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STEAM TURBINE POWER PLANT DESIGN AND IMPUSLE STAGE BLADE'S VON STRESS AND DEFORMATION ANALYSIS

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Abstract

The project was done to have a better understanding of Rankine cycle and its various modified stage to increase the total efficiency of the entire cycle. Initial condition of inlet turbine pressure of 1800 psi, inlet temperature of 1000 F and exhaust quality of steam of 87 percent was assigned and the magnitude of reheat pressure, mass flow rate and efficiency of the cycle, location and number of feed water heaters were determined. The steam turbine bade design is concept based on the stage for conversion of heat to work in which the thermal energy (potential Energy) is first converted to kinetic energy and then to shaft work. Hot air or steam is the medium through which this process takes place. Potential energy (pressure) is the required condition for the motion to occur while thermal effects (temperature) essentially help to enhance the quality of the compression process. The objective of this project is to determine the approximate number of design stages needed to obtain the desired power from the steam turbine and based on the information obtained in design state 1. Assuming that the first two stages are impulse stages, draw the velocity triangles, obtain the length of blades, shape of blades, efficiency and other design parameters to evaluate and mechanically construct a impulse stage of the steam turbine. To visualize this project simple graph, schematic diagrams, and simple calculations will be provided.

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1.0 Nomenclature

$W = work\ per\ unit\ mass\ (Btu/lbm)$

$q_l = heat\ loss\ per\ unit\ mass\ (Btu/lbm)$

$q_h = heat\ supplied\ in\ the\ boiler\ per\ unit\ mass\ (Btu/lbm)$

$h = enthalpy\ (Btu/lbm)$

$h_{fg} = specific\ differecnce\ in\ enthalpy\ (Btu/lbm)$

$h_f = specific\ enthalpy\ of\ fluid\ (Btu/lbm)$

$h_g = specific\ enthalpy\ of\ steam$

$s = specific\ entropy\ (Btu/lbm * R)$

$s_f = specific\ entropy\ of\ fluid\ (Btu/lbm * R)$

$s_g = specific\ entropy\ of\ steam\ (Btu/lbm * R)$

$\eta_{th} = thermal\ efficency\ of\ cycle\ (0 < \eta_{th} < 100$

$x = quality\ of\ steam\ (0 < x < 1)$

$\dot{m} = mass\ flow\ rate\ (lbm/s)$

$P = Pressure, (psi)$

$T = Temperature$ (°F)

$\eta_{th} = Efficiency$

$h = Specific\ enthalpy\ (Btu/lbm)$

$v = Specific\ volume\ (ft^3/lbm)$

$W = Power\ (KW)$

$r_t = Radius\ of\ tip$ **(inch)**

$r_h = Radius\ of\ hub\ (inch)$

$s = Spacing\ between\ blades\ (inch)$

$c = Chord\ length\ (inch)$

$\frac{U}{V0} = Velocity\ ratio$

$V_0 = Adiabatic\ velocity\ (inch/s)$

$U = Blade\ velocity\ (inch/s)$

$V = Absolute\ velocity\ (inch/s)$

$w = Relative\ velocity (inch/s)$

$\alpha = Angles\ made\ by\ absolute\ velocities$

$\beta = Angles\ made\ by\ relative\ velocities$

$g_c = Gravitational\ force\ (ft/s^2)$

$N = Number\ of\ blade$

$r = Radius\ (mch)$

$d = diameter\ (inch)$

$\Psi = Loading\ factor$

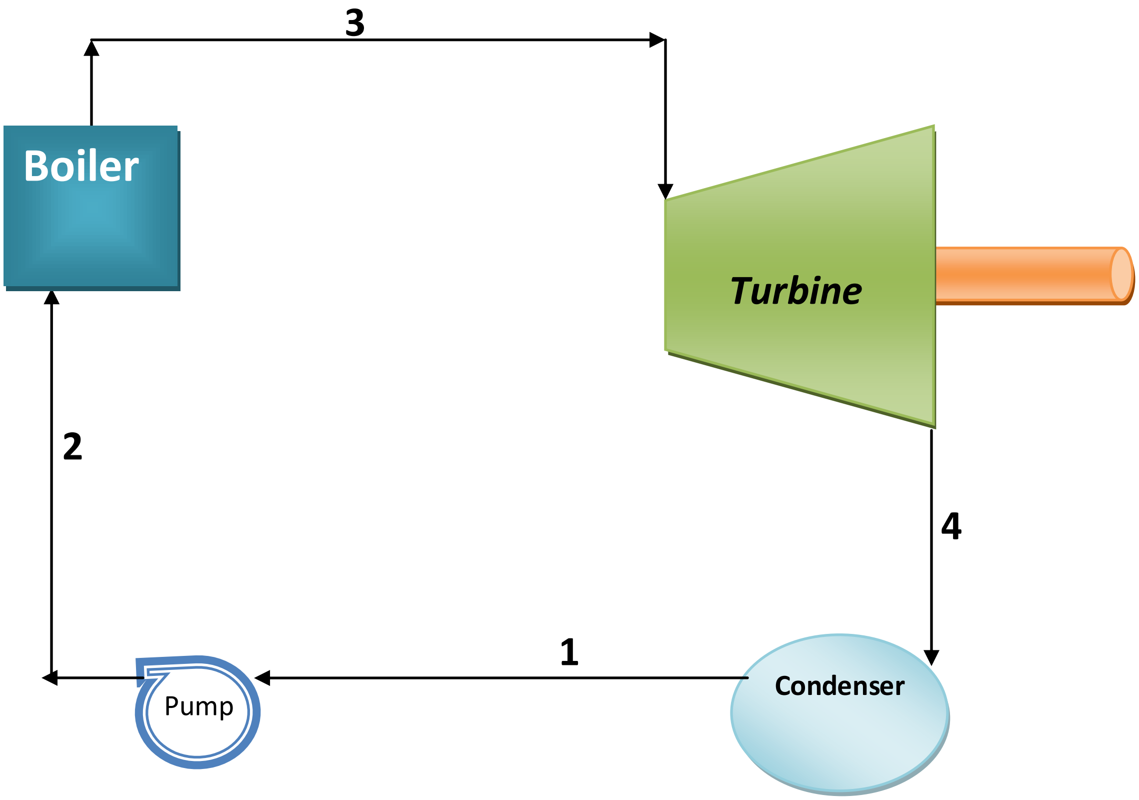
$l = Length\ of\ the\ blade\ (inch)$

$R = Universal\ gas\ constant\ \left(\frac{Btu}{lbmol * R}\right)$

$\rho = Density\ (lbm/ft^3)$

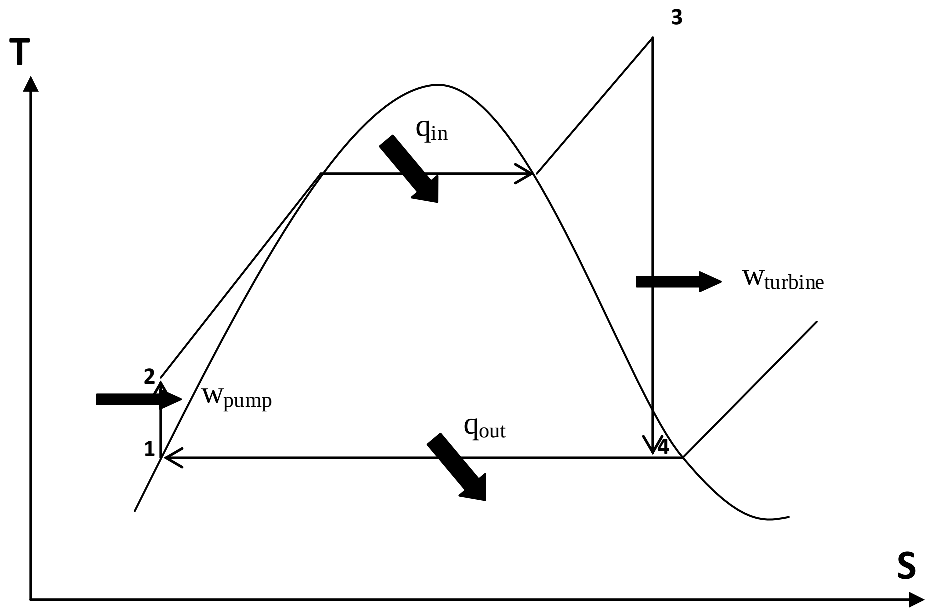
A. Ideal Rankine Cycle

1. Introduction



A Rankine cycle as describes in the figure operates in steam turbine cycle which is most commonly found in power generation plants. Common heat sources for power plants using the Rankine cycle are coal, natural gas, oil, and nuclear. The Rankine cycle is sometimes referred to as a practical Carnot cycle as, when an efficient turbine is used, the TS diagram will begin to resemble the Carnot cycle. The efficiency of a Rankine cycle is usually limited by the working fluid as the working fluid in a Rankine cycle follows a closed loop and is re-used constantly. While many substances could be used in the Rankine cycle, water is usually the fluid of choice due to its favorable properties, such as nontoxic and nonreactive chemistry, abundance, and low cost, as well as its thermodynamic properties. One of the principal advantages it holds over other cycles is that during the compression stage relatively little work is required to drive the pump, due to the working fluid being in its liquid phase at this point.

2. Theory



There are four processes in the Rankine cycle, each changing the state of the working fluid. These states are identified by number in the diagram to the right.

Process 1-2: The working fluid is pumped from low to high pressure, as the fluid is a liquid at this stage the pump requires little input energy.

Process 2-3: The high pressure liquid enters a boiler where it is heated at constant pressure by an external heat source to become a dry saturated vapor.

Process 3-4: The dry saturated vapor expands through a turbine, generating power. This decreases the temperature and pressure of the vapor, and some condensation may occur.

Process 4-1: The wet vapor then enters a condenser where it is condensed at a constant pressure and temperature to become a saturated liquid. The pressure and temperature of the condenser is fixed by the temperature of the cooling coils as the fluid is undergoing a phase-change.

In an ideal Rankine cycle the pump and turbine would be isentropic, i.e., the pump and turbine would generate no entropy and hence maximize the net work output. Processes 1-2 and 3-4 would be represented by vertical lines on the T-s diagram and more closely resemble that of the Carnot cycle. The

Rankine cycle shown here prevents the vapor ending up in the superheat region after the expansion in the turbine, which reduces the energy removed by the condensers.

3. Sample calculation

Inlet temperature (P_3) = 1800 psi

Inlet temperature (T_3) = 1000 ºF

Quality of expanded high pressure steam after expansion (x_4) = .87

Working fluid is ideal gas = water

Assumption for ideal Rankine cycle for ideal gas:

[$P_3 = P_2$]------- (1)

[$P_1 = P_4$]------- (2)

[$s_1 = s_2$]------- (3)

[$s_3 = s_4$]------- (4)

Beginning with condense saturated liquid pressure (P_1) = 5 psi

$h_1 = h_f$ @ P_1 = 130.18 Btu/lbm

$v_1 = v_f$ @ P_1 = 0.01641 ft^3/lbm

State 1 -2: Isentropic compression

$h_2 = v_1*(P_2 - P_1) + h_1$

= 159.636 Btu/lbm

State 2 -3: Constant pressure heat addition

h_3 = h@ P_3 and T_3 = 1481.3 Btu/lbm

s_3 = s@ P_3 and T_3 = 1.5758 Btu/lbm*R

State 3 -4: Isentropic Expansion

$s_3 = s_4$

$x_4 = 0.833418$

h_4 = h @ s_3 and P_1 = 964.015 Btu/lbm

Calculating heat addition:

q_{in}= (h_3 – h_2) = 1321.664 Btu/lbm

Calculating heat rejection:

q_{out} = (h_4 – h_1) = 833.835 Btu/lbm

Calculating Efficiency:

$\eta = (1 - q_{out} / q_{in})*100$

$= 36.91022$

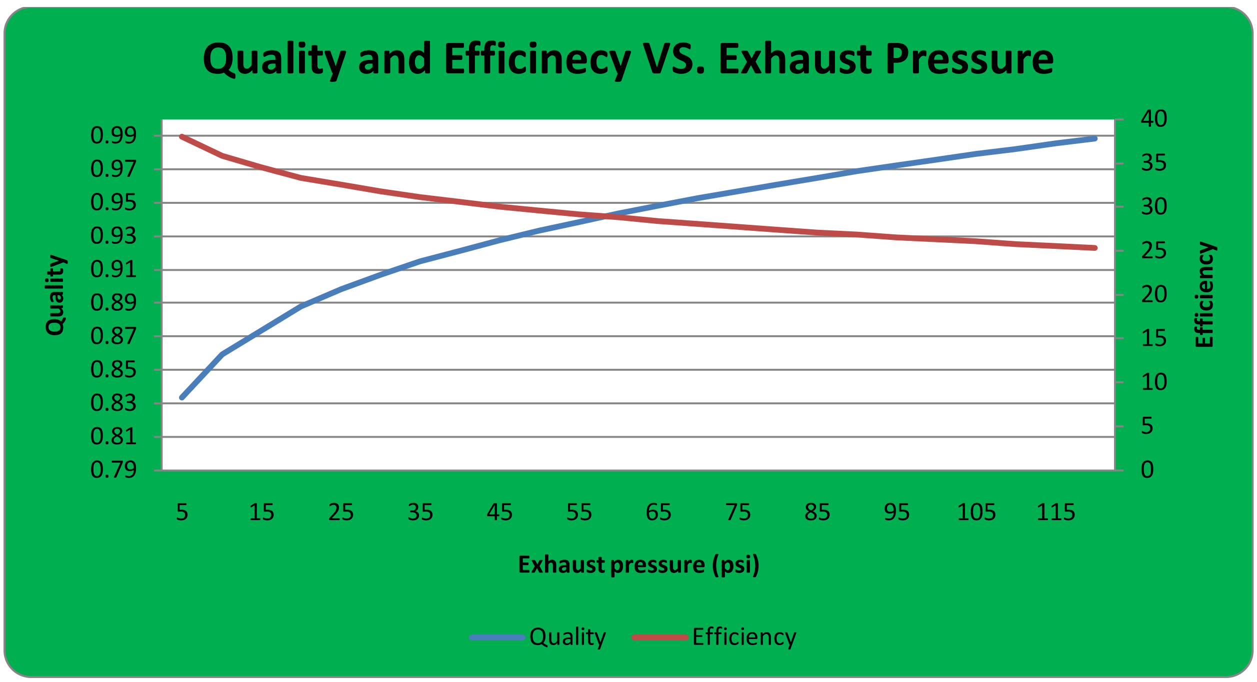
Process repeated for other case @ P_1 consecutive multiple of 5 psi.

4. Results

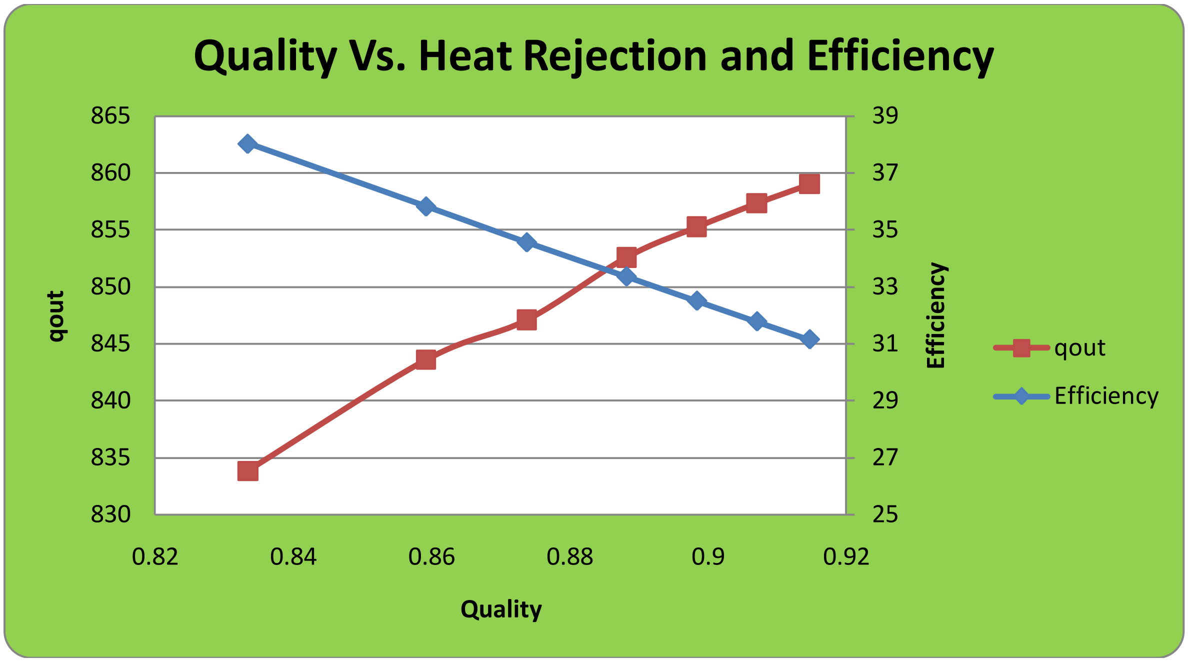
p1 (psi)	h1 Btu/lbm	h2 Btu/lbm	h3 Btu/lbm	h4 Btu/lbm	x4	Qin Btu/lbm	Qout Btu/lbm	efficiency
5	130.18	135.632	1481.3	964.015	0.833418	1345.668	833.835	38.03561
10	161.25	166.7465	1481.3	1004.843	0.859214	1314.554	843.5931	35.82664
15	181.21	186.734	1481.3	1028.316	0.873782	1294.566	847.1058	34.56449
20	196.27	201.8148	1481.3	1048.879	0.8882	1279.485	852.6095	33.36308
25	208.52	214.0788	1481.3	1063.801	0.898376	1267.221	855.2806	32.50739
30	218.93	224.4994	1481.3	1076.294	0.907062	1256.801	857.3642	31.782
35	228.03	233.6098	1481.3	1087.065	0.914684	1247.69	859.0346	31.15001
40	236.14	241.7268	1481.3	1096.247	0.921517	1239.573	860.107	30.61265
45	243.49	249.0804	1481.3	1105.059	0.927735	1232.22	861.5686	30.07995
50	250.21	255.8039	1481.3	1112.757	0.933462	1225.496	862.547	29.6165
55	256.42	262.014	1481.3	1119.816	0.93878	1219.286	863.3958	29.18841
60	262.2	267.7973	1481.3	1126.31	0.943754	1213.503	864.1104	28.79205
65	267.62	273.2173	1481.3	1132.366	0.948446	1208.083	864.7461	28.41996

70	272.72	278.3172	1481.3	1138.019	0.952888	1202.983	865.2986	28.07058
75	277.55	283.1438	1481.3	1143.337	0.957114	1198.156	865.7865	27.7401
80	282.13	287.7235	1481.3	1148.33	0.961142	1193.577	866.2	27.42819
85	286.5	292.0899	1481.3	1153.083	0.965014	1189.21	866.5826	27.12956
90	290.67	296.2563	1481.3	1157.575	0.968728	1185.044	866.9054	26.84612
95	294.67	300.2557	1481.3	1161.855	0.9723	1181.044	867.1847	26.57475
100	298.52	304.1019	1481.3	1165.962	0.975772	1177.198	867.4419	26.313
105	302.15	307.7249	1481.3	1169.741	0.97904	1173.575	867.5905	26.07286
110	305.78	311.351	1481.3	1173.633	0.982356	1169.949	867.8528	25.82131

5. Discussion



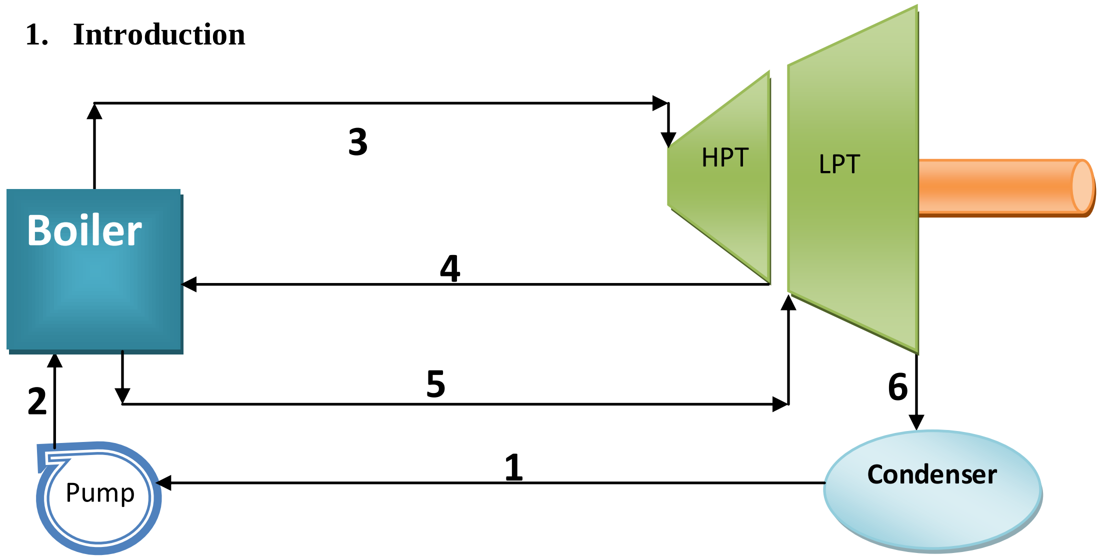
Therefore, from the above plot we can determine at give condition, exhaust pressure was found to be (P_4 = 15psi) @ required quality (x) of 87%.



From above graph we can see that the efficiency of the cycle is high when the quality of the steam is low than the desired 87%. But lower quality imples which fluid content in the expanded working fluid, which are often harmful for the turbine blade and other turbine part and also which quality cause the vapor ending up in the superheat region after the expansion in the turbine, which increases the energy removed by the condensers which create defecate in efficiency. Also to be consider, lower quality cause corrosion and higher efficiency cause erosion to turbine blade hence and optimum quality of exhaust working fluid is to be determined and maintained throughout the re occurrence of the process.

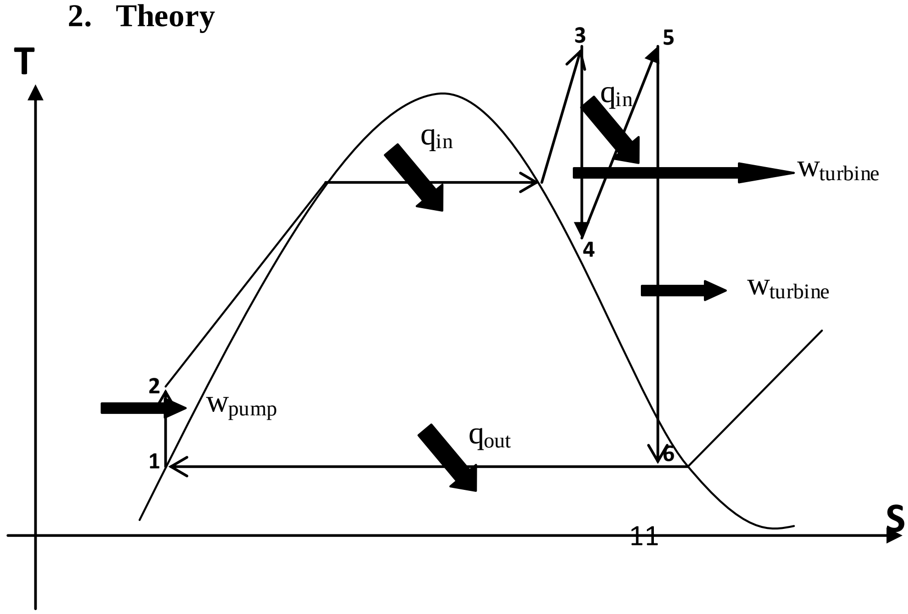
B. Ideal Reheat Rankine Cycle

1. Introduction



The effective temperature of heat addition is increased and the moisture content further reduced by using reheats in the Rankine cycle. A schematic diagram of the cycle appropriate T-s diagram is shown in the figure. High pressure, superheated steam is expanded in a high pressure turbine and the fluid then returned to a second stage boiler and super heater and reheated and is then expanded in a low-pressure turbine to the final exhaust pressure. The moisture content of the working fluid is drastically reduced by use of reheat and this approach is used in all fossil-fuelled and many nuclear power plants. The approach used to compute the work and efficiency of reheat cycles is the same as used in the example problem for the simple Rankine cycle. One calculates the work produced in each turbine separately and the required pumping work. Heat is added to the fluid at two different stages of the cycle and is given by the difference in enthalpy between states.

2. Theory



Process 1-2: The working fluid is pumped from low to high pressure, as the fluid is a liquid at this stage the pump requires little input energy.

Process 2-3: The high pressure liquid enters a boiler where it is heated at constant pressure by an external heat source to become a dry saturated vapor.

Process 3-4: The dry saturated vapor isentropic expands through a high pressure turbine, generating power. This decreases the temperature and pressure of the vapor, and some condensation may occur.

Process 4-5: The low pressure liquid enters to a second stage boiler and super heater and reheated at constant pressure to same temperature of that of inlet temperature of HPT.

Process 5-6: The dry saturated vapor is again goes isentropic expansion through a low pressure turbine, generating power. This decreases the temperature and pressure of the vapor, and more condensation may occur.

Process 6-1: The wet vapor then enters a condenser where it is condensed at a constant pressure and temperature to become a saturated liquid. The pressure and temperature of the condenser is fixed by the temperature of the cooling coils as the fluid is undergoing a phase-change.

3. Sample calculation

Inlet temperature (P_3) = 1800 psi

Inlet temperature (T_3) = 1000 ºF

Working fluid is ideal gas = water

Saturated Liquid pressure (P_1) = 15 psi

Reheat Pressure (P_4) = (0.1~0.4)* (P_3)

Assumption for ideal Reheat Rankine cycle for ideal gas:

[$P_3 = P_2$]------- (1)

[$P_4 = P_5$]------- (2)

[$P_1 = P_6$]------- (3)

[$s_1 = s_2$]------- (4)

[$s_3 = s_4$]------- (5)

[$s_5 = s_6$]------- (6)

Beginning with Fixing condense saturated liquid pressure (P_1) = 15 psi

$h_1 = h_f$ @ P_1 = 181.21 Btu/lbm

$v_1 = v_f$ @ P_1 = 0.01672 ft^3/lbm

State 1 -2: Isentropic compression

$h_2 = v_1 * (P_2 - P_1) + h_1$

= 186.734 Btu/lbm

State 2 -3: Constant pressure heat addition

h_3 = h@ P_3 and T_3 = 1481.3 Btu/lbm

s_3 = s@ P_3 and T_3 = 1.5758 Btu/lbm*R

State 3 -4: Isentropic High Pressure Turbine Expansion

$s_3 = s_4$

h_4 = h @ s_3 and P_4 = 1214.5 Btu/lbm

State 4 -5: Constant pressure heat addition

h_5 = h@ P_4 and T_3 = 1530.1 Btu/lbm

s_5 = s@ P_3 and T_3 = 1.8549 Btu/lbm*R

State 5 - 6: Isentropic Low Pressure Turbine Expansion

$s_5 = s_6$

h_6 = h @ s_6 and P_1 = 1225.528 Btu/lbm

Calculating heat addition:

$q_{in} = (h_3 - h_2) + (h_5 - h_4)$ = 1610.166 Btu/lbm

Calculating heat rejection:

$q_{out} = (h_4 - h_1) = 1044.318$ Btu/lbm

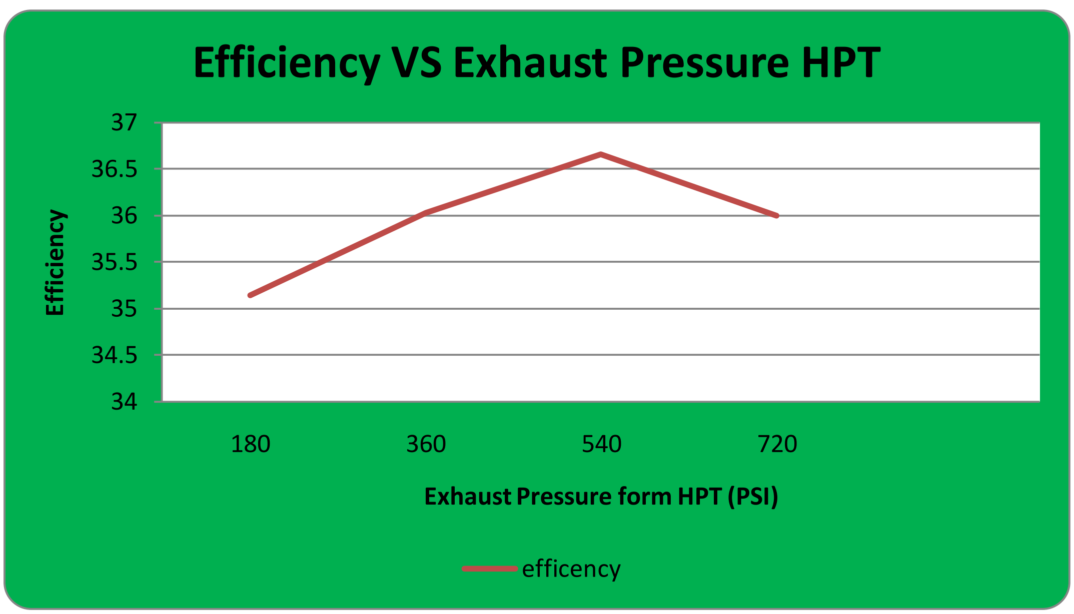
Calculating Efficiency:

$\eta = 1 - q_{out} / q_{in}$

$= 35.1421$

Process repeated for other case @ P_4 = consecutive multiple of (0.1~0.4) psi of P_3.

p1 (psi)	h1 Btu/lbm	h2 Btu/lbm	h3 Btu/lbm	h4 Btu/lbm	h5 Btu/lbm	x6	h6 Btu/lbm	Qout Btu/lbm	Qin Btu/lbm	Efficiency
15	181.21	186.734	1481.3	1214.5	1530.1	0	1225.528	1044.318	1610.166	35.142
15	181.21	186.734	1481.3	1278.149	1525.3	0	1167.429	986.219	1541.717	36.031
15	181.21	186.734	1481.3	1306.808	1521	0.985	1136.914	955.7039	1508.758	**36.656**
15	181.21	186.734	1481.3	1358.571	1515.2	0.959	1109.932	928.7222	1451.195	36.002



In above calculation we found that at P_4 = 540 psi, the reheat Rankine cycle is most efficient.

For optimization to our required criteria of exhaust quality of steam from LPT = 87%.

Assumption:

Inlet temperature (P_3) = 1800 psi

Inlet temperature (T_3) = 1000 ºF

Quality of expanded high pressure steam after expansion (x_6) = .87

Working fluid is ideal gas = water

Reheat Pressure (P_4) = 540 psi

Assumption for ideal Reheat Rankine cycle for ideal gas:

[$P_3 = P_2$]------- (1)

[$P_4 = P_5$]------- (2)

[$P_1 = P_6$]------- (3)

[$s_1 = s_2$]------- (4)

[$s_3 = s_4$]------- (5)

[$s_5 = s_6$]------- (6)

Beginning with assuming condense saturated liquid pressure (P_1) = 0.5 psi

$h_1 = h_f$ @ P_1 = 47.6228 Btu/lbm

$v_1 = v_f$ @ P_1 = 0.0161 ft^3/lbm

State 1 -2: Isentropic compression

$h_2 = v_1 * (P_2 - P_1) + h_1$

= 52.9852 Btu/lbm

State 2 -3: Constant pressure heat addition

h_3 = h@ P_3 and T_3 = 1481.3 Btu/lbm

s_3 = s@ P_3 and T_3 = 1.5758 Btu/lbm*R

State 3 -4: Isentropic High Pressure Turbine Expansion

$s_3 = s_4$

h_4 = h @ s_3 and P_4 = 1214.5 Btu/lbm

State 4 -5: Constant pressure heat addition

h_5 = h@ P_4 and T_3 = 1519.146 Btu/lbm

s_5 = s@ P_3 and T_3 = 1.728 Btu/lbm*R

State 5 - 6: Isentropic Low Pressure Turbine Expansion

$s_5 = s_6$

$s_f = 0.0925$

$s_{fg} = 1.9446$

$x_6 = (s_5 – s_f) / s_{fg} = 0.841$

$h_6 = h_{f\,+}\, x_{6*}\, h_{fg} = 929.57$ Btu/lbm

Calculating heat addition:

$q_{in}= (h_3 – h_2) + (h_5 – h_4) = 1735.695$ Btu/lbm

Calculating heat rejection:

$q_{out} = (h_4 – h_1) = 881.95$ Btu/lbm

Calculating Efficiency:

$\eta = (1 – q_{out} / q_{in})*100$

$= 49.187$

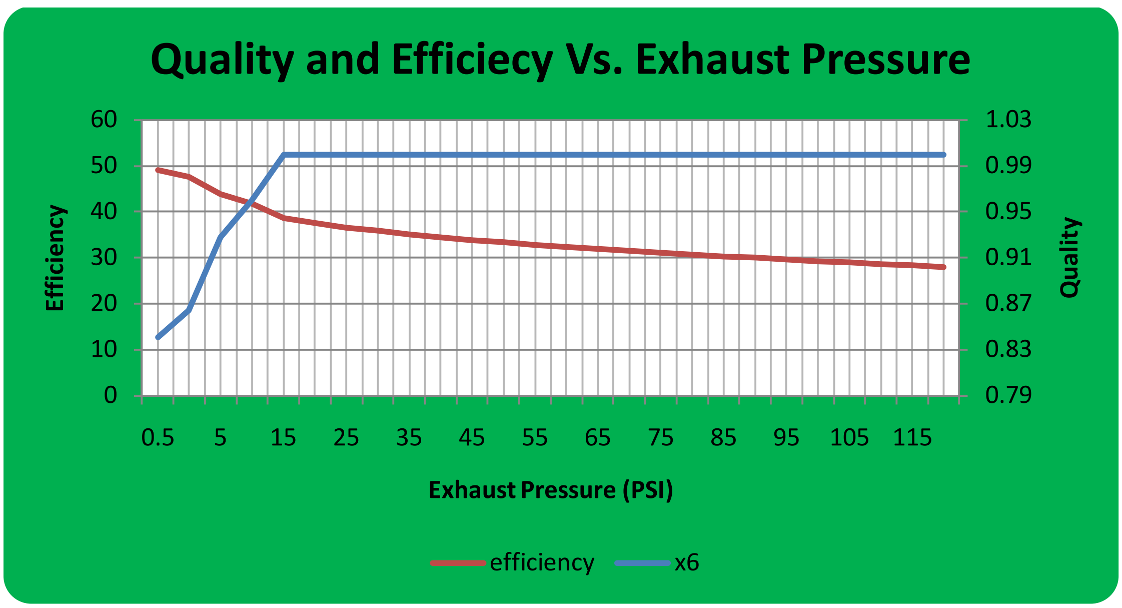
Process repeated for other case of P_1.

4. Results

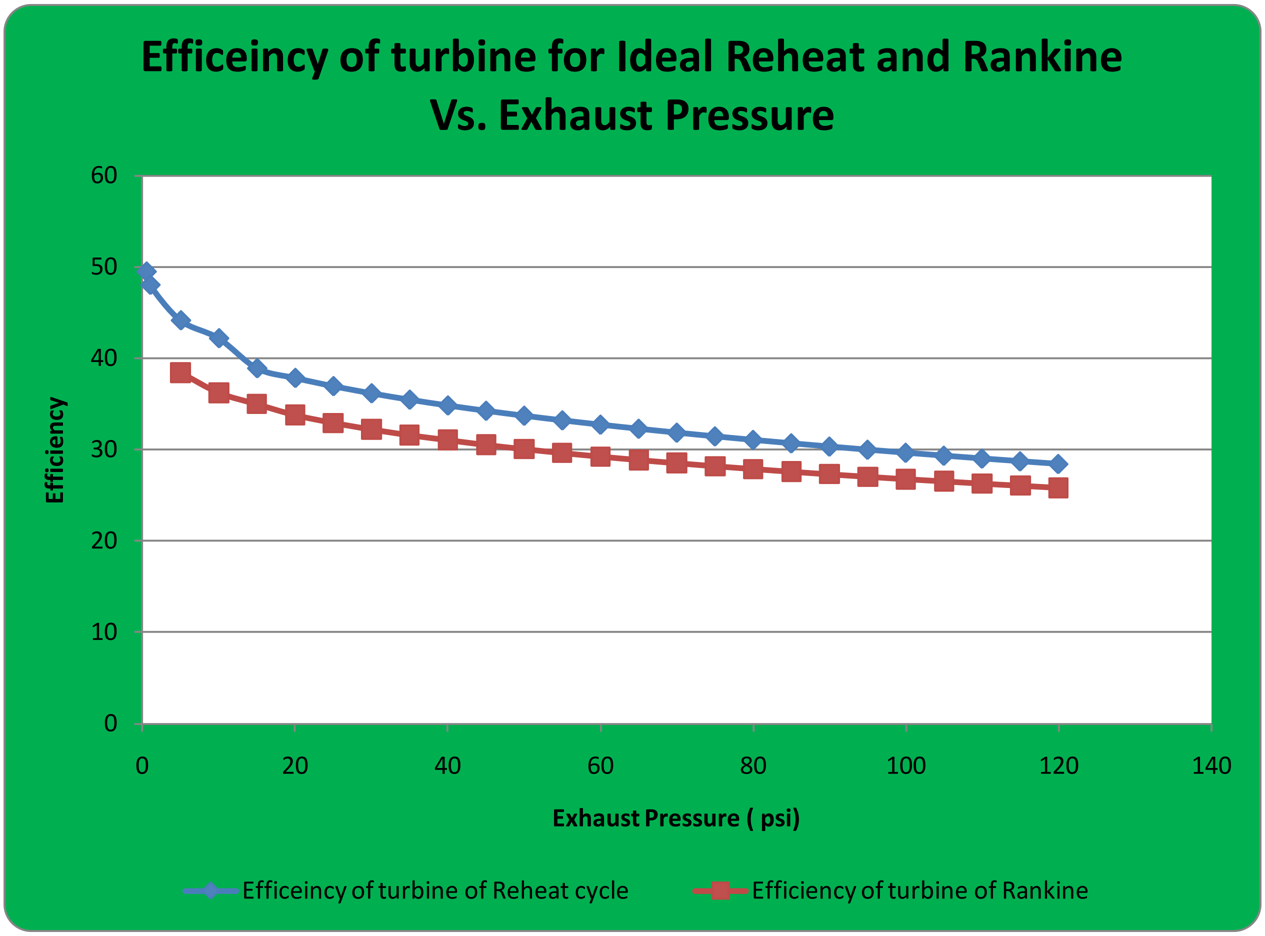
p1	h1	h2	h3	h4	h5	x6	h6	qin	qout	efficiency
0.5	47.6228	52.98522	1481.3	1211.766	1519.146	0.8410	929.5734	1735.695	881.950	49.18747
1	69.7325	75.09343	1481.3	1211.766	1519.146	0.8644	965.4009	1713.587	895.668	47.73137
5	130.18	135.632	1481.3	1211.766	1519.146	0.9280	1058.659	1653.048	928.478	43.83233
10	161.25	166.7465	1481.3	1211.766	1519.146	0.9604	1104.206	1621.934	942.956	41.86223
15	181.21	186.734	1481.3	1211.766	1519.146	1	1165.192	1601.946	983.98	38.57584
20	196.27	201.8148	1481.3	1211.766	1519.146	1	1187.961	1586.865	991.691	37.50629
25	208.52	214.0788	1481.3	1211.766	1519.146	1	1206.738	1574.601	998.218	36.60501
30	218.93	224.4994	1481.3	1211.766	1519.146	1	1222.84	1564.181	1003.91	35.81882
35	228.03	233.6098	1481.3	1211.766	1519.146	1	1237.007	1555.071	1008.97	35.11698
40	236.14	241.7268	1481.3	1211.766	1519.146	1	1249.702	1546.954	1013.56	34.48013
45	243.49	249.0804	1481.3	1211.766	1519.146	1	1261.235	1539.6	1017.74	33.89549
50	250.21	255.8039	1481.3	1211.766	1519.146	1	1271.824	1532.876	1021.61	33.35313
55	256.42	262.014	1481.3	1211.766	1519.146	1	1281.63	1526.666	1025.2	32.84652
60	262.2	267.7973	1481.3	1211.766	1519.146	1	1290.773	1520.883	1028.57	32.37004
65	267.62	273.2173	1481.3	1211.766	1519.146	1	1299.348	1515.463	1031.72	31.91997
70	272.72	278.3172	1481.3	1211.766	1519.146	1	1307.429	1510.363	1034.70	31.49269
75	277.55	283.1438	1481.3	1211.766	1519.146	1	1315.078	1505.537	1037.52	31.08584

80	282.13	287.7235	1481.3	1211.766	1519.146	1	1322.343	1500.957	1040.21	30.69665
85	286.5	292.0899	1481.3	1211.766	1519.146	1	1329.267	1496.59	1042.76	30.32385
90	290.67	296.2563	1481.3	1211.766	1519.146	1	1335.883	1492.424	1045.21	29.96545
95	294.67	300.2557	1481.3	1211.766	1519.146	1	1342.22	1488.425	1047.55	29.62019
100	298.52	304.1019	1481.3	1211.766	1519.146	1	1348.306	1484.578	1049.78	29.28728
105	302.15	307.7249	1481.3	1211.766	1519.146	1	1354.16	1480.955	1052.01	28.96408
110	305.78	311.351	1481.3	1211.766	1519.146	1	1359.803	1477.329	1054.02	28.65348
115	309.16	314.727	1481.3	1211.766	1519.146	1	1365.251	1473.953	1056.09	28.34979
120	312.55	318.1129	1481.3	1211.766	1519.146	1	1370.518	1470.567	1057.96	28.05716

5. Discussion



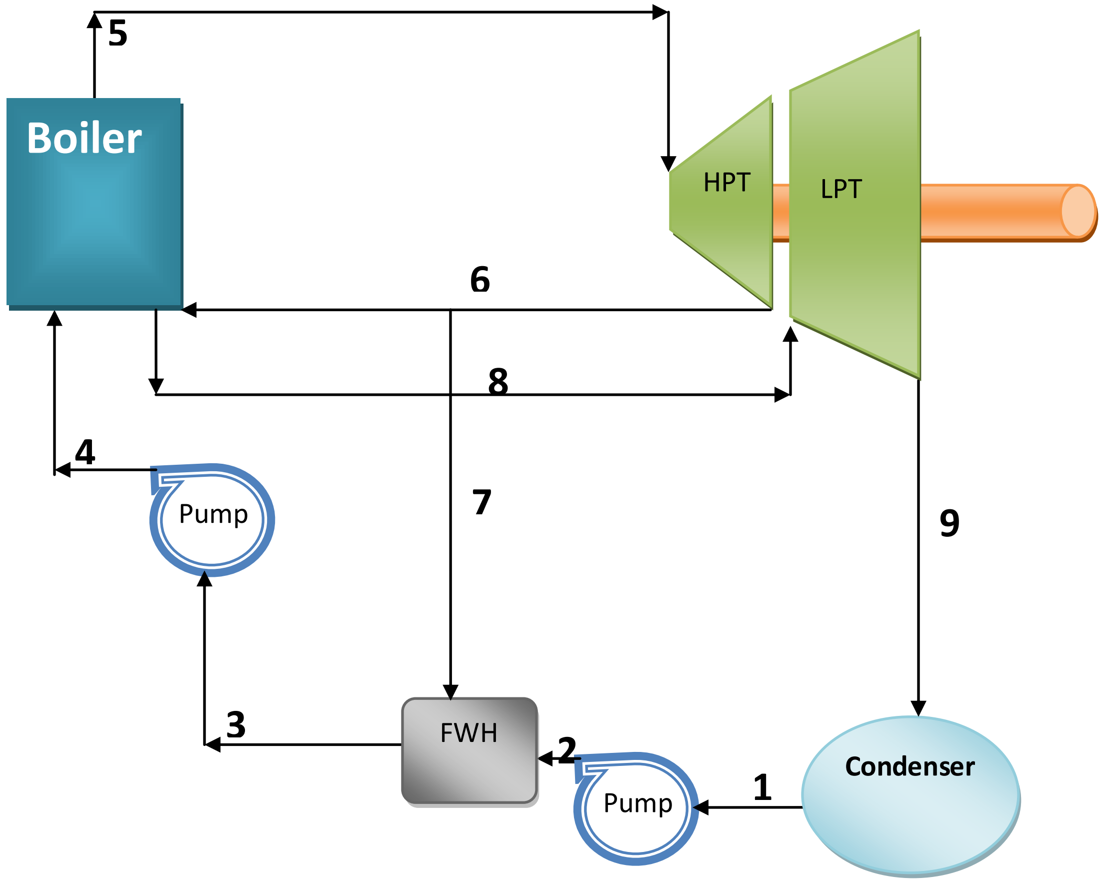
Therefore, from the above plot we can determine at give condition, exhaust pressure was found to be ***<u>(P_6 = 1.33 psi)</u>*** @ required quality (x) of 87% and efficiency of ***<u>46.43%.</u>*** From above graph we can see that the efficiency of the cycle is high when the quality of the steam is low than the desired 87%. At higher pressure the quality of the steam was in super heated region which was undesireable in material and desing point of view.



And also in the graph above we can clearly visualize how the net work of the turbine has increase than that of ideal Rankine cycle in susbstantial range.

C. Regenerative Ideal Rankine Cycle

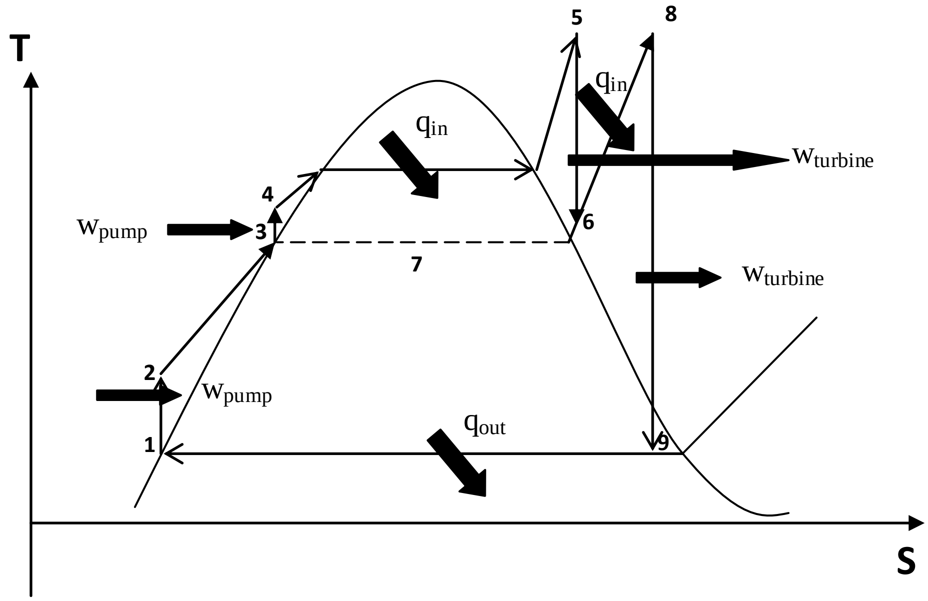
1. Introduction



A regeneration cycle in steam power turbine cycle is a process of extracting steam from the turbine at various point and is heated by a device called feed water heater or regenerator instead, and expand further in turbine to produce more work is called regenerative cycle. Regeneration not only improves cycle efficiency, but also provides a convenient means of decelerating the feed water, removing the air that leaks in at the condenser, to prevent corrosion in the boiler and also help control the large specific volumetric flow rate of the steam at the final stages of the turbine due to low pressure.

For the case study we are considering an open feed water heater which is basically a mixing chamber where the steam extracted from the turbine mixes with the feed water exiting the pump. Ideally, the mixture leaves the heaters as saturated liquid at the heater at the heater pressure. Depending open the requirement, regeneration can be of various stages.

2. Theory



Process 1-2: The working fluid is pumped from low to reheat pressure, as the fluid is a liquid at this stage the pump requires little input energy.

Process 2-3: The pressure liquid enters a feed water heater where it is mixed with steam extracted from HPT at constant pressure to become a saturated liquid at reheat pressure.

Process 3-4: The saturated liquid is pumped to the boiler through an isentropic compression process.

Process 4-5: The pumped liquid is supplied to the boiler where the heated is added to change the phase into superheated steam at constant pressure.

Process 5-6: The dry saturated vapor goes isentropic expansion through a high pressure turbine, generating power.

Process 6-7: Fractional mass of the steam from HPT is extracted and supplied to feed water heater.

Process 6-8: Constant pressure heat addition at low pressure turbine and reheated to the same temperature at that of high pressure turbine inlet temperature.

Process 8-9: The dry saturated vapor goes isentropic expansion through a high pressure turbine, generating power. Here, the steam at low pressure drop more in temperature and pressure and steam condense to some extent to saturated liquid.

Process 9-1: The wet vapor then enters a condenser where it is condensed at a constant pressure and temperature to become a saturated liquid. The pressure and temperature of the condenser is fixed by the temperature of the cooling coils as the fluid is undergoing a phase-change.

3. Calculation

<u>One- open water feed heater:</u>

Inlet temperature (P_5) = 1800 psi

Inlet temperature (T_3) = 1000 ºF

Working fluid is ideal gas = water

Saturated Liquid pressure (P_1) = 1.33 psi

Reheat Pressure (P_6) = 540

Assumption for ideal Reheat Rankine cycle for ideal gas:

[$P_2 = P_3 = P_6 = P_8 = P_7$]------- (1)

[$P_4 = P_5$]------- (2)

[$P_1 = P_9$]------- (3)

[$s_1 = s_2$]------- (4)

[$s_3 = s_4$]------- (5)

[$s_5 = s_6$]------- (6)

[$s_8 = s_9$]------- (6)

Beginning with Fixing saturated liquid at pressure (P_1) = 1.333 psi

$h_1 = h_f$ @ P_1 = 79.5233 Btu/lbm

$v_1 = v_f$ @ P_1 = 0.0162 ft^3/lbm

State 1 -2: Isentropic compression

$h_2 = v_1*(P_2 - P_1) + h_1$

$= 81.1384$ Btu/lbm

State 2 -3: Saturated liquid at pressure (P_3) = 540 psi

$h_3 = h_f$ @ $P_3 = 458.71$ Btu/lbm

$v_3 = v_f$ @ $P_3 = 0.0199$ ft^3/lbm

State 3 -4: Isentropic compression

$h_4 = v_3*(P_4 - P_3) + h_3$

$= 463.354346$ Btu/lbm

State 4 -5: Constant pressure heat addition in boiler

h_5 = h@ P_5 and T_5 = 1480.5856 Btu/lbm

s_5 = s@ P_5 and T_5 = 1.5753 Btu/lbm*R

State 5 - 6: Isentropic high pressure turbine expansion

$s_5 = s_6$

h_6 = h@ P_6 and s_6 = 1324.75 Btu/lbm

State 6 - 8: Constant pressure reheats addition in boiler

h_8 = h@ P_6 and T_5 = 1519.14 Btu/lbm

s_8 = s@ P_6 and T_5 = 1.728 Btu/lbm*R

State 7 - 8: Isentropic low pressure turbine expansion

$s_8 = s_9$

$$x = \frac{(s_8 - s_f)}{s_{fg}} = \frac{(1.728 - 0.1499)}{1.804} = 0.8747$$

h_f and h_{fg} @ P_1

$h_9 = h_f + x* h_{fg} = 79.52 + 0.8747*1030.4752 = 980.96$ Btu/lbm

For unit mass flow rate:

Mass extraction from stage 6 via (7-3) to feed water heater:

$h_3 = m_7 h_7 + (1- m_7)* h_2$…………..[$h_6 = h_7$]

$$m_7 = \frac{(h_3 - h_2)}{(h_7 - h_2)} = 0.303 \text{ per lbm}$$

Calculating heat addition:

$q_{in} = (h_5 - h_4) + (1 - m_7)(h_7 - h_6) = 1152.60$ Btu/lbm

Calculating heat rejection:

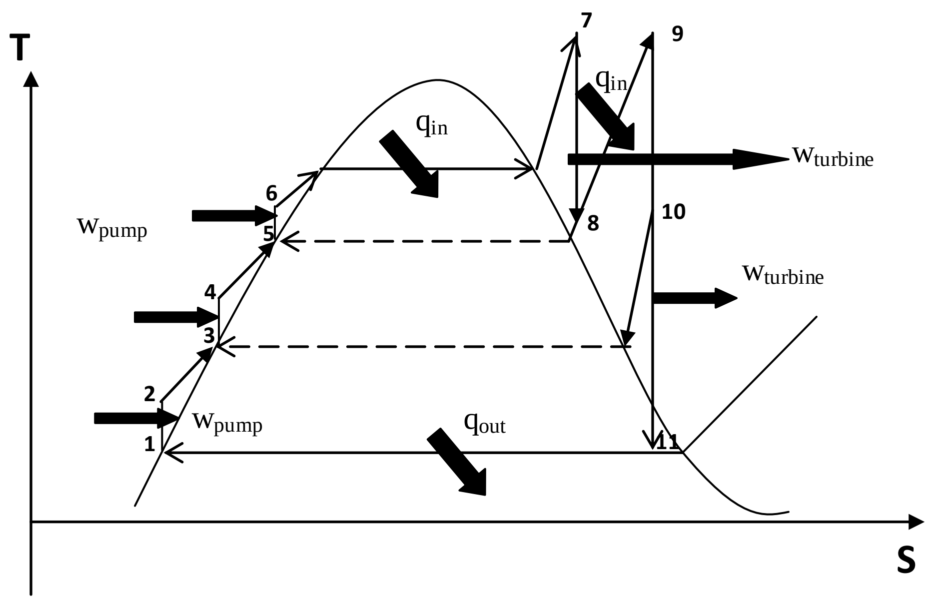
$q_{out} = (1- m_7)(h_9 - h_1) = 627.25$ Btu/lbm

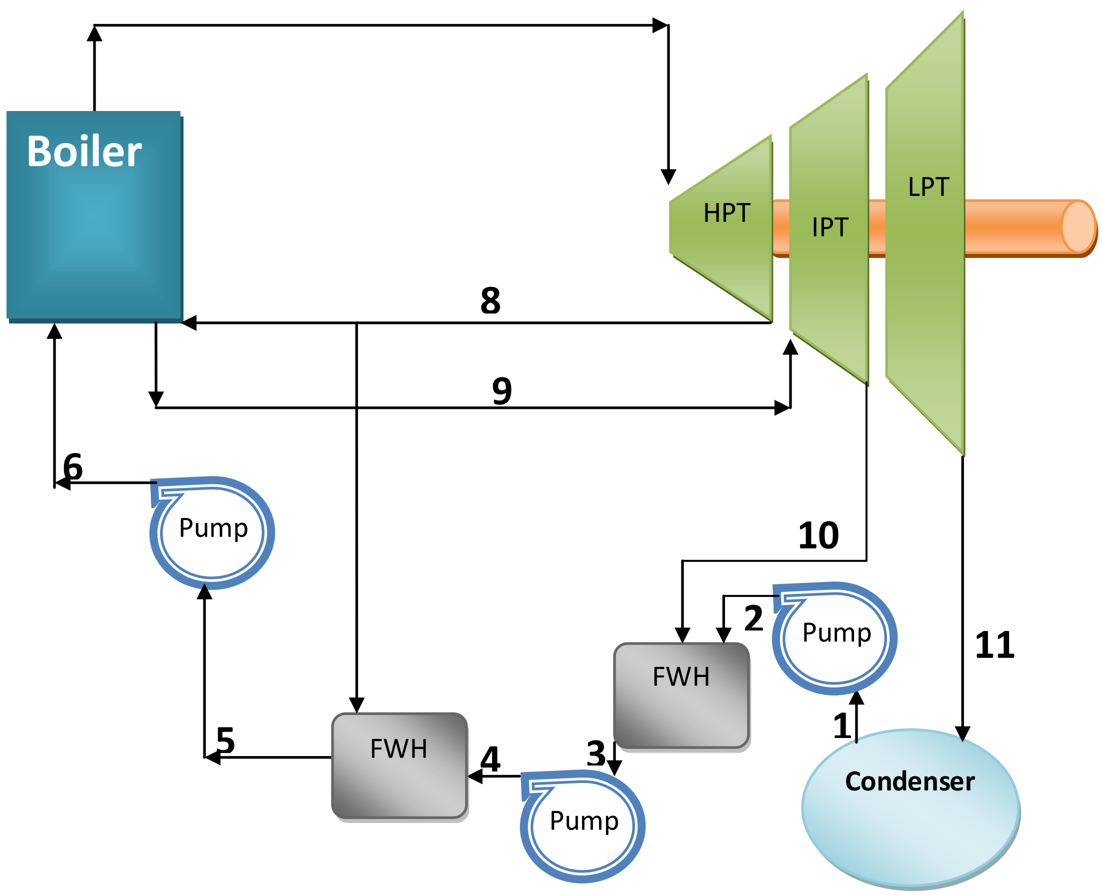
Calculating Efficiency:

$\eta = (1 - q_{out} / q_{in})*100$

$= 45.53$

Two- open feed water heater:





Inlet temperature (P_7) = 1800 psi

Inlet temperature (T_7) = 1000 ºF

Working fluid is ideal gas = water

Saturated Liquid pressure (P_1) = 1.33 psi

Reheat Pressure (P_8) = 540 psi

Intermediate mass extraction Pressure (P_{10}) = 270.5 psi

Assumption for ideal Reheat Rankine cycle for ideal gas:

[$P_2 = P_3 = P_{10}$]------- (1)

[$P_4 = P_5 = P_8 = P_9$]------- (2)

[$P_1 = P_{11}$ and $P_6 = P_7$]------- (3)

[$s_1 = s_2$]------- (4)

[$s_3 = s_4$]------- (5)

[$s_5 = s_6$]------- (6)

[$s_7 = s_8$ and $s_9 = s_{10} = s_{11}$]------- (7)

Beginning with Fixing saturated liquid at pressure (P_1) = 1.333 psi

$h_1 = h_f$ @ P_1 = 79.5233 Btu/lbm

$v_1 = v_f$ @ P_1 = 0.0162 ft^3/lbm

State 1 -2: Isentropic compression

$h_2 = v_1*(P_2 - P_1) + h_1$

= 80.33088508 Btu/lbm

State 2 -3: Saturated liquid at pressure (P_3) = 270.66 psi

$h_3 = h_f$ @ P_3 = 383.801 Btu/lbm

$v_3 = v_f$ @ P_3 = 0.0188 Btu/lbm*R

State 3 -4: Isentropic compression

$h_4 = v_3*(P_4 - P_3) + h_3$

= 384.7391945 Btu/lbm

State 4 -5: Saturated liquid at pressure (P_5) = 540 psi

$h_5 = h_f$ @ P_3 = 458.71 Btu/lbm

$v_5 = v_f$ @ P_3 = 0.0199 ft^3/lbm

State 5 -6: Isentropic compression

$h_6 = v_3*(P_4 - P_3) + h_3$

= 464.34 Btu/lbm

State 6 -7: Constant pressure heat addition in boiler (P_7) = 1800 psi and (T_7) = 1000 F

h_7 = h@ P_7 and T_7 = 1480.5856 Btu/lbm

s_7 = s@ P_7 and T_7 = 1.5753 Btu/lbm*R

State 7 - 8: Isentropic high pressure turbine expansion

$s_7 = s_8$

h_8 = h@ P_8 and s_8 = 1324.75 Btu/lbm

State 8 - 9: Constant pressures reheat addition in boiler (P_9) = 1800 psi and (T_9) = 1000 F

h_9 = h@ P_9 and T_9 = 1519.14 Btu/lbm

s_9 = s@ P_9 and T_9 = 1.728 Btu/lbm*R

State 9- 11: Isentropic Intermediate pressure turbine expansion

$s_9 = s_{11}$

$$x = \frac{(s_8 - s_f)}{s_{fg}} = \frac{(1.728 - 0.1499)}{1.804} = 0.8747$$

h_f and h_{fg} @ P_1

$h_9 = h_f + x* h_{fg}$ = 79.52 + 0.8747*1030.4752 = 980.96 Btu/lbm

For unit mass flow rate:

Mass extraction from stage 8 via (8-5) to feed water heater:

$h_5 = m_8 h_8 + (1- m_8)* h_4$

$$m_8 = \frac{(h_5 - h_4)}{(h_8 - h_4)} = 0.078095 \text{ per lbm}$$

Mass extraction from stage 10 via (10-3) to feed water heater:

$h_3(1- m_8) = m_{10}\, h_{10} + (1- m_8 - m_{10})^{*}\, h_2$

$$m_{10} = (1 - m_8)\,\frac{(h_3 - h_2)}{(h_{10} - h_2)} = 0.2089 \text{ per lbm}$$

Calculating heat addition:

$q_{in} = (h_7 - h_6) + (1 - m_8)(h_9 - h_8) = 1195.335$ Btu/lbm

Calculating heat rejection:

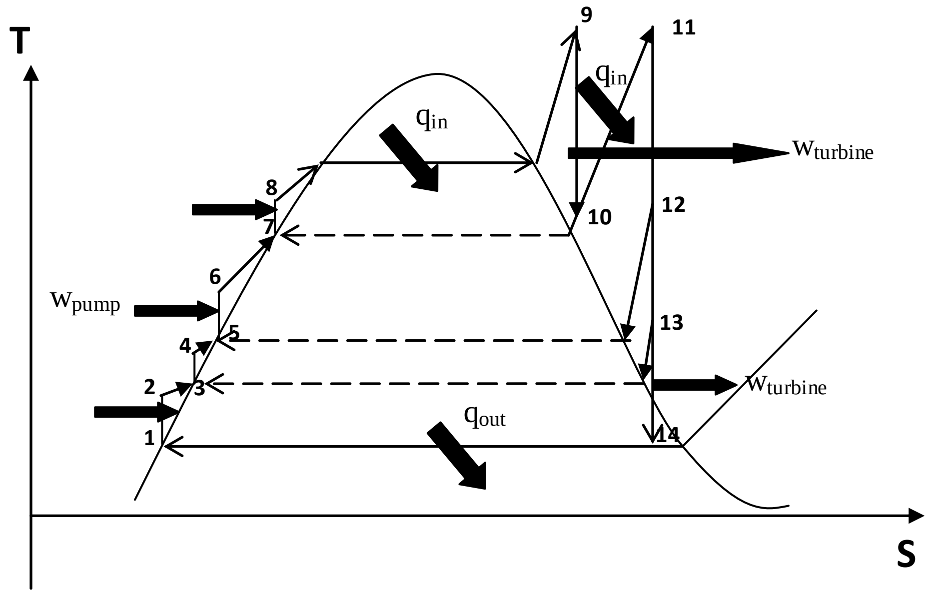
$q_{out} = (1- m_8 - m_{10})(h_{11} - h_1) = 642.125$ Btu/lbm

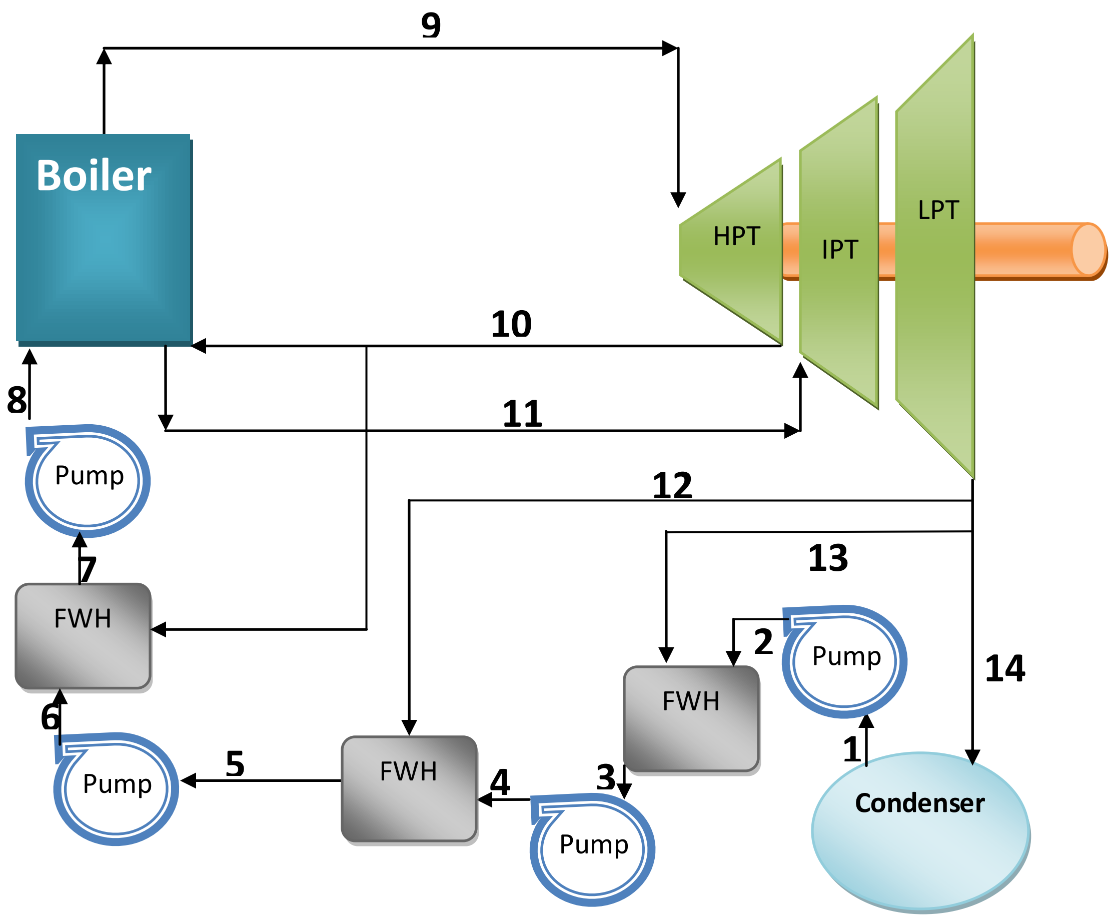
Calculating Efficiency:

$\eta = (1 - q_{out} / q_{in})*100$

$= 46.27$

Three- open feed water heater:





Inlet temperature (P_9) = 1800 psi

Inlet temperature (T_9) = 1000 ºF

Working fluid is ideal gas = water

Saturated Liquid pressure (P_1) = 1.33 psi

Reheat Pressure (P_{10}) = 540 psi

Intermediate mass extraction Pressure (P_{12}) = 270.5 psi

First low pressure mass extraction Pressure (P_{13}) = 135.25 psi

Assumption for ideal Reheat Rankine cycle for ideal gas:

[$P_2 = P_3 = P_{13}$]------- (1)

[$P_4 = P_5 = P_{12}$]------- (2)

[$P_1 = P_{14}$ and $P_6 = P_7 = P_{10} = P_{11}$ and $P_8 = P_9$]------- (3)

[$s_9 = s_{10}$]------- (4)

[$s_{11} = s_{12} = s_{13} = s_{14}$]------- (5)

Beginning with Fixing saturated liquid at pressure (P_1) = 1.333 psi

$h_1 = h_f$ @ P_1 = 79.5233 Btu/lbm

$v_1 = v_f$ @ P_1 = 0.0162 ft^3/lbm

State 1 -2: Isentropic compression

$h_2 = v_1*(P_2 - P_1) + h_1$

= 79.92 Btu/lbm

State 2 -3: Saturated liquid at pressure (P_3) = 135.25 psi

$h_3 = h_f$ @ P_3 = 322.39 Btu/lbm

$v_3 = v_f$ @ P_3 = 0.018 ft^3/lbm

State 3 -4: Isentropic compression

$h_4 = v_3*(P_4 - P_3) + h_3$

= 323.73 Btu/lbm

State 4 -5: Saturated liquid at pressure (P_5) = 270 psi

$h_5 = h_f$ @ P_5 = 383.88 Btu/lbm

$v_5 = v_f$ @ P_5 = 0.0188 ft^3/lbm

State 5-6: Isentropic compression

$h_6 = v_3*(P_4 - P_3) + h_5$

= 384.73 Btu/lbm

State 6-7: Saturated liquid at pressure (P_7) = 540 psi

$h_7 = h_f$ @ P_7 = 458.71 Btu/lbm

$v_7 = v_f$ @ P_7 = 0.01882 Btu/lbm*R

State 7-8: Isentropic compression

$h_8 = v_3*(P_4 - P_3) + h_5$

= 463.35 Btu/lbm

State 8-9: Constant pressure heat addition in boiler

h_9 = h@ P_9 and T_9 = 1480.5856 Btu/lbm

s_9 = s@ P_9 and T_9 = 1.5753 Btu/lbm*R

State 9- 10: Isentropic high pressure turbine expansion

$s_9 = s_{10}$

h_{10} = h@ P_{10} and s_{10} = 1324.75 Btu/lbm

State 10-11: Constant pressure reheats addition in boiler

h_{11} = h@ P_{10} and T_5 = 1519.14 Btu/lbm

s_{11} = s@ P_{10} and T_5 = 1.728 Btu/lbm*R

State 11 - 14: Isentropic low pressure turbine expansion

$s_{11} = s_{14}$

$$x = \frac{(s_8 - s_f)}{s_{fg}} = \frac{(1.728 - 0.1499)}{1.804} = 0.8747$$

h_f and h_{fg} @ P_1

$h_9 = h_f + x^* h_{fg} = 79.52 + 0.8747*1030.4752 = 980.96$ Btu/lbm

For unit mass flow rate:

Mass extraction from stage 10 via (10-7) to feed water heater:

$h_7 = m_{10} h_{10} + (1- m_{10})^* h_6$

$$m_{10} = \frac{(h_7 - h_6)}{(h_{10} - h_6)} = 0.078695 \text{ per lbm}$$

Mass extraction from stage 12 via (12-5) to feed water heater:

$h_5(1- m_{10}) = m_{12} h_{12} + (1- m_{10} - m_{12})^* h_4$

$$m_{12} = (1 - m_{10}) \frac{(h_5 - h_4)}{(h_{12} - h_4)} = 0.0505 \text{ per lbm } [h_{12} = h \text{ @}P_5 \text{ and } S_{11}]$$

Mass extraction from stage 13 via (13-3) to feed water heater:

$h_3(1- m_{10} - m_{12}) = m_{13} h_{13} + (1- m_{12} - m_{10} - m_{13})^* h_2$

$$m_{13} = (1 - m_{10} - m_{12}) \frac{(h_3 - h_2)}{(h_{13} - h_2)} = 0.1686 \text{ per lbm } [h_{13} = h \text{ @}P_3 \text{ and } S_{11}]$$

Calculating heat addition:

$q_{in} = (h_9 - h_8) + (1 - m_{10}) (h_{11} - h_{10}) = 1196.32$ Btu/lbm

Calculating heat rejection:

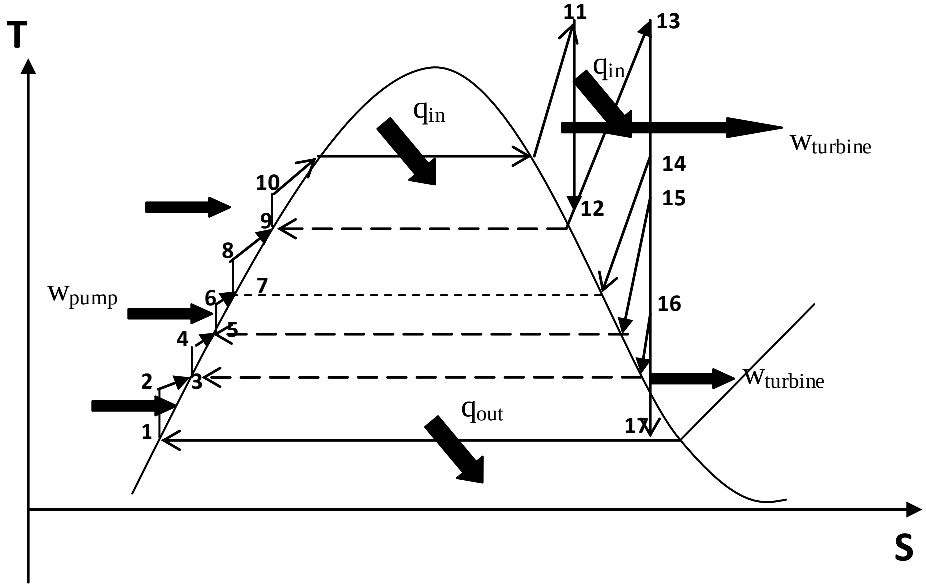
$q_{out} = (1 - m_{10} - m_{12} - m_{13})^* (h_{11} - h_1) = 632.94$ Btu/lbm

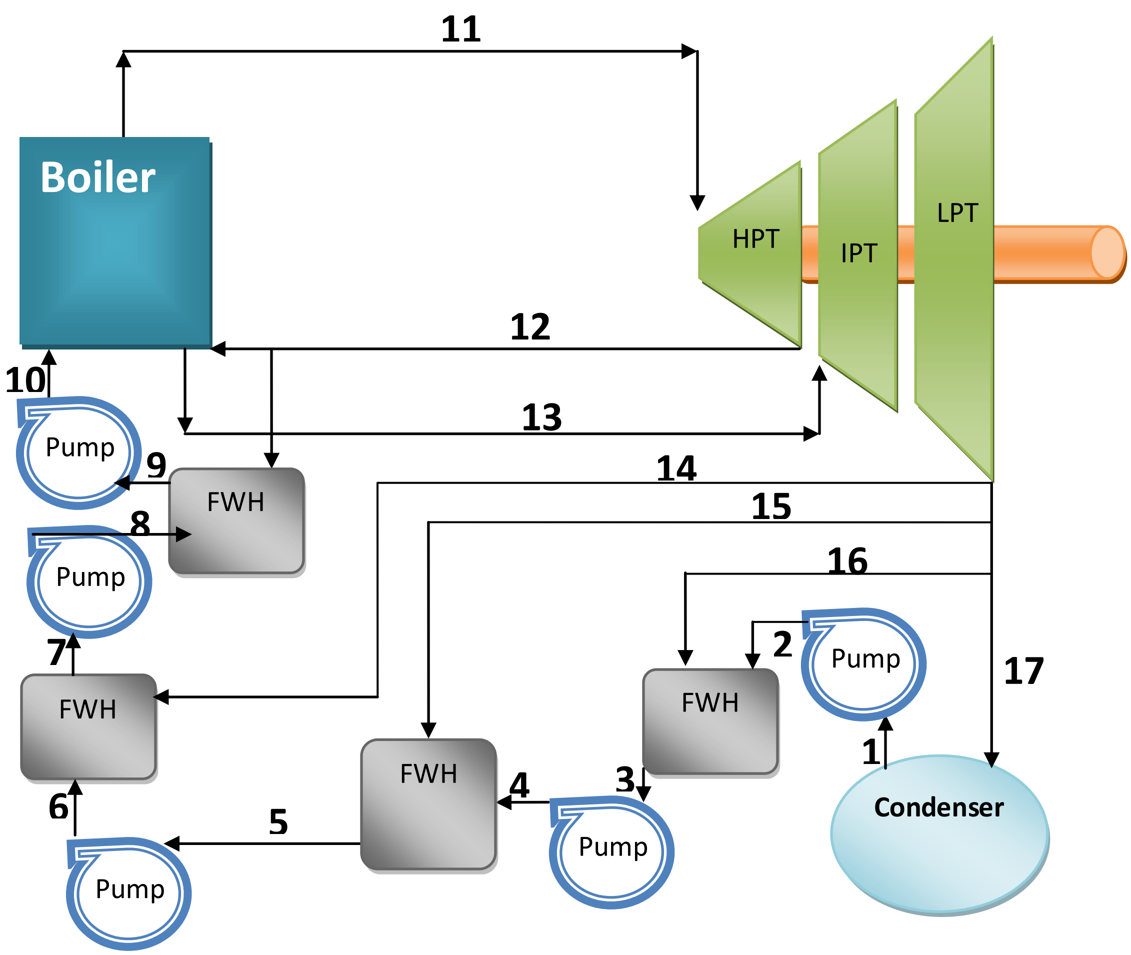
Calculating Efficiency:

$\eta = (1 - q_{out} / q_{in})*100$

$= 47.09$

Four open feed water heater:





Inlet temperature (P_{11}) = 1800 psi

Inlet temperature (T_{11}) = 1000 ºF

Working fluid is ideal gas = water

Saturated Liquid pressure (P_1) = 1.33 psi

Reheat Pressure (P_{12}) = 540 psi

Intermediate mass extraction Pressure (P_{14}) = 270.5 psi

First low pressure mass extraction Pressure (P_{15}) = 135.25 psi

Second low pressure mass extraction Pressure (P_{16}) = 68.66 psi

Assumption for ideal Reheat Rankine cycle for ideal gas:

[$P_2 = P_3 = P_{16}$]------- (1)

[$P_4 = P_5 = P_{15}$ and $P_6 = P_7 = P_{14}$ and $P_8 = P_9 = P_{12} = p_{13}$]------- (2)

[$P_1 = P_{17}$ and $P_{10} = P_{11}$]------- (3)

[$s_5 = s_6$]------- (4)

[$s_{11} = s_{12}$ and $s_{13} = s_{14} = s_{15} = s_{16} = s_{17}$]------- (5)

Beginning with Fixing saturated liquid at pressure (P_1) = 1.333 psi

$h_1 = h_f$ @ P_1 = 79.5233 Btu/lbm

$v_1 = v_f$ @ P_1 = 0.0162 ft^3/lbm

State 1 -2: Isentropic compression

$h_2 = v_1*(P_2 - P_1) + h_1$

= 79.72 Btu/lbm

State 2 -3: Saturated liquid at pressure (P_3) = 68.66 psi

$h_3 = h_f$ @ P_3 = 271.23 Btu/lbm

$v_3 = v_f$ @ P_3 = 0.0175 ft^3/lbm

State 3 -4: Isentropic compression

$h_4 = v_3*(P_4 - P_3) + h_3$

= 271.44 Btu/lbm

State 4 -5: Saturated liquid at pressure

$h_5 = h_f$ @ $P_5 = 322.29$ Btu/lbm

$v_5 = v_f$ @ $P_5 = 0.018$ Btu/lbm*R

State 5 -6: Isentropic compression

$h_6 = v_5*(P_6 - P_5) + h_5$

$= 323.73$ Btu/lbm

State 6-7: Saturated liquid at pressure

$h_7 = h_f$ @ $P_6 = 383.88$ Btu/lbm

$v_7 = v_f$ @ $P_6 = 0.0188$ Btu/lbm*R

State 7-8: Isentropic compression

$h_8 = v_7*(P_8 - P_7) + h_7$

$= 384.73$ Btu/lbm

State 8-9: Saturated liquid at pressure

$h_9 = h_f$ @ $P_8 = 458.71$ Btu/lbm

$v_9 = v_f$ @ $P_8 = 0.01882$ Btu/lbm*R

State 9-10: Isentropic compression

$h_{10} = v_9*(P_{10} - P_9) + h_9$

$= 463.35$ Btu/lbm

State 10-11: Constant pressure heat addition in boiler

$h_{11} = h$@ P_{11} and $T_{11} = 1480.5856$ Btu/lbm

$s_{10} = s$@ P_{11} and $T_{11} = 1.5753$ Btu/lbm*R

State 11- 12: Isentropic high pressure turbine expansion

$s_{11} = s_{12}$

$h_{12} = h$@ P_8 and $s_{10} = 1324.75$ Btu/lbm

State 12-13: Constant pressure reheats addition in boiler

h_{12} = h@ P_8 and T_5 = 1519.14 Btu/lbm

s_{12} = s@ P_8 and T_5 = 1.728 Btu/lbm*R

State 13 - 17: Isentropic low pressure turbine expansion

$s_{11} = s_{17}$

$$x = \frac{(s_8 - s_f)}{s_{fg}} = \frac{(1.728 - 0.1499)}{1.804} = 0.8747$$

h_f and h_{fg} @ P_1

$h_9 = h_f + x^* h_{fg}$ = 79.52 + 0.8747*1030.4752 = 980.96 Btu/lbm

For unit mass flow rate:

Mass extraction from stage 12 via (12-79 to feed water heater:

$h_9 = m_{12} h_{12} + (1 - m_{12})^* h_8$

$$m_{12} = \frac{(h_9 - h_8)}{(h_{12} - h_8)} = 0.07863 \text{ per lbm } [h_{12} = h \text{ @}P_9 \text{ and } S_{11}]$$

Mass extraction from stage 14 via (14-7) to feed water heater:

$h_7(1 - m_{12}) = m_{14} h_{14} + (1 - m_{12} - m_{14})^* h_6$

$$m_{14} = (1 - m_{12}) \frac{(h_7 - h_6)}{(h_{14} - h_6)} = 0.051341 \text{ per lbm } [h_{14} = h \text{ @}P_7 \text{ and } S_{11}]$$

Mass extraction from stage 15 via (15-5) to feed water heater:

$h_5(1 - m_{12} - m_{14}) = m_{15} h_{15} + (1 - m_{12} - m_{14} - m_{15})^* h_4$

$$m_{15} = (1 - m_{12} - m_{14}) \frac{(h_5 - h_4)}{(h_{15} - h_4)} = 0.038562 \text{ per lbm } [h_{13} = h \text{ @}P_3 \text{ and } S_{11}]$$

Mass extraction from stage 15 via (15-5) to feed water heater:

$h_3(1 - m_{12} - m_{14} - m_{15}) = m_{16} h_{16} + (1 - m_{12} - m_{14} - m_{15} - m_{16})^* h_4$

$$m_{16} = (1 - m_{12} - m_{14} - m_{15}) \frac{(h_3 - h_2)}{(h_{16} - h_2)} = 0.138058 \text{ per lbm}$$

$[h_{13} = h$ @P_3 and $S_{11}]$

Calculating heat addition:

$q_{in} = (h_{11} - h_{10}) + (1 - m_{12})\,(h_{13} - h_{12}) = 1196.32$ Btu/lbm

Calculating heat rejection:

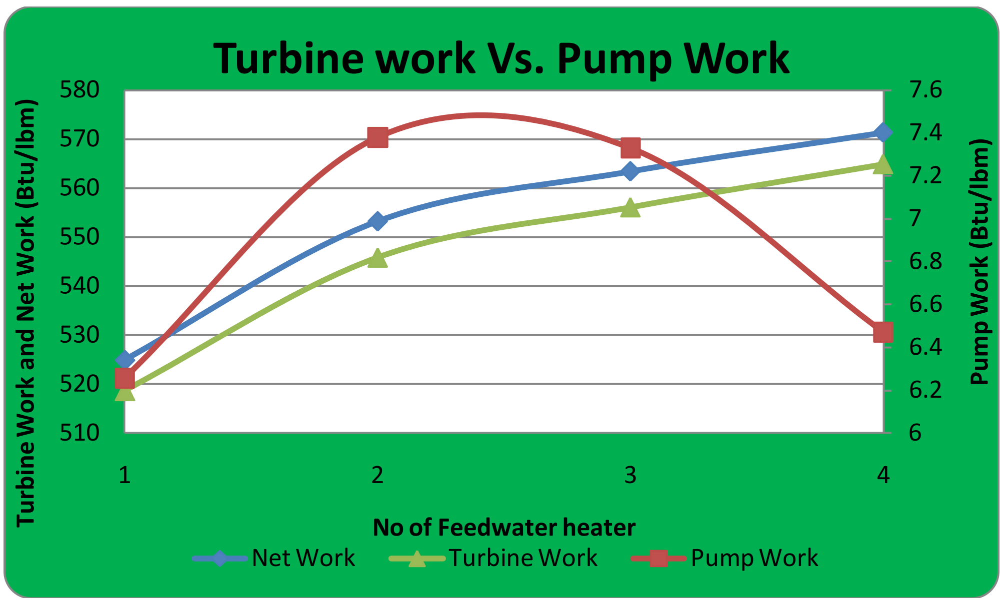
$q_{out} = (1 - m_{10} - m_{12} - m_{13}) * (h_{11} - h_1) = 625.0125$ Btu/lbm

Calculating Efficiency:

$\eta = (1 - q_{out} / q_{in}) * 100$

$= 47.75$

4. Results



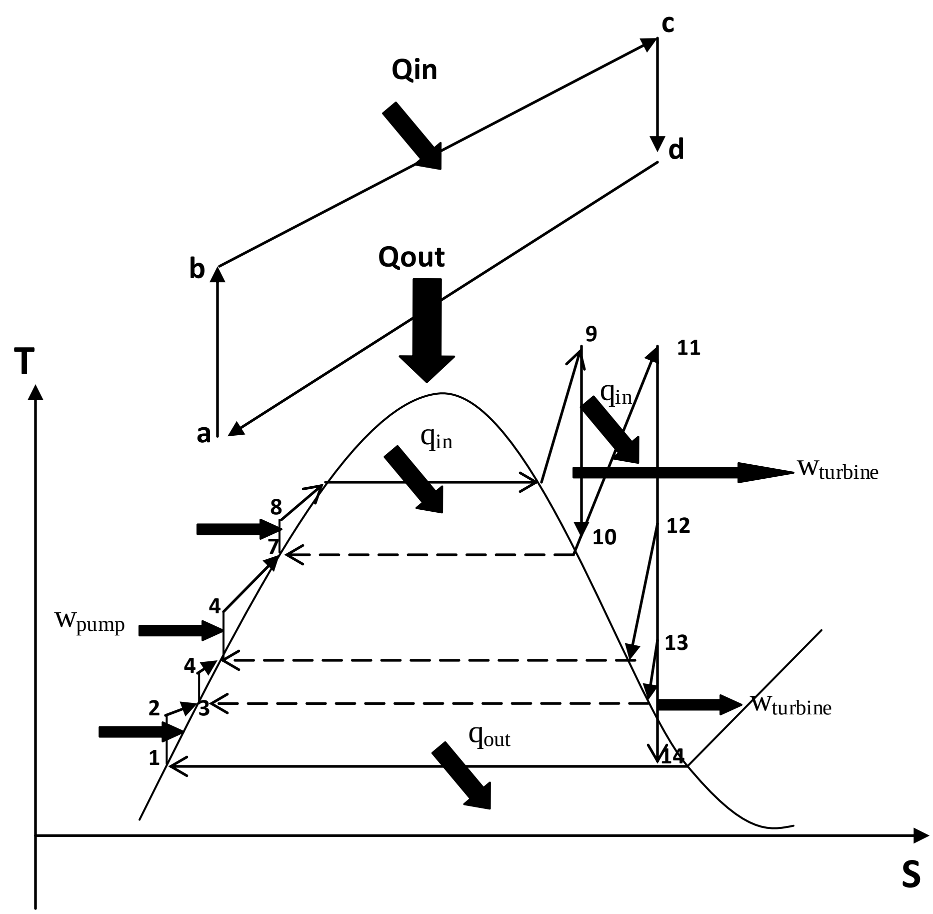
5. Discussion

From the calculation we saw that adding no of open feed water heater has increase efficiency of the cycle while it was also observed that the net work curve is reducing its slope as well. In first it might imply that the efficiency of the cycle should decrease with this, however adding feed water has reduce the pump work while on other hand the heat supplied at the boiler remained same. Hence considering the fuel requirement of boiler to run and the keep the net work output to significant level, the graph shows that three feed water heater is optimum for running regenerative reheat ideal Rankine cycle without reducing the efficiency of the turbine.

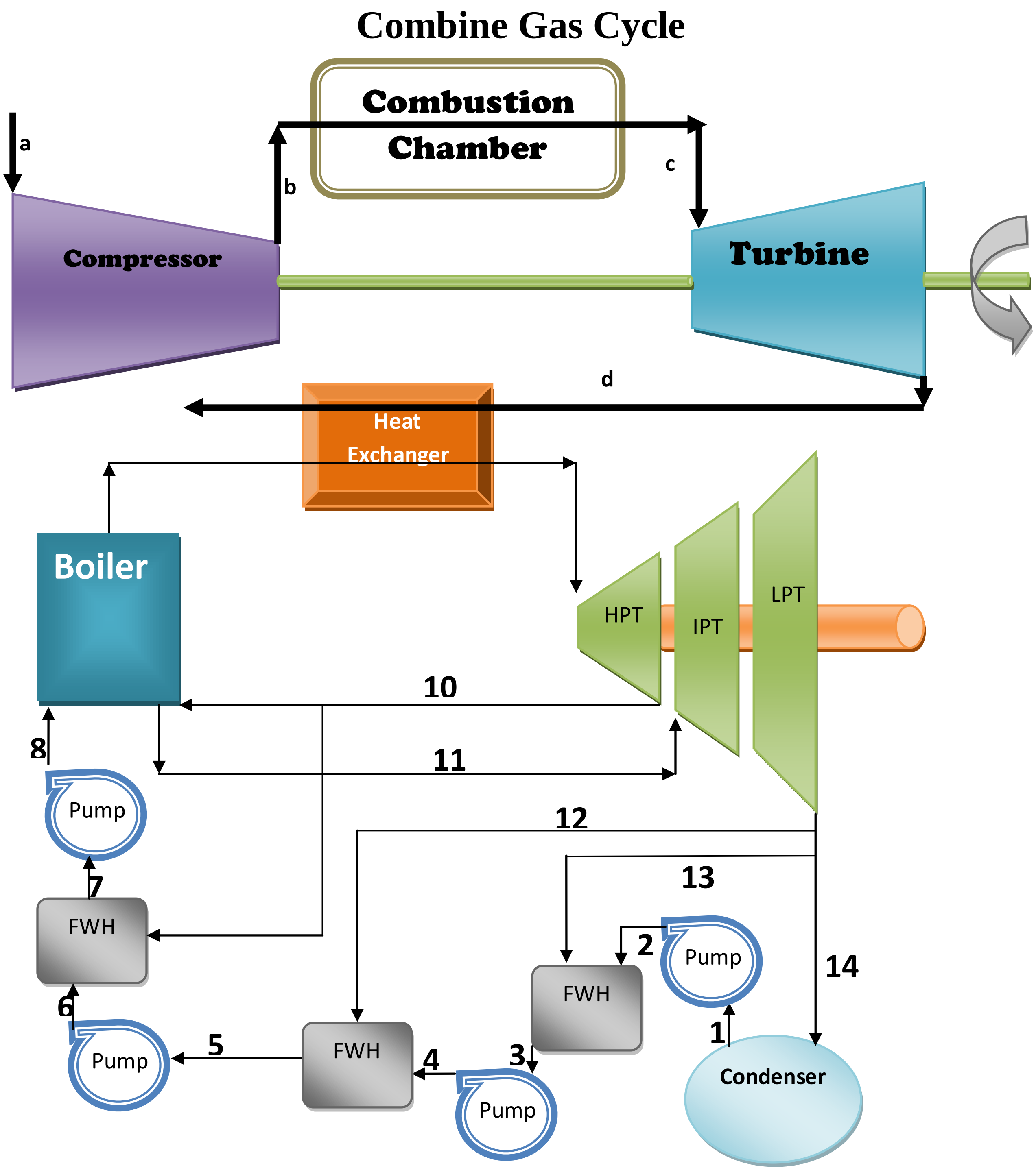
D. Combine Gas-Steam Cycle

1. Introduction

In the quest for the highest thermal efficiency, a new innovative modification to conventional power plant, a gas cycle toping the vapor cycle was introduce, which is now called a combined gas-vapor cycle. The combine gas cycle increases the efficiency without increasing initial greatly.



2. Theory


Combine Gas Cycle
Combustion Chamber
a
b
c
Compressor
Turbine
d
Heat Exchanger
Boiler
HPT
IPT
LPT
10
11
8
Pump
12
13
7
FWH
2
Pump
14
FWH
6
1
5
FWH
4
3
Pump
Condenser
Pump

a- b = ambient air is received and compressed

b- c = compresses air is heated at combustion chamber at constant pressure

c -d = high pressured combustion gases flow in turbine and work is produced

d- e = the exhaust gas is released and use as heat supply in the boiler of the steam cycle.

3. Sample calculation

3 –feed water heater selected Steam Cycle

Power of Steam Turbine = 190 MWT = 180,205.814 Btu/s

q_{in} = 1196.32 Btu/lbm

q_{out} = 632.94 Btu/lbm

$$\dot{m_{steam}} * [q_{in} - q_{out}] = 180{,}205.814$$

$$\dot{m_{steam}} = 319\ lbm/s$$

Gas Cycle

Power of Turbine = 25 MWT = 23711.29 Btu/s

Pressure ratio = 15

Peak temperature (T_c) = 2200 F

Inlet temperature (T_a) = 70 F

Assumptions:

Air is an ideal gas (k=1.4)

Isentropic expansion and compression

$T_a = 529.67R\ [h_a = 126.44\frac{Btu}{lbm}]$

$P_{ra} = 1.2968$

$T_c = 2200^\circ F = 2659.67R\ [h_c = 705.951\frac{Btu}{lbm}]$

$P_{rc} = 1.2968$

$\frac{P_b}{P_a} = \frac{P_{rb}}{P_{ra}} = 15$

$\frac{P_d}{P_c} = \frac{P_{rd}}{P_{rc}} = 15$

$$again, isentropic\ compresion\ \frac{T_b}{T_a} = \left(\frac{P_b}{P_a}\right)^{\frac{k-1}{k}}$$

again, isentropic expanison $\frac{\mathrm{T_d}}{\mathrm{T_c}} = \left(\frac{\mathrm{P_d}}{\mathrm{P_c}}\right)^{\frac{\mathrm{k}-1}{\mathrm{k}}}$

Now,

$$T_b = T_a * (15)^{\frac{1.4-1}{1.4}}$$

$$T_b = 1148.23 \text{ K } [h_b = 278.165 \frac{Btu}{lbm}]$$

similarly

$$T_d = T_c * \left(\frac{1}{15}\right)^{\frac{1.4-1}{1.4}}$$

$$T_d = 1014.83 \text{ K } [h_d = 244.48 \frac{Btu}{lbm}]$$

$$Given\ the\ net\ power\ of\ the\ Gas\ turbine\ = 23711.29\ Btu/s$$

$For\ Steady\ state\ heat\ transfer$

$$q_{out\_GAS} = (h_d - h_a) = 151.725 \frac{Btu}{lbm}$$

$$q_{in\_GAS} = (h_c - h_b) = 427.786 \frac{Btu}{lbm}$$

$$\dot{m}_{air} * [q_{in_{GAS}} - q_{out_{GAS}}] = 23711.29$$

$$\dot{m}_{air} = (23711.29/309.74) = 85.89 \text{ lbm/s}$$

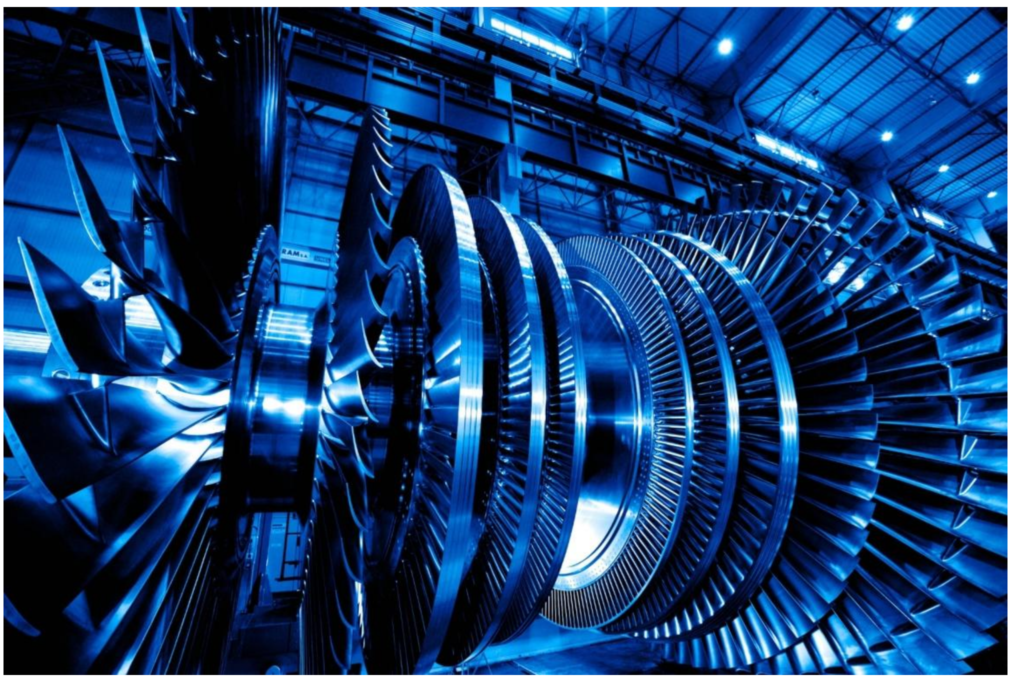
$$Number\ of\ Gas\ Turbine\ = \frac{\dot{m}_{steam} q_{in_{STEAM}}}{\dot{m}_{air} q_{out_{GAS}}} = \frac{319.86 * 1196.32672}{85.89 * 151.725} \cong 17$$

PART 2. IMPULSE STAGE STEAM TURBINE BLADE DESIGN AND CALUALATE VON-MISES STRESS AND TOTAL DEFORMATION OF BLADE

Impulse Stage Turbine Design

1. Introduction

A steam turbine is a mechanical device that extracts thermal energy from pressurized steam, and converts it into rotary motion. Its modern manifestation was invented by Sir Charles Parsons in 1884. It has almost completely replaced the reciprocating piston steam engine primarily because of its greater thermal efficiency and higher power-to-weight ratio. Because the turbine generates rotary motion, it is particularly suited to be used to drive an electrical generator – about 80% of all electricity generation in the world is by use of steam turbines. The steam turbine is a form of heat engine that derives much of its improvement in thermodynamic efficiency through the use of multiple stages in the expansion of the steam, which results in a closer approach to the ideal reversible process.



Source: http://www.power.alstom.com/home/media_centre/newsroom/2010/_files/file_62576_14624.jpg

To maximize turbine efficiency the steam is expanded, generating work, in a number of stages. These stages are characterized by how the energy is extracted from them and are known as either *impulse* or *reaction* turbines. Most steam turbines use a mixture of the reaction and impulse designs: each stage behaves as either one or the other, but the overall turbine uses both. Typically, higher pressure sections are impulse type and lower pressure stages are reaction type.

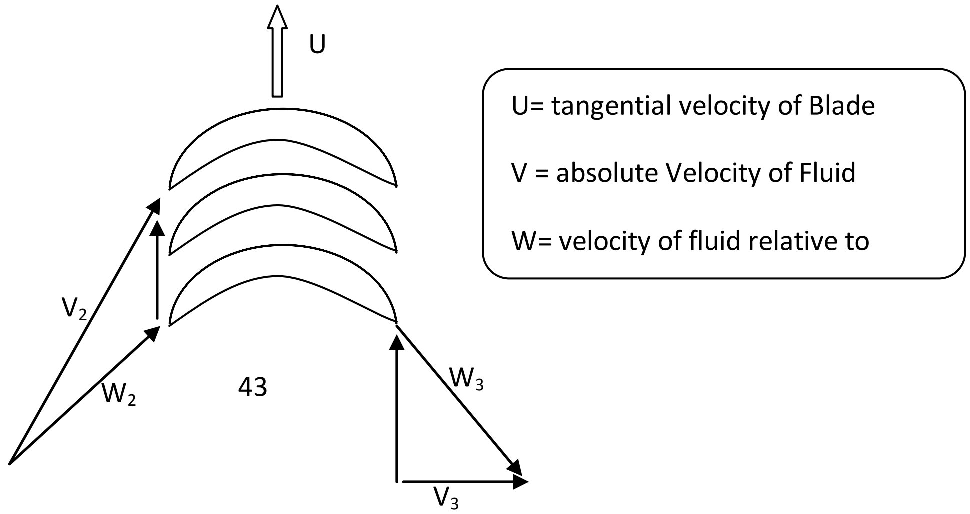
An **impulse turbine** has fixed nozzles that orient the steam flow into high speed jets. These jets contain significant kinetic energy, which the rotor blades, shaped like buckets, convert into shaft rotation as the steam jet changes direction. A pressure drop occurs across only the stationary blades, with a net increase in steam velocity across the stage.

As the steam flows through the nozzle its pressure falls from inlet pressure to the exit pressure (atmospheric pressure, or more usually, the condenser vacuum). Due to this higher ratio of expansion of steam in the nozzle the steam leaves the nozzle with a very high velocity. The steam leaving the moving blades is a large portion of the maximum velocity of the steam when leaving the nozzle. The loss of energy due to this higher exit velocity is commonly called the "carry over velocity" or "leaving loss".

In the **reaction turbine**, the rotor blades themselves are arranged to form convergent nozzles. This type of turbine makes use of the reaction force produced as the steam accelerates through the nozzles formed by the rotor. Steam is directed onto the rotor by the fixed vanes of the stator. It leaves the stator as a jet that fills the entire circumference of the rotor. The steam then changes direction and increases its speed relative to the speed of the blades. A pressure drop occurs across both the stator and the rotor, with steam accelerating through the stator and decelerating through the rotor, with no net change in steam velocity across the stage but with a decrease in both pressure and temperature, reflecting the work performed in the driving of the rotor.

2. Theory

The first stage, including a convergent-divergent inlet nozzle is shown. Ideally there is no change in the magnitude of the relative velocities W_2 and W_3 between inlet and exit respectively. The large inlet absolute velocity V_2 has been reduced to a small absolute velocity V_3, which ideally is in the axial direction.



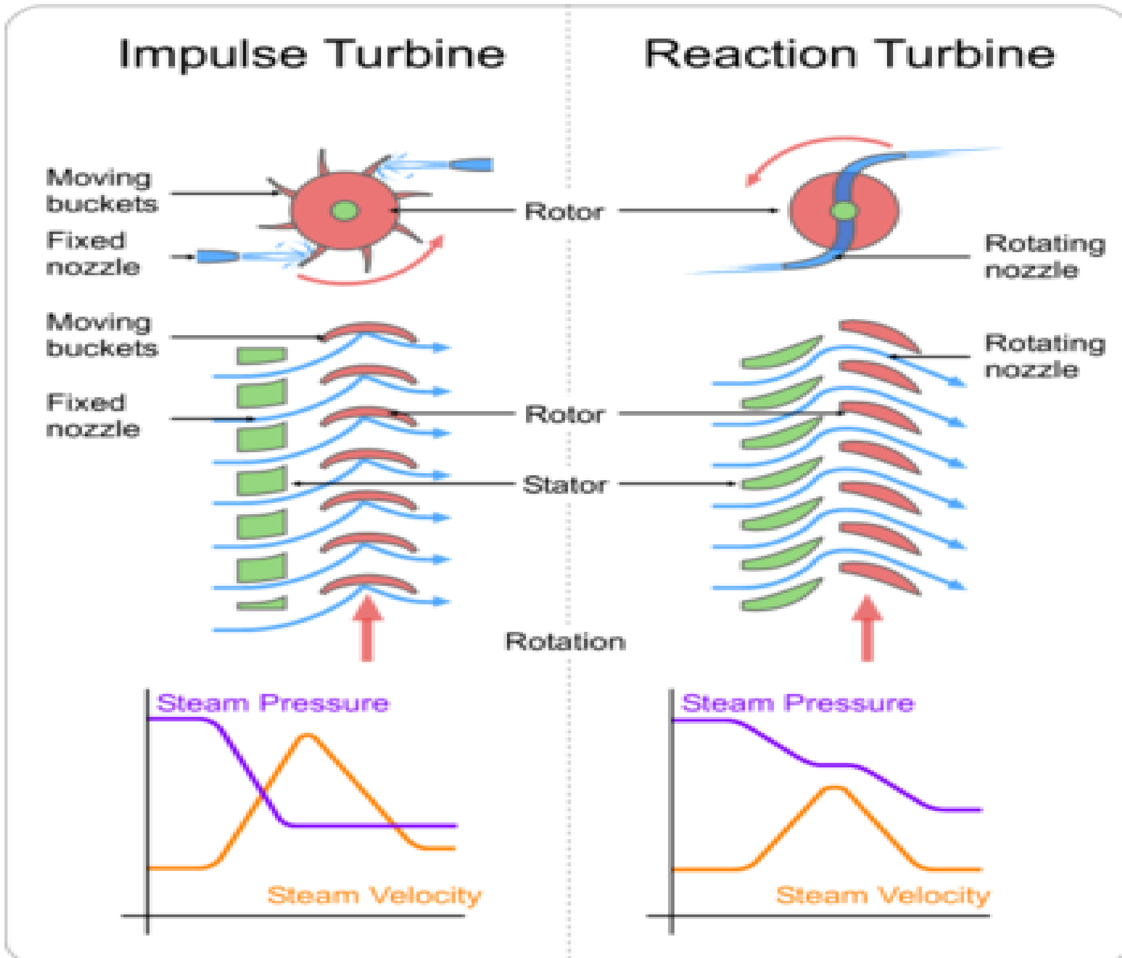


Fig 1.0

Availability of heat in static form, there is a need of some kind of mechanism in between process which creates motion or some form of dynamics. This dynamics will impact kinetic energy to the medium which then impact blades and rotate the shaft. No work can occur when there is some kind of rotation since work is equals to force time distance.

$$[w = \int f.ds]$$

The first part in fig 1 is named as Stator and second part is named as Rotor. It is assumed that there is no loss of heat from casing of the turbine. The work is obtained from rotor and that will be equal to the change in energy. A combination of stator and rotor is called a stage. Generally, a typical steam turbine is made of combination of many stages to obtain ideal process and high thermal efficiency.

3. Design Parameters

- $Impulse\ Stage\ enthalpy\ \Delta h\ =\ 25 \sim 30\ Btu\ /\ lbm$
- $Revolution\ per\ minute\ of\ the\ turbine, RPM\ =\ 1800 \sim 3600$
- $The\ ration\ of, \frac{U}{V0}\ = 0.30 \sim 0.38$
- $Angel\ of\ absolute\ velocity\ \alpha_2\ =\ 10° \sim 15°$
- $Angel\ of\ relative\ velocity\ w_2\ = w_3\ and\ \beta_2\ = \beta_3$
- $Chord\ Length\ {\sim}0.5$
- $\frac{c}{s} \sim 1$
- $Radius\ of\ the\ turbine\ interior\ are\ represented\ in\ ft$

4. Design Assumptions

- $The\ mass\ flow\ rate\ in\ the\ entrance\ region\ of\ the\ turbine = 286.9\ lbm/s$
- $Power\ produced\ in\ the\ Power\ Plant, P\ =\ 161{,}236.78\ Btu/s$
- $The\ Work\ produced\ by\ the\ Turbine = 563.38\ Btu/lbm$
- $Degree\ of\ reaction\ (R) =\ 0$
- $Rotor\ blade\ of\ the\ turbine\ are\ sysmetric$
- $Blades\ of\ the\ turbine\ are\ evenly\ spaced$
- $A\ turbine\ in\ steady\ state\ continiously\ spin$
- $Cascade\ to\ analyse\ flow\ through\ blade\ passages$
- $Relative\ inlet\ velocity\ and\ exit\ velocity\ and\ corresponding\ angle\ are\ equal$

$$w_2 = w_3 and\ \beta_2 = \beta_3$$

5. Design Procedure

5.1 Geometric Modeling

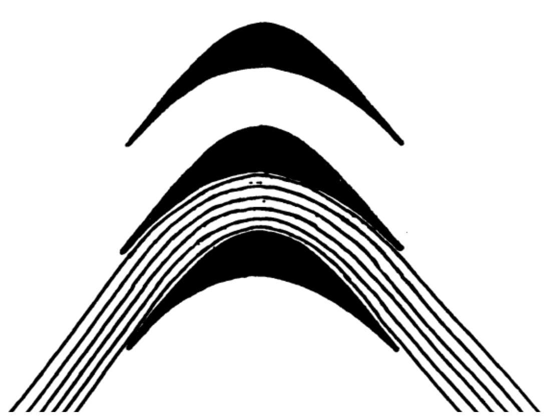
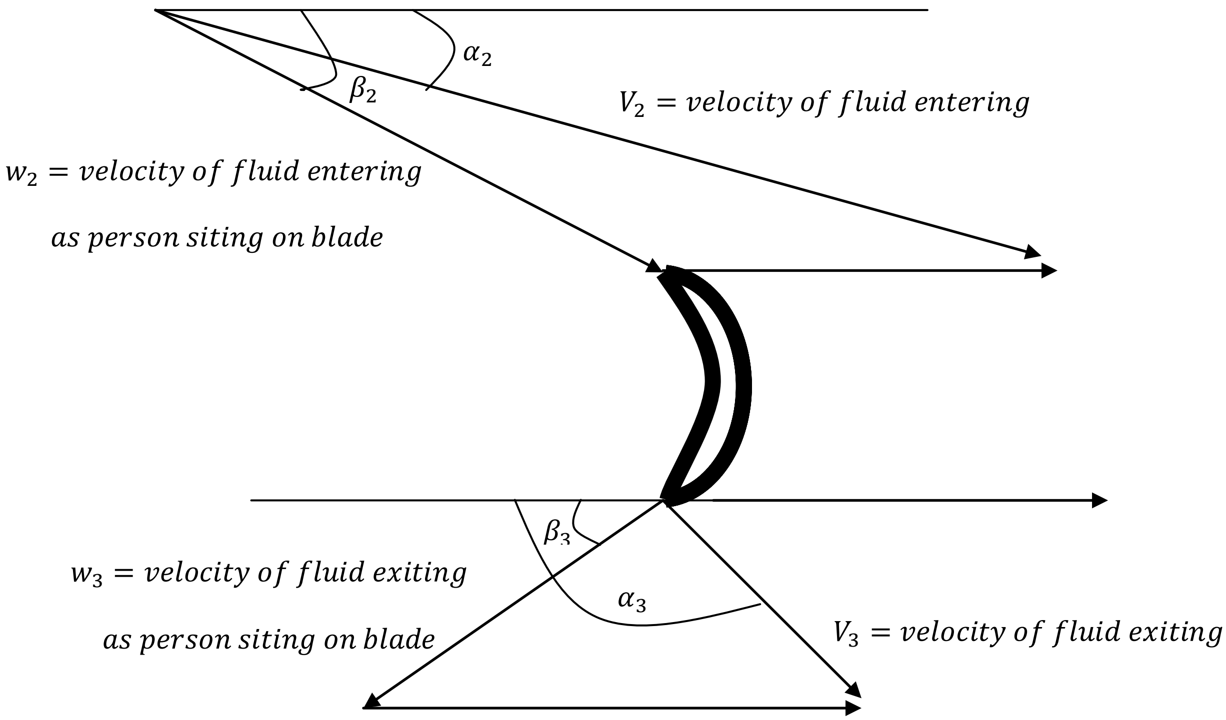


Fig 2. Fluid Flow path



For Inlet:

$$\overrightarrow{V_2} = \overrightarrow{w_2} + \vec{U}$$

Resolving in Rectangular components:

$$V_2 Cos\alpha_2 + V_2 Sin\alpha_2 = w_2 cos\beta_2 + w_2 Sin\beta_2 + U cos\ 0$$

For Exit:

$$\overrightarrow{V_3} = \overrightarrow{w_3} + \vec{U}$$

Resolving in Rectangular components:

$$V_3 Cos\alpha_3 + V_3 Sin\alpha_3 = w_3 cos\beta_3 + w_3 Sin\beta_3 + U cos\ 0$$

5.2 Results

Step 1:

$Assumed\ available\ heat\ (\Delta h)\ =\ 30\ \frac{Btu}{lbm}$

Step 2:

$Absolute\ Velocity\ of\ attack\ of\ fluid\ on\ blade\ \ (V_2)$

$$=\ \sqrt{2 * g_c * \Delta h * 778}\ =\ 1226.0\ \frac{ft}{sec}$$

Step 3:

$Tangential\ Velocity\ of\ the\ Blade\ (U)$

$$= V_2 * 0.38 =\ 0.38\ *\ 1226.00\ =\ 465.88\frac{ft}{sec}$$

Step 4:

Assumed angular velocity $(N)\ =\ 2400\ rpm = \frac{2\pi * 2400}{60}\ = 251.32\ \frac{rad}{s}$

Step 5:

Tangential Velocity $(U)\ =\ N * r_m$

Step 6:

$$Mean\ radius\ (r_m)\ =\ \frac{U}{N}\ =\ \frac{441.36}{251.32}\ =\ 1.85\ ft$$

Step 7:

$Absolute\ velocity\ anlge\ of\ attack\ at\ inlet\ (\alpha_2)\ =\ 12^{\circ}$

Step 8:

$For\ inlet\ in\ Axial\ direction$

$$W_2\ sin\ \beta_2\ =\ V_2\ sin\ \propto_2$$

$$W_2\ sin\ \beta_2\ =\ 1226.0 * \sin(12) =\ 254.89\frac{ft}{s}$$

$For\ inlet\ in\ tangential\ direction$

$$W_2\ cos\ \beta_2\ +\ U\ =\ V_2\ cos\ \propto_2$$

$$W_2\ cos\ \beta_2\ =\ 1226.0\ * cos\ 12\ - 465.88\ \ =\ 733.32\frac{ft}{s}$$

Step 9:

Relative inlet velocity angle (β_2)

$$\frac{W_2 \sin \beta_2}{W_2 \cos \beta_2} = \frac{254.89}{733.32}$$

$$\tan(\beta_2) = 0.35$$

$$\beta_2 = 19.1^{\circ}$$

Step 10:

$$Relative\ inlet\ velocity\ w_2 = \frac{254.89}{\sin\beta 2} = 776.35 fts$$

Step 11:

$For\ exit\ in\ Axial\ direction$

$$W_3 \sin \beta_3 = V_3 \sin \alpha_3\ [w2 = w3\ and\ \beta 2 = \beta 3]$$

$$V_3 \sin \alpha_3 = 776.35 * \sin 19.1 = 254.03 \frac{ft}{s}$$

$For\ exit\ in\ tangential\ direction$

$$U - W_3 \cos \beta_3 = -V_3 \cos \alpha_3$$

$$V_3 \cos \alpha_3 = 776.35 * \cos 19.1 - 465.88 = 267.73 \frac{ft}{s}$$

Step 12:

$$\frac{V_3 * \sin \alpha_3}{V_3 * \cos\alpha_3} = \frac{254.03}{276.55}$$

$$Tan\alpha_3 = 0.91$$

$$\alpha_3 = 42.57^{\circ}$$

Step 13:

$$Relative\ inlet\ velocity\ V_3 = \frac{254.03}{\sin\alpha_3}$$

$$V_3 = 375.51 \frac{ft}{s}$$

Step 14:

$Work\ done$:

$$W = W_2 cos\beta_2 + W_3 cos\beta_3 * \frac{U}{g_c}$$

$$= 2 * 772.35 * cos\ 16 * 465.8832.2\ W$$

$$= 21{,}228.24 \frac{Btu}{lbm}$$

Step 15:

$Efficiency$:

$$\eta = W/(\frac{V_2^2}{2 * gc}) = \frac{21{,}228.24}{\frac{1226.0^2}{2 * 32.2}} = 0.9009 = 90\ \%$$

Step 16:

$Specific\ Volume$:

$Given\ Case: Inlet\ Pressure\ (P_2) = 1800\ psi$

$Inlet\ Temperature(\ T_2) = 1000\ F$

$From, superheated\ Table,$

$$v2 = 0.44471 \frac{ft^3}{lbm}$$

Step 17:

$Length\ of\ blade$

$$l_1 = \frac{(\mathring{m} * v_2)}{2 * \pi * r_m * V_2 * sin\alpha_2}$$

$$= \frac{319 * 0.44471}{2 * \pi * 1.85 * 1226.00 * sin\ 12}$$

$$= 0.57\ inch$$

Step 18:

$Assume\ chord\ length\ (c) = 0.5\ inch$

Step 19:

$Blade\ spacing$:

$$0.85 = 2 * \frac{S}{c} (tan\ \propto_2 + tan\ \propto_3) \cos^2 \propto_3$$

$$S = 0.85 * \frac{0.5}{2 * (tan\ (12)\ + tan(42.57)) * \ cos\ (42.57)^2)}$$

$$= 1.35\ inch$$

Step 20:

$$Number\ of\ blade:$$

$$N * s = 2 * r_m * \pi$$

$$N = \frac{2 * 1.85 * \pi * 12}{1.35} = 104$$

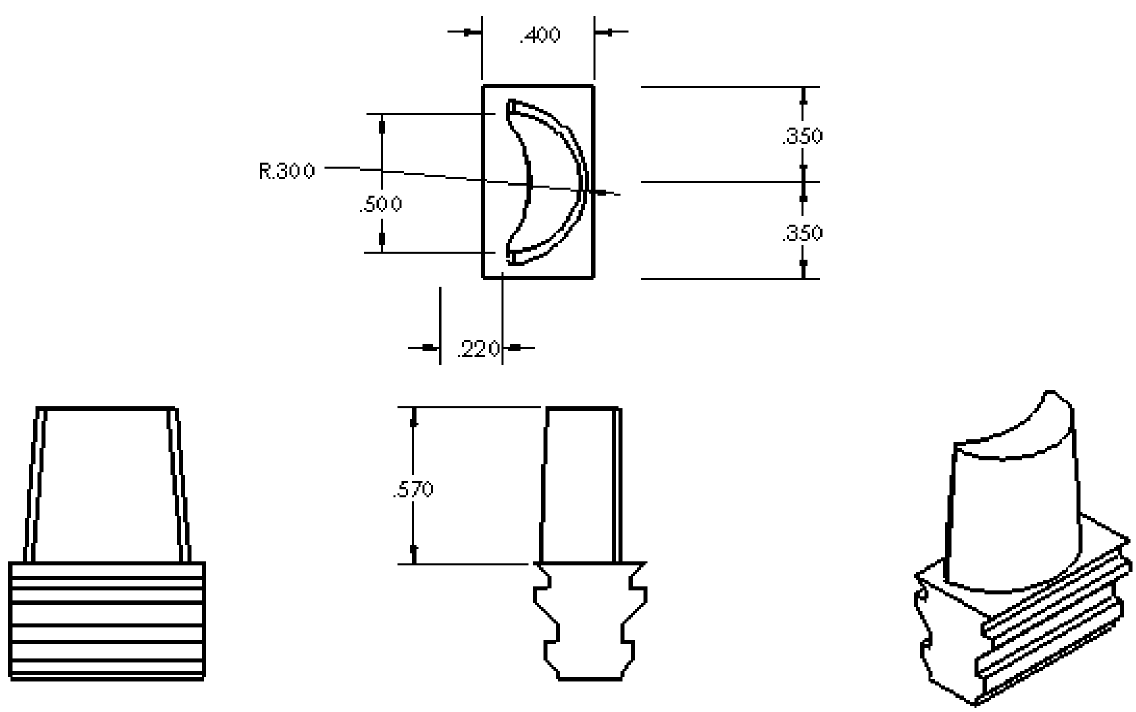
Step 21:

$$Loading\ Factor:$$

$$\psi = \frac{\Delta w}{U^2} = \frac{21228.45 * 32.2}{(465.88)^2} = 3.14 > 3\ (Approximately\ around\ the\ SAFE\ value)$$

6. Optimization and design specifications:

As we experienced from the previous calculation, in impulse stage, the total pressure drop occurs in the stationary rotor blades. This pressure drop increases the velocity of the steam. The shape of the blade is very much critical in the design and performance of the turbine. The shape of the blade in impulse stage is like a bucket and symmetrical in vertical axis, however the blades of the stator or nozzle are like airfoil.



Design specification:

Material: AISI 4340 Steel, normalized

Material Model type: Linear Elastic Isotropic

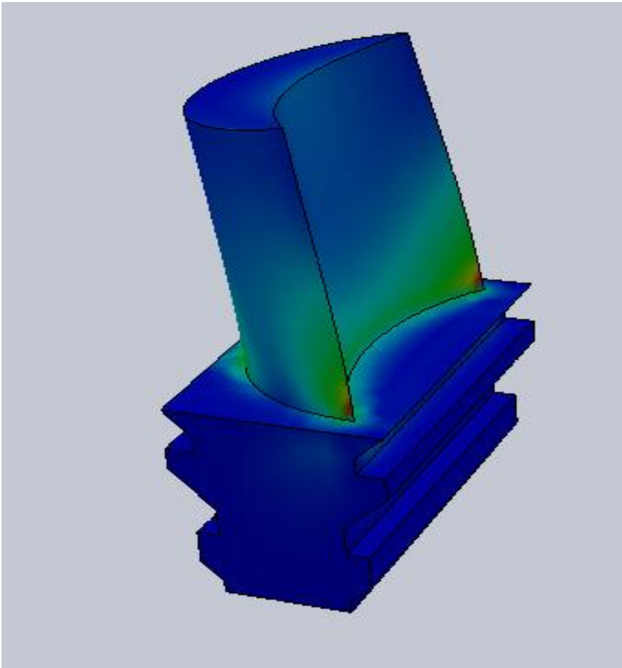
Mechanical Property	Value	Units
Elastic Modulus	2.973*10^7	lb/in^2
Poisson's ratio	0.32	
Shear Modulus	1.1603*10^7	lb/in^2
Thermal Expansion Coefficient	1.23*10^-5	
Density	0.283	lb/in^3
Thermal Conductivity	44.5	W/m K
Specific Heat	475	J/kg K
Tensile Strength	160992	lb/in^2
Yield Strength	102977	lb/in^2
Mass density	7850	kg/m^3

No.	Body Name	Material	Mass	Volume
1	Blade	AISI 4340 Steel, normalized	0.0183391 kg	0.00486 inch^3

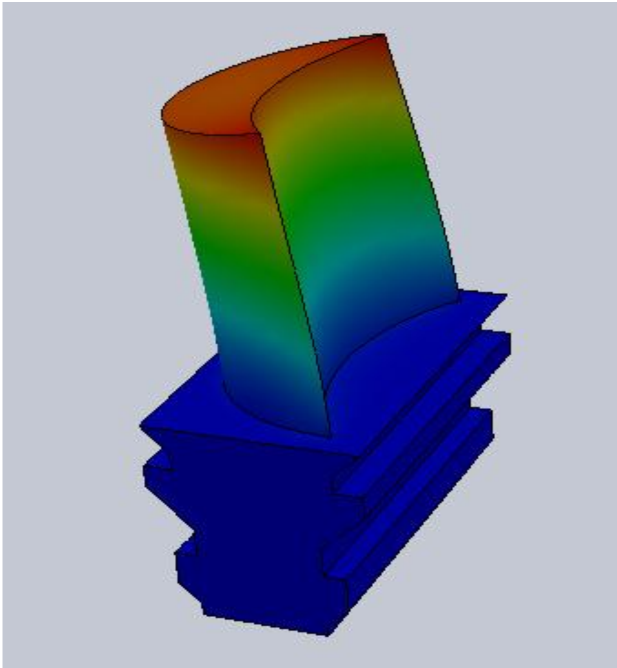
Mesh Information	
Mesh Type:	Solid Mesh
Mesher Used:	Standard mesh
Smooth Surface:	On
Jacobian Check:	4 Points
Element Size:	0.052263 inch
Tolerance:	0.0026131 inch
Quality:	High
Number of elements:	7022
Number of nodes:	11242

Material Type:	Maximum Stress: (Ksi)	Maximum Deflection $(inch)$
AISI 4340 Steel, normalized Preliminary design	79,439.00	0.0015748

Von Mises Simulation

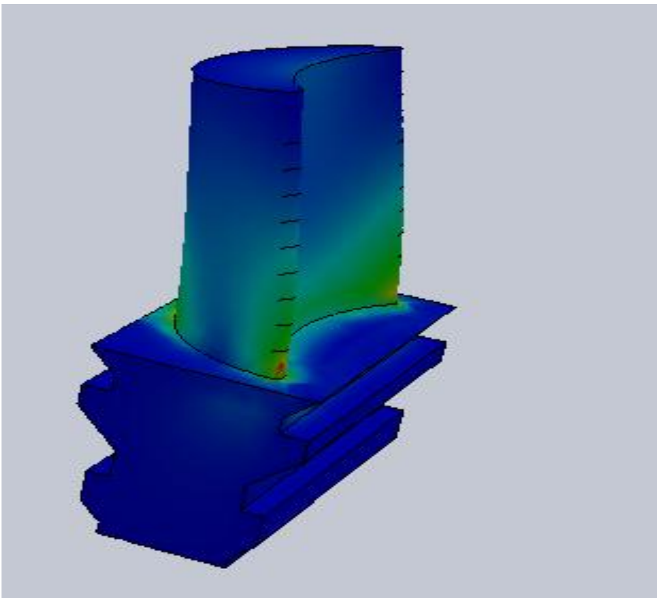


Deformation Simulation

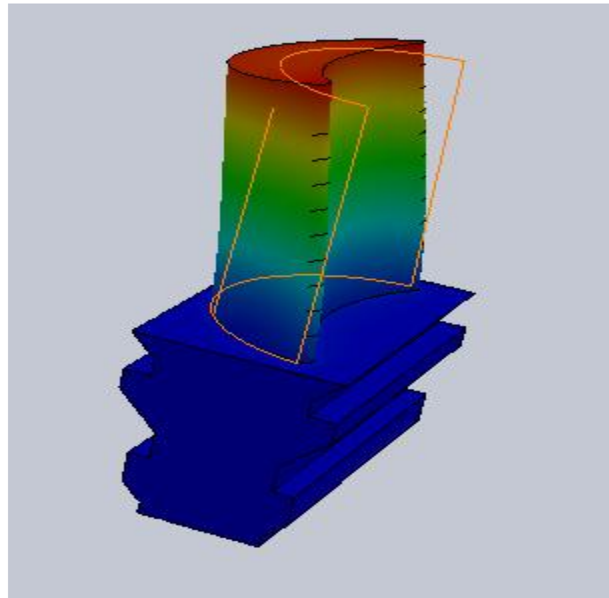


Material Type:	Maximum Stress: (Ksi)	Percentage Change	Maximum Deflection $(inch)$	Percentage Change
AISI 4340 Steel, normalized Final optimized design	65,084.09	18%	0.0013819	12%

Von Mises Simulation



Deformation Simulation



7. Discussion

- The Turbine can develop under the conditions above discussed, but this project is for only one stage of the design.
- The length of the blades seems reasonable for this kind of design because the Toque produce into this blade needs to be transformed into movement in the shaft
- The Power required for keep working the turbine is accurate, but it is not well known the total behavior of the complete turbine in the three section: High Pressure Turbine, Intermediate Turbine, and Low Pressure Turbine, as we are working only on the impulse stage
- The Work Produced by the Turbine can be useful for electric generator of medium Power

- The Stages in the turbine will be around 12 – 13 stages, because the turbine is needed for a Power Plant of the dimensions described in the objective, the stages are supplied because the high power produced by the total turbine in the Power Plant
- The steam mass flow rate must be adequate at the entrance region for better performance in the cycle. The mass flow rate varies from millions of pounds per hour, it depends the total energy produced in the whole Rankine Cycle.
- From the calculation loading factor is less than giving condition so overall calculations and the results are desirable.

8. Conclusion

We can conclude that in a steam turbine there will be two parts, the first will convert heat to kinetic energy and the second part will convert the kinetic so obtained to work. The numerical values found at the calculations are acceptable for this kind of design because the design is based in the entrance region of the Impulse Stage at the Turbine. The obtained values widely varied as there are uncertain changes in other given parameters. The design of turbine must be done with accuracy and precision, as the turbine is one of the most indispensable parts at the Power Plant. The turbine requires more development and precision than the other parts because the blades bears the centrifugal force and any fatigue deformation can created substantial damage to entire structure of the turbine. Nowadays electrical power is one of the essential for human beings. The simple steam power plants are the predominant producer of electricity in the world and most fundamental role shaping human conditions.

Turbine can be expressed as a rotary engine that extracts energy from fluid flow. A turbine consists of a moving part and a stationary part. The Rotary part is composed for the blades, and rotors which are moved by the fluid flow for this case steam. The fluid flow contain a reasonable amount of potential energy and kinetic energy, these fluid flow may be compressed or uncompress depending of the circumstance.

Impulse turbines: These turbines change the direction of flow of a high velocity fluid jet. The resulting impulse spins the turbine and leaves the fluid flow with diminished kinetic energy. There is no pressure change of the fluid in the turbine rotor blades. The fluid flow develops torque while it passes through the turbine structure. Some pressure is required to maintain the system working under the proper circumstances

For the design of this turbine the following requirements are needed such as: the mass flow rate which is found at the turbine entrance, the power required keeping working in excellent conditions the turbine cycle, and perhaps the amount of energy or work produced during the turbine cycle.

9. Reference

Dixon, S.L. *Fluid mechanics, Thermodynamics of Turbomachinery.* Fourth. Madras: Plant A Tree, 1998, 93-133. Print.

Gorla, Rama S.R., and Aijaz A. Khan. *Turbomachinery Design and Theory*. New York: Marcel Dekker, 2003. eBook.

Turton, R.K. *Principles of Turbomachinery*. Second. London: Chapman & Hall, 1995. 160-183. Print.