



Possibility Of Using Back-Pressure Turbine in the Iraqi Petroleum Industry

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ABSTRACT

Dura refinery, Baghdad, utilizes at the same time two forms of energy: electricity and heat. Nowadays, these two forms of energy are supplied by a bottoming cogeneration plant. The present paper describes this cogeneration system in detail. Meanwhile, the theoretical analysis was conducted according to the 1st and 2nd laws of thermodynamics to evaluate system performance. Based on the results of such analysis and considering the advantages of back-pressure turbine properties, two alternative plants based on a true cogeneration system have been designed and analyzed. The results showed a substantial increase in efficiency (6–13%) as well as the achievement of (24%) saving in fuel consumption. Moreover, the findings of the present work indicate that these alternative plants can produce more electricity than is required by the refinery. Such excess power can be effectively sold to the main electric network, improving overall energy utilization.

Keywords: Back-pressure turbine, Cogeneration, Iraqi petroleum industry, Dura refinery, Thermodynamic analysis

الخلاصة

يستخدم مصفى الدورة في بغداد طاقة حرارية و كهربائية في نفس الوقت في الوقت الحاضر هذه الطاقات تجهز من محطة ذات دورة سفلى . البحث الحالي يتضمن وصف لهذه المحطة المركبة و تحليل نظري لها استنادا الى قانوني الترموديناميك الأولى و الثاني .نتيجة الى نتائج التحليل مع اخذ بنظر الاعتبار الاستفادة من مميزات التوربين ذو الضغط الخلفي و ادخاله في محطات مركبة بديلة تم تصميم محطتين و تحليلها .لقد أشارت النتائج إلى زيادة جوهرية في كفاءة المحطة تصل الى 6-13 % اضافة الى امكانية ادخار في الوقود تصل إلى 24 . %لقد أوضحت النتائج انه من خلال هذه المحطات البديلة بالإمكان توليد طاقة كهربائية تفوق احتياج المصفى و هذه الطاقة بالإمكان بيعها إلى الشبكة الرئيسية

الكلمات المفتاحية: التوربينات، الضغط الخلفي، التوليد المشترك، صناعة البترول، مصفى الدورة

1. INTRODUCTION

Simultaneous generation of electricity and steam (cogeneration or CHP) is a long-established practice in large chemical plants (Baughn and Kerwin, 1987). Steam manifolds at different pressure levels are located throughout the plants. This practice has now been adopted by smaller industrial plants due to rising energy prices. However, these plants need smaller equipment. Therefore, a trend exists toward smaller industrial cogeneration systems (Gretciena, 2001; Backlund and Karlsson, 1988; Jose and Vian, 1988; Fageeh et al., 1989). The Dura refinery energy plant consists of two units, supplying the whole refinery with 76.62 kg/s of process steam at 18 bar, 260 °C, and 45 kg/s at 21 bar, 300 °C, as well as 4.5 MW of electric power (Hassan and Hameed, 1988). An additional 11 MW of electricity is purchased from the main electric network. Figs. (1, 2) show unit (2) and unit (1) respectively, while Tables (1, 2) show the quantitative details for these units. Since the Dura refinery was established in the middle of the last century (Hassan and Hameed, 1988), Iraq is now interested in reconstructing its old refineries and is inspired to build new refineries. Therefore, several researchers found that it is necessary to analyze the heat cycle of the current Dura refinery and to propose a new design for a cogeneration plant (Porter and Mastannish, 1983). The back-pressure turbine can generate maximum electricity at approximately the same time as the system's heat is at its peak. Furthermore, it is simple in design, highly reliable, and has a very low heat rate, which permits achieving high fuel savings. Also, the equipment costs and space requirements for back-pressure

(non-condensing) units are much lower than for condensing units (Ivasheva et al., 1992; Richard and Drof, 2000; Jose and Vian, 1988; Melman, 2000; Habib, 1992; Ertesvag, 2007; Kreith and Goswami, 2007). The present study aims to provide a brief description of the present Dura refinery plant and the suggested alternative true cogeneration plant, in which a back-pressure non-condensing turbine is examined. To describe the performance of a cogeneration plant, various thermodynamic parameters were determined.

2. DATA ANALYSIS AND THERMODYNAMICS PARAMETERS

To describe the performance of the present Dura refinery plant and the proposed cogeneration plants, a computer program has been written to run under the Mathcad software package. Manufacturers' design data for proposed back-pressure non-condensing turbines (Catalog of steam turbine, 1960s; Shlyakhin, 1964) and Dura refinery operating conditions (Hassan and Hameed, 1988) were used as input data. In this study, proposed cogeneration plants are designed to always meet the thermal energy requirements of the Dura refinery. For this reason, the proposed back-pressure turbines may operate under off-design conditions rather than at design conditions to meet these thermal requirements. Deviations in the steam flow rate from design conditions (off-design flow rate) affect the heat drop distribution, the pressures prevailing in the turbine stages, the temperature distribution, and the extraction flow rate. So, a system of non-linear differential

equations (Kathem, 2007; Marena, 2001; Sergio et al., 2002a; Sergio et al., 2002b) (such as the Bendman, Stodola, parameters of the new required operating regime were found. The solution results in Figs (3-6), which show the relation between the steam mass flow rate through the proposed back-pressure turbine and different parameters when the turbine operates at various regimes. From these figures, a suitable regime was determined to match the thermal requirements of Dura refinery. The program enables the user to estimate the state variables at each node through the heat cycle using the appropriate thermodynamic relations (Yeza, 1966). Moreover, this suggested program enables analysis of the performance of the present and proposed plants with respect to the following thermodynamic parameters.

2.1 Irreversibility Coefficient

According to the 2nd law of thermodynamics, the irreversibility coefficient (Ω) for each component of the plants is (Moayed, 2007; Rosen et al., 2007; Datta et al., 2007; Kotas, 1985)

$$\Omega_i = \frac{\Phi_i}{m_f \times C.V.} \quad (1)$$

Where,

$$\Phi_i = T_0 \times [\sum_{i=1}^n (m_i \cdot s_i)_{out} - \sum_{i=1}^n (m_i \cdot s_i)_{in}] \quad (2)$$

And the overall irreversibility coefficient of any plant,

$$\Omega_{total} = \frac{\sum_{i=1}^n \Phi}{m_f \cdot C.V.} \quad (3)$$

$$\Omega_{total} = \Omega_1 + \Omega_2 + \Omega_3 \dots \dots \Omega_n = \sum_{i=1}^n \Omega \quad (4)$$

Then, plant thermal efficiency is defined as,

$$\eta = 1 - \sum_{i=1}^n \Omega_i \quad (5)$$

2.2 Efficiency According To 1st Law of Thermodynamics

and heat balance equations) was solved by the Iteration method, and the transient conditions were formulated. The

This efficiency is based on the energy balance of the system, where the useful output (electric power and process heat) is compared to the total energy input from fuel, according to the first law of thermodynamics (Baughn and Kerwin, 1987; Dincer and Rosen, 2007; Yunus and Boles, 2005; Kreith and Goswami, 2007):

(a) Steam efficiency

$$\eta_s = \frac{Q_p}{Q_{in}} \quad (6)$$

(b) Electric efficiency

$$\eta_e = \frac{W_{net}}{Q_{in}} \quad (7)$$

(c) Plant efficiency

$$\eta_{plant} = \eta_s + \eta_e \quad (8)$$

2.3 Power to Heat Ratio (PHR)

The power-to-heat ratio (PHR) represents the relationship between the electrical power generated and the useful thermal energy produced in a cogeneration system, and it is an important parameter for evaluating system performance and energy utilization (Baughn and Kerwin, 1987)

$$PHR = \frac{W_g}{Q_p} \quad (9)$$

2.4 Fuel Saving Rate (FSR)

The fuel saving rate (FSR) represents the reduction in fuel consumption achieved by the cogeneration system compared to separate production of heat and power, and it is used as an indicator of the economic and energetic advantage of the system (Baughn and Kerwin, 1987):

$$FSR = \frac{(m_f)_{present} - (m_f)_{proposed}}{(m_f)_{present}} \quad (10)$$

Where, mf is fuel consumed for the same output (heat + electric) from both plants. Considering the lack of electric power

(generated in the present plant) are supplied from another condensing plant with $\eta=0.35$.

2.5 Specific Fuel Consumption

In a cogeneration plant, the input fuel must be divided between fuel for power generation and fuel for process steam generation (Bruce and John, 1980; Muhammed Nusrat Saeed, 1992; Sergio et al., 2002). Throughout the current study, this parameter was estimated according to 1st and 2nd law of thermodynamics (Pustovalov, 1987) as follows:

2.5.1 Physics method (1st law)

2.5.1.1 Specific fuel consumption for electric (g/kW.h)

$$(s.f.c.)_e = \frac{Q_{add} - Q_{hp}}{W_g \cdot C.V. \cdot \eta_{boiler}} \quad (11)$$

2.5.1.2 Specific fuel consumption for heat (kg/GJ)

$$(s.f.c.)_h = \frac{1}{C.V. \cdot \eta_{boiler}} \quad (12)$$

2.5.2 Exergy method (2nd law)

2.5.2.1 Specific fuel for electric (g/kW.h)

$$(s.f.c.)_e = s.f.c. \quad (13)$$

$$s.f.c. = \frac{m_f}{(W_g + Q_{h.p} \cdot \eta_{carnot})} \quad (14)$$

Where,

$$\eta_{carnot} = 1 - \frac{T_0}{T_{in}} \quad (15)$$

2.5.2.2 Specific fuel for heat (kg/GJ)

$$(s.f.c.)_h = (s.f.c.)_e \cdot \eta_{carnot} \quad (16)$$

3. ALTERNATIVE PLANTS FOR DURA REFINERY

As the Dura existing plant for units (1) and (2) represents an old system and it is not of the true cogeneration type (Hassan and Hameed, 1988), alternative plants were chosen to cover refinery requirements in heat and electric power. In this study, three types of back-pressure non-condensing turbines were suggested. These were R-50-127-13, R-25-90-31 and

R-25-90-18. Where the first number refers to power (MW), the middle one refers to inlet turbine pressure (bar), and the last number refers to turbine exit pressure (bar). The operation design conditions for the above turbines are listed in Tables 3-5. The results showed that turbine R-25-90-18 cannot meet the steam demand and the conditions for refinery process heat; therefore, the study described and analyzed only the other two turbines for units (1) and (2): **(1) R-50-127-13**

Unit 1: In this cogeneration system, a boiler was selected based on the turbine's operating conditions (Gost, 1989). To meet the heat demand of unit (1), the turbine must be operated continuously at partial load with an exit pressure of 18 bar. The temperature of the steam from the turbine exhaust is around 316 °C, so most of the superheat is removed by water injection. Fig. 7 represents a simple layout for this unit, and the quantitative details are listed in Table 6. This plant can cover the process heat and produce 21 MW of surplus electricity, which can be sold to the grid at a profit.

Unit 2: As unit (1), turbine R-50-127-13 operated continuously at partial load with exit pressure 21 bar and 336 °C. To regulate steam properties (p&t), the demand for heat processes requires the use of water injection and expansion valves. From this unit, 35 MW of surplus electrical power can be produced. The general plant flow diagram for this unit is shown in Fig. 8, and its quantitative details are listed in Table 7. **(2) R-25-90-31 Unit 1:** The layout for this unit with a back-pressure turbine R-25-90-31 is shown in Fig. 9. The steam exhausted from the turbine is at 31 bar and 403 °C, as shown in Table 8. Steam properties are regulated using water injections and expansion

valves. From this unit, 3 MW of surplus electrical power can be produced.

Unit 2: Fig. 10 represents a flow diagram for Unit 2 with this turbine. As unit (1), the steam properties must be regulated to match the unit demand. The quantitative details of this plant are listed in Table 9. From this unit, 20 MW of surplus electrical power can be produced.

4. RESULTS AND DISCUSSION

The results for the present Dura refinery units (1) and (2) and their suggested are listed in Tables (10-15), respectively. The results showed that the present plant has lower efficiency than the alternative plants. This can be explained as: the cogeneration plant for unit (1) depends on the Bottoming cycle, which causes wasting exergy without producing mechanical work (El-Wakil, 1984). Whereas in unit (2), the steam was used only to meet the heat demand, and the concept of cogeneration was not used. The two alternative plants with back-pressure (non-condensing turbines) improve refinery plant performance by around 13% and 6%, according to the 1st and 2nd laws of thermodynamics, respectively. In the present plant, the specific fuel consumption for electric power is very high, even higher than that for separate plants. Although increasing the capacity of a back-pressure turbine surely led to an increase in initial cost on one side, it also increased the power-to-heat ratio (PHR). For this reason, this turbine leads to an increase in fuel saving rate (FSR) of around 4% compared with the smallest back-pressure turbine. The results of the current study showed that the suggested plants can produce surplus electric power, which can be sold to the grid.

5. CONCLUSIONS

1-According to 1st and 2nd thermodynamics law, the suggested plants

can improve thermal efficiency by around 13% and 6%, respectively.

2- Although increasing the capacity of back-pressure leads to an increase in initial cost, it also leads to an increase in PHR and FSR.

3- The price of electric and heat power from cogeneration plants depends on the method of estimation of specific fuel consumption. Therefore, this estimation method should be chosen to promote the construction of new cogeneration plants in Iraq.

4- The present suggested alternative plants offer the ability to choose an operation regime, which can match exactly the heat demand. This is in contrast with what happens in the presently used plant.

5-At present, the true industrial cogeneration plants are recommended in Iraq, especially in national industry sectors, to contribute with ministry of Electricity to solve the present crisis of electric power generation.

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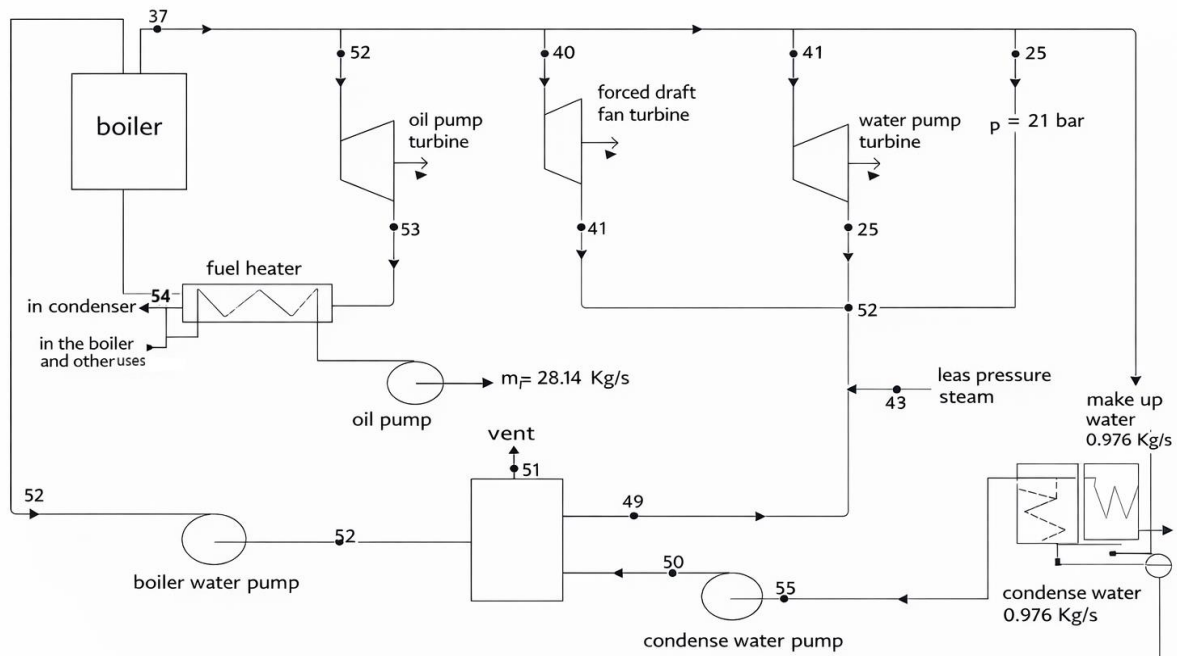


Figure 1. Schematic diagram for Unit 2 of the present Dura refinery

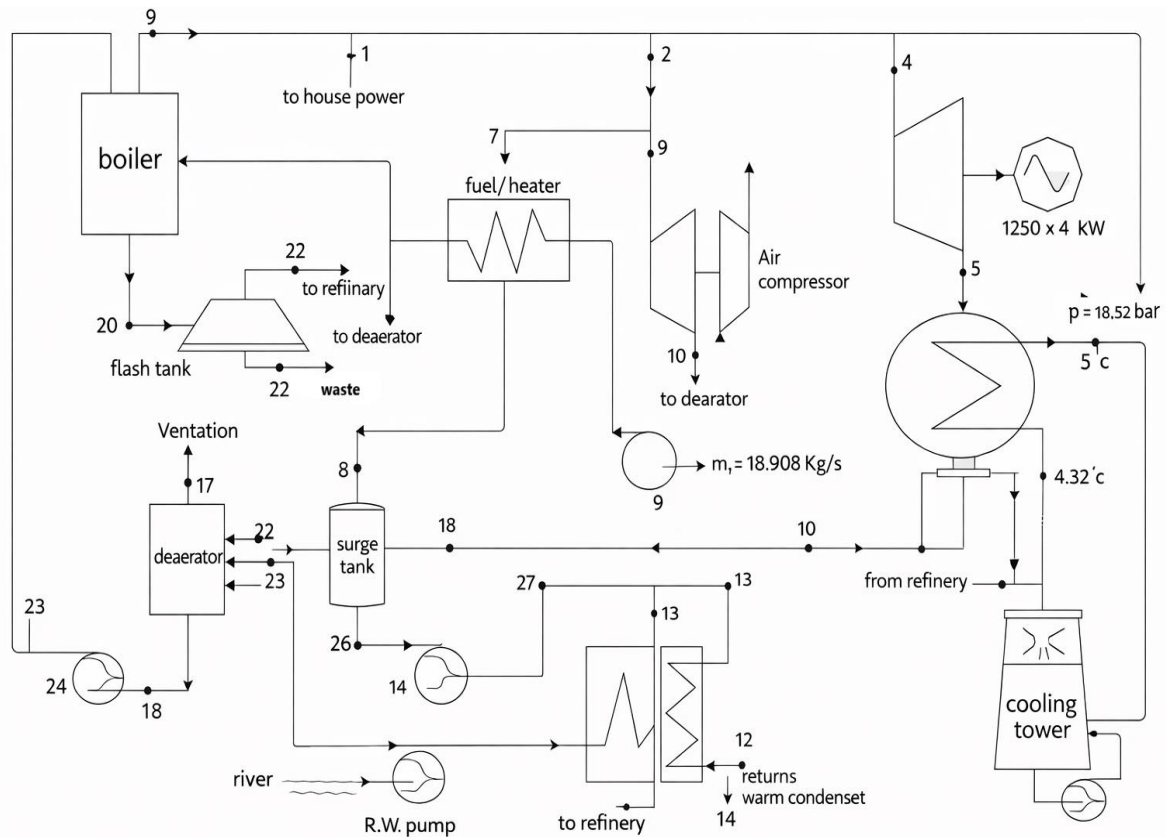


Figure 2. Schematic diagram for Unit 1 of the present Dura refinery

Table 1: Description data of the steam circuit for unit (1) Dura's refinery

Pt.	P (bar)	t (°C)	m (kg/s)	Pt.	P (bar)	t (°C)	m (kg/s)
0	18.93	260	75.6	19	1.36	108.4	6.433
1	18.24	260	3.143	20	18.39	209.6	9.639
2	18.24	260	4.187	21	1.36	108.4	7.796
3	18.24	260	57.182	22	1.36	108.4	1.843
4	18.24	260	11.088	23	22.37	86.7	85.239
5	0.1	45.8	11.088	24	18.24	248.9	0.547
6	0.1	46.1	11.088	25	1.7	115.2	0.512
7	17.55	248.9	1.462	26	3.01	34	66.125
8	17.48	96.5	1.462	27	1.01	33.9	66.125
9	18.24	260	2.725	28	18.24	248.9	0.296
10	1.7	115.2	2.7125	29	1.43	148.9	0.296
11	0.1	46.1	11.088	30	18.24	248.9	0.176
12	1.01	71.1	11.494	31	1.7	160.6	0.176
13	1.01	26.1	54.631	32	16.17	232.2	5.579
14	1.01	93.3	76.627	33	0.1	45.8	5.579

15	2.01	39.9	78.675	34	0.1	46.1	5.578
16	18.24	260	0.887	35	17.55	260	1.035
17	1.36	108.4	0.756	36	1.7	165.6	1.035
18	1.8	86	85.239				

Table 2: Description data of the steam circuit for unit (2) Dura's refinery

Pt.	P (bar)	t (°C)	m (kg/s)	Pt.	P (bar)	t (°C)	m (kg/s)
37	23.35	268	78.04	49	3.34	137.4	78.04
38	18.24	260	19.445	50	1.36	27.4	64.522
39	22.57	268	0.357	51	1.36	108.8	0.412
40	20.65	260	6.596	52	21.55	255	0.515
41	3.77	151.7	6.596	53	3.77	193.9	0.515
42	20.65	260	4.799	54	3.77	141.5	0.515
43	3.77	168.3	4.799	55	36.29	139.3	78.04
44	21.55	265	0.767	56	18.24	260	76.627
45	21.55	265	45.561	57	3.76	165.6	12.162
46	1.01	60	9.365	58	1.01	27.4	64.522
47	3.76	170	1.768	59	1.01	26.1	55.521
48	3.76	165.8	13.93				

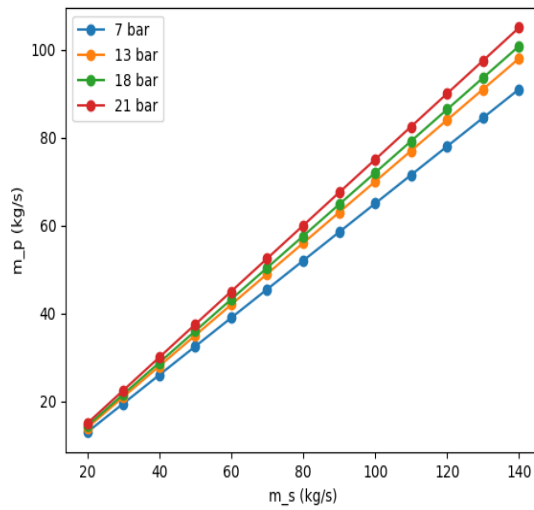


Figure 4. Relationship between process steam demand and steam flow rate through the turbine

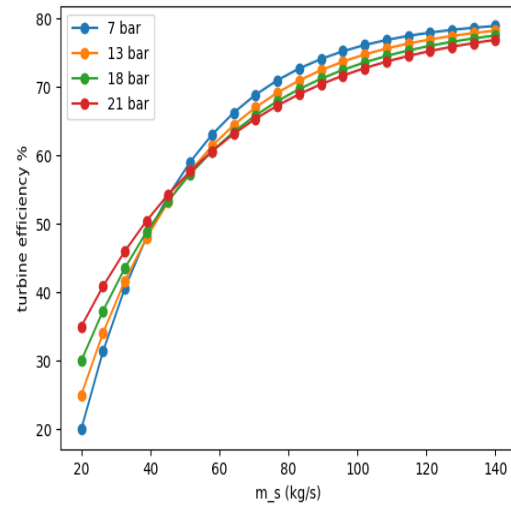


Figure 3. Variation of steam turbine efficiency with steam flow rate at different back pressures.

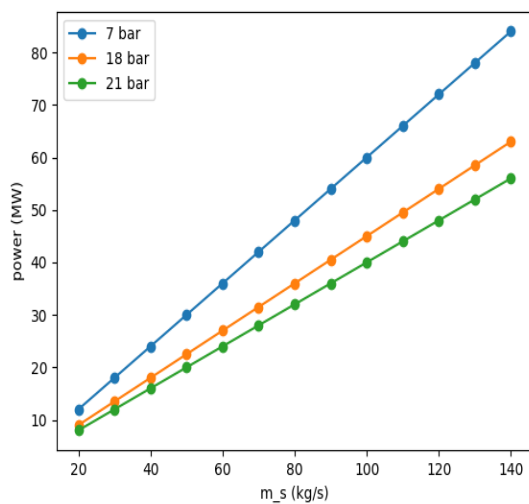


Figure 5. Variation of turbine power output with steam flow rate at different back pressures.

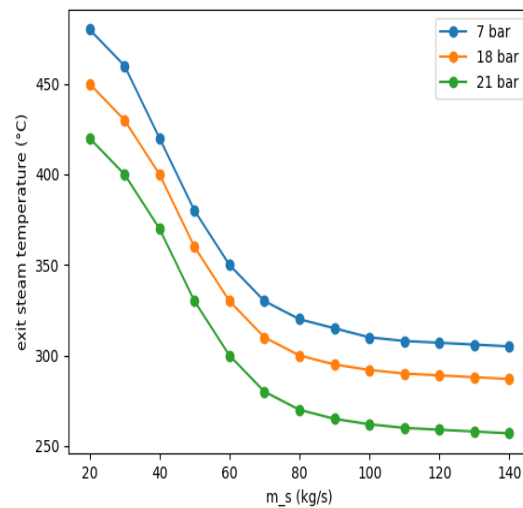


Figure 6. Variation of exhaust steam temperature after the turbine with steam flow rate at different back pressures.

Table 3. Operation design conditions for turbine type R50-127-13

P MW	p _{in} (bar)	t _{in} (°C)	m _{max.} (kg/s)	m _o (kg/s)	m _{1st.b} (kg/s)	m _{2nd.b} (kg/s)	m _{3rd.b} (kg/s)	p _{c.s.} (bar)	p _{1st.b} (bar)	p _{2nd.b} (bar)	p _{s.4} (bar)	p _{exit} (bar)
50	127	565	136	102	5.5	6	9.58	95	37	22	65	13

Table 4. Operation design conditions for turbine type R25-90-31

P MW	p_{in} (bar)	t_{in} (°C)	$m_{max.}$ (kg/s)	m_o (kg/s)	$m_{1st.b}$ (kg/s)	$m_{2nd.b}$ (kg/s)	$m_{3rd.b}$ (kg/s)	$p_{c.s.}$ (bar)	$p_{1st.b}$ (bar)	$p_{2nd.b}$ (bar)	p_{exit} (bar)
25	90	535	105	105	7.526	1.414	16.84	60	43.36	38.87	31

Table 5. operation design conditions for turbine type R25-90-18

P MW	p_{in} (bar)	t_{in} (°C)	$m_{max.}$ (kg/s)	m_o (kg/s)	$m_{1st.b}$ (kg/s)	$m_{2nd.b}$ (kg/s)	$m_{3rd.b}$ (kg/s)	$p_{c.s.}$ (bar)	$p_{1st.b}$ (bar)	$p_{2nd.b}$ (bar)	p_{exit} (bar)
25	90	535	74	74	9.936	1.978	17.36	60	49.64	42.64	18

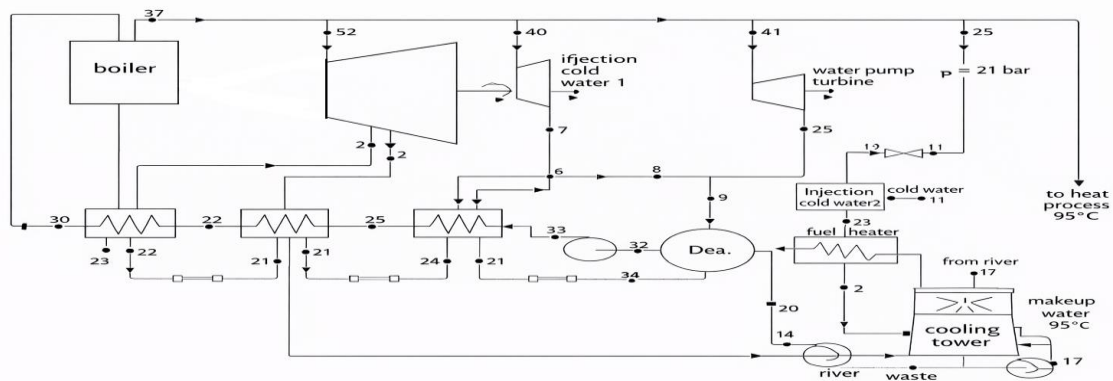


Figure 7. Schematic diagram for unit 1 with turbine R-50-127-13

Table 6. Description data of the steam circuit for unit (1) Dura's refinery with the turbine type R 50-127-13

Pt.	P (bar)	t (°C)	m (kg/s)	Pt.	P (bar)	t (°C)	m (kg/s)
0	130	565	83.337	17	1.01	93	51.115
1	34	389.7	3.571	18	1.01	27	48.232
2	23.61	343.3	2.232	19	1.01	38.94	58.695
3	18.24	316.18	77.534	20	6	38.96	58.695
4	18.24	316.18	7.232	21	18.24	260	1.216
5	18.24	316.18	70.302	22	6	158.8	83.337
6	18.24	316.18	11.607	23	130	160.3	83.337
7	18.24	316.18	58.605	24	130	204.5	83.337
8	1.01	27	2.871	25	130	218.2	83.337
9	18.24	260	61.566	26	130	238.1	83.337
10	17.55	258.9	1.216	27	34	240.9	3.571
11	18.24	260	60.35	28	23.61	221	3.571
12	1.01	27	0.012	29	23.61	221	5.803
13	17.15	248	1.228	30	18.24	207.8	5.803
14	1.01	93	60.35	31	18.24	207.8	13.035
15	1.01	93	9.235	32	6	158.9	13.035
16	17.48	96	1.228				

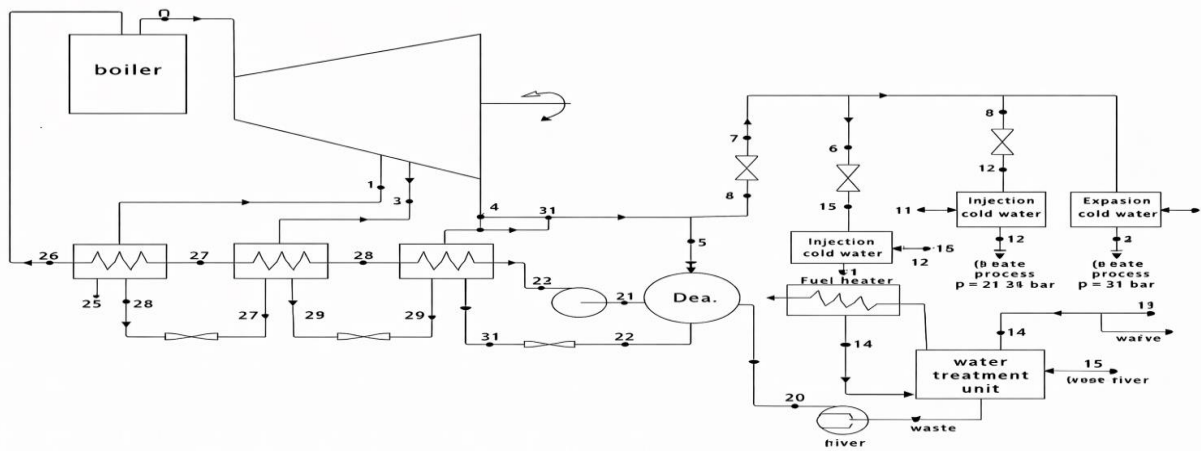


Figure 8. Schematic diagram for unit 2 with turbine R-50-127-13

Table 7. Description data of the steam circuit for units (2) Dura's refinery with the turbine type R 50-127-13

Pt.	P (bar)	t (°C)	m (kg/s)	Pt.	P (bar)	t (°C)	m (kg/s)
0	130	565	87.38	17	1.01	60	9.751
1	35.65	387.6	3.571	18	1.01	27	2.592
2	24.7	342.1	1.114	19	1.01	27	51.513
3	21.55	332.9	9.09	20	1.01	33.2	61.798
4	21.55	332.9	11.663	21	6	33.3	61.798
5	21.55	332.9	0.49	22	6	155	87.38
6	21.55	332.9	18.339	23	130	157	87.38
7	21.55	332.9	42.969	24	130	210.4	87.38
8	18.24	329.5	18.339	25	130	217.2	87.38
9	21.55	260	45.561	26	130	236.9	87.38
10	3.77	313.8	0.49	27	35.65	243.6	3.571
11	1.01	60	45.561	28	24.7	223.3	3.571
12	18.24	260	19.445	29	24.7	223.3	4.829
13	3.77	194	0.534	30	21.55	207.8	4.829
14	1.01	27	1.106	31	21.55	207.8	13.919
15	1.01	27	0.044	32	6	158.9	13.919
16	3.77	141	0.534	33	21.55	332.9	61.798
				34	21.55	332.9	82.551

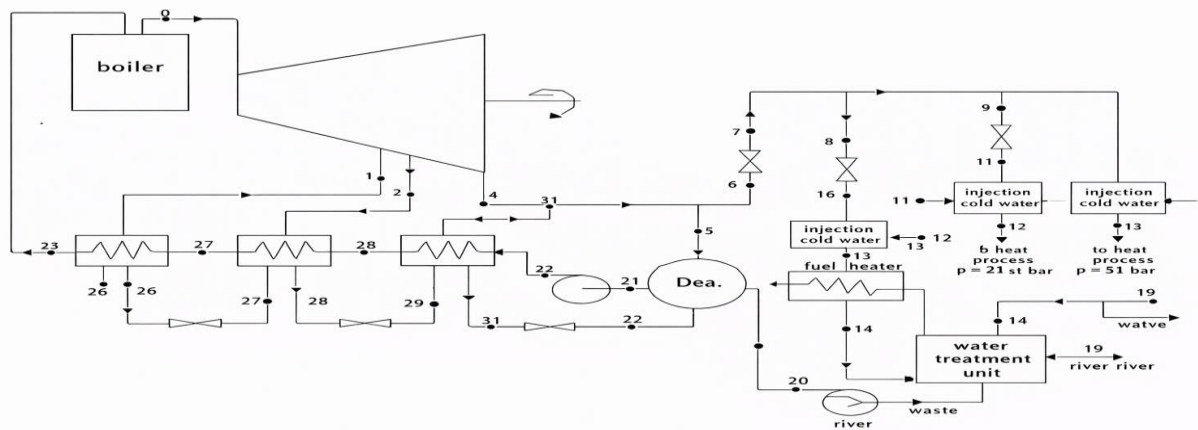


Figure 9. Schematic diagram for unit 1 with turbine R-25-90-31

Table 8. Description data of the steam circuit for unit (1) Dura's refinery with the turbine type R 25-90-31

Pt.	P (bar)	t (°C)	m (kg/s)	Pt.	P (bar)	t (°C)	m (kg/s)
0	90	535	80.012	17	1.01	96	9.205
1	45.77	450	4.015	18	1.01	27	45.163
2	39.28	434.1	2.222	19	1.01	40	55.557
3	31	402.8	73.775	20	6	40	55.557
4	31	402.8	55.557	21	90	146.9	80.012
5	31	402.8	6.901	22	90	147.8	80.012
6	31	402.8	11.317	23	90	220	80.012
7	18.24	393.3	55.557	24	90	233.6	80.012
8	18.24	393.3	1.063	25	90	256.6	80.012
9	18.24	393.3	54.494	26	45.77	258.4	4.015
10	1.01	27	5.856	27	39.28	249.3	4.015
11	18.24	260	60.35	28	39.28	249.3	6.237
12	1.01	27	0.126	29	31	257	6.237
13	17.55	248	1.189	30	31	230.1	17.554
14	17.48	96	1.189	31	6	158.8	17.554
15	18.24	94	61.369	32	1.01	94	52.164
16	18.24	393.2	1.063				

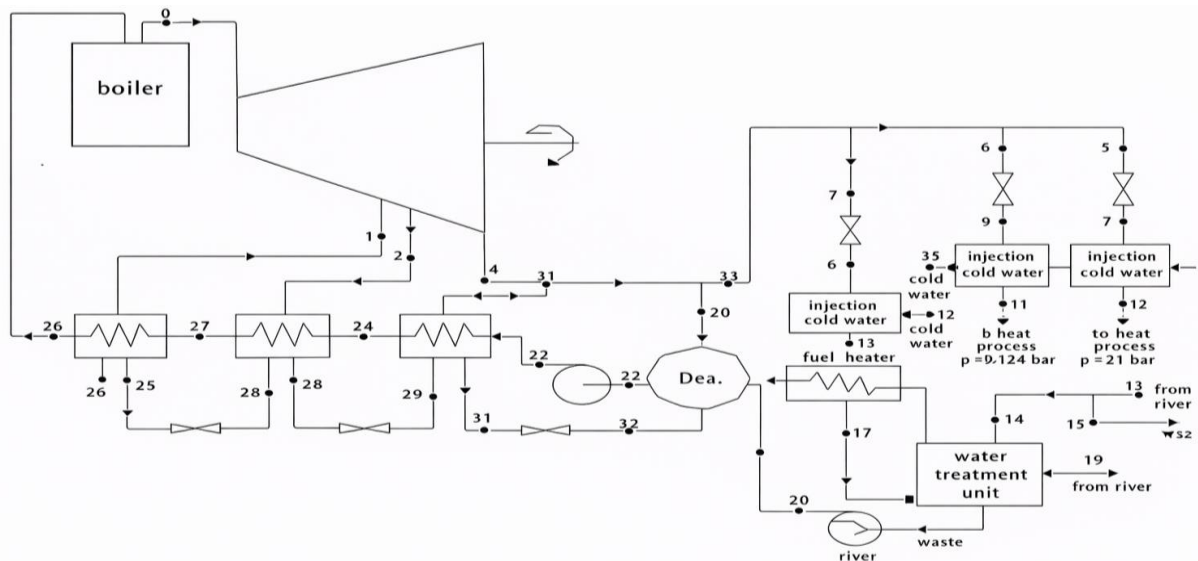


Figure 10. Schematic diagram for unit 2 with turbine R-25-90-31

Table 9. Description data of the steam circuit for unit (2) Dura's refinery with the turbine type R 25-90-31

Pt.	P (bar)	t (°C)	m (kg/s)	Pt.	P (bar)	t (°C)	m (kg/s)
0	90	535	85.62	19	6	54.9	58.924
1	48.18	447.2	4.776	20	31	400	7.177
2	40.74	429.1	2.698	21	31	400	12.045
3	31	400	78.146	22	6	158	85.62
4	31	400	0.438	23	90	159	85.62
5	31	400	17.552	24	90	230	85.62
6	31	400	40.934	25	90	245	85.62
7	21.5	392.8	40.934	26	90	269.9	85.62
8	18.24	392.2	17.552	27	48.18	261.6	4.776
9	3.73	378.5	0.438	28	40.74	251.6	4.776
10	21.55	260	45.561	29	40.74	251.4	7.474
11	18.24	260	19.445	30	31	235.7	7.474
12	3.77	194	0.5	31	31	235.7	19.519
13	1.01	60	45.561	32	6	158.9	19.519
14	1.01	94	19.445	33	31	400	58.924
15	1.01	60	9.751	34	1.01	60	0.062
16	1.01	27	48.673	35	1.01	60	1.893
17	3.77	141	0.5	36	1.01	60	4.627
18	1.01	54.87	58.924				

Table 10. The results of thermodynamic analysis for the present unit (1) Dura's refinery
 (Kathem, 2007)

No.	Equipment	1 st law		Entropy method	
		Efficiency %	Losses to Energy supplied %	Effectiveness %	Losses to availability % (Ω)
1	Elect. gen. turbine	56	-	57.4	1.578
2	First Cond.	100	-	100	0.6236
3	Fuel heater	99.89	0.00132	45.266	0.2524
4	Main storage tank	99.85	0.0075	36.78	0.09205
5	Purity tank	98.5	0.05466	75.044	0.1742
6	deaertor	99.9	0.01354	52.55	0.8442
7	turbine of the F.W. pump	56.3	-	62.321	0.03342
8	F.W. pump	46	-	45.478	0.04842
9	Air comp.	76.4	-	85.5	0.1346
10	turbine of air comp.	76.4	-	80.7	0.091
11	turbine of an oil pump	29.4	-	34.9	0.031
12	turbine of cond. Water pump	26.1	-	31.9	0.03568
13	Cond. Water pump	46	-	40	0.00739
14	Boiler blower turbine	31.44	-	37	0.09809
15	turbine of the cooling tower pump	47.73	-	45.4	0.9632
16	Secondary cond.	100	-	100	0.3191
17	Steam boiler	82.5	17.5	29.62	70.38
18	Rejected from the distillation tower	-	5.6	-	0.544
19	η_e			2.19%	
20	η_s			61.43%	
21	η_{plant}	63.628%		23.66%	
22	PHR	0.028			
23	Specific fuel consumption for process heat (kg/GJ) (physics method)	29.601		Specific fuel consumption for electric (g/kW.hr) (physical method)	1115.746
24	Specific fuel consumption for process heat (kg/GJ) (availability method)	36.45		Specific fuel consumption for electric (g/kW.hr) (availability method)	363.512

Table 11. The result of thermodynamic analysis for the present unit (2) Dura's refinery (Kathem, 2007)

1 st law				Entropy method	
No.	Equipment	Efficiency %	Losses to Energy supplied %	Effectiveness %	Losses to availability % (Ω)
1	Oil pump turbine	18	-	23.7	0.04459
2	Air blower turbine	50.3	-	58.4	0.3491
3	turbine of the F.W. pump	39.2	0.001384	47.7	0.3071
4	F.W. pump	44.1	-	39.5	0.2261
5	Fuel heater	96.9	0.01606	57	0.0602
6	deaerator	99.7	0.06198	59	2.4
7	Treated water pump	32.1	-	100	0
8	Steam boiler	81.7	18.3	31.5	68.5
9	Rejected from the distillation tower	-	4.6	-	0.658
10	PHR	0			
11	Thermal eff.			77%	
12	Overall eff.	77.1%		28.15%	

Table 12, the result of thermodynamic analysis for units (1) with turbine type R 50-127-13 (Kathem, 2007)

1 st law				Entropy method	
No.	Equipment	Efficiency %	Losses to Energy supplied %	Effectiveness %	Losses to availability % (Ω)
1	boiler pump	95.5	-	68.3	0.1496
2	Cond. pump	97.57	-	84.12	0.00191
3	Fuel heater	99.1	0.1122	46.9	0.234
4	Elect. gen. turbine	77.6	-	86.7	2.2
5	Steam boiler	82.5	17.5	42.9	57.1
6	deaertor	98.3	0.3848	57.4	2.366
7	1st heater	99.9	0.02224	98.2	0.1375
8	2nd heater	99.9	0.0273	98.9	0.0738
9	3rd heater	100	0	95.2	0.3322
10	1st ejector	100	0	99.7	0.358
11	2nd ejector	100	0	99.7	0.001159
12	Air comp.	76.178	-	87.13	0.0473
13	valves	100	0	97.4	0.0542
14	Rejected from distillation tower	2	5.6	-	0.546
15	η_e	14.384%			
16	η_s	62.507%			

17	η_{plant}	76.891%		36.378%	
18	PHR	0.227			
19	Specific fuel consumption for process heat (kg/GJ) (physics method)	29.6		Specific fuel consumption for electric (g/kW.hr) (physical method)	146.903
20	Specific fuel consumption for process heat (kg/GJ) (availability method)	23.78		Specific fuel consumption for electric (g/kW.hr) (availability method)	237.186
21	saving rate %			24.4	

Table 13. The result of the thermodynamic analysis for unit (2) with turbine type R 50-127-13 (Kathem, 2007)

1 st law				Entropy method	
No.	Equipment	Efficiency %	Loss to Energy supplied %	Effectiveness %	Losses to availability % (Ω)
1	boiler pump	83	-	63.3	0.2012
2	Cond. pump	60.8	-	52.2	0.009645
3	Fuel heater	100	0.001278	58.5	0.0531
4	Elect. gen. turbine	77.9	-	87.3	2
5	Steam boiler	82.9	17.1	43.1	56.9
6	deaertor	99.9	0.02179	54.6	2.5
7	1st heater	99.9	0.03483	93.5	0.4797
8	2nd heater	99.9	0.03291	99.4	0.04202
9	3rd heater	99.9	0.03564	98.3	0.1346
10	1st ejector	99.8	0.1017	97.6	0.4073
11	2nd ejector	100	0	94.7	0.008072
12	3rd ejector	99.9	0.000581	98.1	0.1357
13	valves	100	0	97.4	0.02406
14	Rejected from distillation tower	-	4	-	0.3064
15	η_e	13.1%			
16	η_s	65.7%			
17	η_{plant}	78.8%		36.5%	
18	PHR	0.195			
19	Specific fuel consumption for	29.458		Specific fuel consumption for electric	123.00

	process heat (kg/GJ) (physics method)			(g/kW.hr) (physical method)	
20	Specific fuel consumption for process heat (kg/GJ) (availability method)	23.367		Specific fuel consumption for electric (g/kW.hr) (availability method)	232.97
21	saving rate %			18.6	

Table 14. The result of the thermodynamic analysis for unit (1) with turbine type R 25-29-31 (Kathem, 2007)

1 st Law				Entropy method	
No.	Equipment	Efficiency %	Loss to Energy supplied %	Effectiveness %	Losses to availability % (Ω)
1	boiler pump	97.7	-	70.5	0.09647
2	Cond. pump	93.6	-	78.4	0.002917
3	Fuel heater	99.4	0.001278	46.6	0.2493
4	Elect. gen. turbine	71	-	84.3	1.5
5	Steam boiler	82.5	17.5	41.4	58.6
6	deaertor	99.9	0.02165	54.9	2.3
7	1st heater	99.8	0.1023	89.6	0.9917
8	2nd heater	99.9	0.05184	98.6	0.1156
9	3rd heater	99.8	0.06269	98.1	0.1877
10	1st ejector	100	0	95.5	1.2
11	2nd ejector	100	0	94.7	0.02746
12	Air comp.	99.9	0	87.1	0.05224
13	valves	100	0	94.1	1.9
14	Rejected from distillation tower	-	6.4	-	0.6449
15	η_e	7.7%			
16	η_s	68.1%			
17	η_{plant}	75.8%		32%	
18	PHR	0.115			
19	Specific fuel consumption for process heat (kg/GJ) (physics method)	29.149		Specific fuel consumption for electric (g/kW.hr) (physical method)	190.046
20	Specific fuel consumption for process heat	26.149		Specific fuel consumption for electric	268.654

	(kg/GJ) (availability method)			(g/kW.hr) (availability method)	
21	saving rate %			21.7	

Table 15. The result of the thermodynamic analysis for unit (2) with turbine type R 25-29-31 (Kathem, 2007)

No.	Equipment	1 st		Entropy method	
		Efficiency %	Loss Energy supplied % to	Effectiveness %	Losses to availability % (Ω)
1	boiler pump	86.5	-	65.4	0.1296
2	Cond. pump	71.3	-	60.9	0.006848
3	Fuel heater	99.7	0.001384	57.9	0.05414
4	Elect. gen. turbine	73.2	-	85.7	1.4
5	Steam boiler	82.7	17.3	41.4	58.6
6	deaertor	99.9	0.01878	52.5	2.4
7	1st heater	99.7	0.0383	89	1
8	2nd heater	99.9	0.03291	98	0.172
9	3rd heater	99.9	0.05783	98	0.1986
10	1st ejector	99.8	0.1299	95.9	0.7667
11	2nd ejector	99.8	0.05004	95.8	0.329
12	3rd ejector	99.6	0.002519	91.4	0.01349
13	valves	100	0	95	1.3
14	Rejected from distillation tower	-	4.2	-	0.4783
15	η_e	7.7%			
16	η_s	70.4%			
17	η_{plant}	78.1%		33.3%	
18	PHR	0.11			
19	Specific fuel consumption for process heat (kg/GJ) (physics method)	29.529		Specific fuel consumption for electric (g/kW.hr) (physical method)	190.03
20	Specific fuel consumption for process heat (kg/GJ) (availability method)	26.89		Specific fuel consumption for electric (g/kW.hr) (availability method)	268.2
	saving rate %			12.5	