Energy Conversion and Management 52 (2011) 703–712

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Exergy analysis of a steam production and distribution system including alternatives to throttling and the single pressure steam production

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ARTICLE INFO

Article history:
Received 2 December 2009
Accepted 30 July 2010
Available online 21 August 2010

Keywords:
Exergy
Marginal efficiency
Throttling
Steam turbine
Boiler
HRSG

ABSTRACT

The operative steam production and distribution system of Kårstø natural gas processing plant at four operating conditions was studied by exergy analysis. The eight boilers, of which two are direct fired produce steam at a single pressure. The steam (at 59 bar, 420 °C) is then distributed by extensive use of throttling. Most of the steam is utilized at low pressure (7 bar, 200 °C). The effect of implementing steam turbines in steam distribution system, increase of steam production pressure (from 59 to 120 bar) or two-stage pressure steam production (120/59 bar) were examined. The exergy efficiency of the existing system was 44.3%. Implementing steam turbines or elevation of production pressure (single/dual) resulted in marginal exergy efficiencies of 92%, 89.8% and 98.7%, respectively. Combinations of steam turbines and elevated pressure gave a ratio of 90.9%. In terms of lower heating value, the marginal electric efficiencies ranged from 89% to 104%. The single most fuel exergy demanding alternative was the elevated pressure, with 3.6% increase, which resulted in 18.5 MW of extra electric power. The corresponding figures for steam turbines and two-stage pressure were 3.0%/16.2 MW and 2.6%/15.6 MW. Thus, the study provided an example from an existing, industrial steam system that illustrates both the losses in throttling and low-pressure steam production, and the practical potentials for improvement.

1. Introduction

Kårstø natural gas processing plant was built in the early 1980s on the south-western coast of Norway to receive natural gas from the northern part of the North Sea. It has got a nominal capacity of handling 88 million standard cubic meter rich gas per day [1], which made Kårstø the third largest shipping terminal worldwide in 2004, and the largest one in Europe. The plant was extended in −93 and −05, and at present it delivers ethane, propane, iso-butane, normal-butane and naphta by boat and methane-rich sales gas through pipelines. The steam production is an important part of the utility system of the gas processing plant, mainly for heating purposes, but also for steam turbines, cleaning and pollutant reduction in combustion.

Throttling of superheated steam and mixing with water are a widespread practice in the industry. High-pressure steam is more convenient to transport compared to low-pressure steam with the same enthalpy due to the lower volumetric flow. Mixing with water is a simple and instant way of increasing the mass flow, and also to control the quality. Moreover, both processes are relatively easy to regulate. At Kårstø the steam is produced at one and relatively low pressure level only, while even lower pressure steam is utilized the most.

From an energetic viewpoint, throttling and mixing give no losses as the enthalpy is maintained. It is well known, however, that throttling and mixing of flows with different temperatures are irreversible processes that involve entropy production and destruction of exergy (irreversibility). Design guidelines based on the Second Law of Thermodynamics (e.g. [2,3]) include clear recommendations on avoiding throttling and mixing whenever possible. Producing steam at a relatively low pressure level seems somewhat modest given the small pump work compared to the power achieved by steam turbines. Two-stage (and more) pressure steam production give better fitted heating curves, thus reducing the irreversibilities [4].

It is common knowledge in thermodynamics that the majority of the exergy losses is located in combustion and heat exchange over finite (huge) temperature differences [3,4]. Naturally the main focus exergywise is normally put on these areas. We, on the other hand, have studied an old-established operating steam production and distribution system that uses throttling extensively. Gas turbines (GTs), heat recovery steam generators (HRSGs) and boilers could of course have been replaced to improve the efficiency. However, the focus of attention in addition to mapping the operational status today, is to examine the actual effects of reducing the throttling in steam distribution system (SDS) and careful elevation of

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doi:10.1016/j.enconman.2010.07.049

the pressure level of steam production or two-stage pressure steam production with steam turbines with moderate efficiencies. By examining these implementations, we get a good sense of how such relatively cautious actions will affect the exergy efficiency of Kårstø steam production and steam distribution system.

The question to be answered is what, compared to the existing system, can be achieved by using one of the alternatives. From the viewpoint of the owner and operator, this can be expressed in terms of additional electric energy production (i.e. saved purchase) divided by the additional fuel energy consumed. This is the marginal electric efficiency of the modification, and it can be compared to a typical figure for electric efficiency of a new conventional power plant, approximately 55–58%. From a thermodynamic viewpoint it is also interesting to explain the changes, for which the exergy flows and distribution of the degradation are important inputs.

Initial studies were presented by Wølneberg [5] and Rian et al. [6].

2. Process description and problem specifications

The steam production system, including the return water of the users and the make up water, is illustrated in Fig. 1. It consists of eight boilers, of which only two are directly fired. The others (HRSGs) utilize the exhaust of the gas turbines (GTs) to produce high pressure steam. Supplementary firing (SF) is necessary due to quantity and quality control. The compositions of the fuels vary, as they depend on the type of gas which is available at the site. The boilers deliver steam at minimum pressure and temperature of 59 bar and 420 °C, respectively.

The six cogen-units are three Rolls Royce (RR) Avon GTs each connected to a Foster-Wheeler (FW) HRSG, one GE Frame 6 GT connected to a Moss HRSG and two GE LM2500 GTs each connected to an Aalborg HRSG. KEP and Sleipner are the direct fired boilers.

The HRSGs/boilers have a continuous blowdown to remove impurities. The GE Frame 6 GT produces at most 38 MW of power, while the RR Avon GT and the GE LM2500 GT give 12.32 MW and 28 MW of mechanical work, respectively. The three Avon-FW units constitute the CHP of the original plant in 1983. Alternatives for replacement of these are investigated in [7].

For presentation purposes the water pumps and the STs utilizing the expansion of the elevated boiler/HRSG pressure (see below) were regarded as parts of the steam production system (CHPs/boilers).

Fig. 2 gives a flowsheet of the existing steam distribution system. Most of the high pressure steam is mixed with water to in-

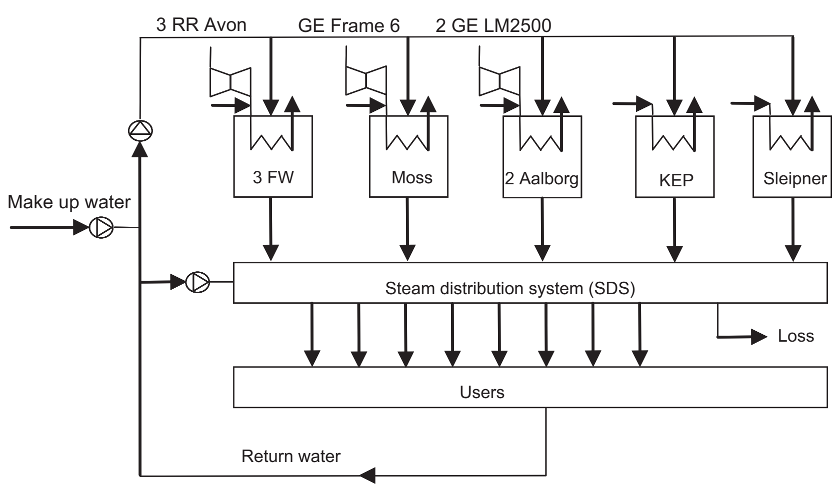

Fig. 1. The existing steam production system (CHP/boilers) and the accompanying steam flows.

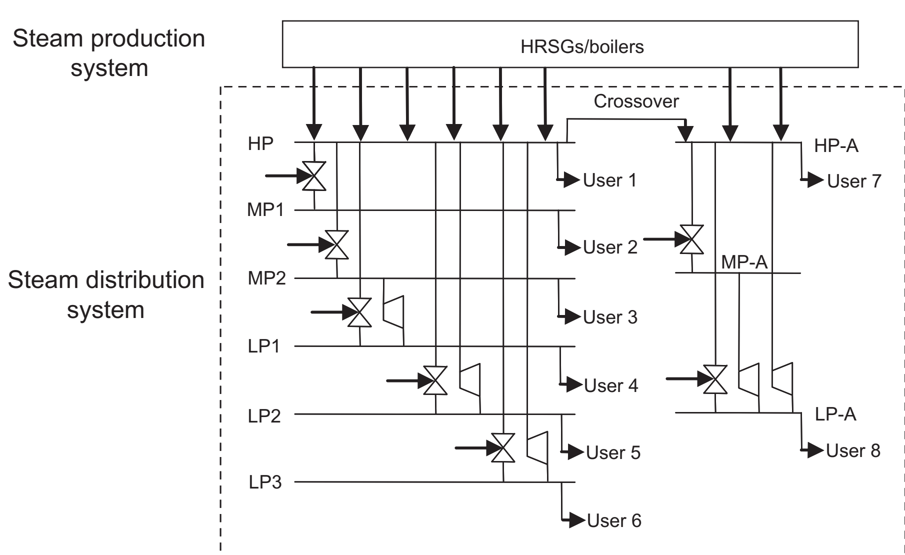

Fig. 2. The existing steam distribution system (SDS). The user flows go out of the sub-system.

Table 1
States and mass flow rates (t/h) through the existing SDS.

Header	HP	MP1	MP2	LP1	LP2	LP3	HP-A + crossover	LP-A	Total
p (bar)	59	44	38	7	7	7	59	7	
T (°C)	420	267	370.4	239.4	199.7	174	420	200	
Delivery to pressure level	335.7	29.1	52.2	71.4	157.1	46.2	169.1	162.7	335.7
Extra water injected		4.2	1.6	2.7	9.7	4.3		3.8	26.3
Crossover from HP							39.1		
Lost							3.0		3.0
Throttled from HP		24.9	50.6	21.5	58.1	21.7		19.2	
Expanded from HP					89.3	20.2			
Expanded from MP2				45.6					
Expanded from MP-A								121.4	
Throttled/expanded from MP-A								18.3	
Utilized (t/h)	10.3	29.1	5.0	71.4	157.1	46.2	7.2	162.7	489.0
User	User 1	User 2	User 3	User 4	User 5	User 6	User 7	User 8	$\sum$User

crease the amount of steam and to control the quality. It is then throttled to the specific state, which is mostly low pressure. Steam is delivered to different parts of the processing plant at three main levels of pressure: High (HP), intermediate and low pressures. There are two levels of intermediate pressure (MP1, MP2) and three states of low pressure (LP1, LP2, LP3). No steam is utilized at MP-A (38 bar, 368.6 °C). Table 1 shows the states and mass flows of the existing SDS for a particular case (A1). Most (97%) of the steam is passed onto the lower pressure levels from the HP-header. Some of the extra water injected to SDS (to increase the amount/ quality purposes), is transferred between levels.

The return water for the boilers was pumped up to 69.9 bar (125 °C), whereas the return water for SDS was pumped up to 93 bar (125 °C). There are some STs in the existing SDS, which provide mechanical work for cooling compressors, pumps and fans.

The existing steam production system (CHPs/boilers) and steam distribution system (SDS) at two delivery rates, denoted A and B, and two production distributions among the boilers/HRSGs, denoted 1 and 2, were analysed. Thus, for the cases A1 and BI, every boiler/HRSG was on duty, whereas for the cases A2 and B2, one HRSG (FW) was inoperative, which is a probable scenario as those boilers are old and often require maintenance. The existing plant was referred to as Alt. 0.

Then some means for improving both the SDS and the steam production system were investigated in terms of the following alternatives:

Alt. 1 Additional steam turbines in a modified SDS.
- Throttling valves were replaced by STs in the SDS.
- This required extra steam production, which was assumed to be produced by the FW HRSGs, Moss HRSG, Aalborg HRSGs, Sleipner boiler or KEP boiler (termed ST I, ST II, ST III, ST IV and ST V, respectively).
- This alternative was investigated for all the four cases, A1, A2, B1 and B2, while the remaining alternatives (Alt. 2–5) were conducted for case A1 only.

Alt. 2 All steam produced at an elevated pressure, 120 bar (termed HP).
- The steam was expanded in new STs from 120 to 59 bar and then fed into the existing SDS.

Alt. 3 Two-stage pressure steam production, 120 bar/59 bar (termed 2-P).
- The steam produced at 120 bar was expanded to 59 bar through a ST and then all of the steam was fed into the existing SDS.

Alt. 4 Steam production at an elevated pressure, 120 bar, and STs in SDS (termed HP/ST).
- All steam was produced at 120 bar and expanded through STs to 59 bar and fed into the modified SDS.
- Throttling valves were replaced by STs in the SDS.

Alt. 5 Two-stage pressure steam production, 120 bar/59 bar, and STs in SDS (termed 2-P/ST).
- The steam which was produced at 120 bar was expanded to 59 bar through a ST, fed into the modified SDS.
- Throttling valves were replaced by STs in the SDS.
- The extra steam needed was produced at the Aalborg HRSGs only.

Hence, there were two changes from the existing system (Alt. 0): First, the SDS was modified (Alt. 1, 4 and 5) or not modified (Alt. 2 and 3). Second, the steam production pressure level was maintained (Alt. 1) or changed (Alt. 2–5). The alternatives with elevated pressure would require some remodeling of the HRSGs/boilers. It was assumed that the replaced boilers/HRSGs had the same steam production capacity rates as the existing boilers.

For all alternatives the steam deliveries to the users were unchanged and equal to those of the existing system. It was out of the scope of this study to evaluate the use of steam at the processing plant.

3. Theory and method

The thermal enthalpy was defined as the enthalpy at the actual state relative to the chosen ambient temperature and pressure (T_0, p_0) as

$$h_{\text{th}} = h - h_0 = h(T,p) - h(T_0,p_0) \tag{1}$$

The total enthalpy was determined as the sum of the thermal enthalpy and the lower heating value (LHV) of the substance. LHVs of Kotas [8] were utilized. The fuel, air and exhaust at ambient pressure were regarded ideal mixtures. Thus, enthalpies were calculated as weighed sums of component enthalpies [4].

The exergy balance is developed by combining the balances of mass, energy and entropy [8,9]. For a steady state, non-expanding (sub-)system, the balance can be formulated as

$$0 = \dot{E}^Q - \dot{W} + \sum_{in} \dot{m}_k \varepsilon_k - \sum_{out} \dot{m}_k \varepsilon_k - \dot{I} \tag{2}$$

where $\dot{E}^Q$ is the rate of exergy transferred with heat to the control volume (CV), $\dot{I}$ is the rate of irreversibility (exergy destruction) in the CV, and ε_k is the specific flow exergy (per kg) [8,9] of flow k across the CV-boundary. The flow exergy may be split into a thermomechanical and a chemical exergy component, $\varepsilon = \varepsilon_{\text{th}} + \varepsilon_0$. The thermomechanical exergy is determined from

$$\varepsilon_{\text{th}} = h - h_0 - T_0(s - s_0), \tag{3}$$

where $h_0 = h(T_0, p_0)$ and $s_0 = s(T_0, p_0)$ are the values at the restricted dead state for the relevant flow (mixture).

For a single, gaseous component present in the atmosphere, the chemical exergy is determined as

$$\bar{\varepsilon}_{0,i} = \bar{R}T_0 \ln(p_0/p_{i,0}) = -\bar{R}T_0 \ln(x_{i,0}) \tag{4}$$

where $\bar{R}$ is the universal gas constant, x_i^e is the mole fraction of the species i in the atmosphere and $p_{i,0}$ is the corresponding partial pressure. The overbars denote molar quantities. Data for the chemical exergy of other species was obtained from Kotas [8] which is given at a reference state of 1 atm, 25 °C and 28% relative humidity (RH). In the present analysis, they were corrected for deviating ambient conditions according to [9,10] as

$$\bar{\varepsilon}_{0,i} = \bar{\varepsilon}_i^0 \frac{T_0}{T^0} + \bar{h}^0_{\mathrm{LHV}} \frac{T^0 - T_0}{T^0} + T_0\bar{R} \sum_{j \neq i} \nu_j \ln \frac{x_j^0}{x_{j,0}} \tag{5}$$

Here $\bar{\varepsilon}_i^0$ and $\bar{h}^0_{\mathrm{LHV}}$ are the molar chemical exergy and the molar lower heating value, respectively, determined at the reference state of 1 atm, 25 °C, 28% RH. The superscript ^{0} denotes this reference state, while the subscript $_0$ denotes the ambient state chosen for this analysis. The index j denotes the co-reactants and the products of the reference reaction while ν_j is the stoichiometric coefficient of each species in the reaction of fuel and atmospheric oxygen. Thus, x_j^0 and $x_{j,0}$ denote the atmospheric mole fractions of oxygen and reaction products in, respectively, the reference and ambient states.

The chemical exergy of a mixture is determined from

$$\bar{\varepsilon}_{0,\mathrm{mix}} = \sum x_i \bar{\varepsilon}_{0,i} + \bar{R}T_0 \sum x_i \ln x_i, \tag{6}$$

where x_i is the actual mole fraction of species i in the mixture. The last term represents the reduced exergy due to the mixing of the components.

The analyses of the overall system and the sub-systems were based on steady state rate balances of mass, amounts of species or elements, energy and exergy.

The commercially available program PRO/II (ver 8.0) [11] provided enthalpy and entropy differences of the flows, with a Soave–Redlich–Kwong (SRK) equation of state [12] for the water/steam and SRK Kabadi–Danner [13] for the fuel, the air and the exhaust. The corresponding exergy differences were then calculated from these differences and balanced in a spreadsheet. Hence the exergy calculator of PRO/II was not used.

The exergy efficiencies of the (sub-)system(s) are determined from

$$\psi_i = \frac{\dot{E}_{Utilized}}{\dot{E}_{Supplied}} \tag{7}$$

For the CHP $\dot{E}_{Utilized}$ comprises net work rate (electrical and mechanical, as delivered by the system) and rate of thermomechanical flow exergy increase in water/steam (including thermomechanical increase for the make up water), while the $\dot{E}_{Supplied}$ is the fuel exergy rate.

For the SDS $\dot{E}_{Utilized}$ is the net work rate (electrical and mechanical) and the thermomechanical exergy rate of the users. $\dot{E}_{Supplied}$ is the rate of thermomechanical exergy of steam and rate of thermomechanical exergy of water delivered to the SDS.

For the total system (i.e. CHP and SDS) $\dot{E}_{Utilized}$ is the net work rate and the rate of thermomechanical exergy increase for the users. $\dot{E}_{Supplied}$ is the fuel exergy rate.

Thus, the marginal efficiency for the total system is then expressed as the ratio of net increase of power to increase in fuel consumption.

The sum of exergy efficiency and irreversibility ratio thus equals unity. The irreversibility ratio of SDS is then the ratio of the lost exergy to thermomechanical exergy supplied to the high pressure header.

4. Present assumptions

The following was specified prior to the analysis:

- Air fuel ratios (kg/kg) for each GT combustor were kept constant for every alternative: 68 (RR Avon), 49.23 (GE Frame 6), 51.49 (GE LM 2500), 16.49 (Sleipner) and 5.75 (KEP) which corresponded to excess air ratios of 4.22, 3.05, 3.26, 1.15 and 1.14, respectively. Thus, the efficiencies of GTs and HRSGs/boilers were regarded constant for high part load.
- The supplementary firing (in SF combustor) was in accordance with the temperatures of exhaust gases discharged to environment, 185 °C (FW), 200 °C (Moss), 180 °C (Aalborg), 180 °C (Sleipner), 140 °C (KEP). These temperatures corresponded to figures given by the plant and fulfilled the physical conditions (i.e. no intersecting heating curves).
- The rate of Fuel 4 was fixed (17.7 t/h) and corresponded to the original system for case A1. All of Fuel 4 was utilized. Thus, to implement the different alternatives, for KEP Fuel 2 was used in addition to Fuel 4. The excess air ratio, corresponding to Fuel 4, was kept constant (1.14).
- Air composistion on molar basis: 77.09% N_2, 20.69% O_2, 0.93% Ar, 0.03% CO_2 and 1.26% H_2O.
- Ambient temperature and pressure of 15 °C and 1.013 bar, respectively, relative humidity of 75%. From meteorological data [14] for a nearby location (Haugesund airport) this appeared to be a representative atmospheric state in the summer.
- Efficiency of electric generator was 97%.
- Stream losses in SDS (leakages, dumped steam, etc.) were assumed to 3 t/h. They were replaced by water at ambient conditions, which was pumped up to 5 bars then heated by cold exhaust (200 °C) from the Moss boiler to the state of the return water.
- Assumed state of return water was 122 °C, 5 bar. It was then pumped to requisite pressure levels in the boilers/HRSGs and SDS.
- Blowdown in the boilers was neglected.
- All components were assumed to be adiabatic.
- Changes in kinetic and potential energy were neglected.
- The extra power from SDS due to the additional STs was electrical, while the already existing STs delivered mechanical work.
- Efficiency of new STs in SDS: 59.9–80%. Chosen to maintain the throttled state of steam. (Water injection was required in two instances to achieve the correct temperature.)
- Isentropic efficiency for steam turbines at elevated pressure of steam production, 120 bar was 70%.
- 10.9 bar pressure drop for two-stage pressure steam production (120/59 bar).

The compositions of the fuel mixtures used in the calculations are defined in Table 2. Fuel 4 consisted of components removed from the processed gas to meet the specifications of the sales gas. Thus, using Fuel 4 in the direct fired boilers solved one of

Table 2
Compositions (%) of fuel mixtures.

Component	Fuel 1	Fuel 2	Fuel 3	Fuel 4
N_2	0.71	0.92	0.61	0.02
CO_2	1.76	6.32	2.55	57.48
CH_4	92.05	53.41	87.91	10.92
C_2H_6	4.58	37.60	8.18	31.60
C_3H_8	0.79	1.58	0.70	0.00
C_4H_{10}	0.10	0.18	0.06	0.00

Table 3
Distribution of steam production (t/h) among the boilers.

Boiler/HRSG	A1	A2	B1	B2
Foster-Wheeler	120.0	72.0	100.0	72.0
Moss	80.0	50.0	80.0	50.0
Aalborg	130.0	139.0	130.0	130.0
Sleipner	45.7	102.5	47.3	97.5
KEP2005	90.0	102.5	90.0	97.5
Sum	465.7	466.0	447.3	447.0

the waste problems on the site, as discharge would not be allowed by the Norwegian authorities.

Table 3 shows both the distribution when all boilers were on duty (A1 and B1) as well as the cases where one FW was inoperative (A2 and B2). For the latter cases, the steam productions at the remaining two FWs were increased and the loads at Sleipner and KEP were increased considerably. Normally, it is thermodynamically beneficial to run those HRSGs with minimum additional heating instead of using boilers when rearranging the distribution of the steam production rate among the boilers. However, the direct fired boilers, Sleipner and KEP, are base load units due to the low NO_x-emissions and the fuel.

In day-to-day operation the process flows and hence the steam required will vary. However, the specified flow rates and compositions are realistic cases for the plant.

Steam turbines were operated at high load. Thus, efficiency of steam turbines at (low) part load; including finding the critical (minimum) volume flow, was not covered by the analysis. By inserting STs in parallell rather than replacing the throttling valves in real life, shut down could be avoided when the amount of water is below the critical amount for the STs. Assumed mass ratio of high-pressure steam to low-pressure steam was 5:1 for the two-stage pressure steam production. Optimal distribution of mass ratios of steam at two levels was neither a part of the analysis.

5. Results and discussion

The chosen means, i.e. steam turbines, elevated pressure/two-stage steam production, for improving this system are well known from thermodynamics. They will surely increase the fuel consumption, amount of steam production as well as power. The additional needed steam production does not exceed max capacity of the HRSGs/boilers. The system solution for this plant from the 1980s seemed at first sight robust, yet simple and poorly exergetic efficient. It is thus important to keep in mind that this is a definite, real industrial plant built at a time where the fuel used on-site was virtually free of charge.

5.1. Performance of the existing steam production system

Table 4 presents the steam production for one of the cases (A1) for the existing system and the corresponding fuel consumption.

Table 5 describes the exergy rates through the steam production system, including the GTs, for two production rates (where A > B) and two steam production distributions among the boilers/HRSGs (1 and 2, where the latter indicates that one FW HRSG is inoperative). The fuel exergy, and thus the irreversibilities, increased when one of the FW HRSGs was inoperative. As the steam production rate was larger for case A1 compared to B1, the irreversibility rate was also larger.

The performances of the CHPs and boilers are given in Table 6. By comparing exergy efficiencies for the cogen-units and the direct fired boilers, the latter was outnumbered as expected, at most with 15%-points. The exergy efficiency of RR Avon-FW dropped from 45.3% (A1) to 37.2% (A2) when one HRSG was inoperative, as the exhaust from one turbine was not utilized and more steam was then produced by supplementary firing. Since steam production by heat recovery was replaced by direct firing and supplementary firing, the overall exergy efficiency was also reduced, close to 2%-points.

Table 7 summarizes the irreversibilities in the total system in terms of absolute figures (MW), distributions (%) and also irreversibility ratios (%). The steam distribution system contributed to only

Table 4
Steam production (t/h) and fuel consumption (t/h) from both exhaust gas (EG) and supplementary firing (SF) for case A1[a].

Boiler(s)/HRSG(s)	Steam production	Fuel consumption	
		EG	SF
3 Foster-Wheeler	120	9.07[1]	2.87[2]
1 Moss	80	7.23[1]	1.91[2]
2 Aalborg	130	12.10[3]	2.14[3]
1 Sleipner	45.7	–	3.20[2]
1 KEP2005	90	–	17.68[4]
Sum	465.7		

[a] Superscripts denote fuel type, see Table 2.

Table 5
Rates of exergy converted (MW) in CHPs/boilers and rates of irreversibility, four cases in the existing system.

Case	A1	A2	B1	B2
Chemical fuel exergy	585.3	609.9	563.9	587.2
Thermomech. fuel exergy	4.6	4.6	4.3	4.4
Thermomech exergy to water				
Foster-Wheeler HRSGs	41.8	25.1	34.8	25.1
Moss HRSG	27.9	17.5	27.9	17.5
Aalborg HRSGs	45.3	48.4	45.3	45.3
Sleipner boiler	15.9	35.7	16.5	34.0
KEP2005 boiler	31.4	35.7	31.4	34.0
Water injected into SDS	0.1	0.1	0.1	0.1
Total to HRSGs/boilers	162.4	162.5	156.0	155.9
Work from gas turbines				
RR Avon GTs (MW)	31.0	31.0	30.9	30.9
GE Frame 6 GT (MW)	36.9	36.9	36.9	36.9
GE LM2500 GTs (MW)	53.7	53.7	49.6	49.6
Total work from GTs	121.6	121.6	117.3	117.3
Work to pumps				
Total pump work needed	2.4	2.4	2.3	2.3
Total exergy (heat and work)	281.6	281.7	271.0	270.9
Irreversibilities				
RR Avon-Foster-Wheeler	87.3	85.6	78.2	85.4
GE Frame 6-Moss	58.0	44.4	58.0	44.4
GE LM2500-Aalborg	92.6	96.5	89.8	89.8
Sleipner	23.7	53.1	24.5	50.5
KEP2005	46.7	53.0	46.7	50.4
Water injected to SDS	0.1	0.1	0.0	0.0
Total	308.3	332.8	297.3	320.7

Table 6
Exergy efficiencies (%) for CHPs and boilers, four cases of the exisiting system.

Case	A1	A2	B1	B2
RR Avon-Foster-Wheeler	45.3	37.2	45.5	37.2
GE Frame 6-Moss	52.6	54.9	52.6	54.9
GE LM2500-Aalborg	51.5	51.3	51.2	51.2
Sleipner	39.9	39.9	39.9	39.9
KEP2005	39.8	39.9	39.8	39.9
Total exergy efficiency	47.7	45.8	47.7	45.8

Table 7
Distribution (%) of total irreversibilities of CHPs/boilers and SDS, four cases of the exisiting system.

Case	A1	A2	B1	B2
RR Avon-Foster-Wheeler CHP	26.6	24.3	24.7	25.2
GE Frame 6-Moss CHP	17.7	12.6	18.4	13.1
GE LM2500-Aalborg CHP	28.2	27.3	28.4	26.5
Sleipner boiler	7.2	15.0	7.8	14.9
KEP2005 boiler	14.2	15.0	14.8	14.9
Steam distribution system	6.2	5.7	5.9	5.5
Total irreversibilities (MW)	328.6	353.0	315.8	339.3
Ratio of total irrev to fuel exergy (%)	55.7	57.5	55.6	57.3

6.2% of the total irreversibilities in case A1, as there were no heat exchange or combustion included in this sub-system. The irreversibility ratio increased close to 2%-points when one FW HRSG was inoperative.

5.2. Effects of alternative 1 – additional steam turbines in steam distribution system

Table 8 shows the distributions of both losses and utilized exergies of the transferred exergies from the boilers, as well as exergy efficiencies and irreversibility ratios. The steam turbines increased the exergy utilization of the thermomechanical exergy supplied to the water from 88.2% to 93.4% for case A1. The steam turbines were also the head contributors to irreversibilities due to the moderate efficiencies. Thus, the exergy of the steam would have been utilized even better if the STs were incorporated into the system at the planning stage of the plant. The utilized exergy of the users did not change when doing the implementation, it is only the percentage distribution that varied. For case A1 the irreversibilities of throttling amounted to 6.6% of the transferred exergy to the SDS, whereas the mixer irreversibilities amounted to 0.9%. The STs reduced those irreversibility rates with 100.0% and 88.9%, respectively. Steam production needed to increase 22.25 t/h in total for A1 and A2, and 19.22 t/h for B1 and B2 when new STs were implemented in the SDS, as most of the water injection was removed. Moreover, the implementation moved the irreversibilities from the SDS to the steam production system, due to the increased fuel consumption. For presentation purposes the cases A2 and B2 were left out in the table, as for the SDS these cases were equal to A1 and B1, respectively.

Table 9 clearly shows that throttling was the main contributor to losses in the steam distribution system with 55.6% of the irreversibilities for case A1, which corresponded to 11.26 MW.

Table 8
Effects of throttling and additional steam turbines on the utilization (%) of the thermomechanical exergy to the HP-header.

	Throttling		Additional STs	
Case	A1	B1	A1	B1
Thermomech. Exergy to HP-header (MW)	171.6	164.8	179.3	171.4
Lost (%) in				
Throttling	6.6	5.9	0.0	0.0
Steam turbines	3.7	3.9	5.6	5.5
Electric generator	0.0	0.0	0.3	0.3
Mixers	0.9	0.8	0.1	0.1
Stream losses	0.6	0.7	0.6	0.6
Irreversibility ratio (%)	11.8	11.3	6.6	6.5
Utilized (%) by				
Users	71.1	70.9	68.1	68.1
Steam turbines	17.1	17.8	25.4	25.3
Exergy efficiency (%)	88.2	88.7	93.4	93.5

Table 9
Distribution of irreversibilitites (%) in steam distribution system.

	With throttling		With additional STs	
Component	A1	B1	A1	B1
Throttling	55.6	52.6	0.0	0.0
Steam turbines	31.5	34.3	84.8	84.7
Electric generator	0.0	0.0	4.3	3.9
Mixers	7.5	7.2	1.6	1.6
Stream losses	5.4	5.9	9.4	9.8
Total irreversibility rate (MW)	20.25	18.59	11.75	11.22

The irreversibility rate for A1 decreased 42.0%, from 20.25 MW to 11.75 MW when doing the implementation of additional STs. The losses due to throttling decreased of course the most, 55.6%-points for case A1. The mixers only contributed originally to 7.5% of the irreversibilities, and this amount was reduced by close to 6%-points. The percentage distribution also changed dramatically as expected, whereas steam turbines now held the major part of the losses. This was due to those modest isentropic efficiencies.

5.3. Effects of alternatives 1–5

Exergy efficiency is a typical measure of performance. These figures are given in Fig. 3 for the five alternatives. The existing system as of today gave an exergy efficiency of 44.3% for case A1. STs in SDS would, as expected, have the smallest impact on this ratio, increasing it by only 1.4%-points. The effect of implementing the other single means (i.e. no combinations) was similar to the former. Combinations of elevated pressure level/two-stage pressure steam production and STs in SDS gave an increase of reasonable 2.9%-points and 2.7%-points, respectively, on the exergy efficiency. The ST alternatives involving other CHPs/boilers than GE LM2500-Aalborg were left out of the presentation as they gave similar results as the chosen CHP.

Using marginal ratios as shown in Fig. 4, i.e. changes to changes instead of absolute values, made it more convenient to both see the actual gain compared to the extra supplied exergy and also to spot the differences, if any, between the alternatives. As opposed to Fig. 3 we expected ratios of extra power to extra fuel exergy above 55%, which is a typical exergy efficiency of a conventional separate gas-fired power plant as of today. Solutions with ratios below this figure is thus not interesting to implement.

The marginal ratios were all very high (Fig. 4), 89.8–98.7% for case A1, and thus certainly very promising. The roman numbering in Fig. 4, i.e. I, II, III, IV and V indicate that the extra steam needed was produced at either FW HRSGs, Moss HRSG, Aalborg HRSGs, Sleipner boiler or KEP boiler, respectively. The somewhat conservative efficiency (70%) chosen for the new STs at elevated pressure indicate that the performance could have been even better. The two-pressure level steam production stood out with the highest figure, 98.7%. The contribution from the steam turbines was also significant, with 92.0%. Again, the utilization of the steam and hence the performance, would have been even better if the new STs were implemented before building the plant. Alternatives involving the direct fired boilers gave somewhat poorer ratios, though, ranging from 84.4% to 86.5% for steam turbines and 87.1–88.0% for higher pressures/steam turbines.

Concerning gain in terms of extra power, it was the high pressure steam production combined with STs in SDS that for sure contributed the most, 35.5 MW or 23.9% increase compared to the existing system, as seen in Fig. 5. The results do not change for the other ST alternatives as well as HP/ST alternatives. Depending on the existing design, the alternative involving the elevated pressure level of steam production might be the one which requires the

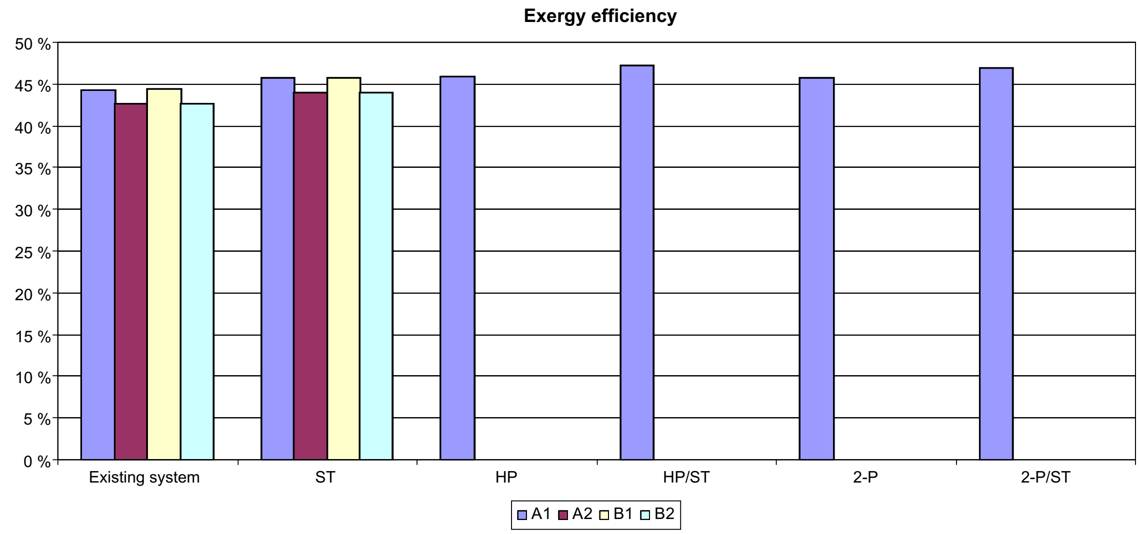

Fig. 3. Exergy efficiency in terms of ratio of useful exergy to fuel exergy.

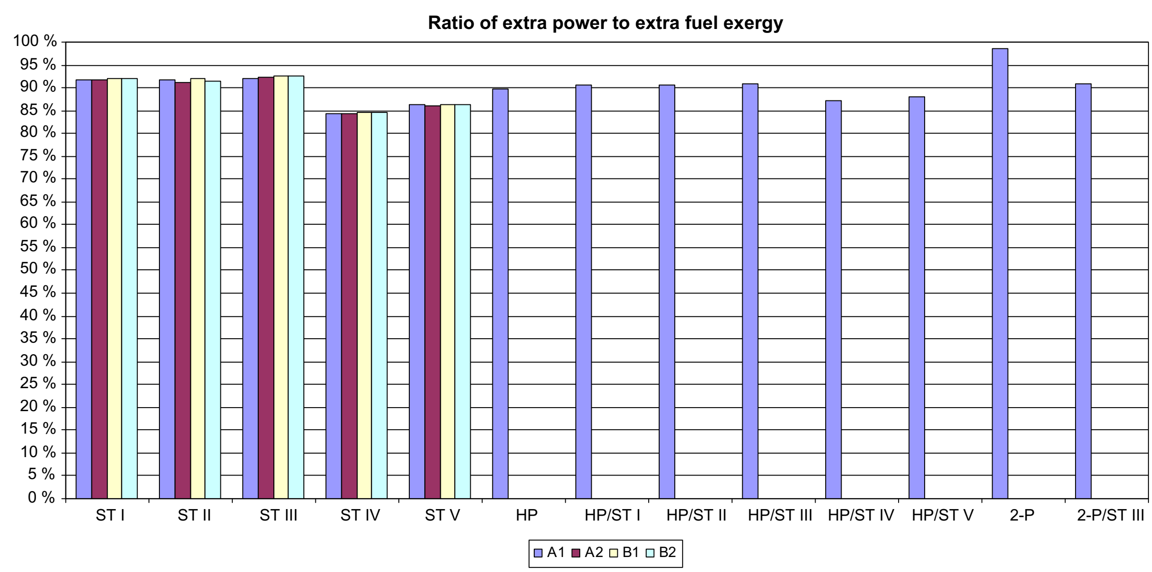

Fig. 4. Marginal exergy efficiency in terms of ratio of extra power to extra fuel exergy.

least of remodeling/upgrading, hence the most favourable one in terms of reducing the amounts of shut downs. The contribution of solid 18.5 MW was thus quite good in spite of the moderate ST efficiency of 70%. This represented in fact close to 60% of the power demand of the sales gas compressors connected to the three RR Avon GTs.

Regarding extra fuel consumption, Fig. 6 shows that the two-stage pressure steam production costed the least, only 2.6% of the original fuel consumption of the existing system or 15.58 MW. Additional STs in SDS was the second best alternative. Elevated pressure steam production (from 59 to 120 bar) represented the single most expensive option with 3.5% or 20.54 MW. Combination of steam production at elevated pressure and STs in SDS was thus the most fuel demanding alternative, amounted to 6.6% or 39.09 MW. The alternatives with the direct fired boilers were 0.2%-points and 0.1%-point higher for steam turbines and increased pressure level, respectively. This is due to reduction of access air, thus higher temperatures for CHPs and less fuel consumption. The air–fuel ratio was maintained for the direct fired boilers, due to the fixed exhaust temperatures. The extra fuel consumption of Sleipner compared to KEP was caused by lower exhaust temperature for KEP and also a decrease of access air in KEP.

The total irreversibilities did not change much for the entire system (CHPs/boilers and SDS) as shown in Fig. 7. The alternatives involving direct fired boilers were a bit bigger, ranging from 0.3% to 0.5%, whereas the other CHPs gave similar results as GE LM2500-Aalborg CHP. Two-stage pressure steam production was the only alternative that resulted in an actual reduction.

Table 10 summarizes the main findings. Alternative 0 refers to the existing system as of today, whereas the other alternatives were thoroughly described in Section 2. Here, the extra steam needed was produced at the Aalborg HRSGs only, as they were the most exergy efficient unit of the boilers/HRSGs. In addition, the results of a stepwise improvement were also included. The marginal exergy efficiency of two-stage pressure (59/120 bar) steam production stood out with 98.7%. This was due to better adjusted temperature, as some heat then was transferred at a lower

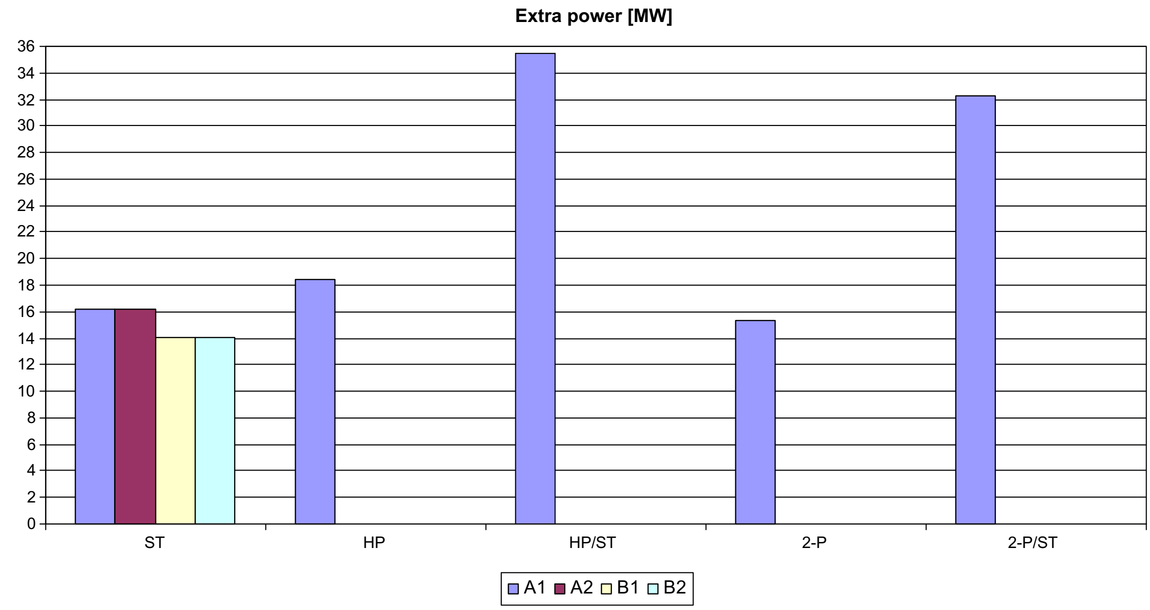

Fig. 5. Extra power (MW) compared to the existing system.

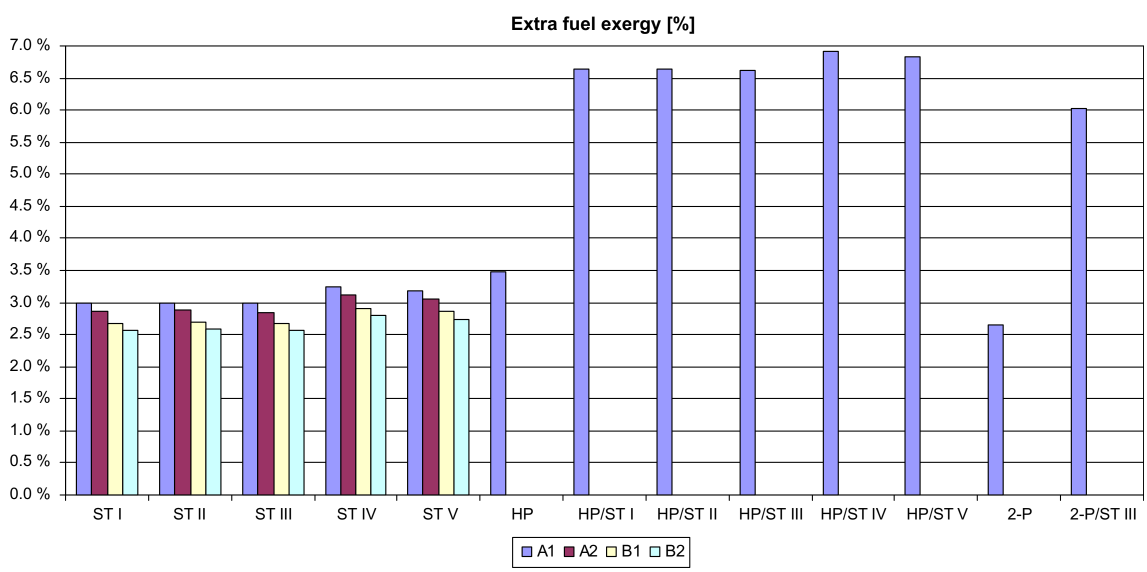

Fig. 6. Extra fuel (%) compared to the exisiting system.

presure level. The other alternatives ranged from noteworthy 89.8 to 92.0%.

There was a significant difference in marginal exergy efficiency between implementing STs for elevated pressure steam production and for two-stage pressure steam production. This was due to both less fuel needed and more power delivered. The ratio of extra power to extra fuel exergy was 84.9% for the two-stage pressure steam production (from Alt. 3 to Alt. 5) and increased by 7.1%-points for the elevated, single-stage pressure option (from Alt. 2 to Alt. 4) The lower marginal efficiency was again due to better adjusted temperature curves for the two-stage pressure steam production alternative. Doing a stepwise improvement from Alt. 1 to Alt. 4 and from Alt. 1 to Alt. 5 gave similar results.

The marginal electric efficiency as defined in terms of the lower heating values, can be obtained by multiplying the marginal exergetic efficiency by 1.05 [8]. This gave results from 89.1% to 103.6%, which were promising.

5.4. Accuracy

All mass flows were either specified or simple sums of such quantities. The elemental balances were satisfied for the units without chemical reactions. The combustors showed small deviations in the elemental flow rates between inflows and outflows. The relative deviations were 2×10^{-4} or less for both hydrogen and carbon.

The energy balances were satisfied for the units without chemical reactions, as the work of pumps and turbines were calculated from the differences between the inflow and outflow. This was also the case for the heat exchangers. The deviations between inflow

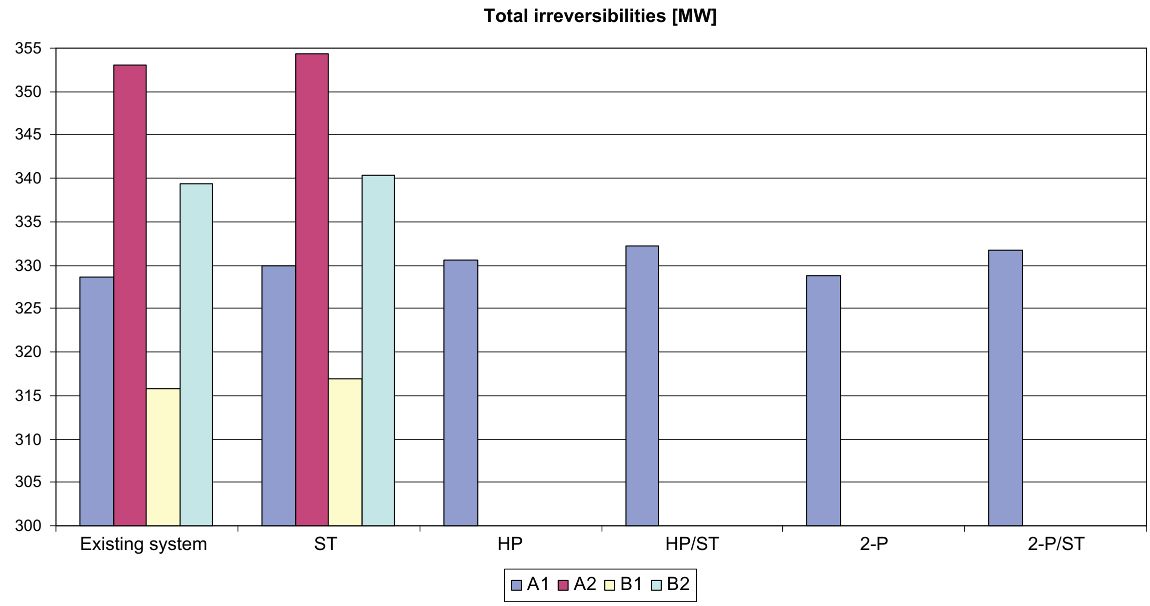

Fig. 7. Total irreversibilities (MW).

Table 10
Effects of Alt. 1–5.

Alt. From → To	Change in system	Add. consumed fuel exergy ΔE_{fuel}	Add. produced power ΔW_{el}	Marginal exergetic efficiency $\Delta W_{el}/\Delta E_{fuel}$ (%)	Total exergy efficiency $(W_{net} + Eth)/E_{fuel}$ (%)
0 → 1	Existing → ST	17.58 (3.0%)	16.18 (10.9%)	92.0	45.7
0 → 2	Existing → HP	20.54 (3.5%)	18.45 (12.4%)	89.8	45.8
0 → 3	Existing → 2-P	15.58 (2.6%)	15.37 (10.3%)	98.7	45.7
0 → 4	Existing → HP/ST	39.09 (6.6%)	35.51 (23.9%)	90.8	47.2
0 → 5	Existing → 2-P/ST	35.51 (6.0%)	32.29 (21.7%)	90.9	47.0
2 → 4	HP → HP/ST	18.55 (3.0%)	17.07 (10.2%)	92.0	47.2
3 → 5	2-P → 2-P/ST	19.93 (3.3%)	16.92 (10.3%)	84.9	47.0
1 → 4	ST → HP/ST	21.51 (3.5%)	19.33 (11.7%)	89.9	47.2
1 → 5	ST → 2-P/ST	17.92 (3.0%)	16.11 (9.8%)	89.9	47.0

and outflow for the combustors were at most 0.34%, thus due to inaccuracies in PRO/II [11].

For water/steam properties, the Soave–Redlich–Kwong (SRK) [12] equation of state was used in PRO/II. To assess the inaccuracies in the water/steam calculations, all enthalpy and entropy differences were also compared to the presumably more accurate, multi-parameter model of Haar et al. [15], using the Engineering Equation Solver (EES) [16]. The boiling (saturation) temperatures at the relevant pressures were also calculated with both approximations, resulting in differences of 0.15 K (for 59 bar) or less. The largest relative deviation between the two models were 0.49% and 1.21% in enthalpy and entropy differences, respectively, found between the states of HP and User 3 (involving approximately 10% of the steam). For the difference between the states of HP steam and the users, the deviation was 0.2% or less for enthalpy and 0.3% or less for entropy, whereas the deviations for the remaining differences were of order of 10^{-4} or less. The exergy differences were approximately 0.2% between the states of HP and User 2 as well as the states of HP and User 3, The deviations of the remaining exergy differences were also of order of 10^{-4} or less. The deviations for enthalpy, entropy and exergy increased for states closer to the saturation temperatures. Since the model of Haar et al. [15] is a multi-parameter curve-fit to experimental data, the found deviations indicate the magnitude of the calculation inaccuracies.

These inaccuracies had minor impact on the calculated figures, and the main findings and conclusions were not affected.

5.5. Overall discussion

Given that the steam distribution system contributed to only 6% of the irreversibilities, whereas the CHPs/boilers were responsible for 94%, as shown in Table 7, it does not seem logical that we focus on the throttling for improving the system. The potentials for improvement in the power and steam production is an open-and-shut-case, i.e. this will be accomplished by replacing the direct fired boilers with GTs and HRSGs and by replacing the old CHPs with new, modern units dimensioned to produce the steam with less supplementary firing. This would increase both the power production and the exergy efficiency considerably, as concluded for the Avon-FW CHPs in [7]. However, when, or if, this is done, the irreversibilities in the steam distribution system due to the extensive use of throttling, will still remain. The potential for this particular improvement was examined in the present study.

Another question that might be apparent is why the plant was built in this way, with steam production at one single pressure, a relatively low pressure compared to those found in power plants, although high compared to most of the user requirements. There are several reasons for this. First, the plant has gone through multiple extensions. The power and steam production is four times larger than in the original plant of 1983. Each extension has had its own requirements to satisfy. Second, the original plant was built at a time when natural gas used on-site had virtually no economic value, which means that low investment, short construction time, simple and flexible operation, etc. were preferred rather than thermodynamic efficiency. Third, regularity and flexibility have been and still are important priorities. The gas export from Kårstø has had a very high regularity, typically 98–100% in the recent years. A few days of stoppage may thus be more expensive than the savings of an improved efficiency. The compositions and the amount of incoming raw gas streams for processing change during the lifetime of a gas-field. Thus, flexibility in operation is also important. Fourth, the original system has been normative for the extended system. As a result, the various extensions have had to give compatible steam states and connections to ensure flexibility in the supply and use of steam.

Nevertheless, implementing steam turbines – and actually using them – are two relatively simple actions. The elevated pressure in the HRSGs/boilers will most likely require new equipment. However, the study demonstrated that when new HRSGs/boilers are installed, there is a considerable potential for improvement if the pressure is increased from the present level. The investigated alternatives with both steam turbines and elevated pressure gave an additional power production larger than that of the original plant.

Another "mental barrier" against implementing steam turbines is that the throttling does not have any energy losses. "We know that there is entropy generation, but it is the enthalpy we make use of" is an authentic statement of an industrial engineer. This study presents a quantification of the losses in an existing, industrial system – not only in terms of entropy production or irreversibility, but also in terms of a potential extra power production and the associated (marginal) efficiency. Although the percentual improvement may seem small with respect to the entire system, the extra power production corresponds to one or two gas turbines with close to 100% electric efficiency. In this respect, the study presents an example from an existing industrial plant that illustrates the losses in throttling and the practical possibilities for improvement.

6. Conclusions

The existing steam production and distribution system for Kårstø natural gas processing plant was analysed at four operating conditions by means of the exergy analysis. Then some means for improving the performance were studied. Extensive throttling of steam and mixing with water were replaced by steam turbine expansion. Furthermore, steam was produced at higher (single and dual) pressure for back-pressure steam turbine utilization before directed into the steam distribution system. Combinations of the modifications were also investigated.

The exergy efficiency of the existing system was 44.3%, and it dropped 1.8% when one HRSG was inoperative. The direct fired boilers were outnumbered by the CHPs as regards efficiency, at most with 15%-points. The steam distribution system contributed to only 6.2% of the total irreversibilities.

Every alternative increased the fuel consumption compared to the existing system, from 2.6% to 6.6%. The total irreversibilities also increased, except from the two-stage pressure steam production. However, the additional power production was considerable and resulted in a remarkable 89–104% marginal electric efficiency, even with conservative assumptions on steam turbine efficiency and moderate modifications of the system.

Replacing throttling with steam turbine expansion in the distribution system gave an additional electric power production of 16.2 MW at an additional consumption of 17.6 MW fuel exergy, hence a marginal exergetic efficiency of 92%. An increase of steam production pressure from 59 to 120 bar gave 18.5 MW at 89.8% marginal exergetic efficiency (single pressure) or 15.4 MW at 98.7% marginal exergetic efficiency (59/120 bar dual pressure). The combination of additional steam turbines in the steam distribution system and elevated production pressure (120 bar) gave 35.5 MW at a marginal exergetic efficiency of 90.8%.

The study provided an example from an existing, industrial steam system that illustrates both the losses in throttling and low-pressure steam production, and the practical potentials for improvement.

Acknowledgements

The authors are grateful for discussions, ideas and information from Dr. Hans Jørgen Dahl of Gassco, Alf Martinsen and Jon Magne Flo Hvidsten of the Kårstø Technical Service Provider at Statoil.

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