23 Pressure at start of expansion line, Fig. (12*a*) = [(1) + 15 psi] × 0.96, psia 1455
24 Enthalpy at start of expansion line (same as 17), Btu per lb 1489.6

The expansion line can now be constructed as shown in Fig. 12 (*a*)

Example of Calculation of Expansion Lines for Reheat Turbine

Assume items (1) to (8) inclusive:

1	Throttle pressure, psig	1500
2	Throttle temperature, deg F	1000
3	High-pressure section exhaust pressure, psia	400
4	Reheat temperature, deg F	1000
5	Reheater and piping pressure loss, psi	35
6	Condenser pressure, in. Hg abs	1.0
7	Top feedwater-heater extracting from h-p section exhaust	
8	Tandem-compound triple flow arrangement like Type 6, Fig. 7, rpm	3600
9	Output of generator, kw	100000
10	Estimated throttle steam rate, from Fig. 15 = 8.0 × 0.83 = 6.6 lb per kwhr.	
11	Estimated h-p section throttle flow = (9) × (10) = 660000 lb per hr	
12	Estimated flow through reheater, 0.91 × (11) = 600000 lb per hr	
13	H-p section equivalent flow per sq in. nozzle area = 58000 lb per hr	
14	H-p section equivalent nozzle area = (11) ÷ (13)	11.4 sq in.
15	H-p section pressure ratio, $\frac{(1) + 15 \text{ psi}}{(3)} = 3.8$	
16	H-p section efficiency, from Fig. 6 at (14) and (15) = 82.2 per cent	
17	Throttle enthalpy of h-p section, from Steam Table, Btu per lb	1489.6
18	Throttle entropy, from Steam Table	1.5988
19	Theoretical h-p section end-point enthalpy at (3) and (18), from Steam Table, Btu per lb	1316.9
20	H-p section available energy X = (17) — (19) = 172.7 Btu per lb	
21	H-p section used energy Y = (20) × (16) = 142.0 Btu per lb	
22	H-p section expansion line end-point enthalpy, = (17) — (21) = 1347.6 Btu per lb	
23	Steam pressure at turbine after reheater = (3) — (5) = 365 psi	
24	Enthalpy of steam after reheater, at (4) and (23) = 1523.4 Btu per lb	
25	Entropy of steam after reheater, at (4) and (23) = 1.7729	
26	Theoretical l-p section exhaust enthalpy, at (6) and (25) = 953.0 Btu per lb	
27	L-p section available energy X = (24) — (26) = 570.4 Btu per lb	
28	Equivalent flow per sq in. at l-p section inlet from Fig. 4, at (23) and (4) = 13800 lb per hr	
29	Equivalent nozzle area for l-p section = (12) ÷ (28) = 43.5 sq in.	
30	L-p section efficiency from Fig. 7, at (29) and (8) = 88.7 × .998 = 88.5 per cent	
31	L-p section initial superheat, from (4) and (23) = 564 F	
32	Correction factor from Fig. 8, at (31) and (23) = 1.028	
33	L-p section expansion-line efficiency = (30) × (32) = 91.0 per cent	
34	L-p section used energy Y = (27) × (33) = 519.3 Btu per lb	
35	L-p section expansion-line end point = (24) — (34) = 1004.1 Btu per lb	
36	Pressure at start of l-p section expansion line = 0.98 × (23) = 358 psia	
37	Enthalpy at start of expansion line (same as 24) = 1523.4 Btu per lb	

The expansion lines can now be constructed as shown in Fig. 12(*b*).

Discussion

B. C. Mallory[9] and W. F. Allen, Jr.[10] The data and procedures presented in this paper will be of great assistance to steam-power-plant designers. The earlier paper by Warren and Knowlton[11] was the initial step in bridging the gap between the turbine designer and the power-plant designer as it made available to the plant designer hitherto restricted basic information on turbine performance, and outlined a systematic procedure for utilization of this information in predicting performance.

The present paper, which confirms and supplements data and procedures of the earlier paper, is of particular interest because it includes the comparison of results of turbine tests and guarantee values of the past decade for unit sizes up to 150 mw, for initial steam conditions as high as 2350 psig, 1050 F, and for reheat units. The method given for predicting over-all performance of reheat machines, which was not included in the earlier paper, will be valuable, and the use of curves of internal efficiencies versus equivalent nozzle areas, instead of over-all engine efficiencies against rated load megawatts as given in the earlier paper, will simplify the performance calculations.

The procedure in the paper involves the use of approximate steam rates to obtain equivalent nozzle areas and in addition, for reheat units, a fixed factor of either 0.86 or 0.91 in conjunction with high-pressure turbine-throttle flow to obtain equivalent nozzle area for the low-pressure unit. After construction of the expansion lines, the estimated throttle flow may be used for the heat-balance calculations. The authors perhaps may have considered it too obvious to mention, but it may be necessary to use a revised throttle flow and repeat the entire procedure, including construction of new expansion lines, in order to obtain the desired megawatt output. For reheat units also, it is not likely that the flow to the low-pressure turbine obtained from the heat-balance calculation will coincide exactly with the 86 or 91 per cent of the high-pressure turbine-throttle flow used for estimating low-pressure turbine internal efficiencies. It would be of value if the authors would comment on whether it is worth while to recompute low-pressure turbine internal efficiency for the variation in flow to the reheater or whether the accuracy of the data is such that this refinement is unnecessary.

No new data are presented for performance at partial loads. It may be necessary to investigate partial-load performance when making complete studies for size of unit, initial steam conditions, and feed-heating arrangements. The authors state that curve A from the Warren-Knowlton paper[11] applies fairly well to modern units, but curves B and C do not. In the paper, each curve covers a range of throttle volume flows in cubic feet per second. It is not clear if curve A may be used over the entire range, or if it is valid only for the range previously defined. If curve A covers only a restricted range, information for other throttle flows would be helpful.

Authors' Closure

Messrs. Mallory and Allen have raised questions regarding the detailed procedure for using the data in the paper. The first of these questions has to do with the possible necessity for making a repeat calculation in case it proves that the first set of assumptions do not result in the desired output.

It is entirely possible, of course, to make a wrong estimate of the required throttle flow for a given power output from the generator. The information given in Fig. 15 is the best simple

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[11] Authors' reference 3.

guide for estimating purposes that the authors were able to devise. It is to be expected that particular cases, when worked out in detail, will be a few per cent different from the data given in Fig. 15. Depending upon result desired it may or may not be advisable to repeat the detailed calculation in order to achieve a closer approximation. It seems unlikely that the original estimate of flows through the various sections of the turbine will be far enough off to require a new calculation of the turbine internal efficiencies and a redrawing of the expansion lines.

With respect to the calculation of performance at partial load, the authors have not made a complete study of this as would be required if new data were to be published. The curve *A* from Fig. 7, reference 3, mentioned in the paper and also by the discussers, refers to the performance of a turbine without any by-passing; this is typical of the large high-pressure, high-temperature machines being designed at present by the authors' company.

In the absence of a careful study of means for presenting in simple fashion the complicated design calculations, the authors believe it best not to attempt to compare machines at partial load by the methods outlined in the paper. It may be possible in the future to present some simplified data describing part-load performance more exactly.

The authors wish to point out that in the present Fig. 10 an exhaust loss curve is shown for a 39.8 sq ft annulus area at 3600 rpm and reference is also made to this annulus area on Fig. 13. Since the paper was written, it has been decided that instead of the 39.8 sq ft annulus, a somewhat larger one of 41 sq ft at 3600 rpm would be offered instead. The curve No. 7 in Fig. 10 will apply to this 41 sq ft annulus if the equivalent annulus velocity is figured for the 41 sq ft annulus instead of for 39.8 sq ft.