intention is to use as a reference the cycle defined by Selvey and Knowlton,[5] the theoretical heat rejection would be increased by an amount given in the writer's paper[6] of 1945.

It is not clear to the writer why, in Table 1, a fictitious entropy quantity representing turbine output is added to total entropy rejection in the cycle. This total does not appear to provide any additional information. Thermal efficiency of the turbine cycle may be found by dividing kw output by boiler output in kw, i.e., 11940/35122, or 34.0 per cent. Plant efficiency would be found by deducting auxiliary power before making this division, then multiplying the quotient by boiler efficiency.

The question of radiation loss is side-stepped by the assumption of zero. However, radiation can be handled quite readily in this type of computation. It may be of value to consider two types of entropy change, one by transfer, the other by generation. Entropy is increased by transfer in the boiler, and decreased by the same means wherever radiation or conduction losses occur. Entropy generation, as by pressure drop or internal heat transfer with a temperature difference, always results in an increase. Heat rate of the cycle becomes

$$\frac{3412.75\ \Delta H_B}{\Delta H_B - (\Sigma T\,\Delta S)_r - (T\,\Delta S)_c} \qquad [4]$$

where ΔH_B is the heat received in the boiler, $(\Sigma T\,\Delta S)_r$ is the summation of all entropy losses through radiation or conduction, multiplied by the temperature at which the loss occurs, and $(T\,\Delta S)_c$ is the entropy rejection in condenser multiplied by condensing temperature. Where radiation and pressure drop occur simultaneously, the entropy rejection due to radiation must be computed separately from the entropy generation due to pressure drop, in order to evaluate $(\Sigma T\,\Delta S)_r$; the net change affects $(T\,\Delta S)_c$.

The author's evaluation of the effect of an individual loss is substantially correct in so far as capacity value is concerned, unless capacity limitation lies in the generator. There is an additional value, however, due to fuel savings. For example, with a load factor such that 6000 kwhr can be generated annually from each kw of capacity, and where energy cost due to fuel averages 1 mill per kwhr the fuel saving amounts to $60 annually. Assuming 12 per cent fixed charges, capitalized value of the saving becomes $500 per kw, or $4900 in the example cited by the author. This value is obtained regardless of capacity limitation, as fuel input may be reduced at the burner. Both values must of course be corrected to a plant value, by considering the effect of boiler efficiency and auxiliary loss, which increases the value of the savings.

[5] "Theoretical Regenerative-Steam-Cycle Heat Rates," by A. M. Selvey and P. H. Knowlton, Trans. ASME, vol. 66, 1944, pp. 489–512.

[6] "Irreversibility in the Theoretical Regenerative Steam Cycle," by R. E. Hansen, Trans. ASME, vol. 67, 1945, pp. 557–560.

Whenever entropy is discussed, the question arises as to what it is and why it is useful. Most forms of energy represent the product of a potential, such as force or voltage, by a quantity, such as distance or current. In the case of heat, temperature is the potential, but there is no readily perceived quantity by which to multiply it. This difficulty is solved by inferring the existence of entropy, and computing its value mathematically. There is no need to know any more than this about its nature, and no probability that we ever will.

Author's Closure

The comments of R. E. Hansen are all quite pertinent.

The theoretical cycle the author used as a reference on page 950 is the completely reversible regenerative cycle in which the feedwater is heated by extracting heat from the turbine at the same temperature as the feedwater. This could be accomplished, theoretically, by flowing the feedwater, in a counterflow direction, through a water jacket around the turbine shell and heating the feedwater by transferring heat at constant temperature through the turbine shell. The losses relative to the foregoing cycle caused by extracting superheated steam, as discussed on page 951 of the author's paper, are the same as those discussed in the R. E. Hansen paper,[6] except that they are evaluated for an actual cycle with a finite number of feedwater heaters and a less than 100 per cent efficient turbine.

In Table 1 the fictitious entropy quantity representing turbine output was inserted for checking purposes only. The text on page 950 explains how the 217,377.5-Btu/F/hr entropy increase assigned to the boiler was calculated. By calculating a similar entropy flow value which, at condenser temperature, would be equivalent to the turbine power output, a check was established that turbine-power output plus turbine, condenser, and feedwater-heating-system losses equaled the boiler output, as expressed in entropy units.

Radiation as well as many other considerations were sidestepped to keep the paper short. For example, the entropy balance concept can be used to evaluate boiler feed-pump losses, which was also sidestepped by stating that boiler feed-pump power had been neglected. The purpose of the paper is to present a concept for isolating and evaluating different kinds of losses to which each engineer can fill in the necessary detail to adapt the concept to his own needs.

Much has already been written on the subject of steam-power-plant efficiencies and heat balances. Mr. J. K. Salisbury has given much thought to this subject, and his 1949 ASME paper[7] presents another approach to the problem of evaluating power-plant losses. The entropy balance concept is just another tool which a power-plant designer may use in his work.

[7] "Power-Plant Cycle Evaluation," by J. K. Salisbury, Trans. ASME, vol. 71, 1949, pp. 593–604.