Effect of Reheating Cycle on Efficiency of Rankine Cycle and its Practical Significance

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Abstract. The Rankine cycle performance and efficiency can be improved with the reheat cycle as well as by regenerative cycle. It can be improved by increasing the thermodynamic properties of the steam entering into the turbine. At the initial stage, increased pressure results in increased expansion ratio. But as gradually progressing almost at the end of expansion wet steam is observed. This observation is due to increased moisture content of the steam. This is not desirable because it leads to wearing away of turbine blades and its loss which finally results in reduction of efficiency of nozzle and blade. Reheating cycle play a role in solving these problems. This paper discusses in briefly about methodology, basic principles in reheating cycle and comparison of heat rate and steam flow is made between ideal cycle and practical cycle.

INTRODUCTION

Steam Pressure and Temperature has a continual increase in turbine-inlet because of the improved thermal efficiency resulting in the greater capacity from a given physical size of the machine. On large units, 100,000 kw and up, pressures of 1,800 to 2,400 psi are usual, along with temperatures of 1000 to 1050F, with 2,400 psi-1100F in use in one case. A turbine for 4,500 psi-115F with two reheating, first to 1050F, second to 1000F, is currently in operation; another for 5,000 psi-1200F with double reheat to 1050F is being designed. Increases in temperature above 1200F apparently must await the development of higher-strength materials of reasonable cost and availability.

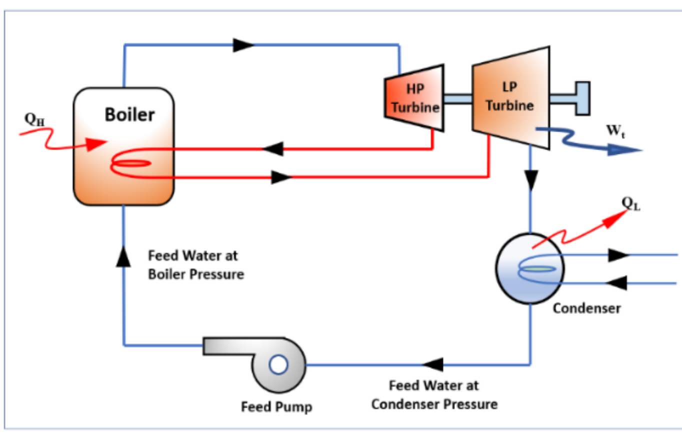

FIGURE.1 Reheat Cycle

On the other hand, some large steam plants being designed for nuclear fuel will use saturated steam without superheating or reheating because of temperature limitations in the reactors. In order to limit the moisture content of the steam in the lower pressure stages, it will be necessary to use moisture extraction devices in the steam path or in the crossover connections between turbine casings, or both; otherwise, excessive erosion could be expected in the low-pressure stages. With such devices, however, it appears entirely possible to reduce the moisture content to values not greater than have been successfully handled in the past. In reheat cycle, the steam is extracted from a suitable point in the turbine and is reheated with the help of flue gases in the boiler furnace as shown in Fig.1

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The main reason of reheating is to change thermodynamic properties of the exhaust steam of initial turbine. As stages of the turbine passes moisture content of steam decreases resulting in dryness. It has to be remembered that the dryness fraction of steam should not decrease below the value of 0.88. In addition, there is a decreased specific steam consumption. The thermal efficiency is enhanced which depends upon the ratio between reheat pressure of steam and original pressure of steam which is taken as 0.4.

In this cycle initially there is a partial expansion steam then it has to undergo reheating cycle and finally returns to the lower pressure section of the turbine. Reheating results in approximately 5% decrease of the turbine heat rate. The precise improvement is multi-factorial dependent, thermodynamic gain results in about 40% improvement and enhanced thermal efficiency leads to 60% improvement. Reheating can be done any number of times theoretically, but because of economic loss and complexity of plant cycle number of reheating cycles are limited practically as shown in the Fig.2.

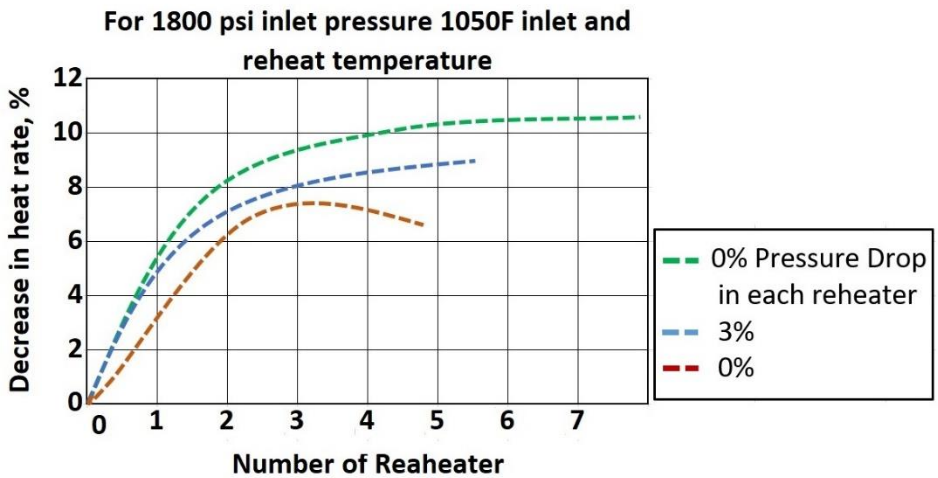


FIGURE2 Approximate gains due to reheating

At the similar initial thermodynamic conditions of steam, by reheating upto the initial temperature, there is a 17% reduction of throttle steam-flow rate and 13% reduction of condenser steam-flow rates when compared with no reheat cycle. Practically, the reheat pressure is 20 to 30% times initial pressure, Because the reheater and piping contain a relatively large volume of steam, it is necessary to provide quick-closing valves like intercept valves to shut off this steam from the turbine in the event of sudden loss of load. These valves are operated by the turbine speed governor system. In extreme cases of a combination of large stored volume and relatively low turbine and generator rotor inertia, another set of valves may be used to bypass this volume of the steams to the condensers.

LITERATURE SURVEY

Steam power plant exergy and energy analysis has been done. Analysis carried out with regenerative Rankine cycle with different heaters and energy and exergies are analyzed with the help of HYSYS[1]. Thermodynamic analysis has been carried out with the combination of reheat and regeneration of Rankine cycle and it is observed that due to the reheat and regeneration Thermal efficiency of the plant is increased [2]. Steam power plant reheat and regeneration cycle comparison has been done by using the c program thermal efficiencies are calculated by using different process parameters after that it is observed that regenerative cycle is more efficient then reheat cycle [3]. To improve the thermal efficiency of regenerative cycle are used i.e. heat rejected by the condenser is gain to heat the boiler water. This process is achieved by using Absorption Heat Transforms Technique [4]. Feed water heaters are used in regenerative Rankine cycle to improve the thermal efficiency and it is observed that based on the no. of feed water heaters the efficiency is changing [5]. For reducing the energy requirement of steam turbine water rate of turbine is improved. Energy consumption is a key parameter for development so prevention of energy has been done with new technologies [6]. In steam turbine plant reheat and regenerative cycles pressure levels are enhanced and quality of the steam generation irreversibility has been analyzed [7]. Pollution factors of environment has been analyzed and it is observed that conventional Rankine cycle consumes more fuel in place of fuel fossil fuel are used and these fossil fuels are used to reduce the pollution by using ORCS method [8]. Different reheater types of Rankine cycle has been observed and for estimating the properties of steam MAT LAB program used and it is mainly focused at entry and exit of the turbine temperatures and pressures. Exergy losses also analyzed [9]. Thermodynamic analysis has been done for different cycles with various organic fluids and it is investigated that out of all different methods reheat combined with regeneration plant is more economical [10].

METHODOLOGY

An advantage of multistage turbines is that the energy losses in initial stages can be partially recovered for producing useful work in the subsequent stages. The energy losses in a stage are converted to heat and thus increase the steam enthalpy behind the stage. This results in an increase of the steam temperature in the region of superheated steam or in an increase of dryness fraction of steam in the region of wet steam. Because of an increase of the temperature of dryness fraction of steam, the heat drops of a stage increases compared with the heat drop of that stage calculated along the main isentropic of ideal expansion of steam. As may be seen from Fig.3.

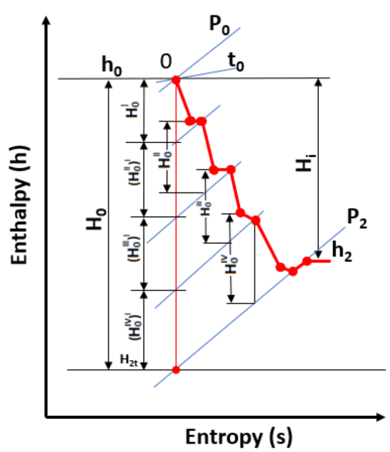


FIGURE3 shows h-s diagram of steam expansion in a multi-stage turbine

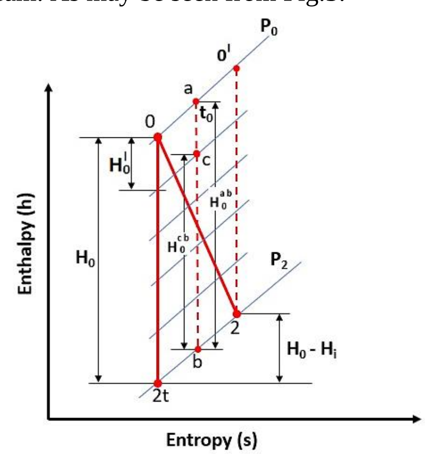


FIGURE 4 shows h-s diagram for deriving the formula of reheat factor

$$H_0^{II} > (H^{II})', H_0^{III} > (H^{III})' \text{etc.}$$

This gain in heat drops is known to be caused by divergence of isobars in the h-s diagram towards increasing entropy. Thus, if the heat drops of turbine stages are stages

$$H_0^{I}, H_0^{II}, H_0^{III}$$

are added, their sum turns out to be greater than the heat drops H0 of the turbine along the main isentropic, i.e.

$$\sum_{i=1}^{z} H_0^{i} - H_0 = Q \tag{1}$$

Where Q is the recovered heat of the energy losses in the stages, which increases the available energy of the stages of a multistage turbine compared with that of a single-stage turbine. Calculate the efficiency of a multistage turbine assuming that it consists of stages having the same efficiency(η) represents in Fig.5

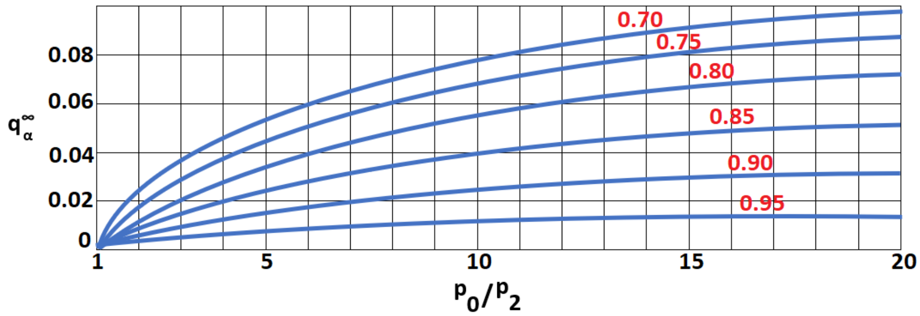


Fig.5 Reheat factor of a turbine with infinite number of stages and k=1.3.

$$\boldsymbol{\eta_{ri} = \frac{H_i}{H_0} = \frac{\sum_{i=1}^{n} H_i^n}{H_0} = \frac{\sum_{i=1}^{n} H_0^n \eta_{ri}^{si}}{H_0} = \eta_{ri}^{si}\frac{H_0+Q}{H_0}} \text{(2)}$$

(or)

$$\boldsymbol{\eta_{ri} = \eta_{ri}^{si}(1+q_t)}$$

Where $q_t = \frac{Q}{H_0}$ is the heating factor follows from above formula, the effect of reheating (heat recovery) increases the internal relative efficiency of a turbine compared with that of a single stage. This increase is determined by the reheat factor which may vary between 0.02 and 0.10 depending on H0, the number of stages, and efficiency. From a turbine with the known h-s diagram of a steam expansion and a given number of stages, the reheat factor can be easily found by adding the heat drops of a turbine stages by formulae (1) and (2) estimation can be done by a number of methods for calculating the refactor.

One of them is based on determination of the recovered heat in the h-s diagram represents in Fig.4. Assuming that the actual process in a turbine with an infinite number of stages follows line 0-2, the sum of heat drops of all stages should be somewhere between the heat drops determined by sections 0-2t and 0'-2. The heat drop determined by section 0'-2 corresponds to a hypothetical case when the temperature of steam before the first stage increases due to recovery of the energy losses in all subsequent stages, Since the total energy loss increases from one stage to another practically linearly, the real sum of heat drops is determined by the isentropic a-b which passes through the midpoints of sections 2t-2 and 0-0'. Thus, the reheat factor for a turbine with infinite number of stages can be found by the formula

$$\boldsymbol{q_t^\infty = \frac{H_0^{ab} - H_0}{H_0}} \text{(3)}$$

Let us use the equation of available heat drop:

$$\boldsymbol{H_0 = \frac{k}{k-1} R T_0 \left[1 - \left(\frac{p_2}{p_0}\right)^{k-\frac{1}{k}}\right]} \text{(4)}$$

$$\boldsymbol{= \frac{k}{k-1} p_0 v_0 \left[1 - \left(\frac{p_2}{p_0}\right)^{k-\frac{1}{k}}\right]}$$

Where $v_0 = v^n x$ (for wet steam).

The recovered heat Q∞=Hab0-Hi =(1-ηri) can be found as follows

$$\boldsymbol{Q_\infty = \frac{k}{k-1} R(T_{0a} - T_0)\left[1 - \left(\frac{p_2}{p_0}\right)^{k-\frac{1}{k}}\right]}$$

Since $\boldsymbol{T_{0a} = T_0 + \frac{(H_0 - H_1)}{2_{c_p}}}$ and $\boldsymbol{H - H_i = (1-\eta_{ri})\, x\, H_0}$

Finally $$q_t^{\infty} = \frac{A}{T_0}(1 - \eta_{ri})H_0 \quad (5)$$

Where A is a constant factor depending on the isentropic exponent k and gas constant R. Formula (5) for an infinite number of stages is convenient for calculations on the h-s diagram of steam expansion when k and R are variables. For cases when the equations for ideal gases are applicable, this formula can be written in a different form Fig.5

$$q_t^{\infty} = A'\left(1 - \eta_{ri}^{si}\right)\left[1 - \left(\frac{p_2}{p_0}\right)^{k - \frac{1}{k}}\right] \quad \textbf{(6)}$$

With a finite number of turbine stages, the reheat factor decreases, since the heat drop of the first stage is not increased due to heat recovery. Therefore, with a finite number of stages, the recovered heat Q diminishes to a value which can be found by reference to Fig.4

$$Q = H_0^{cb} - (H_0 - H_0')\textbf{(7)}$$

If the heat drop is the same in all stages, we have $H_0^{cb} = H_0^{ab}\frac{z-1}{z}$ and $H_0 - H_0' = H_0\frac{z-1}{z}$and the reheat factor for a finite number of stages can be represented in the form:

$$q_t = \frac{(H_0^b - H_0^1)^{z - \frac{1}{2}}}{H_0^1} = q_1^{\infty}\frac{z-1}{z} \quad \textbf{(8)}$$

In practical calculations the formula is usually modified:

$$q_t = k_t(1 - \eta_{ri})H_0\frac{z-1}{z} \quad \textbf{(9)}$$

Where the coefficient k, is taken to be:

$k_t = 4.8x10^{-4}$ for a group of stages operating in the superheated steam region; $k_t = 2.8x10^{-4}$ for group of stages in the region of wet steam and $k_t = 3.2- 4.3x10^{-4}$ for group of stages which operate partially on superheated and partially on wet steam. The indicated value of k_t relate to H_0 measured in kilojoules per kilogram.

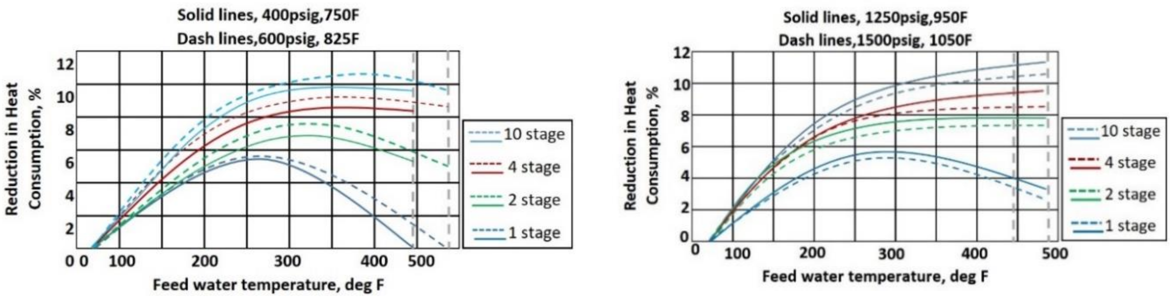
RESULT AND DISCUSSION

In high capacity plants with power of 50000kw units and above and high-pressure steam as high as100kg/cm^2, reheating cycle is most preferred. Comparison is made between ideal cycle and practical cycle. In the ideal cycle mixing heaters are used with discharged from each pumped back. Whereas closed heaters are used in practical cycle which arranged as upper one is cascaded drained and the lowest heater is pumped drained.

Table.1 Comparison of efficiency for simple and reheating cycles.
(Reheat when steam is dry saturated exit pressure 0.04bar)

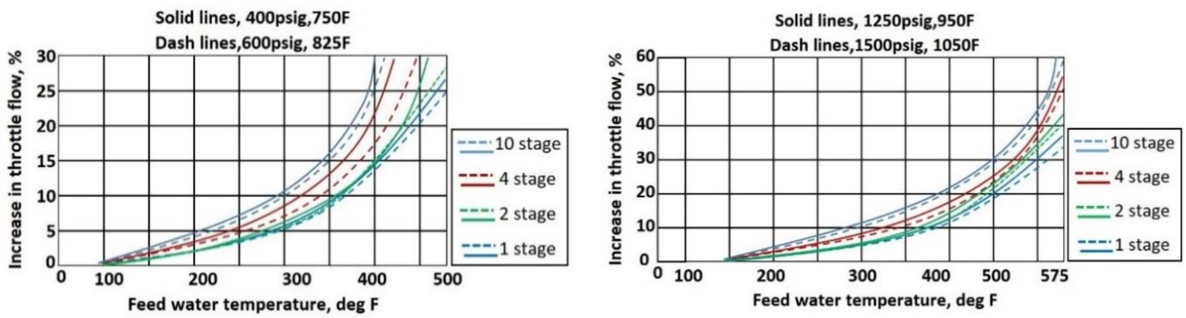
	Super-heated to 600ºC		**Super-heated and reheat to 600ºC**	
	160bar	350bar	160bar	350bar
Efficiency	45.3	46.8	46.8	48.5
Dryness at exit	77.2	70.8	95.4	87.1

The difference between these two cycles is usually about 1½ %. The heat rate reduction and rate of increase steam flow of both cycles are represented in Fig.7(a) & (b) and Fig.8 (a) & (b).



(a) (b)

FIGURE.6 (a) & (b) Heat ratereduction in by use of ideal cycle, with 1 in. Hg back pressure.



(a) (b)

FIGURE7 (a) & (b) Rate of increased steam flow necessary to maintain the same power output when using the ideal cycle, with 1 in. Hg back Pressure.

No allowance is made for the loss due to the feed-pump power which is about two-thirds the percent which the feed-pump power is of the net output because of the partial regain of this energy in the system. Fig.7 shows the reduction in the heat rate for various initial pressures and temperatures, and for 1 in. Hg back pressure, for various feed-water temperatures.

CONCLUSIONS

- Reheating cycle plays a vital role in decreasing the moisture content of steam and increases dryness fraction, thereby prevents erosion and increases efficiency blade as well as boiler efficiency indirectly in the thermal power station.
- Reheating cycle is used to improve the overall cycle efficiency and to reduce the size of the plant
- Reheating can be done any number of times theoretically, but because of economic loss and complexity of plant cycle number of reheating cycles are limited practically.

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