Analysis of Regenerative System in Steam Power Plant

S. Naga Raju, Dr. N. Hari Babu, P. Dilip Kumar

***Abstract*— The development of any country directly relates on capital energy consumption. The demand for power generation on the large scale is increasing day by day. Owing to their major contribution towards power production, thermal power plants have a vital role to play in the development of nation. Due to the scarcity of power, every power plant needs to be operated at maximum level of efficiency. In case of thermal power plants this applies equally to all its auxiliaries.**

The feed water heaters form a part of the regenerative system to increase the overall thermal efficiency of the plant. In the operation and maintenance of a power plant the feed water heaters are virtually neglected compared with other components. To realize the effect of feed water heating and an attempt is made in this project work to find the improvement in cycle efficiency due to FWH. 2 units of different configuration with same capacity (210MW) are considered for the analysis. In general an improvement of 5-6% in efficiency is possible. The actual gain in efficiency depends on the number of FWH's and the performance of heater.

The improvement in efficiency of the 2 units considered for the analysis indicated the above factor. Hence an effective analysis is required to the number of FWH to be used. Also the maintenance of heater in the form of removal of scale (or) corrosion plays important role in the overall performance of the plant.

***Index Terms*— performance, Feed water heater, power plants**

I. INTRODUCTION

A. RANKINE CYCLE

For each process in the vapour power cycle, it is possible to assume a hypothetical or ideal process which represents the basic intended operation and involves no extraneous effects. For the steam boiler, this would be a reversible constant pressure heating process of water to form steam, for the turbine the ideal process would be a reversible constant pressure heat rejection as the steam condenses till it becomes saturated liquid, and for the pump, the ideal process would be the reversible adiabatic compression of this liquid ending at the initial pressure. When all these four processes are ideal, the cycle is an ideal cycle, called a Rankine cycle.

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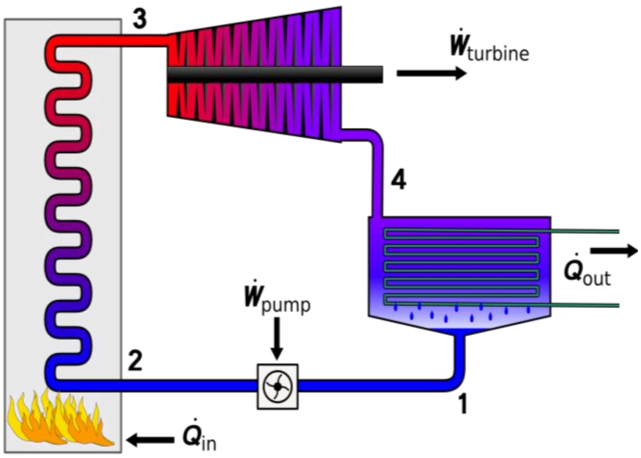

Fig.1.1.Flow diagram of Rankine cycle.

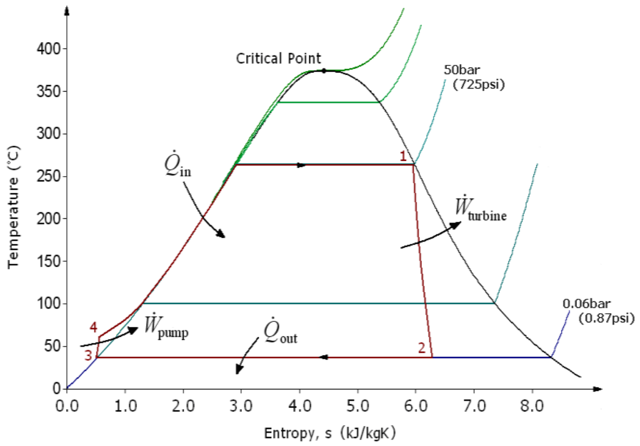

Fig.1.2. Graphical representation of Efficiency of Rankine cycle

For purpose of analysis, the Rankine cycle is assumed to be carried out in a steady flow operation. Applying the steady flow energy equation to each of the process on the basis of unit mass of fluid, and neglecting changes in kinetic and potential energy, the work and heat quantities can be evaluated in terms of the properties of the fluid.

The pump consists of liquid water which is incompressible, i.e. its density or specific volume under goes little change with an increase in pressure. Usually, the pump work is quite small compared to the turbine wok and is sometimes neglected. Then $h_4 = h_3$ and the cycle efficiency approximately becomes.

$$\eta = \frac{h_1 - h_2}{h_1 - h_4}$$

The efficiency of the Rankine cycle is presented graphically in the T-s plot in figure. 1.2.

B. REGENERATIVE CYCLE:

In the regenerative cycle the dry saturated steam from the boiler enters the turbine at a higher temperature and then expands entropic ally to a lower temperature in the same way as that of Rankine and Carnot cycle. Now the condensate from condenser is pumped back and circulated around the casing, in the direction opposite to the steam flow in turbine. The steam is thus heated before entering into the boiler. Such a system of heating is known as regenerative heating, as the steam is used to heat the steam itself.

The ideal regenerative cycle has efficiency equal to that efficiency of Carnot cycle with the same heat supply and heat rejection temperatures. It is impossible to achieve the cycle, in actual practice due to not possible to affect the necessary heat transfer from the steam in the turbine to the liquid feed water and moisture content of the steam leaving the turbine is considerably increased as a result of the heat transfer.

In order to increase the mean temperature of heat addition (T_{m1}), attention was so far confined to increasing the amount of heat supplied at high temperatures, such as increasing superheat, using temperature of steam, and using reheat. The mean temperature of heat addition can also be increased by decreasing the amount of heat added at low temperatures. In a saturated steam Rankine cycle shown in figure 1.3. Considerable part of the total heat supplied is in the liquid phase when heating up water from 4 to 4' at a temperature lower than T_1, the maximum temperature of the cycle.

For maximum efficiency, all heat should be supplied at T_1 and feed water should enter the boiler at state 4'. This may be accomplished in what is known as an ideal regenerative cycle.

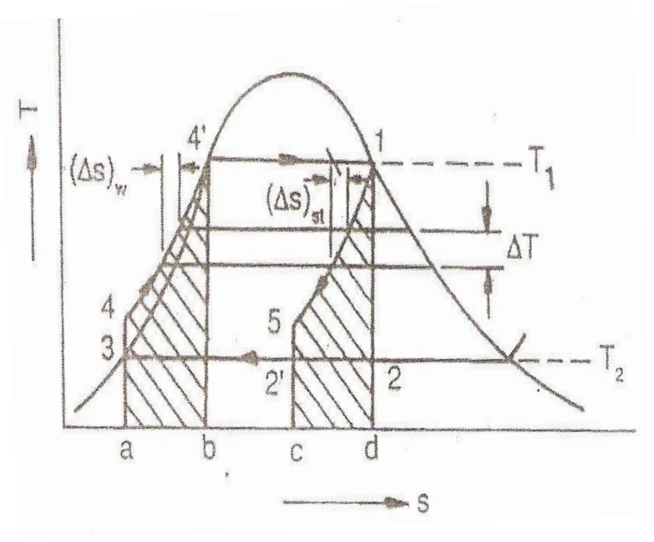

Fig.1.3. Ideal regenerative cycle on T-s plot

The unique feature of the ideal regenerative cycle is that the condensate, after leaving the pump circulates around the turbine casing counter flow to the direction of vapour flow in the turbine. Thus it is possible to transfer heat from the vapour as it flows through the turbine to the liquid flowing around the turbine. It may be assumed that this is a reversible heat transfer, i.e. at each point the temperature of the vapour is only infinitesimally higher than the temperature of the liquid. The process 1-2' thus represents reversible expansion of steam in the turbine with reversible heat rejection for any small step in the process of heating the water.

PARAMETERS	**PRESSURE**	**TEMPERATURE**	**ENTHALPY**
	(Kg/cm^2)	**(^{O}C)**	**(KJ/Kg)**
H.P.TURBINE			
INLET	116	538	3454.87
OUTLET	25.5	337	3095.88
I.P.TURBINE			
INLET	22.27	536	3545.68
OUTLET	1.2848	185	2844.22
L.P.TURBINE			
INLET	1.2848	185	2844.22
OUTLET	-0.9	52	2595.82

Table.1.1. specifications of steam turbines

The slopes of lines 1-2 and 4'-3 (fig) will be identical at every temperature and the lines will be identical in the contour. Areas 4-4'-b-a-4 and 2'-1-d-c-2' are not equally but congruous. Therefore, all the heat added from an external source (Q_1) is at the constant temperature T_1, and all the heat rejected (Q_2) is at the constant temperature T_2 both being reversible then,

$Q_1 = h_1 - h_4' = T_1 (s_1 - s_4)$ $\quad$ $s_4 - s_3 = s_1 - s_2$

$Q_2 = h_2 - h_3 = T_2 (s_2 - s_3)$ $\quad$ $s_1 - s_4 = s_2 - s_3$

$$\eta = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2}{T_1}$$

The efficiency of the ideal regenerative cycle is thus equal to the Carnot cycle efficiency then the steady flow energy equation for the turbine

$H_1 - W_t - h_2 + h_4 - h_4 = 0$

$W_t = (h_1 - h_2) - (h_4 - h_4)$

The pump work remains the same as in the Rankine cycle, i.e.

$W_p = h_4 - h_3$

The net work output of the ideal regenerative cycle is thus less, and hence its steam rate will be more, although it is more efficient, when compared with the Rankine cycle. However, the cycle is not practicable for the following reason.

II. CALCULATIONS

A. STAGE-1 CYCLE EFFICIENCY (With Regeneration)

Technical Data (Stage 1) : Load: 210MW Total mass of steam entering into H.P.Turbine= 636 tons/hour
Total Feed water flow in the circuit= 689 tons/hour

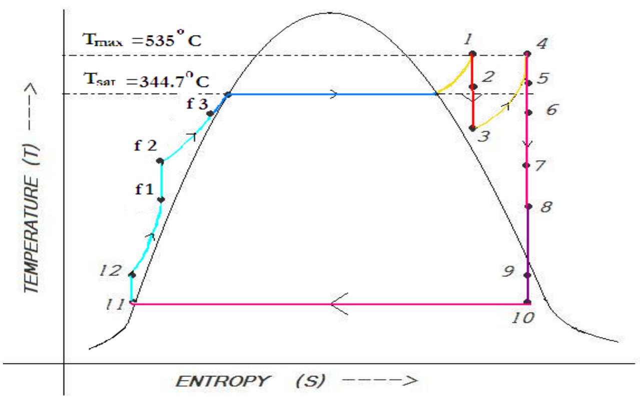

Fig.2.1.Rankine cycle with regeneration and reheating.

Let assume that equivalent amount of steam entering into the turbine

m=mass of steam at the inlet of H.P.Turbine
m_1=mass of extraction steam in H.P.H7
m_2=mass of extraction steam in H.P.H6
m_3=mass of extraction steam to Deareator
m_4=mass of extraction steam in H.P.H5
m_5=mass of extraction steam in L.P.H4
m_6=mass of extraction steam in L.P.H3
m_7=mass of extraction steam in L.P.H2
m_8=mass of extraction steam in L.P.H1
H_1=Enthalpy of steam at the inlet of H.P.Turbine
H_2=Enthalpy of extraction steam in H.P.H7
H_3=Enthalpy of extraction steam in H.P.H6 & Enthalpy of extraction steam to Deareator & Enthalpy of steam at the exit of H.P.Turbine
H_4=Enthalpy of steam at the inlet of I.P.Turbine
H_5=Enthalpy of extraction steam for H.P.H 5
H_6=Enthalpy of extraction steam in L.P.H4
H_7=Enthalpy of extraction steam in L.P.H3
H_8=Enthalpy of extraction steam in L.P.H2 & Enthalpy of steam at the inlet of L.P.Turbine
H_9=Enthalpy of extraction steam in L.P.H1
H_{10}=Enthalpy of steam at the outlet of L.P.Turbine

HEATERS	Quantity of Extraction steam (Tons/hr)	Mass Fraction	Pressure (Kg/cm^2)	Temperature (^{O}C)	Enthalpy (KJ/Kg)
H.P.H7	32.447	m_1=0.0510	39.5	382	3172.54
H.P.H6	47.278	m_2=0.0743	26.5	325	3065.32
H.P.H5	17.661	m_4=0.0277	11.5	447.3	3363.5
DEAERATOR	6.5	m_3=0.0102	22	325	3076.5
L.P.H4	25.4	m_5=0.040	9.4	363.5	3187.9
L.P.H3	22.53	m_6=0.0354	1	263.5	3001.41
L.P.H2	26.67	m_7=0.0419	0.4	183	2845.06
L.P.H1	8	m_8=0.0125	-0.7	66.8	2622.12

Table.2.1. parameters of Heaters

Work done by H.P.Turbine:

$W_{H.P.T} = m*(H_1-H_2) + (m-m_1)*(H_2-H_3)$
$W_{H.P.T} = 1*(3454.87-3172.54) + (1-0.0510)*(3172.54-3065.32)$
$W_{H.P.T} = 383.96$ KJ/Kg

Work done by I.P.Turbine:

$\mathbf{W_{I.P.T}} = (m-m_1-m_2-m_3)*(H_4-H_5) + (m-m_1-m_2-m_3-m_4)*(H_5-H_6) + (m-m_1-m_2-m_3-m_4-m_5)*(H_6-H_7) + (m-m_1-m_2-m_3-m_4-m_5-m_6)*(H_7-H_8)$
$\mathbf{W_{I.P.T}} = (1-0.0510-0.0743-0.0102)*(3545.68-3363.5) + (1-0.0510-0.0743-0.0102)*(3363.5-3187.9) + (1-0.0510-0.0743-0.0102-0.0277+0.0401)*(3187.9-3001.41) + (1-0.0510-0.0743-0.0102-0.0277-0.0401-0.0354)*(3001.41-2845.06)$
$W_{I.P.T}$ = 594.47 KJ/Kg

Work done by L.P.Turbine:

$\mathbf{W_{L.P.T}} = (m-m_1-m_2-m_3-m_4-m_5-m_6-m_7)*(H_8-H_9) + (m-m_1-m_2-m_3-m_4-m_5-m_6-m_7-m_8)*(H_9-H_{10})$

$\mathbf{W_{L.P.T}} = (1-0.0510-0.0743-0.0102-0.0277-0.0401-0.0354-0.0419)*(2845.06-2622.12) + (1-0.0510-0.0743-0.0102-0.0277-0.0401-0.0354-0.0419-0.0125)*(2622.12-2595.82)$
$W_{L.P.T}$ = 186.07 KJ/Kg

Total Work done by Turbine $W_{Turbine} = W_{H.P.T} + W_{I.P.T} + W_{L.P.T}$
=383.9+594.4+186.07
$W_{turbine}$ = 1164.37 KJ/Kg

Calculations for Heat Input (Q_{in}):

Heat Input $Q_{in}=Q_{f3\text{-}1} + Q_{3\text{-}4}$
Where
Q_{in}= Total Heat Supplied
$Q_{f3\text{-}1}$= Heat Supplied to rise the Feed Water temperature to Super Heat Condition
$Q_{3\text{-}4}$= Heat Supplied during Reheating of Steam
Mass of Feed water at the inlet of Economiser = 1
Temperature of Feed water before Economiser = 255 ^{o}C
Pressure of Feed water before Economiser =137 Kg/cm^2

PARAMETERS	PRESSURE (Kg/cm^2)	TEMPERATURE (^{O}C)	ENTHALPY (KJ/Kg)
H.P.TURBINE			
INLET	116	538	3454.87
OUTLET	25.5	337	3095.88
I.P.TURBINE			
INLET	22.27	536	3545.68
OUTLET	1.2848	185	2844.22
L.P.TURBINE			
INLET	1.2848	185	2844.22
OUTLET	-0.9	52	2595.82

Table.2.2. Parameters of Turbines

Enthalpy =1109.89 KJ/Kg
Temperature of Steam after Super Heating = 538 ^oC
Pressure of Steam after Super Heating = 116 Kg/cm^2
Enthalpy= 3454.87 KJ/Kg

Q_{f3-1}=m*(H_{f3}-H_1)
=1*(3454.87-1109.89)
Q_{f3-1}= 2344.98 KJ/Kg
Mass of Steam at the inlet of Reheating =(1-0.052147-0.069-0.04601)
Temperature of Steam before Reheating = 337 ^oC
Pressure of Steam before Reheating =25.5 Kg/cm^2
Enthalpy=3095.88 KJ/Kg
Temperature of Steam after Reheating = 536 ^oC
Pressure of Steam after Reheating =22.27 Kg/cm^2
Enthalpy=3545.68 KJ/Kg
Q_{3-4}= (m-m_1-m_2-m_3)*(H_{f3}-H_1)
=(1-0.0510-0.0743-0.0102)*(3545.68-3095.88)
Q_{3-4}= 388.17 KJ/Kg
Total heat supplied
Q_{in}=Q_{f3-1} + Q_{3-4}
= 2344.98+388.17
Qin = 2733.15 KJ/Kg
Efficiency (η) = W_{net}/Q_{in}
= 1164.37/2733.15
η = 42.6 %
Efficiency of Cycle with Regeneration of Stage 1 = 42.6%

B. STAGE-1 CYCLE EFFICIENCY (Without Regeneration)

Technical Data (Stage 1): Load: 210MW

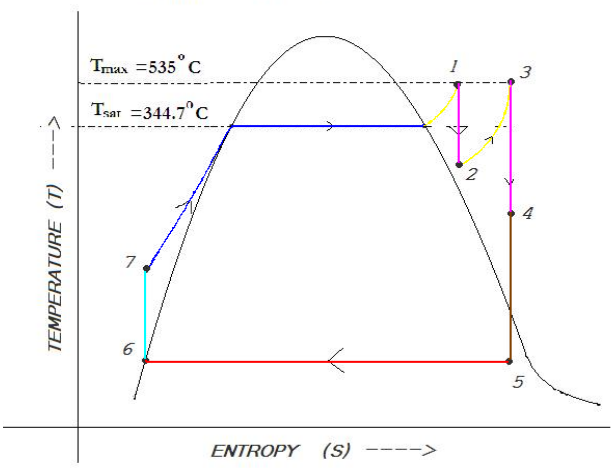

Fig.2.2.Rankine cycle without regeneration and with Reheating

H_1=Enthalpy of steam at the inlet of H.P.Turbine
H_2=Enthalpy of steam at the exit of H.P.Turbine
H_3=Enthalpy of steam at the inlet of I.P.Turbine
H_4= Enthalpy of steam at the outlet of I.P.Turbine & Enthalpy of steam at the inlet of L.P.Turbine
H_5=Enthalpy of steam at the outlet of L.P.Turbine

Work done by H.P.Turbine:
$W_{H.P.T}$ = m*(H_1-H_2)
$W_{H.P.T}$ = 1*(3454.87-3095.88)
$W_{H.P.T}$ = 358.99KJ/Kg

Work done by I.P.Turbine:
$W_{I.P.T}$ = m*(H_3-H_4)
$W_{I.P.T}$ = 1*(3545.68-2844.22)
$W_{I.P.T}$ = 701.46 KJ/Kg

Work done by L.P.Turbine:
$W_{L.P.T}$ = m*(H_4-H_5)
$W_{L.P.T}$= 1*(2844.22-2595.82)
$W_{L.P.T}$ = 248.4 KJ/Kg
Total Work done by Turbine
$W_{Turbine}$= $W_{H.P.T}$ + $W_{I.P.T}$ + $W_{L.P.T}$
= 358.99+701.46+248.4
$W_{turbine}$ = 1308.85 KJ/Kg

Calculations for Heat Input (Q_{in}):
Q_{in}=Q_{7-1} + Q_{2-3}
Where
Q_{in} = Total Heat Supplied
Q_{7-1}= Heat Supplied to rise the Feed Water temperature to Super Heat Condition
Q_{2-3}= Heat Supplied during Reheating of Steam
Mass of Feed water at the inlet of Economiser = 1
Temperature of Feed water before Economiser = 62 ^oC
Pressure of Feed water before Economiser = 160 Kg/cm^2
Enthalpy= **272.877 KJ/Kg**
Temperature of Steam after Super Heating = 538 ^oC
Pressure of Steam after Super Heating = 116 Kg/cm^2
Enthalpy= **3454.87 KJ/Kg**

Q_{7-1}=m*(H_{f3}-H_1)
=1*(3454.87-272.877)
Q_{7-1}= **3181.99 KJ/Kg**

Mass of Steam at the inlet of Reheating = 1
Temperature of Steam before Reheating = 337 ^oC
Pressure of Steam before Reheating = 25.5 Kg/cm^2
Enthalpy = **3095.88 KJ/Kg**
Temperature of Steam after Reheating = 536 ^oC
Pressure of Steam after Reheating = 22.27 Kg/cm^2
Enthalpy = **3542.3 KJ/Kg**

Q_{2-3}= m*(H_{f3}-H_1)
= 1*(3545.68-3095.88)
= **449.80 KJ/Kg**

Total heat supplied Q_{in} = Q_{7-1} + Q_{2-3}
= 3181.99+449.80
Qin = 3631.79 KJ/Kg

Efficiency (η) = W_{net}/Q_{ins}
= 1308.85/3631.79
η = 36.03 %
Efficiency of Cycle without Regeneration of Stage 1 = 42.6%

III. 3D MODELING OF STEAM BOILER TUBE

Boiler assembly model created using NX 7.5 software. Drawing inputs of the model are taken from design calculations.

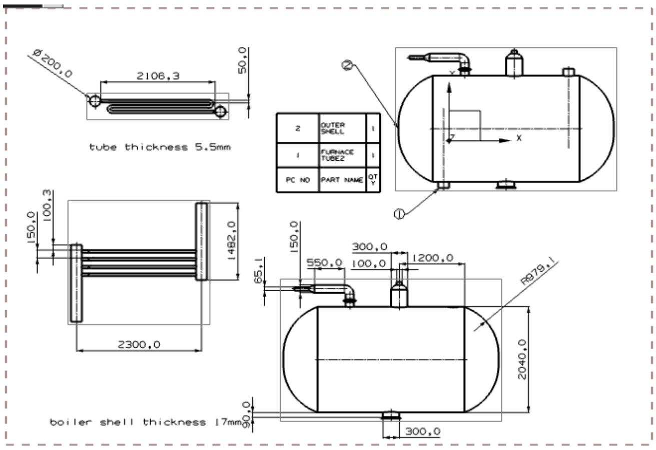

Fig.3.1. 2D input of boiler

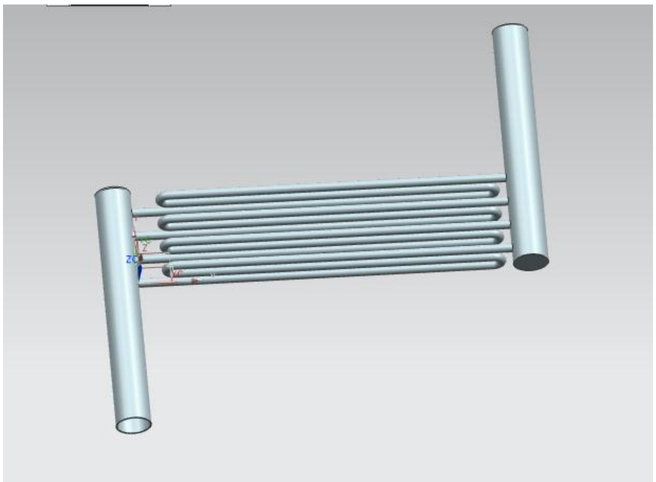

Fig.3.2. 3D model of boiler tube (Fire).

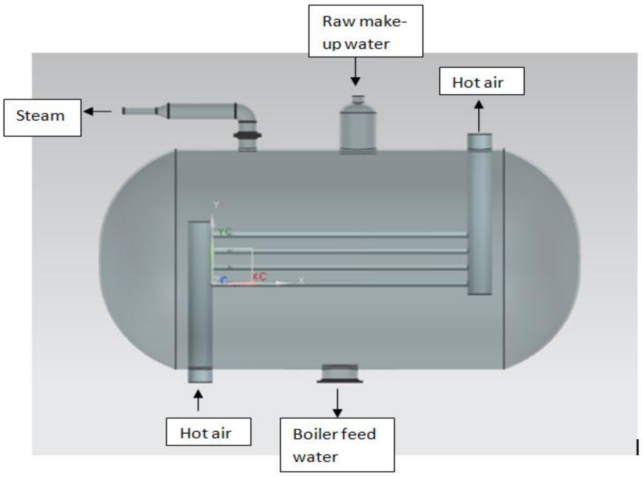

Fig.3.3. inlet and outlet of boiler details

A. THE MODELING PROCESS:

The modeling process consists of fluid geometry and replicating this in the virtual environment. Mesh can be created to divide the fluid up into discrete sections. Boundary conditions must then be entered into the model to designate parameters Conventional geometries, with their characteristics and applications of fluids to be modeled or the details of any solid edges or flow inlets/outlets. The simulation is then ready to be run and when a converged solution is found. The same is analyzed to establish whether the mesh is appropriately modeling the flow conditions. Generally, some form of mesh refinement will be necessary to put in further detail around the areas of interest. In this case diameter of tube is 50mm.

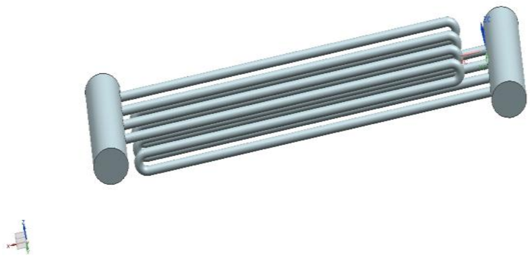

Fig.3.4. Shows the Geometry of steam boiler furnace tubes

IV. RESULTS

CONTOUR PRESSURE:

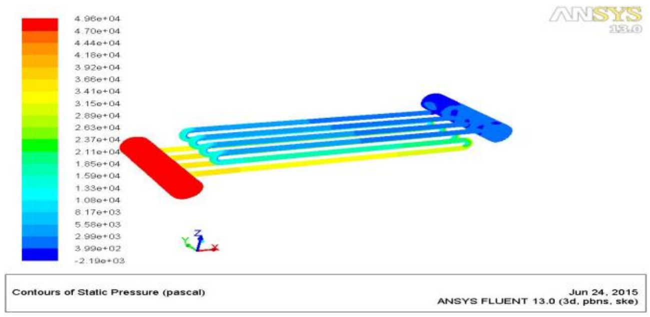

Fig.4.1. Contour pressure.

The maximum contour output Pressure occurred on the steam boiler furnace tube

VECTOR PRESSURE:

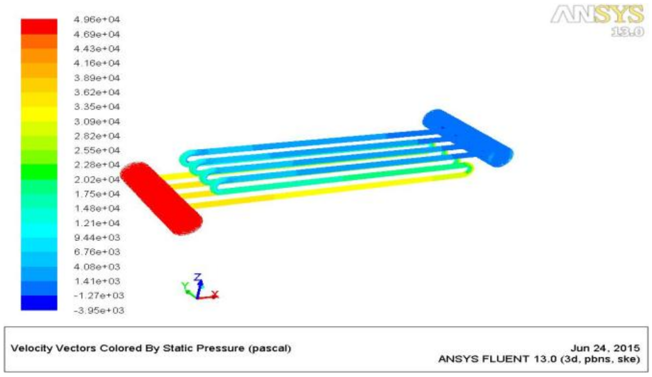

Fig.4.2. Vector Pressure

The maximum Vector Pressure occurred on the steam boiler furnace tube.

The observation is that the maximum pressure (49620Pa) occurred on the steam boiler furnace tube inlet areas.

VELOCITY
COUNTOUR:

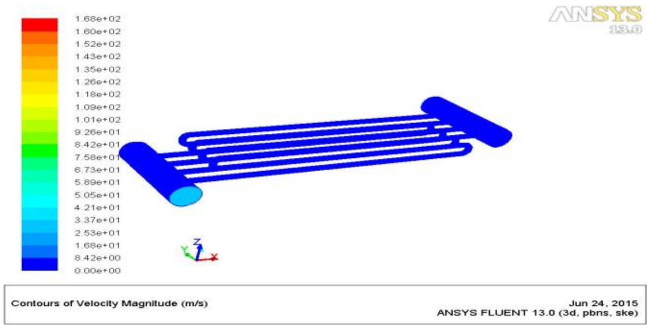

Fig.4.3. Shows the maximum Contour velocity occurred on the steam boiler furnace tube

VECTOR VELOCITY:

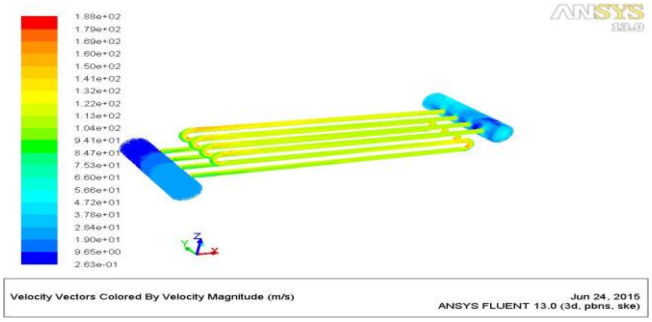

Fig.4.4. Shows the maximum Vector velocity occurred on the steam boiler furnace tube

The observations are the maximum velocity (168m/s) occurred on the steam boiler furnace tube circular areas. The inlet velocity is 28m/s, and outlet velocity is 42m/s.

COUNTOUR TEMPERATURE:

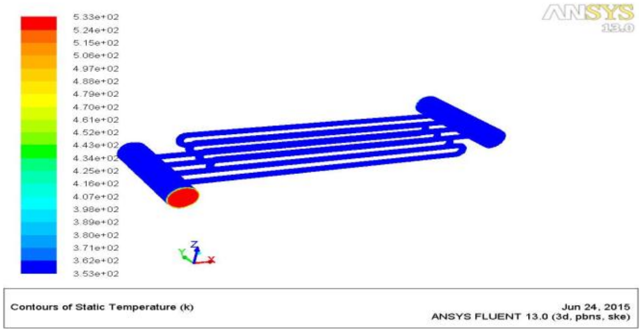

Fig.4.5. Shows the max. Contour temperature occurred on the steam boiler furnace tube

VECTOR TEMPERATURE:

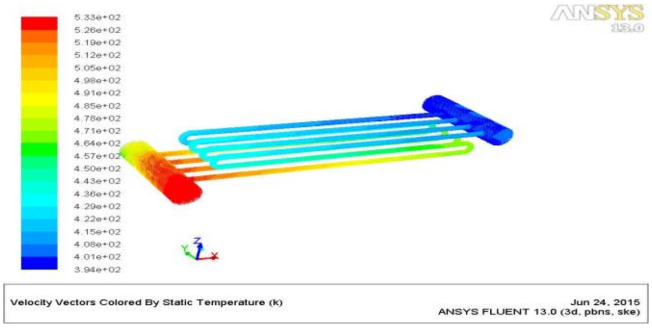

Fig.4.6. Shows the max. Vector temperature occurred on the steam boiler furnace tube

The observations are the inlet and outlet temperature is 533K, 394.49K respectively occurred on the steam boiler furnace tube inlet and outlet areas

One unit (210 M.W) of stage -I is considered for the present analysis.

Efficiency Stage/Unit	η (With regeneration)I	H (Without regeneration)
Stage-1	42.6%	36.03%

It is clearly observed from the above results that there is considerable increment in the efficiency due to regenerative feed water heating.

Hence an effective analysis required to optimize the number of feed water heaters to be used in the system. Also the maintenance of the heaters in the form of scale or corrosion plays an important role in the overall performance of the plant.

S.NO	NAME	INLET	OUTLET	VARIATION
1	pressure	49640Pa	4080Pa	45560
2	velocity	28m/s	42m/s	14
3	temperature	533K	394 K	-139
4	Mass Flow Rate	1.0760864(kg/s)	-1.0759337 (kg/s)	0.00015265535

Table4.1. shows the inlet and outlet values

V. CONCLUSIONS

The following conclusions can be made

- The unit of stage -I has gained an efficiency of 3.36%
- The tubes of the FW heaters must be periodically cleaned using water jets & chemicals
- Replacement of damage or worn-out components
- The excessive tube dummies may be avoided, so as to increase the effective heat transfer.
- It is suggested to replace the tubes if the number of tubes dummied is more than 10% of the total number of tubes.
- Effective instrumentation is needed to evaluate the total system & sub systems performance.
- Optimization of Deaerator performance is needed.
- By taking proper steps for the above mentioned reasons, the overall efficiency of the plant will be improved considerably & components can be made compact & also the loads on condenser can be reduced and hence there will be an increased power output.

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