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A Method to Improve Exergtic Efficiency of Power Plant Cycle by Heat Pipes

T. Mallikharjuna Rao
Joint Director, Central Power Research Institute, Bangalore- 560080, India.

Orcid ID : 0000-0002-1992-2246

Dr. S.S. Rao
Professor, Mechanical Engineering Department, National Institute of Technology, Warangal-506004, India.

Abstract

In the recent decades, exergy analysis has found increasingly widespread acceptance as a useful tool in the design, assessment, optimisation and improvement of energy systems. This paper explores the possibility of reduction in exergy destruction by using Heat pipes in condenser and thus improving the exergtic efficiency a thermal power plant. The detailed explanation of exergy and exergy efficiency of different components of a thermal power plant was presented. Experimental setup and results of steam condenser loaded with heat pipes were presented.

Keywords: Exergy, Heat pipes, Condenser, Thermal Power plant etc.

INTRODUCTION

The gap between energy demands to energy supply narrowing down day to day around the world. Energy consumption is one of the most important indicator showing the development stages of countries and living standards of communities. In order to meet these energy demands, the capacity addition is inevitable. This growing demand of power has made the power plants of scientific interest, but most of the power plants are designed by the energetic performance criteria based on the first law of thermodynamics only. Here, the efficiency indicates how well an energy conversion or process accomplished. This word "Efficiency" is one of most frequently misused in thermodynamics and often used without being properly defined. The real useful energy loss cannot be justified by the first law of thermodynamics, because it does not differentiate between the quality and quantity of energy. First law of Thermodynamics is a simple energy balance without taking into account the quality of energy used. Exergetic analysis, based on second law of thermodynamics, takes in to account the energy quality. The energetic and exergetic analysis will provide a complete picture to improve the plant efficiency. Hence this exergy analysis can be generally applied to energy and other systems, it appears to be more powerful tool than energy analysis for power cycles because of the fact that it helps determine the true magnitudes of losses and their causes and locations, and improve the overall system and its components.

ANALYSIS OF STEAM POWER PLANT

Steam Power Plants (SPPs) are based on the Rankine cycle. However, after a century of research and development, current SPPs have become more complex than ideal Rankine cycles, in order to achieve thermal efficiencies above 40%, based on the Low Heating Value (LHV) of the fuel (Ataei, 2009). The SPP is known to feature high flexibility, a long lifetime, high reliability without complexity, and commercial applicability; SPPs have become quite popular. The recent increase in fuel prices, the necessity for better environmental performance, and the curbing of air pollution and greenhouse gases have stimulated the search for further improvements. The efficiency of the Rankine cycle can be improved by varying cycle parameters such as the turbine inlet pressure, inlet temperature, reheat pressure, reheat temperature, extraction pressure, and condenser pressure, with respect to their optimum values (Azhdari et al., 2009). In steam power plants, it can be observed from cycle thermodynamics that synthesis of an optimal heat exchanger network with minimization of utilities may reduce Exergy losses and improve the cycle efficiency (Kwak, 2003; Rosen and Dincer, 2003; Sanjay, 2007) .

In the ideal cycle for vapor power cycle, many of the impracticalities associated with the Carnot cycle can be eliminated by superheating the steam in the boiler and condensing it completely in the condenser, as shown schematically on a T-s diagram in **Figure 1**. The ideal Rankine cycle does not involve any internal irreversibility and consists of the following four processes:

1-2 Isentropic compression in a pump;

2-3 Constant pressure heat addition in a boiler;

3-4 Isentropic expansion in a turbine;

4-1 Constant pressure heat rejection in a condenser

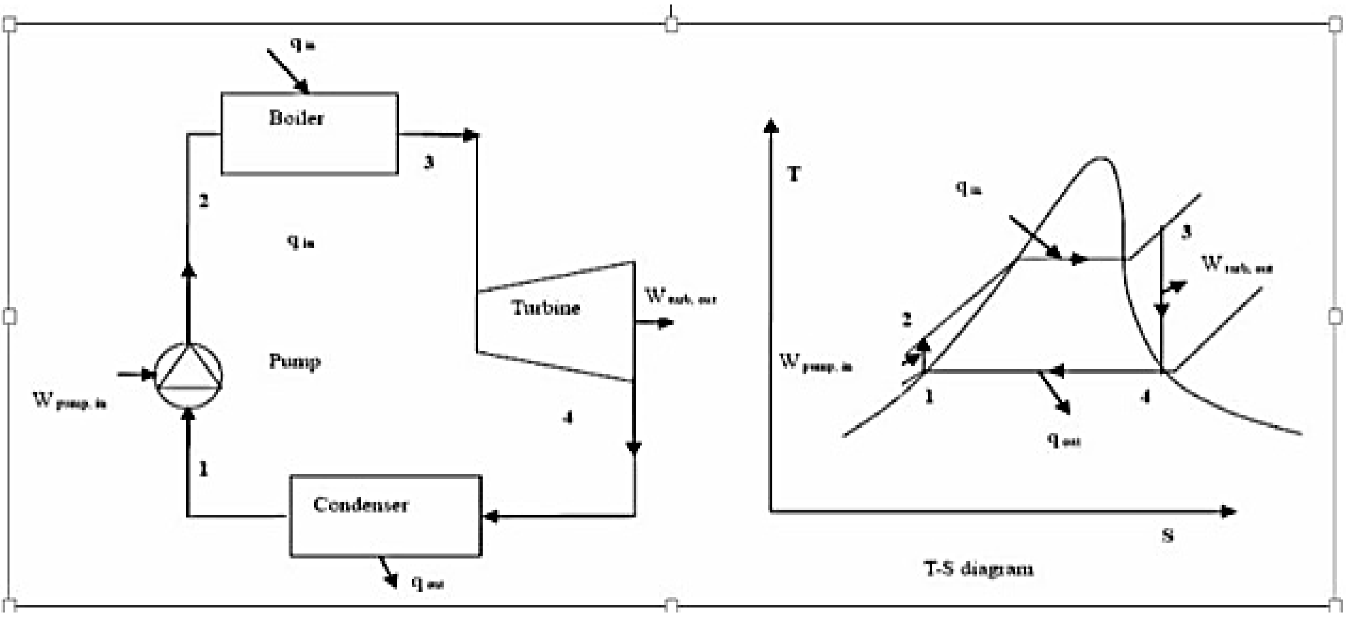

Figure 1. Schematic representation of Rankine Cycle

Energy Performance Analysis of the Power Plant :

Energy performance analysis is based on the First Law of Thermodynamics. Using this law, the main performance criteria like power output and thermal efficiency can be analyzed. In this analysis, the input and output values of the plant components are determined using the measured or calculated thermodynamic variables such as enthalpy, pressure, temperature, entropy, mass flow rate and quality. Each device in the power plant forms a control volume, with associated equations for energy analysis as described further. From a standard book on Power Plant engineering, it can written as,

Continuity Equation :

$$\sum m_{in} = \sum m_{out}$$

Where m is the mass flow rate and subscripts in and out represent inlet and outlet conditions, respectively. The energy balance equation:

$$-W + Q = \sum m_i.(h_{out.i} - h_{in.i})$$

where Q is the heat transfer rate to the control volume, W is the given work per unit of time and h is the enthalpy. Kinetic and potential energy changes, considered negligibly small compared to the changes of enthalpy, are neglected. The power output of a steam turbine is calculated by:

$$W_T = m_{in}(h_{in} - h_1) + (m_{in} - \sum_{j=1}^{n} m_j).(h_j - h_{j+1})$$

Where the subscript n represent the number of steam extraction in the steam turbine. The power consumed by pumps is the only internally consumed power considered in the plants model. This power is calculated by:

$$W_P = m_{in}.(h_{out} - h_{in})/\eta_p$$

where η_p is the pump efficiency. Net electrical power output is given by,

$$W_{net} = \sum W_T - \sum W_P$$

The thermal efficiency of the plant can be calculated by as follows,

$$\eta_{th} = \frac{W_{net}}{m_f.LHV}$$

Where LHV is lower heating value of the coal and m_f is the fuel consumption rate.

EXERGY PERFORMANCE ANALYSIS OF THE POWER PLANT

Exergy performance analysis is based on Second Law of Thermodynamics. The results obtained from this analysis can be used for determining the irreversibility components in the power plant. Exergy is a thermodynamic indicator which shows the transformation potential and conversion limit of an energy carrier to maximum theoretical work under the conditions imposed by an environment at given pressure and temperature.

For a control volume of any plant's component at steady-state conditions, a general equation of exergy destruction rate derived from the exergy (Γ) balance can be given as:

$$\Gamma_{in} + \Gamma_Q = \Gamma_{out} + \Gamma_W + \Gamma_D + \Gamma_L$$

Where subscripts *in* and *out* referee to inlet and out let flows with respect to the control volume, and

$$\Gamma_Q = (1 - \frac{T_o}{T_i}).\,Q_i$$

$$\Gamma_w = W$$

where T is the absolute temperature, whilst subscripts i and o refer to the surface and environment conditions, respectively. Exergy destruction Γ_D and exergy loss Γ_L represent a measure of the inefficiency of the irreversible processes occurring in the K^{th} component of the plant. When considering a single component of a thermal system, the exergy losses are usually equal to zero as shown by Ameri *et al.* [6]:

$$\Gamma_L = 0$$

Exergy flow rate of a system consists of a kinetic, potential, physical and a chemical one:

$$\Gamma^* = \Gamma^*_{PH} + \Gamma^*_{KN} + \Gamma^*_{PT} + \Gamma^*_{CH}$$

Where Γ^*_{PH} , $\Gamma^*_{KN}, \Gamma^*_{PT}$ *and* Γ^*_{CH} are the physical exergy , kinetic exergy , potential exergy and chemical exergy respectively , formulation of which are described by Bejan et al. [7].

By crirically reviewing the performance of the power plants , is found that , in the Thermal Power Plant major exergy destruction is taking place in the Turbine, Boiler and Condenser. Among these three components, if exergytic efficiency of turbine and boiler improved, then the net output will increase. But in the condenser, which is basically an energy dumper, the exergy destructions are more and this exergy destruction enters into atmosphere as undesired pollution. The Irreversibility in condenser is also significant, because the condensate that is condenser output is the input to the cycle again.

WHAT IS THE METHOD TO REDUCE THIS EXERGY?

A complete exergetic analysis of an existing Condenser could help to identify sources of exergy loss and possible improvements. Analyzing exergy in a condenser, either for design or for analysis, is not straightforward. The three main causes of irreversibility are heat transfer between the flows, pressure losses due to fluid friction, and dissipation of energy to the environment; the three phenomena can also occur simultaneously. There exists other problems, which have minor effects, such as stream wise conduction in the walls of the heat exchanger The work of Bejan [9] is regarded as the basis of exergetic analysis of heat exchangers. The problem has been studied enthusiastically by other authors, leading to a number of methods and indicators for expressing exergetic efficiency; these are summarized in [10]. But the available literature lacks exact solution to reduce or minimize exergy destruction in the condensers.

Here, it is proposed that the exergy reduction is possible in the condenser by successive cooling the dumped steam that the dumped steam is not directly cooled by cooling water. The heat energy from the steam is transferred to a third liquid which is a higher temperature than the cooling water and then the heat energy from this third liquid is transferred to the cooling water . The scheme can be depicted as below.

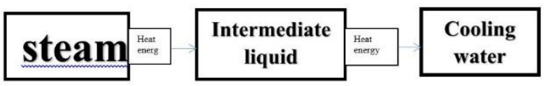

Figure 2. Scheme of proposed heat transfer

It is known, that the exergy destruction depends on the temperatures of the fluid and environmental. Now according to Fig 2 , the steam is not exchanging the temperature directly with environment , hence there will be comparatively less exergy destruction in this proposed method.

This method can be achieved by the use of heat pipes. Hence the possibility of heat pipes for using steam condensation is explored, which is not explored till today. Hence a heat pipe designed for this purpose and presented in Fig 3.

The details of Heat Pipe designed for this purpose are as follows. *(Fig 3)*

Length of the Heat pipe = L = 4.3m

Length of evaporator section = L_e = 2m

Length of the condenser section = L_c = 2m

Length of the adiabatic section = l_a = 0.3 m

Tube is 'K' type Nominal Standard Size is 2 inch

Outside diameter of the heat pipe = d_o = 2.125 inch=0.0540 m

Inside diameter of the heat pipe = d_i=1.959 inch= 0.0497 m

Thickness of the heat pipe wall = t = 0.083 inch = 2.1082 x 10^{-3} m

<u>The heat pipe is a wickless, gravity assisted that is strictly it is a two phase closed thermosiphon</u>

Vacuum inside heat pipe = 0.07 bar

Hence Saturation temperature for water inside heat pipe = 39°C

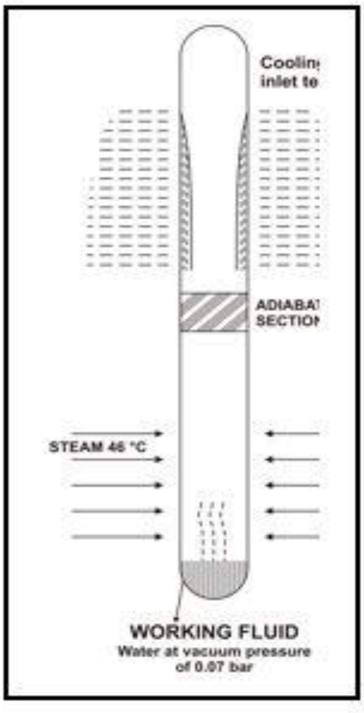

Figure 3. Designed Heat Pipe

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The fabricated heat pipes are shown in Fig 4.

Figure 4. Fabricated heat pipes

The Thermodynamic limits of the designed heat pipe are as follows.

Sl.No	Parameters	Desired requirements of Heat Pipes in the proposed HPHE	Designed Heat pipes characteristics as per different calculations
1	Maximum heat transfer limit from the Boiling point of view	30 kW	71 kW from (3)
2	Maximum heat transfer limit from the Flooding point of view	30kW	59.3 kW from (4)

With these developed heat pipes a laboratory experiment conducted for steam condensation and presented below.

Line Diagram for the above Experimental Setup:

The line diagram, actual photograph of the experimental set is shown in Fig 5.

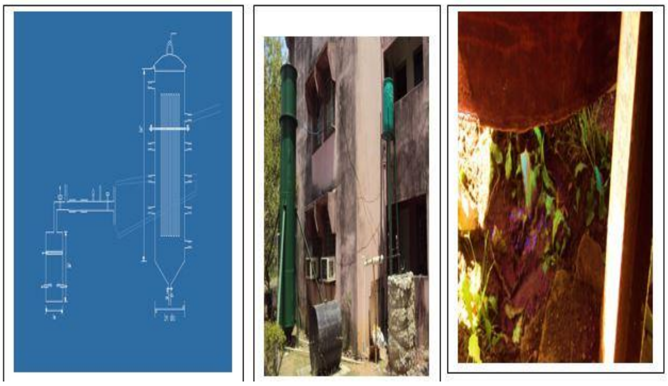

Figure 5. Line Diagram of Experimental Set up

<u>Instruments Used during Experiment:</u> Flow Meters, Digital temperature indicators, compound pressure gauge, vacuum pump. Bucket and beaker (to measure the quantity of cooling water) Power Analysers to measure the power input to boiler heaters.

<u>Description of the Experiment</u>: Steam inlet entry to the condenser will be from the both sides as shown in the figure. The cooling water will be entered inside the condenser just above separator plate and exited from the top portion .The condensate will be flooded to the bottom of the condenser.

No of heat pipes used are 16. A miniature steam generator whose heating capacity will be 33 kW made use to supply the steam. A superheating system is arranged to superheat the generated steam. Then the superheated steam and throttled and fed into the heat pipe condenser. The temperature and pressure of the steam measured before throttle valve. The condensate is collected at bottom tank. The cooling water is supplied from a overhead tank. The inlet and out let temperature of cooling water is determined by thermometers. The actual photo experimental set up is given above. After reaching the steady state conditions, the readings were taken. Ten different trials were conducted and results are presented in Table 1 and also the performance shown in the in the Fig.6

Table 1

Sl. No	$M_{condensate}$ Lit/min	T_1 of Condensate in°C	T_2 of steam inlet in°C	P of steam inlet in bar (gauge pressure)	ṁ of cooling water in lit/min	T_3 of inlet cooling water°C	T_4 of outlet cooling water°C	Q_1 heat input by steam in kW	Q_2 heat energy carried out by condensate in kW	Q_3 Heat output of cooling water in kW	Difference of heat input and output in kW (Q_1-{Q_2+Q_3})
1	0.32	118.0	119.0	0.90	35	26.5	30.0	11.80	2.64	8.53	0.63
2	0.34	117.5	118.0	0.85	35	26.5	30.2	12.50	2.82	9.02	0.66
3	0.35	119.2	120.0	0.95	35	26.5	30.5	12.80	2.91	9.75	0.14
4	0.74	102.0	103.0	0.09	35	26.0	35.0	27.91	5.27	21.95	0.69
5	0.77	102.0	103.0	0.10	35	26.0	35.3	28.73	5.50	22.68	0.55
6	0.78	104.0	105.0	0.18	33	26.0	36.1	29.30	5.71	23.33	0.37
7	0.88	104.5	105.0	0.18	33	26.0	36.2	29.88	5.84	23.45	0.59
8	0.83	119.5	120.0	0.94	33	26.0	36.0	30.50	6.92	22.99	0.59
9	0.84	118.5	119.0	0.90	35	26.0	35.5	31.20	6.97	23.20	1.03
10	0.85	117.0	118.0	0.80	35	26.0	35.8	31.31	6.98	23.90	0.43

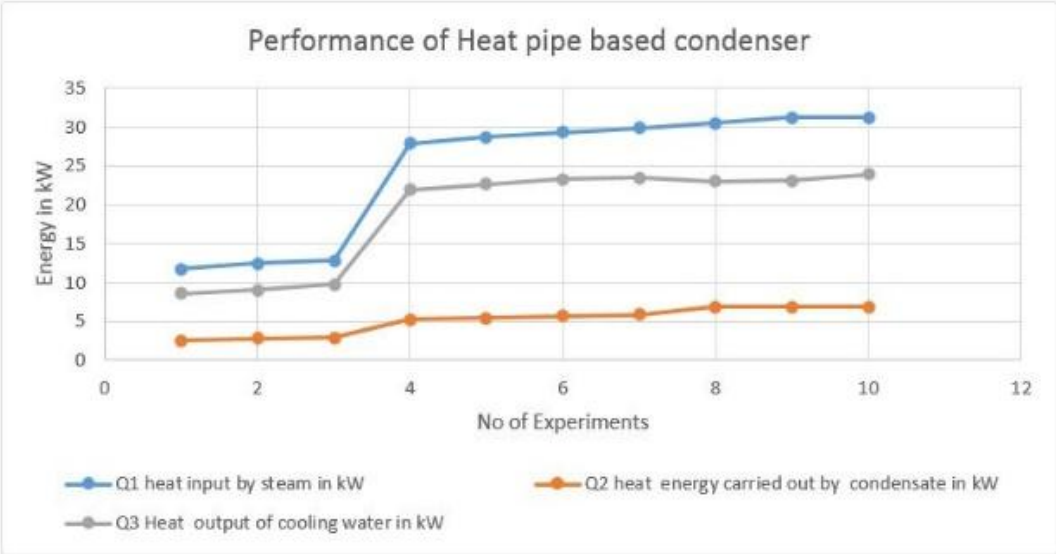

Figure 6. Performance of condenser

The above picture clearly indicates that the heat pipe based condenser consistently transfer the dumped steam energy to the cooling fluid.

Now the case study described in 4.2 proves that the exergy destruction is possible with the heat pipe based condenser.

CASE Study for Demonstration of Reduction in Destruction of Exergy :

Considering the case of a 210 MW plant and its existing condenser. The details of Condenser of the plant for this case study conducted are as follows.

Sl.No	Parameter	Numerical value
1	Unit Load	191 MW
2	Condenser Steam inlet temperature	46 ° C
3	Condenser steam inlet pressure	0.09 bar
4	Condenser cooling water inlet Temp	26.62 ° C
5	Condenser cooling water outlet Temp	36.65 ° C
6	No of Condenser Tubes	19,208
7	Condenser Tube OD	25.4 mm
8	Condenser Tube ID	24.0 mm
9	Condenser Tube Length	11.28 m
10	Load on Condenser	221171743.8 Kcal/hr,260 MW
11	Cooling water Flow	21033.95 t/hr ,5842.76 kg/s
12	No of Condenser tubes	19,200

Now, this conventional condenser is proposed to replace with a heat pipe based condenser. The technical details of the proposed heat pipe based condenser will be as follows.

Sl.No	Parameter	Numerical value
1	No of Heat Pipes in HPHE (Dimensions of Heat pipe given in justification)	9025
2	Arrangement of Heat pipes in HPHPE	Staggered , 95 x 95
3	Length of Heat Pipe	4.3 m
4	Material of Heat Pipe	Copper
5	Vacuum inside the Heat Pipe	0.07 bar
6	Working fluid inside heat pipe	Distilled water
7	Saturation temperature of water inside the heat pipe	39 ° C
8	Wick material	Wickless heat pipe
9	Total Number of Heat Pipes	9025
10	Load on each Heat Pipe	28.8 kW ≈ 30 kW

Calculation of Exergy Destruction for both type condensers:

For Conventional Condenser

$\Delta \Gamma = T_{env} [C_1 \ln (T_1''/T_1') + C_c (T_1' - T_1'')/T_2']$ (Reference 11)

Where $\Delta \Gamma$ is destructed exergy.

Now , C_1 = Cooling water quantity = 5843 kg/s

T_{env} = Temp. Of the Environment = 28 °C = 301 K

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T_1'' = Cooling water out let Temp = 37.62 ºC = 310.62 K

T_1' = Cooling water inlet Temp = 26.62 ºC = 299.62 K

T_2' = Steam inlet temperature = 46 ºC =319 K

Applying numerical,

Hence , $\Delta\Gamma$ = 301 x 5843 x4.18 x[ln (310.62/299.62 + (299.62-310.62/319)]

= 303 x 5843 x 4.18 x (0.0361-0.0345)

= 11,761 kW

For HPHE Condenser

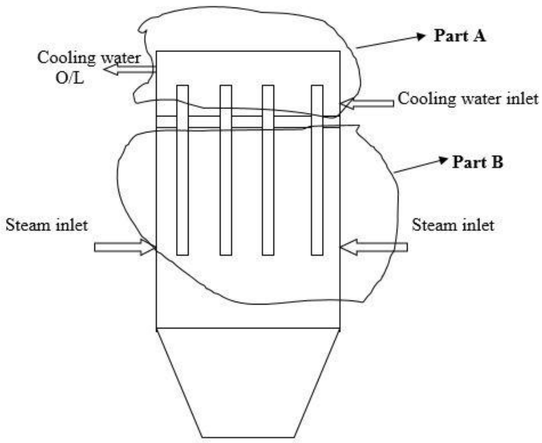

Figure 7: Line Diagram of Heat pipe condenser

For Part B

Where steam condenses into water and fluid inside the heat pipe evaporates.

$\Delta\Gamma = U_oA_o (\pi_T - 1)^2 / \pi_T$] (From Advanced Thermodynamics , Bejan)

U_o = Overall heat transfer coefficient = 2406 $W/m^2.K$

A_o = Overall heat transfer area = 1.3 m^2

π_T = the ratio of input thermodynamic temperature of the streams = 46/39.02 = 1.179

Applying numerical,

Hence , $\Delta\Gamma = 2406 \times 1.3 \times (1.179-1)^2 / 1.179$ = 85 W

For Part A

Where cooling water gets heated and vapor inside heat pipe condenses into liquid.

$\Delta\Gamma = T_{env} [C_1 \ln (T_1''/T_1') + C_1 (T_1' - T_1'')/T_2']$

C_c = Heat capacity of water stream , W/k

T_1' = temperature in K of cooling water at inlet = 299.62 K 302.62

T_1'' = temperature in K of cooling water at outlet = 310.26 K 312.65

T_2' = temperature in K of vapor inside heat pipe before condensation.= 312.02

Hence , $\Delta\Gamma$ = 301 x 5843 x 4.178 x [ln (310.26/299.62) + (299.62-310.26)/312.02]

= 7348028.3 x (0.0349-0.0341)

= 5878.4 kW

Total Exergy in Part A and Part B = 5878.4 kW + 85 kW = 5963.4 ≈ 5963 kW

Reduction of exergy reduction by using Heat pipe condenser = (11,761-5963) kW

= 5,798 kW

That is almost 50 % reduction is possible with Heat pipe based condenser.

CONCLUSION

It can be concluded that the Exergtic efficiency improvement in the Thermal power plant cycle is possible by reducing the exergy destruction in the condenser. This exergy destruction in the condenser is possible by successive cooling of the dumped steam in the condenser. This cooling method is possible by employing the heat pipes in place of conventional tubes in the condenser. Thus if successive cooling method is applied by employing heat pipes in the conventional condenser around 50 % of exergy destruction is possible . Hence the exgertic efficiency of the cycle improves.

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NOMENCLATURE

A cross sectional area, m^2

C Heat Capacity

d diameter, m

g gravitational acceleration, $m.s^{-2}$

h enthalpy

L length, m

m mass flow rate, kg/s

Q Steam Load

T Temperature

t thickness

ρ density, kg.m-3

ε Effectiveness

Γ Exergy

Subscripts

g gas phase

l liquid phase

TP two-phase

p constant pressure

L coupling liquid

o outside

i inside

t total

c condenser

e evaporator

a adiabatic

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