# STEAM PARAMETERS FOR ECONOMIC CO-GENERATION PLANT

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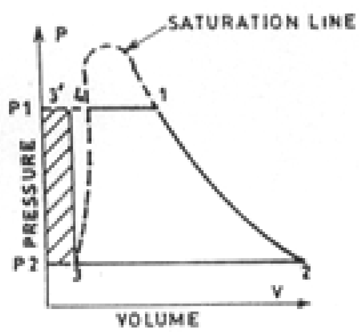
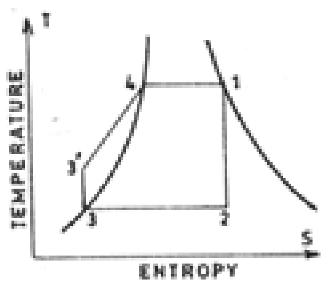
***Abstract***

*Profit margin of the sugar factories are squeezing because of price of the sugar in domestic and international market is less and cost of the sugar cane is increasing. Survival of sugar industry without co-generation is very difficult. Optimum co-generation capacity depends upon the crushing capacity and steam demand of the plant. Steam parameter plays vital role for selection of the Turbine and Boilers. In this paper author's focus is on the topic of selection of the steam parameter and thermal cycle for the most economic co-generation plant.*

## Introduction

In Steam power plant steam is used as working substance to convert heat into work. Power plant is working on Rankine cycle. Rankine power cycle is sketched on P-V and T-S co-ordinate below.




1-2 The steam is fed to the engine or turbine where it expands isentropically to the lower pressure P2.

2-3 At point2, the wet steam enters the condenser where its latent heat is removed by circulating water. The wet steam is condensed completely at pressure P2. Horizontal line 2-3 represents the process of condensation.

3-3 The liquid as it enters the pump at state point 3 is at the condenser pressure P2 and corresponding saturation temperature. The liquid is entropically raised in pressure to that boiler pressureP1. The process 3-3' during which the water is being fed to the boiler is almost isochonic due to low compressibility of the liquid. At state point 3, the fluid has a pressure equal that of the pressure P1 but its temperature is lower than the saturation temperature for the boiler pressure P1 and according the liquid is known as sub-cooled or compressed.

3-4 The temperature of the feed water is raised to that of saturation temperature corresponding to boiler pressure by supplying heat energy. The operation 3'-4 can by heat take place in any of the following devices.

1. Economiser, which may form an integral part of the boiler.
2. Special feed heater between feed pump and boiler

$$\text{Thermal Efficiency of Rankine cycle} = \frac{Q1 - Q2}{Q1}$$

Where Q1- Q2 = W is the useful work output

Q1= Heat supplied to working fluid in the boiler

Q2= Heat rejected by the fluid in the condenser

Now for Rankine cycle

Q1= H1-hw3', Q2=H2-hw3

Rankine thermal efficiency is given by

$$\text{Efficiency} = \frac{(H1\text{-}hw3') - (H2\text{-}hw2)}{H1\text{-}hw3'}$$

$$= \frac{(H1\text{-}H2) - (hw3'\text{-}hw3)}{(H1\text{-}hw3)-(hw3'\text{-}hw3)}$$

Pump work hw3'- hw3 is very small can be neglected.

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Then thermal efficiency of Rankine cycle

$$= \frac{H1-H2}{H1-hw3} = \frac{H1-H2}{H1-hw2}$$

This is because hw3= hw2 as each equal to the sensible heat corresponding to condenser pressure.

Feed pump work (hw3'-hw3) can be evaluated by using expression

$$\frac{v\,(p3'-p3)}{J}$$

**Selection of the steam Parameters**

**Factors affecting selection of Steam temperature**

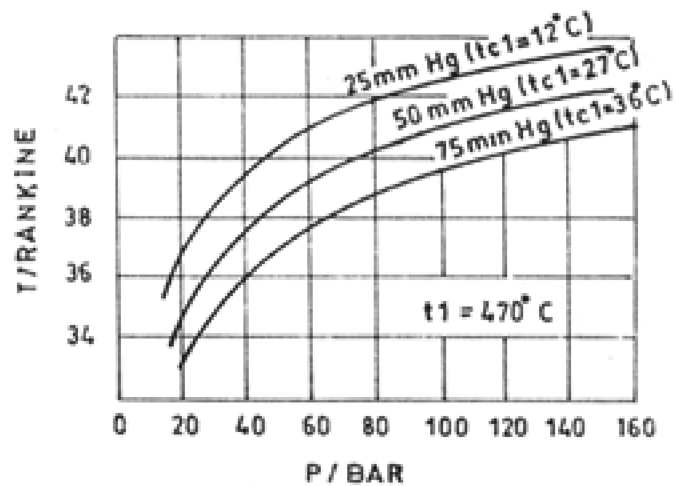
Superheating has an advantage of using the higher operating temperature without increasing the maximum pressure of the cycle and keeping the condition of the exhaust steam within limits. Moreover with superheating there is saving of fuel. First 40°C superheat gives a saving of order 6-7%, for next 40° C superheat, gain is only 4-5%. This gain is due to more heat content of the superheated steam and consequently less steam consumption for a given output. In case of condensing steam turbine there is 10-15 % saving steam consumption for about 40°C superheat. Raising the inlet steam temperature also reduces the wetness of the steam in the later stage of the turbine and improve turbine internal efficiency.

The Maximum temperature of steam that can be used is fixed from metallurgical consideration i.e. materials used for the manufacturing of the components which are subjected to the high pressure, and high temperature steam like superheater, valves, pipelines, inlet stage of the turbine and so on. It is called metallurgical limit. The ultimate strength of unalloyed steel drops by about 30% as the steam temperature is raised from 400 to 500°C. Alloying with chromium and molybdenum and eventually, the use of austenitic instead of ferric steels increases the strength at high temperatures. Steam power plants generally limits steam temperature to 538°C and in few cases to 565°C. By considering the project cost of cogeneration plant, inlet steam temperature of the turbine is limited to 480-490°C is most economically viable.

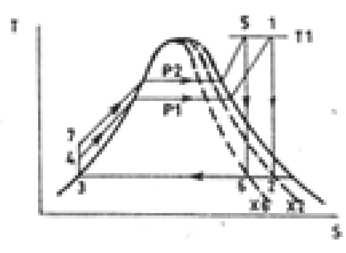
**Factors affecting selection of Condenser Pressure (Turbine Exhaust)**

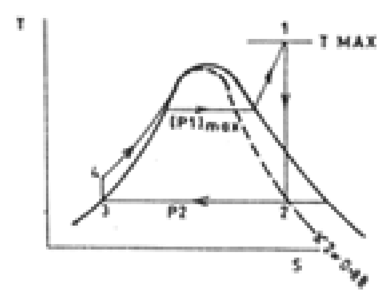
Considerable improvement in cycle efficiency is observed /noticed with the decrease of condenser pressure. Such a decrease mainly depends on the availability of cooling water temperature and thus climatic condition. A lower cooling water temperature gives lower condenser pressure (Higher vacuum). For available cooling water temp of 30-32°C, vacuum can be maintained 0.12 to 0.18 kg/cm².

Effect of inlet steam pressure (P1) and condenser pressure on Rankine efficiency with constant inlet steam temperature of 470°C shown in the below.

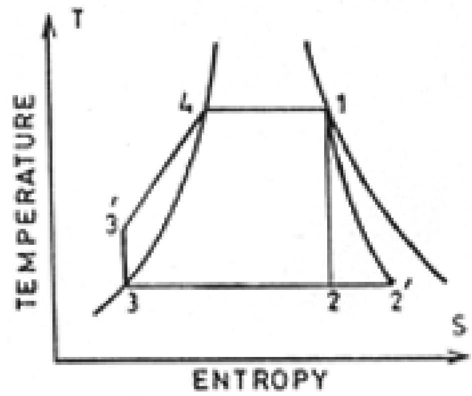



**Factors affecting selection of steam pressure:**






When the turbine inlet pressure increases from p1 to p2, the ideal expansion line of steam shift to the left and moisture content of the turbine exhaust increases. If the moisture content of the steam in the later stage of the turbine is high, the entrained water particle along with high velocity strike the blade and erode their blades in the later stages of the turbine. The maximum moisture content of the turbine exhaust is not allowed to exceed 12% or dryness factor quality of steam should not be less than 88%.




**Case study**

Consideration metallurgical limit maximum inlet steam temperature of the turbine is 480-485°C, Steam outlet from boiler is 490-495°C.

Availability of cooling water in South India is 30-32°C so exhaust pressure of the turbine is 0.14 to 0.18 Kg/Cm².

For selection of the pressure, author is limiting to the following three options for easy understanding.

1. Turbine inlet pressure 66 Kg/Cm² and temperature 480°C
2. Turbine inlet pressure 45 Kg/Cm² and temperature 480°C
3. Turbine inlet pressure 82 Kg/Cm² and temperature 510°C

Following are the assumptions for calculation purpose

1) Adiabatic Expansion Efficiency (Turbine internal efficiency) – 85 %
2) Dryness factor of steam more than 88%
3) Condenser Pressure is 0.18 Kg/Cm²
4) Boiler efficiency (Thermal) – 71 % on GCV of bagasse
5) GCV of fuel (Bagasse)- 2276 Kcal/kg
6) Capacity of Boiler- 80.0 MT/hr

**Option I ( 66 kg/cm² and temp 480°C)**

Enthalpy of steam at 66 kg/cm² and temp 480°C, H1 = 807 Kcal/Kg

Enthalpy of steam at 0.18Kg/cm², H2 = 532 Kcal/Kg

Enthalpy drop H1-H2 = 807-532

Actual enthalpy drop adiabatic efficiency (H1-H2) =0.85(807-532) =234 Kcal/Kg

Enthalpy of steam at H2' =807-234= 573 Kcal/Kg

H2' = hw+XL

573 = 57.4 +X 564.6

X = 0.913 i.e dryness fraction is within limit.

$$\text{Rankine cycle efficienty} = \frac{\text{Turbine work}}{\text{Heat supplied}} = \frac{807-573}{807-57.4} = 0.312 \text{ i.e } 31.2\ \%$$

Steam generated by one tonne of bagasse =

$$\frac{0.71 X 2276}{794.6} = 2.15 \text{ MT}$$

Quantity of bagasse requied for 80 MT boiler

$$= \frac{80}{2.15} = 37.2 \text{ MT}$$

**Specific steam consumption of turbine**

$$= \frac{860}{(H1\text{-}H2)} \text{ X adibatic efficiency of turbine X}$$

Mech efficiency of turbineX efficiency alt generator X efficiency of reduction gearbox

Adiabatic efficiency of turbine= 0.85

Mechanical efficiency of the turbine= 0.985

Efficiency of the alternator generator= 0.94-0.985

Efficiency of reduction gearbox = 0.97-0.985

Specific steam consumption $= \frac{860}{234 \times 0.93} = 4.03$ kg/kw-hr

Power generated by 80 Mt of steam $= \frac{80}{4.03} = 19.8$ MW

**Option II**

Steam parameter 45 Kg/Cm² and Temperature 480°C

Enthalpy of steam at 45 kg/cm² and temp 480°C, H1

= 819 Kcal/Kg

Enthalpy of steam at 0.18Kg/cm², H2

= 550 Kcal/Kg

Enthalpy drop H1-H2 = 819-550

Actual enthalpy drop adiabatic efficiency (H1-H2)

= 0.85(819-550)

= 228.65 Kcal/Kg

Enthalpy of steam at H2' =819-228.65

= 590 Kcal/Kg

H2' = hw+XL

590= 57.4 +X 564.6

X = 0.94 i.e dryness fraction is within limit.

Rankine cycle efficiency $= \frac{\text{Turbine work}}{\text{Heat supplied}}$

$$= \frac{228.65}{819-57.4}$$

= 0.30 %

Steam generated by one tonne of bagasse

$$= \frac{0.71 \times 2276}{761.5} = 2.12 \text{ M.T}$$

Quantity of bagasse requied for 80 MT boiler

$$= \frac{80}{2.12} = 37.7 \text{ MT}$$

Specific steam consumption of turbine

$$= \frac{860}{(H1\text{-}H2)} \text{ X}$$

adiabatic efficiency of turbine X Mech efficiency of turbine X efficiency of alternator generator X efficiency of reduction gearbox

Specific steam consumption$= \frac{860}{228 \times 0.93} = 4.14$ kg/kw-hr

Power generated by 80 Mt of steam

$$= \frac{80}{4.14} = 19.3 \text{ MW}$$

**Option III**

Steam parameter 82 Kg/Cm² and temperature 505°C

Enthalpy of steam at 82 kg/cm² and temp 480°C, H1

= 812 Kcal/Kg

Enthalpy of steam at 0.18Kg/cm², H2

= 514 Kcal/Kg

Enthalpy drop H1-H2 = 812-514

Actual enthalpy drop adiabatic efficiency (H1-H2)

=0.85(812-514)

=253 Kcal/Kg

Enthalpy of steam at H2' =812-253.5= 559 Kcal/Kg

H2' = hw+XL

559 = 57.4 +X 564.6

X = 0.88 i.e dryness fraction is within limit.

Rankine cycle efficiency $= \frac{\text{Turbine work}}{\text{Heat supplied}}$

$$= \frac{253.3}{812-57.4}$$

$= 0.33\ \%$

Steam generated by one tonne of bagasse

$$= \frac{0.71 \times 2276}{754.6} = 2.15\text{M.T}$$

Quantity of bagasse required for 80 MT boiler

$$= \frac{80}{2.15} = 37.2 \text{ MT}$$

Specific steam consumption of turbine

$$= \frac{860}{(H1\text{-}H2) \times \text{adiabatic efficiency of turbine} \times \text{Mech efficiency of turbine} \times \text{efficiency of alternator generator} \times \text{efficiency of reduction gearbox}}$$

$$\text{Specific steam consumption} = \frac{860}{253 \times 0.93} = 3.72 \text{ kg/kw-hr}$$

$$\text{Power generated by 80 Mt of steam} = \frac{80}{3.72} = 21.5 \text{ MW}$$

**Conclusion**

| Option | Boiler Pressure Kg/cm² (g) | Cycle Efficiency % | Steam fuel ratio | Specific steam Consumption Kg/kw-hr | Power generated in M.W for 80MT of steam |
|---|---|---|---|---|---|
| Case1 | 66 | 31.2 | 2.15 | 4.03 | 19.8 |
| Case2 | 45 | 30 | 2.12 | 4.14 | 19.3 |
| Case3 | 32 | 33 | 2.28 | 3.72 | 21.5 |

Capital Cost of boiler and turbine for option 1 is 40-50% more as compared to option 2 and for option 3 capital cost is 70 % more than option 2. By comparing the pay back period and return on the assets option 2 is most economically viable.

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