Doctoral theses at NTNU, 2011:80

Anne Berit Rian

On exergy analysis of industrial plants and significance of ambient temperature

NTNU
Norwegian University of Science and Technology
Thesis for the degree of philosophiae doctor
Faculty of Engineering Science and Technology
Department of Energy and Process Engineering

Doctoral Thesis

Anne Berit Rian

Doctoral theses at NTNU, 2011:80

ISBN 978-82-471-2690-5 (printed ver.)
ISBN 978-82-471-2689-9 (electronic ver.)
ISSN 1503-8181

NTNU
Norwegian University of Science and Technology

NTNU
Norwegian University of Science and Technology

NTNU

Anne Berit Rian

On exergy analysis of industrial plants and significance of ambient temperature

Thesis for the degree of philosophiae doctor

Trondheim, March 2011

Norwegian University of
Science and Technology
Faculty of Engineering Science and Technology
Department of Energy and Process Engineering

NTNU
Norwegian University of
Science and Technology

NTNU
Norwegian University of Science and Technology

Thesis for the degree of philosophiae doctor

Faculty of Engineering Science and Technology
Department of Energy and Process Engineering

ISBN 978-82-471-2690-5 (printed ver.)
ISBN 978-82-471-2689-9 (electronic ver.)
ISSN 1503-8181

Doctoral Theses at NTNU, 2011:80

Printed by Tapir Uttrykk

Abstract

The exergy analysis has been a relatively mature theory for more than 30 years. However, it is not that developed in terms of procedures for optimizing systems, which partly explains why it is not that common. Misconceptions and prejudices, even among scientists, are also partly to blame.

The main objective of this work was to contribute to the development of an understanding and methodology of the exergy analysis. The thesis was mainly based on three papers, two of which provided very different examples from existing industrial systems in Norway, thus showing the societal perspective in terms of resource utilization and thermodynamics. The last paper and the following investigation were limited to certain aspects of ambient conditions. Two Norwegian operational plants have been studied, one operative for close to 30 years (Kårstø steam production and distribution system), while the other has just started its expected 30 years of production (Snøhvit LNG plant). In addition to mapping the current operational status of these plants, the study of the Kårstø steam production and distribution system concluded that the potential for increasing the thermodynamic performance by rather cautious actions was significant, whereas the study of the Snøhvit LNG plant showed the considerable profit which the Arctic location provided in terms of reduced fuel consumption. The significance of the ambient temperature led to the study of systems with two ambient bodies (i.e. ambient water and ambient air) of different temperatures, here three different systems were investigated: A regenerative steam injection gas turbine (RSTIG), a simple Linde air liquefaction plant (Air Liq) and an air-source heat pump water heater (HPWH). In particular, the effect of the chosen environment on exergy analysis was negligible for RSTIG, modest for Air Liq and critical for HPWH. It was found that the amount of exergy received from the alternative ambient body, compared to the main exergy flow of the system gave an indication of whether the choice of environment would affect the exergy results or not. Furthermore, the additional study, where the effect on exergy results due to a fixed environmental state versus a natural environment was investigated, also suggested that neither calculations nor software tools should uncritically be based on the fixed environment when it does not correspond with the natural environment.

The findings in these studies can be useful at different levels, such as for further studies and optimization of similar plants, for the authorities to encourage or demand even better performances from future plants and for developing methodical engineering tools for exergy.

Acknowledgements

The work presented in this thesis has been performed at the Department of Energy and Process Engineering, Faculty of Engineering Science and Technology, Norwegian University of Science and Technology.

I acknowledge both the financial support from the Faculty of Engineering Science and Technology and the additional funding from the Department of Energy and Process Engineering. In particular, I appreciate the support and guidance of Administrative Leader, Gerd Randi Fremstad and Head of Department, Professor Olav Bolland.

I would like to thank my supervisor Professor Ivar S. Ertesvåg of the Department of Energy and Process Engineering, Faculty of Engineering Science and Technology, Norwegian University of Science and Technology for his valuable assistance throughout the years.

I am thankful to Dr. Jostein Pettersen and Dr. Geir Owren at Statoil for providing information about the Snøhvit LNG processing plant, Dr. Hans Jørgen Dahl of Gassco, Alf Martinsen and Jon Magne Flo Hvidsten of the Kårstø Technical Service Provider at Statoil for process data about the steam production and distribution system at the Kårstø processing plant and Dr. Yves Ladam and Dr. Geir Skaugen at SINTEF Energy Research for lending me their Span-Wagner EOS macro in order to model the CO_2 circuit for the heat pump water heater. The assistance of you all has been vital for conducting the studies.

Special thanks are given to fellow PhD students and graduates I got to know during the three last years of this journey. I appreciate the cheerful working environment each of you created.

I am very grateful for the unconditional love and support my kind parents always have given. You have never failed to believe in me, which I appreciate sincerely.

My deepest gratitude goes to Inge Saanum for making me believe in myself as well as unwinding me when needed by means of a great sense of humour, easy-going approach to life and numerous hikes to the forests and mountains, where

the comfort-to-weight ratio as regards equipment may now be considered close to fully optimized.

Trondheim, March 2011

Anne Berit Rian

List of papers

The thesis is mainly based on the following papers:

1. Rian, A.B., Ertesvåg, I.S., “Exergy evaluation of the Arctic Snøhvit LNG processing plant in northern Norway – significance of ambient temperature.”
   Submitted to an international journal.

2. Rian, A.B., Ertesvåg, I.S., “Exergy analysis of a steam production and distribution system including alternatives to throttling and the single pressure steam production,” Energy Convers. Managn. 2011 52 (1), 703–712.

3. Rian, A.B., Ertesvåg, I.S., “Exergy analysis of systems interacting simultaneously with two ambient bodies of different temperatures.”
   Submitted to an international journal.

The author’s contribution:

The author of the PhD thesis has:

- Performed all the analysis of data in the work.
- Written the main parts of the papers.

Contents

1 Introduction

1.1 Background

The high standard of living in the western world is achieved and maintained through the exploitation of natural resources. Norway developed into one of the richest countries in the world by means of its great oil discoveries which started with the Ekofisk field in 1969. Forty years and more than 3000 wells at the Norwegian Continental Shelf later have resulted in a gross income of NOK 8 trillion[1]. Still, less than half of the Norwegian oil and gas resources are exploited (Norwegian Petroleum Directorate, 2010). Nowadays, the world is headed towards an energy shortage as consumption is exceeding the exploitation level of non-renewable resources. Many have predicted a crude oil shortage in the 21st century, eg. Aleklett et al. (2010), Höök et al. (2009) and Campbell and Laherrere (1998). The focus of the last 10 years has turned to renewable energy which is both essential and wise, as regards environmental issues. An additional important means, which has not received much attention, would be to adjust the quality of the energy to its use.

The energy balance has been the traditional method to evaluate and design engineering systems involving physical and chemical processing of materials and transfer/transformation of energy, e.g. resource utilization. The energy analysis is based on the mass balance and the First Law of Thermodynamics, which basically is a conservation principle. As regards degradation of energy which always occurs in any actual process to a varying extent or quantification of the usefulness of the heat content of streams leaving the system, i.e. waste or product, the energy method is unfortunately insufficient. However, the exergy analysis overcomes these obstacles, as it includes the Second Law of Thermodynamics. In contrast to energy, exergy can be destroyed, that is, exergy is not conserved. The exergy analysis is particularly suited for developing more efficient use of resources, evaluating existing systems and minimizing losses of existing systems as it locates and identifies the causes and magnitudes of exergy losses. Moreover, the exergy analysis pinpoints the parts which offer the best opportunity for improvement (e.g. Szargut et al. (1988)).

[1] Trillion = 10^{12}

The exergy analysis has been a relatively mature theory for more than 30 years now. However, it is not that developed in terms of procedures for optimizing systems, which partly explains why it is not utilized even more. Misconceptions and prejudices, even in scientific environments, are also partly to blame.

Norway has exploited oil and gas since the early 1970s and as a result large processing plants have been built. Two facilities connected to this field have been considered in the thesis work which also could be interesting to Norwegians and the authorities as they are familiar installations. Moreover, the findings can be used by the authorities to encourage or/and specify better performance for new plants.

Everybody familiar with the laws of thermodynamics is well aware that throttling implies irreversibilities, even though there is no energy loss. Unfortunately, to many engineers, the exergy analysis is a challenging academic exercise or a vague, distant and unpleasant memory from mandatory university level courses. In the second paper of the thesis work, this “academic exercise” is translated into possible additional power production. Thus this is far easier to both comprehend and to relate to than irreversibility figures. In the first paper, which is about the Arctic Snøhvit LNG plant, the exergy analysis represents an objective reference which makes it possible to evaluate the performance of this facility and assess how much the Arctic conditions give for free.

As opposed to the energy balance, the environment constitutes an important part in the exergy analysis. The results from the first paper about the Arctic plant and curiosity led to the investigation of the effects of known simplifications of the environment on exergy results.

Norwegian citizens have started using the principle of adjusting the energy quality to its use, though rather unconsciously, by replacing electric heaters with heat pumps for room heating purposes. A survey performed by Statistics Norway (2008) demonstrated an increase of heat pumps in Norwegian households from 4% in 2004 to 8% in 2006. Since then the development has been even more rapid. According to Norsk Varmepumpeforening (2010) there are now more than 500,000 heat pumps in Norwegian households, which is significant, given a population of a modest 4.9 million (Statistics Norway, 2010). The interest in heat pumps is, however, due to an economic incentive (i.e. saving money) rather than an awareness of thermodynamics.

Given the principle of energy conservation, i.e. energy cannot be created, or disappear, it seems like a paradox that we are experiencing a scarcity of energy resources. This “paradox” is because there are two meanings of the word “energy” (Szargut et al., 1988): The scientific one which means no destruction, and the other which refers to the ability to drive processes or machines. The energy situation today is a great opportunity to introduce, demystify and establish the concept of exergy for the public and reluctant engineers. In fact, as all natural processes involve degradation of energy quality, it ought to be easier to comprehend than the energy conservation principle.

1.2 Scope of work

The main objective of the thesis is to contribute to the development of an understanding and methodology of exergy analysis in order to – in future – develop methodical means, i.e. engineering tools, for optimization of processing plants.

The content for the PhD work was:

- Evaluate Snøhvit LNG processing plant in northern Norway
  - Map the energy utilization, make visible and quantify the useful energy resources at the plant
  - Discuss and find suitable measures of performance
  - Investigate/quantify the advantages of cold climate by performing the analysis for an identical plant at increased environmental temperatures
- Analyse the steam production and steam distribution system at Kårstø processing plant for natural gas
  - Map the energy utilization, make visible and quantify the useful energy resources (steam production, electricity production) for different production rates and kinds of distribution among the boilers
  - Introduce known means and combinations of such to reduce the extensive use of throttling and single pressure steam production
  - Investigate/quantify the advantages in terms of suitable measures
- Study systems which are interacting simultaneously with two ambient bodies of different temperatures with respect to exergy analysis and the choice of environment; for three different exemplary systems
  - Quantify this effect in terms of exergy utilization (efficiency)
  - Map any similarities/differences/trends for the chosen systems
  - Discuss and suggest a measure of significance
  - Evaluate the effect of a fixed environment versus an environment defined from the actual surroundings

– Present the results of the analyses as illustration examples for useful utilization of the exergy method. Focus on supplementary benefits compared to a conventional mass and energy analysis or entropy analysis

– Attempt a critical presentation of the results, pinpointing possible pitfalls which could give rise to misunderstandings or misleading information, pursue formulations of performance ratios as a means for identifying and locating irreversibilities easily

1.3 Limitations

The basis for the analysis and the development of the methodology were two Norwegian operational processing plants and three well known industrial systems from the literature. The components, compositions and thermal properties of these components which constitute a part of the analyses were known.

The reference systems of the real industrial plants were chosen from their natural environment. The importance of these choices was emphasized.

No experiments were performed in this work. An exergy analysis cannot state whether the possible improvement is practicable. An economic analysis would be needed to address that issue, and it is not included here. Nor was thermo-economics a part of the study.

It was beyond the scope of this project to establish an engineering tool. However, the work was expected to contribute to the improvement of such a tool.

1.4 Thesis outline

Chapter 1 gives the background for the thesis, describes the scope of work, limitations and concludes with a thesis outline.

Chapter 2 defines the concept of exergy, introduces the main idea of the exergy analysis with basic concepts and equations, and includes a brief historical perspective on both the exergy concept and exergy efficiency.

Chapter 3 presents the background and main findings of Paper 1 – "Exergy evaluation of the Arctic Snøhvit LNG processing plant in northern Norway – significance of ambient temperature."

Chapter 4 addresses the background and main findings of Paper 2 – "Exergy analysis of a steam production and distribution system including alternatives to throttling and the single pressure steam production."

Chapter 5 contains the background and main findings of Paper 3 – "Exergy analysis of systems interacting simultaneously with two ambient bodies of different temperatures."

Chapter 6 presents the investigation of significance of fixed environmental state versus actual environment on exergy results.

Chapter 7 summarizes the conclusions of the studies.

The papers in their complete forms are placed in Appendices A–C. They constitute the main part of the thesis.

2 Exergy analysis

2.1 Basic theory

Some important aspects about the exergy concept are addressed (based on e.g. Moran and Shapiro, 2004; Szargut et al., 1988), and basic equations are then described in the following.

Exergy is the maximum theoretical work obtained from a system when this system is brought from a state to equilibrium with the environment while interacting only with environment. The state of a system is defined by temperature, pressure and composition. Also, exergy is the minimum theoretical work needed to bring the system from equilibrium with the environment to the given state. This means that exergy is a measure of the departure of the state of a system from the state of the environment. This makes exergy an attribute of both the system and environment together. The environment is large in extent, uniform in temperature, pressure and composition, and it is regarded as free of irreversibilities, i.e. there is no exergy destruction within the environment. All significant irreversibilities are located within the system and its immediate surroundings. Once the environment is specified, the exergy values can be assigned, and accordingly, exergy becomes a property of the system. Exergy values cannot be negative, as opposed to exergy change. Moreover, the numerical values for exergy depend on the state of the system and the environment. The conservation principle of energy is not valid for exergy due to irreversibilities, which refers to destruction of exergy within the system. In other words, the main reasons for thermodynamical imperfection of thermal and chemical processes cannot be detected by an energy analysis. Processes such as throttling, irreversible heat transfer, adiabatic combustion do not result in any energy loss, though they are irreversible processes which reduce the energy quality.

The exergy balance was developed by combining the balances of masses and the First and Second Law of Thermodynamics, e.g. Szargut et al. (1988), Moran and Shapiro (2004) and Kotas (1995).

The control volume exergy rate balance can then be formulated as (Moran, Shapiro, 2004)

$$\frac{dE_{\mathrm{CV}}}{dt} = \sum (1 - \frac{T_0}{T_j})\dot{Q}_j - (\dot{W}_{\mathrm{CV}} - p_0 \cdot \frac{dV_{\mathrm{CV}}}{dt}) + \sum_{in} \dot{m}_k \varepsilon_k - \sum_{out} \dot{m}_k \varepsilon_k - \dot{I}_{\mathrm{CV}} \tag{1}$$

where $\frac{dE_{\mathrm{CV}}}{dt}$ represents the rate change of the exergy of the control volume.
$\dot{Q}_j$ gives the rate of heat transfer at the location on the boundary where the temperature is T_j, whereas the accompanying exergy transfer is represented by $(1 - \frac{T_0}{T_j})\dot{Q}_j$, and T_0 is the ambient temperature.
$\dot{W}_{\mathrm{CV}}$ accounts for the rate of energy transfer by work other than flow work, whereas accompanying exergy transfer is given by $\dot{W}_{\mathrm{CV}} - p_0 \cdot \frac{dV_{\mathrm{CV}}}{dt}$, here $\frac{dV_{\mathrm{CV}}}{dt}$ is the rate of volume change.
$\dot{m}_k \varepsilon_k$ represents the rate of exergy transfer accompanying both mass flow and flow work at inlet “in” or exit “out.”
$\dot{I}_{\mathrm{CV}}$ accounts for the rate of exergy destruction due to irreversibilities within the control volume.

For a steady state, non-expanding (sub-)system, the balance can be reduced to

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

where $\dot{E}^Q$ is the rate of exergy transferred with heat to the control volume (CV), $\dot{W}$ is the rate of work, $\dot{I}$ is the rate of irreversibility (exergy destruction) in the CV, and $\dot{m}_k$ and ε_k are the mass flow rate and the specific flow exergy (Szargut et al., 1988; Kotas, 1995), respectively, of flow no. k across the boundary of CV.

The specific flow exergy is composed of a thermomechanical and a chemical exergy component,

$$\varepsilon = \varepsilon_{\mathrm{tm}} + \varepsilon_0 \tag{3}$$

The specific thermomechanical exergy, where the kinetic and potential contributions are here neglected, is determined from

$$\varepsilon_{\mathrm{tm}} = h - h_0 - T_0\left(s - s_0\right) \tag{4}$$

where h is the specific enthalpy, s is the specific entropy, while $h_0 = h\left(T_0, p_0\right)$ and $s_0 = s\left(T_0, p_0\right)$ are the values at the restricted dead state (i.e. environment) for the relevant flow (mixture).

The molar chemical exergy of a single, gaseous component present in the atmosphere (environment) is given by

$$\bar{\varepsilon}_{0,i} = \bar{R}T_0 \ln\ (p_0 / p_{i,0}) = -\bar{R}T_0 \ln\ (x_{i,0}) \qquad \textbf{(5)}$$

where $\bar{R}$ is the universal gas constant, $x_{i,0}$ is the mole fraction of the species *i* in the environment and $p_{i,0}$ is the corresponding partial pressure. The overbars denote molar quantities.

The chemical exergies of hydrocarbons were corrected for deviating ambient conditions according to Szargut et al. (1988) and Ertesvåg (2007) as

$$\bar{\varepsilon}_{0,i} = \bar{\varepsilon}_i^0 \frac{T_0}{T^0} + \bar{h}_{\mathrm{LHV}}^0 \frac{T^0 - T_0}{T^0} + T_0 \bar{R} \sum_{j \neq i} \nu_j \ln \frac{x_j^0}{x_{j,0}} \qquad \textbf{(6)}$$

where $\bar{\varepsilon}_i^0$ and $\bar{h}_{\mathrm{LHV}}^0$ are the molar chemical exergy and the molar lower heating value, respectively, determined at the reference state of 1 atm, 25ºC, 28% relative humidity (RH) (Kotas, 1995). The superscript ^{0} denotes this reference state, while the subscript $_0$ denotes the ambient state chosen as the environment 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.

Then, for an ideal mixture, the molar chemical exergy is determined from

$$\bar{\varepsilon}_{0,\mathrm{mix}} = \sum x_i \bar{\varepsilon}_{0,i} + \bar{R}T_0 \sum x_i \ln x_i \qquad \textbf{(7)}$$

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.

For an incompressible water stream the molar chemical exergy is given (Szargut et al., 1988) by

$$\bar{\varepsilon}_{0,\mathrm{water}} = \bar{v}_{f,T_0} \cdot (p_0 - p_{g,T_0}) - \bar{R}T_0 \ln \varphi_0 \qquad \textbf{(8)}$$

where $\varphi_0 = x_{\mathrm{H_2O,\,0}} \cdot p_0 / p_{g,T_0}$, φ represents the relative humidity (RH) and $\bar{v}$ denotes the molar volume. The subscripts *f* and *g* represent the saturated liquid and saturated vapour states, respectively, while subscript T_0 denotes that the quantity is determined at the temperature of the environment.

2.2 Reference system

There are two main approaches in defining the reference systems. Ahrendts (1980) suggested a reference system in total equilibrium formed by the atmosphere, the oceans and a portion of the earth's crust (1m) at 25°C, 1.019 atm, 100% relative humidity (RH). The idea of total equilibrium is based on the understanding that the environment should not be able to produce any work, and also that the exergy values should be unique. Relative humidity of unity is, however, a rather rare condition for the atmosphere. The figures also depend a great deal on the chosen thickness of the crust, as Ahrendts also pointed out.

In contrast, Szargut et al. (1988) emphasized that the natural environment is not in equilibrium, thus the reference should be defined accordingly. Therefore the reference needs to be founded on correct calculations of external exergy losses. Moreover, the reference substances chosen for each chemical element have to be present in huge amounts in the natural environment. Szargut's standard reference condition is 25°C, 1.01325 bar, 70% RH. However, this model gives negative figures for the nitrates $Ca(NO_3)_2$, $NaNO_3$ and KNO_3, which implies that they will form spontaneously. This formation is, however, blocked kinetically.

An earlier approach (1965) of Szargut and Petela was utilized by Kotas (1995), where the standard reference state was given as 25°C, 1 atm, and a partial water pressure of 0.0088 bar. The latter gives 28% RH, which in turn results in an unrealistic low RH at higher temperatures and subcooling (supersaturation) at temperatures below 5.1°C. Using an unstable condition as the environment at low temperatures seems somewhat awkward, which strongly supports the common principle of utilizing relative humidity instead, i.e. the actual water content depends on the chosen environmental temperature.

In addition to the two main approaches in defining the reference systems, there are also various standard atmospheric conditions defined for technical equipment, such as the ISO standard for gas turbines, which is 15°C, 1.01325 bar, 60% RH (ISO, 2009).

2.3 Use of the exergy concept – a historical perspective

A full historical review of the concept of exergy and its applications is beyond the scope of this thesis. However, some milestones in the development will be presented.

The exergy analysis may be regarded as a new technique, though the first attempts at assessing various energy forms from their convertibility to other forms are directly connected to the first formulations of the Second Law of Thermodynamics (Kotas, 1995). In 1824 Carnot stated that the equivalent amount of work cannot be obtained from a given amount of heat (e.g. Kotas

(1995), Szargut et al. (1988) and Sciubba and Wall (2007)). Furthermore, the extracted work from a heat engine is proportional to the temperature difference between the hot and cold reservoir. The Second Law of Thermodynamics is thus based on Carnot's reflections, while Joule proved the conservation principle of energy, i.e. the First Law of Thermodynamics through his many laboratory experiments during the 1840s (Szargut et al., 1988).

There was a rapid development in the use of these thermodynamical laws. It included introducing thermodynamic functions, defining internal energy and entropy functions, from which the functions of enthalpy, Helmholtz and Gibbs free energy were developed. These functions were very valuable in terms of comprehending the consequences of the First and Second Laws of Thermodynamics and for actually using them in solving practical problems (Szargut et al., 1988).

The earliest contributions (from 1868) to the exergy concept were mainly due to work of Clausius, Tait, Thomson, Maxwell and Gibbs (Szargut et al., 1988; Kotas, 1995; Sciubba, Wall, 2007). Gibbs was the first to use the term "available work", prior to that he defined a thermodynamic function he named "available energy", and today's definition of exergy in fact corresponds with his equation (Sciubba, Wall, 2007).

Gouy and Stodola (cf. Szargut et al. 1988) discovered independently of each other, in 1889 and 1898, respectively, the effect of ambient temperatures on obtainable work and the law of loss of maximum work. The latter states that the extracted work always is less than the maximum work due to irreversibilities. Today it is referred to as the Gouy-Stodola relation[2] (e.g. Kotas, 1995).

The first publications related to exergy did not get much attention. Actually, further development was slow until the 1930s when industrial growth and new technical developments stimulated practical applications. Bosnajakovic (1938) started a new era in the development of the exergy method as he declared: "Fight against the irreversibilities!" Most of the definitions of the "modern" exergy concept and its application took place in the years between 1950 and 1970.

The term "exergy" was proposed by Rant (1956) as the capacity to do work relative to the state of common components in nature. The ambient components were given zero exergy value. The ability of an energy carrier to do work also expresses its general ability to be converted into other forms of energy (Szargut et al., 1988).

From this viewpoint both Rant and Baehr proposed the concept of "anergy" as the untransformable part of energy in 1964 and 1965, respectively (Szargut et al., 1988). Anergy was perceived as the difference between energy and exergy.

[2] The irreversibility rate in an open control volume equals the product of the ambient temperature and the entropy production rate.

This concept is limited to temperatures above the environment. Due to its limitations anergy is not used in this thesis work.

In 1963 Tribus (cf. Ahern (1980)) proposed that exergy should not be considered as a kind of energy, but instead as a measure of departure from equilibrium.

There was a 15-year debate about the exergy concept mainly in Germany. First, the focus was to develop problem solving procedures in terms of entropy and exergy. Then, discussions on the definition of efficiency followed. Finally, the work towards a standard notation system started in the 1960s. At the end of that decade the theory of exergy was more or less completed. However, only a few practical applications were published (Sciubba, Wall, 2007).

During these years Russian and Eastern European scientists published fundamental contributions which were not available to the wider world, i.e. there was a parallel development on the topic at this point (Sciubba, Wall, 2007). In fact, the more extensive use of exergy analysis in, at the time, the Soviet Union and Europe compared to other countries may have been due to the significant results from analyses of cryogenic and power systems by Brodyanskii, Bosnajakovic in 1960 (Ahern, 1980). Especially, application of exergy analysis in cryogenics had a great impact on its continued use. Early examples of technical improvements due to the exergy analysis are Kapitsa's turboexpander and Brodyanskii's secondary coolant loop for liquefaction systems (Ahern, 1980).

From the 1970s, the development of the theory and especially the number of theoretical publications escalated due to the oil crisis (e.g. Kotas, 1995; Sciubba, Wall, 2007). In addition, Sciubba and Wall (2007) stressed the importance of some textbooks which provided valuable discussions on the topic of exergy: Baehr, Schmidt, Obert, Hatsoupoulos and Keenan. Thus, for the next 30 years most of the publications on exergy dealt with procedures for optimization. During this period the first international workshops were organized, which have been important for broadening and deepening in the field of exergy (Sciubba, Wall, 2007). Also, the interest in design tools grew as industrial researchers became more interested in the exergy analysis.

2.4 Development of the concept of exergy efficiency

The most common performance criteria for energy systems are energy based (Lior and Zhang, 2007). They are useful for determining the efficiency of energy use. In addition, they are also easy to convert into energy cost efficiencies, given that the prices of the energy forms of the useful outputs and the paid inputs are known. Obviously, energy-based criteria do not account for the quality of energy, thus exergy-based criteria are better, as they account for the use of energy resources and give much better guidance for improving the system. Naturally, this concept has been discussed throughout the development of the exergy analysis. Some historical aspects as regards performance criteria will now be presented.

Bosnajakovic (1938) used the term *"Gütesgrad"* which referred to $\eta = 1 - \frac{\Delta L}{L_{\text{Theor}}}$ where ΔL was the lost work and L_{Theor} was the theoretical work. This definition of *Gütesgrad* should not be confused with thermal efficiency. The concept was originally derived from the concept of degree of irreversibility, which he defined as the ratio of lost work to theoretical work. Both *Gütesgrad* and degree of irreversibility could be given any value ($\pm\infty$) according to Bosnajakovic.

Grassmann (1950) suggested a general definition for efficiencies based on the concept of *"Arbeitsfähigkeit"* which is German for the ability to do work and was equivalent to the exergy concept. He proposed the ratio of the increased ability to do work to the used ability to do work (i.e. decrease in exergy). In contrast to Bosnajakovic, Grassmann emphasized that this performance ratio should have values between zero and unity in order to make sense, e.g. an increase which is larger than the decrease would imply a *perpetuum mobile.*

Baehr (1968) acknowledged that there are numerous definitions of exergy efficiencies as the concept relies on the authors' views and opinions and what the authors want to achieve. He performed a theoretical survey based on exergy flows to identify every possible form of exergy efficiency. Baehr then related his findings to different authors and their use of the concept. Baehr emphasized that exergy efficiencies that include differences are considerably smaller compared to common efficiencies, where all supplied flows and all outflows are defined as "used exergy" and "useful exergy", respectively. And hence, differences are especially suited for describing a desired exergy increase.

An important feature of systems that affects the perceived performance was identified by Kostenko (cf. Fratzscher et al., 1986) as "transit exergy" (*Transitexergie*). Transit exergy is the part of the exergy flow in or/and out of a system that remains constant during a process. If this part dominates the outflow, it will thus give a better performance. In 1964 Kostenko proposed his *"technologischen Gütesgrade"* (a technological degree of quality) or *"Güte"* in short, as the ratio of total exergy into the system without the transit exergy to the total exergy into the system with transit exergy. He also introduced a loss ratio which he defined as the ratio of the lost exergy to the total exergy into the system without the transit exergy (cf. Fratzscher et al., 1986).

In recent years, Lior and Zhang (2007) made an attempt to clarify both the definitions and the use of performance ratios based on energy and exergy. They made a distinction between exergy-based and Second Law-based criteria, where the latter is the ratio of energy-based performance to reversible performance operating between the same states, i.e. Second Law-based efficiency is not equal to isentropic efficiencies. However, the literature has not, by and large, been consistent in this area, as many authors use the notation "Second Law-based efficiencies" for exergy-based criteria. Today, the notation "rational efficiency" often refers to an exergy efficiency to describe the degree of reversibility (Szargut et al., 1988; Kotas, 1995).

There are numerous ways to describe performances. It is thus difficult to develop any standard definitions for exergy efficiency as the efficiency is, as Baehr (1968) also recognized, defined in accordance to what the author considers "useful" or "supplied"/"used" for the system that is analysed. It is solely the responsibility of the author to clarify the definition which is used and for what purpose, and it is also the responsibility of the reader to be aware of the definitions which are used rather than make assumptions.

3 Summary of Paper 1: Exergy evaluation of the Arctic Snøhvit LNG processing plant in northern Norway – significance of ambient temperature

In the following, the background for the paper is presented with the main findings. The complete paper is given in Appendix A.

3.1 Motivation

Liquefied natural gas (LNG) plants are special as regards thermodynamics, as they spend exergy (fuel, work) in order to remove energy (heat) from the processed natural gas. Even though liquefaction of natural gas has been analysed before, this work is interesting on several levels.

There are a modest number of LNG plant exergy analyses available in the archival journals. Much of the literature on these processes tends to focus on capacity, construction costs and construction time. Here, the societal perspective is taken in terms of resource utilization and thermodynamics. Thus, detailed information about the internal processes is of less interest. Still, having access to the main process data of Snøhvit LNG plant represented a unique opportunity to evaluate this operating facility.

There are multiple ways of specifying performance. However, specific fuel consumption or specific power will vary with raw gas composition, degree of separation, exchange of other exergy forms and ambient temperature. Accordingly, two plants operating under different conditions cannot be compared directly. Approaches which eliminated these obstacles were pursued.

This is the very first Arctic LNG plant. Naturally, the development of this field started political controversy in Norway, as it is the first field in the vulnerable Barents Sea to be exploited.

When the developer in addition claims a 50–70% reduction of energy consumption compared to other LNG plants due to new technology (Statoil, 2001), it becomes interesting to quantify how much is for free, i.e. due to the cold climate. The study will reveal whether there is a potential for improvement. Based on such knowledge, the authorities also have a means for encouraging or simply demanding better performance in future plants.

3.2 Main findings

The exergy of the products in comparison to the feed stream exergy was, as expected, high due to the nearly unchanged chemical exergy, whereas the exergy efficiency expressed in terms of the desired exergy change to the consumed exergy amounted to 23.2%. Here, compression of CO_2, separation, and cooling accounted for 0.7%-points, 1.9%-points and 20.6%-points, respectively. The exergy losses were distributed as 37% in the processing plant, 52% in the gas turbines and 11% in the heat recovery unit. The Arctic location reduced the fuel consumption significantly. A comparison of the LNG plant at 4°C with a "twin plant" with the same overall exergy efficiency showed 10.9% and 19.9% less consumption at 4°C compared to ambient temperatures of 20°C and 36°C, respectively. On the other hand, maintaining the material and energy flows required an increased exergy efficiency of 25.6% at 20°C and 28.1% at 36°C.

4 Summary of Paper 2: Exergy analysis of a steam production and distribution system including alternatives to throttling and the single pressure steam production

In the following, the background for the paper is presented with the main findings. The complete paper is given in Appendix B.

4.1 Motivation

The oldest parts of this utility system for the Kårstø natural gas processing plant are nearly 30 years old and still operating. It produces steam at a relatively low pressure level, while the steam distribution system utilizes mixing and throttling extensively. These processes may be considered thermodynamically ignorant, though both mixing and throttling in fact represent a widespread practice throughout the industry due to a number of practical reasons. From the energetic viewpoint, mixing and throttling do not represent any losses, as the enthalpy is maintained. However, design guidelines based on the Second Law of Thermodynamics (e.g. Boem, 1997; Leites et al., 2003) include clear recommendations on avoiding both throttling and mixing. Again, being fortunate to have access to the operational data of this plant makes it possible to map its status today. The main exergy losses are located in combustion and heat exchange. Normally, the focus is placed on these areas. Gas turbines, heat recovery steam generators and boilers could, of course, be replaced to improve efficiency. However, that is an open-and-shut case, which has been confirmed numerous times in both textbooks and archival journals, e.g. Moran and Shapiro (2004), Wølneberg and Ertesvåg (2008). Instead, examining the actual effects on performance of rather cautious means such as reducing throttling and careful elevation of steam pressure or two-stage pressure steam production with steam turbines, would be interesting.

4.2 Main findings

The exergy efficiency of the operating utility plant was 44.3%. The practical potential for improvement was significant in terms of marginal efficiencies, expressed as the ratio of extra power to additional fuel exergy. Implementing steam turbines in the steam distribution system or elevation of production pressure (single or dual from 59 to 120 bar) resulted in marginal exergy efficiencies of 92.0%, 89.8% and 98.7%, respectively. Moreover, the combination of steam turbines in the distribution system and elevated pressure gave a corresponding ratio of 90.9%. With respect to the lower heating value, the marginal electric efficiencies ranged from 89–104%. These figures may be compared to the typical figure for electric efficiency of a new conventional power plant, which is approximately 55–58%. Elevating the pressure was the single most exergy-demanding alternative with a 3.6% increase in fuel exergy and 18.5 MW extra power, whereas steam turbines and two-stage pressure resulted in a 3.0% increase/16.2 MW extra power and 2.6% increase/15.6 MW extra power, respectively.

5 Summary of Paper 3: Exergy analysis of systems interacting simultaneously with two ambient bodies of different temperatures

In the following, the background for the paper is presented with the main findings. The complete paper is given in Appendix C.

5.1 Motivation

In the exergy analysis as opposed to the energy analysis it is vital to define the environment properly before actually starting to investigate the system in question. What about those systems that simultaneously interact with two ambient bodies of different temperatures? How should they be treated? In other studies, they are either defined to be equal to each other, or one of the ambient temperatures is basically ignored. Is the effect of this rather common simplification too significant to be left out, or is it negligible?

It was the significance of the ambient temperature investigated in Paper 1 that led to this more theoretical issue. This is not another exercise in doing an exergy analysis, rather it is an investigation of the exergy method. Nor should it be mistaken for a study of the effect of different ambient temperatures.

5.2 Descriptions of the systems

In this study a regenerative steam injection gas turbine (RSTIG), a simple Linde air liquefaction system (Air Liq) and a heat pump water heater (HPWH) were investigated. The following gives a description of these systems, which was left out due to space considerations in the submitted paper.

5.2.1 Regenerative steam injection gas turbine (RSTIG)

RSTIG is a gas turbine system which injects steam into the combustor in order to improve performance. The system is based on one of the systems in the work of Nishida et al. (2005), where they compared the performances of two RSTIG systems and simple regenerative water injection and steam-injected gas turbine systems. There are several types of water and steam injection gas turbine cycles. In steam injection gas turbine systems, the steam, which is generated in the heat recovery steam generator (HRSG), is returned to the gas turbine (GT) and used as the working fluid. The optimum pressure ratios for maximum efficiency of RSTIG systems are relatively low. This type of system is developed primarily for small-scale gas turbine systems. The steam injection configuration can even be used in flexible combined heat and power (CHP) systems. According to Nishida et al. (2005), 70% of higher heating value (HHV) may be achieved.

The regenerative steam injection gas turbine is illustrated in Fig. 5-1. Water is pumped up to 12.2 bar before entering the HRSG, where it is turned into steam by heat exchange with the exhaust. Then the exhaust temperature decreases to 100°C. The fuel and air are compressed and preheated by the exhaust subsequent to the GT before entering the combustor. The warm exhaust from the combustor is fed to the GT, where work is extracted. This particular configuration is a combination of a regenerative cycle and a STIG system.

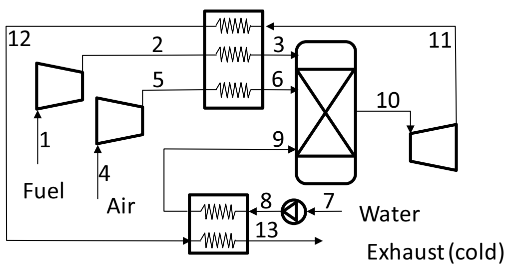

Fig. 5-1 Regenerative steam injection gas turbine (RSTIG)

5.2.2 Simple Linde air liquefaction system (Air Liq)

Liquefaction of gas may be achieved in several ways, usually by cooling, i.e. heat transfer to a cold reservoir, or by expanding the gas, i.e. the fluid performs work on the environment. The former method is suitable for systems at temperatures which are not very low, while the latter method, which is utilized in this study, covers the low temperature region.

Fig. 5-2 illustrates the simple Linde process (e.g. Winterbone (1996)) which starts with ambient (superheated) air which is fed to the compressor, then compressed to 200 bar, which is well above the critical pressure of 37.7 bar. The compression involves interstage cooling with water. Here, the eight compression stages chosen for the study are not shown in the figure for presentation purposes. Liquid water is removed from the flow after each compressor stage. Then the air enters a counter-flow heat exchanger, with no external heat losses or friction, where it is cooled by the return stream from the flash chamber. The air cools down during expansion due to the Joule-Thomson effect and enters the two-phase region. The gas which does not liquefy re-enters the system at the compressor, where additional air from the environment is also added. This is a continuous process, as opposed to the simplified system presented by Winterbone (1996). The yield of liquid gas is determined by the quality of state after throttling. The simplification of the process, hence the label Simple Linde, includes the omission of the cascade process, which implies that the liquefaction here occurs in a single process as opposed to two throttling units and an extra container which would return some of the air back to the compressor at an earlier point.

Szargut et al. (1988) named the throttling which occurs in a Simple Linde process as a "structural exergy loss." In contrast to structural exergy losses, which cannot be reduced or removed without changing the principle of the process, there are "technical exergy losses", which are due to the imperfection of parts of the facility, and hence these can be improved. It was Brodyanskyi (cf. Szargut et al. (1988)) who identified these two forms of internal exergy loss.

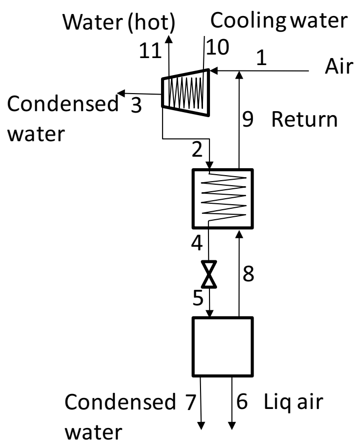

Fig. 5-2 Simple Linde air liquefaction system (Air Liq)

5.2.3 Heat pump water heater (HPWH)

HPWH is short for a heat pump which is used to heat domestic tap water, i.e heat pump water heater. The process is transcritical for this particular system, which means that it operates around the critical point. In the following both the system configuration and the working medium are addressed.

The configuration of the system was based on Nekså et al. (1998) and is illustrated in Fig. 5-3. The cycle is characterized by operation around the critical point for the working fluid, CO_2, with a super-critical high pressure and a sub-critical low pressure. Above the critical point it is no longer possible to distinguish between the gaseous phase and the liquid phase. The compressed (supercritical) CO_2 (1–2) in the figure is led to the gas cooler where heat is rejected to the water (9–10) with a certain temperature glide as the working fluid (CO_2) undergoes a continuously increase in density (2–3) from vapour state to a liquid-like dense gaseous state. Then it passes through the internal heat exchanger (3–4) to heat the counter-flow CO_2 (4–1) before the compression. The throttling follows next (4–5) before the two-phase flow evaporates (5–6) by heat exchange with ambient air in the evaporator. The glycol circuit of Nekså's system was thus replaced by ambient air in this analysis.

CO_2 is a very interesting working fluid in HPWH due to its characteristics, safety reasons and costs. CO_2 has excellent thermophysical properties which gives good heat transfer, efficient compression and high volumetric capacity, which in turn results in a compact system design. In addition, it is non-flammable and non-toxic, as well as inexpensive and easily accessed. According to Nekså et al. (1998), CO_2 was also found to be one of the best suited working fluids for an HPWH application. CO_2 is particularly interesting due to its relatively low critical temperature of 31.1°C. Due to this supercritical state, the condensation is replaced by gas cooling which gives good temperature adaption at counter-flow heat exchange due to the temperature glide. Optimum pressure varies with temperature before throttling and not by high pressure which is typical for conventional vapour compression cycles. The system of Nekså et al. (1998) also includes a low-pressure liquid receiver for CO_2. This unit is necessary in real life in order to change the high pressure (optimum) and at the same time ensure correct feeding of the evaporator. A system without this liquid reservoir will have poor performance at off-design, and it will be very sensitive to CO_2 leakages. The unit is however left out in the presentation as a controller in the system modelled in PRO/II ensured the required state of the working fluid, and there were no leakages in the modelled system. Initially water was heated to 60°C. However, this configuration is fully capable of heating water up to 90°C. This has been confirmed by several studies. Nekså et al. (1998) also maintained the evaporation temperature of 0°C during the parameter change, which was not adequate for the study of systems with two ambient temperatures, as the idea was that the characteristics of the system remained constant.

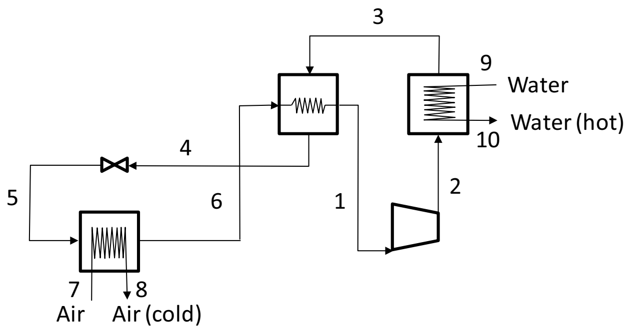

Fig. 5-3 Heat pump water heater (HPWH)

5.3 Main findings

Three systems were investigated: An air-source heat pump water heater (HPWH), a simple Linde air liquefaction plant (Air Liq) and a regenerative steam injection gas turbine (RSTIG). As regards exergy efficiency, the HPWH was very sensitive to the choice of environment, the RSTIG was vaguely affected, and Air Liq was moderately affected. At an ambient temperature difference of 10°C the relative deviations were 61%, -0.22% and -2.0%, respectively. Some cases of less effective versions of these systems were also investigated. It was found that the choice of environment became more important for more effective versions of each system. Special features of the systems were discussed. Indicators for the given significance for other such systems may be the ratio of thermomechanical exergy of ambient water to supplied (or net produced) work when ambient air is chosen as the environment, and the corresponding ratio with thermomechanical exergy of ambient air to supplied (or net produced) work when ambient water is the environment. In short, it may be expressed as the ratio of exergy from the ambient body (not chosen as the environment) to the main exergy flow of the system. The results indicated that systems with either such ratio above 1% should be given special attention.

6 Significance of fixed environment versus natural environment

6.1 Introduction

Engineering tools for calculating the exergy have not been utilized during the work, as there were difficulties in finding (controlling) models and the theory that they were based upon. The thermomechanical exergy values were thus calculated from differences in enthalpy and entropy while the chemical exergy values were based on the compositions. The basic expressions were given in Chapter 2, while additional expressions necessary for the studies were presented in the corresponding papers, see Appendices A–C.

There are commercial programs that have been developed to simulate processing plants that fix the environmental state to 25°C, 70% relative humidity (RH) and 1 atm independently of the natural environment. There are also authors that follow this principle in their calculations. For systems operating in climates which correspond well to this defined environmental state, this might be a good procedure. However, the question remains whether it works everywhere.

The objective is to evaluate the effect on the exergy results for a fixed environmental state of 25°C, 70% RH and 1 atm versus an environment defined from the natural surroundings. For the present study the simple Linde air liquefaction system (Air Liq) and air-source heat pump water heater (HPWH) which were described in Appendix C, "Exergy analysis of systems interacting simultaneously with two ambient bodies of different temperatures", were investigated. The two systems were chosen as they had shown sensitivity to various extents to the choice of the environment. On the other hand, the RSTIG from the paper was left out as the results showed that the choice of environment had a negligible effect on the perceived performance for that system.

The HPWH and Air Liq systems interact simultaneously with two ambient bodies of different temperatures. In other studies these temperatures are either defined to be equal to one another, or one of the temperatures is basically ignored in the analysis when the environment is defined. As opposed to the previous investigation (cf. Appendix C), the ambient water will not be defined as the environment in this study.

In the following, the problem is specified, then a shortened description of the processes, theory, method and present assumptions are presented. Subsequently, the results are given and discussed. Finally, the main conclusions are drawn.

6.2 Problem specification and process description

The purpose of the work is to quantify the effect on the exergy analysis for a fixed environmental state of 25°C, 70% RH and 1 atm instead of a natural ambient body as the environment. Two different systems are chosen: A simple Linde air liquefaction system (Air Liq) and an air-source heat pump water heater (HPWH). When moving from the natural ambient body as the environment to a theoretically defined environment, the system itself remains unchanged. Do the systems' characteristics seem to influence the results in accordance with the findings in Appendix C? Could any general statements be made, i.e. what is state dependent, and what is system dependent? The systems are described in Appendix C. The background with a further description was presented in Chapter 5.

The temperature range is extended, and different levels of relative humidity are also included. The Air Liq and HPWH systems are studied for ambient air temperatures of -20°C, 0°C, 15°C, 25°C and 45°C and relative humidity of 10%, 60% and 90%. The ambient water temperature and ambient pressure remain at 10°C and 1 atm, respectively. The systems at the given ambient states are compared to the same systems and ambient bodies, though the environment is fixed at 25°C and 70% RH.

6.3 Theory and method

The models of the Air Liq and HPWH which were developed by means of PRO/II ver. 8.0 (Simsci Inc.) for the Paper, "Exergy analysis of systems interacting simultaneously with two ambient bodies of different temperatures," see Appendix C, were used in this study.

It is convenient to repeat the utilized definitions of the exergy efficiencies of Air Liq and HPWH.

$$\psi_{\text{Air Liq}} = \frac{\dot{E}_{\text{Liq}} - \dot{E}_{\text{Air in}}}{\dot{W}_{\text{Compr}} + \Delta\dot{E}_{\text{Cooling}}} \quad \textbf{(9)}$$

$$\psi_{\text{HPWH}} = \frac{\dot{E}_{\text{th,Hot water}}}{\dot{W} + \dot{E}_{\text{th,Water}}} \qquad \textbf{(10)}$$

The theory was in accordance with the paper in Appendix C, with the following exceptions:

A useful quantity in the presentation of the results was the difference in %-points between the two exergy efficiencies, which was denoted as $\Delta\psi$ and defined as $\psi_{\text{T}_0=25°\text{C}} - \psi_{\text{T}_0=\text{Ambient air}}$

In the analysis, the relative deviations in exergy efficiency (Eq. (9) and Eq. (10)) were formulated as

$$\text{RD}_{\psi} = \frac{\psi_{T_0=25°\text{C}} - \psi_{T_{0,\text{ Ambient air}}}}{\psi_{T_{0,\text{ Ambient air}}}} = \frac{\psi_{T_0=25°\text{C}}}{\psi_{T_{0,\text{ Ambient air}}}} - 1 \qquad \textbf{(11)}$$

Given that irreversibility is defined as entropy production multiplied by temperature of the environment, the relation between (system/subsystem) irreversibilities is expressed as

$$I_{T_{0,\text{Ambient air}}} = \frac{T_{0,\text{Ambient air}}}{298.15\text{K}} \cdot I_{T_0=\ 298.15\text{K}} \qquad \textbf{(12)}$$

The corresponding relative deviation in irreversibility is thus a constant figure for a given combination of ambient air and constant temperature of 25°C, and it is independent of the system.

$$\text{RD}_{I} = \frac{I_{T_0=\ 298.15\text{K}} - I_{T_{0,\text{ Ambient air}}}}{I_{T_{0,\text{ Ambient air}}}} = \frac{I_{T_0=\ 298.15\text{K}}}{I_{T_{0,\text{ Ambient air}}}} - 1 = \frac{298.15\text{K}}{T_{0,\text{Ambient air}}} - 1 \qquad \textbf{(13)}$$

For instance, for $T_{0,\text{ Ambient air}}$ = 15°C, this relative figure amounts to 3.47%.

6.4 Present assumptions

The assumptions outlined in Appendix C were also valid for this study.

The temperature of water which was fed into both systems was maintained at 10°C, while the temperature of the hot water for HPWH was kept at 60°C.

6.5 Results and discussion

The two systems, Air Liq and HPWH, are presented separately as their characteristics differ a lot. Mutual findings for the systems are then presented, and an overall discussion is conducted.

The actual exergy efficiencies are presented merely to give an idea of how effective the investigated systems are. A simple difference of the exergy efficiencies for different environmental states (ambient air vs. 25°C) is also system dependent, but easy to comprehend. A relative difference takes care of the mentioned obstacle, i.e. gives system independency, and will also be used throughout the presentation of the results. Irreversibility rates are not given, as actual figures or simple differences depend on the size of the plant. The corresponding relative figures are redundant, as they are constants for given combinations of ambient air temperature and the fixed temperature.

6.5.1 Simple Linde air liquefaction system – Air Liq

The temperatures, pressures and mass flow rates in the simple Linde system for T_{Air} = 15°C and 60% RH are presented in Table 6-1. The hyphenated numbers for streams 2, 3, 10 and 11 in this table refer to the eight stages in the compressor. The initial mass flow rate of air was set to 1000 kg/s. In this particular case, no condensed water was extracted from the first four stages of compression (streams 3-1 to 3-4). The total compressor work rate amounted to 620.92 MW. For T_0 = *25°C* the exergy rate of inflowing air was 36.85 kW.

Table 6-1 Temperatures, pressures and mass flow rates in the Air Liq at T_{Air} = 15°C, RH = 60%

Stream	T [°C]	p [bar]	FR [kg/s]
1	15	1.01	70.63
2-1	15	1.97	1000.02
2-2	15	3.81	1000.02
2-3	15	7.39	1000.02
2-4	15	14.33	1000.02
2-5	15	27.79	1000.02
2-6	15	53.89	99.96
2-7	15	104.50	999.78
2-8	15	202.64	999.68
3-5	15	27.79	0.06
3-6	15	53.89	0.19
3-7	15	104.50	0.10
3-8	15	202.64	0.05
4	-105.70	202.64	999.63
5	-194.18	1.01	999.63
6	-194.18	1.01	70.16
7	-194.18	1.01	0.06
8	-194.18	1.01	929.41
9	10	1.01	929.41

10-i [a]	10	1.01	1000
11-1	27.00	1.01	1000
11-2	28.50	1.01	1000
11-3	28.60	1.01	1000
11-4	28.80	1.01	1000
11-5	29.22	1.01	1000
11-6	29.98	1.01	1000
11-7	31.04	1.01	1000
11-8	32.36	1.01	1000

[a] $i = 1, 2, \dots, 8$

Different ambient air temperatures did not affect the air liquefaction exergy efficiencies much for ambient air temperatures from -20°C to 45°C, as shown in Fig. 6-1. For $T_0 = T_{0,\text{ Ambient air}}$ the exergy efficiency increased from 7.5% to 8.5% in the given temperature range. The impact of relative humidity was negligible.

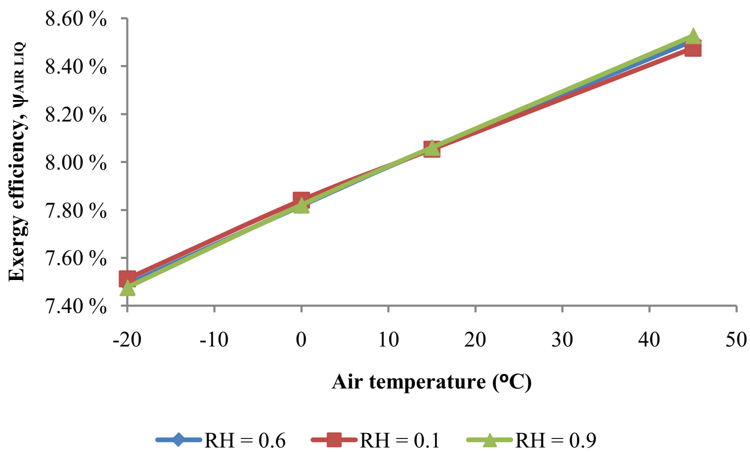

Fig. 6-1 Air Liq exergy efficiencies at different air temperatures when $T_0 = T_{\text{Ambient air}}$

The deviation (in %-points) in exergy efficiency for the air liquefaction system, $\Delta\psi$, varied from 0.7%-points to -0.3%-points between -20°C and 45°C, while the corresponding relative deviations, RD_ψ, were considerable, ranging from 9.4% to -3.9%. The impact of relative humidity was minute. However, small deviations in exergy efficiency were registered at air temperature of 45°C. This was probably due to differences in actual water content in the incoming air, as the relative irreversibility remained constant. This is shown in Fig. 6-2 and Fig. 6-3.

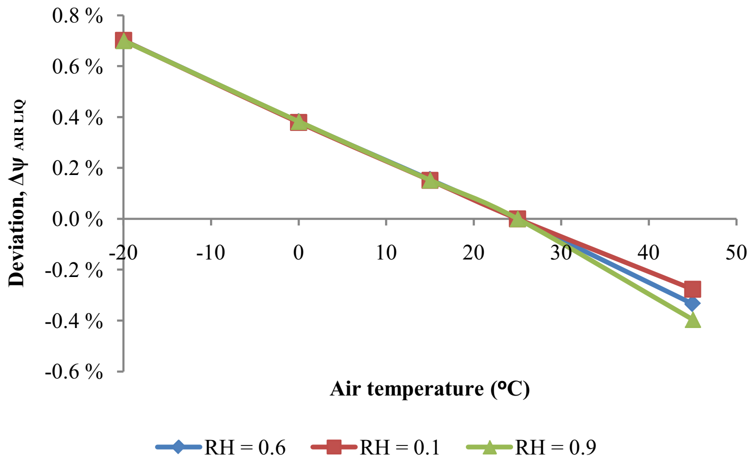

Fig. 6-2 Deviation in exergy efficiency (%-points) for Air Liq at different ambient air temperatures

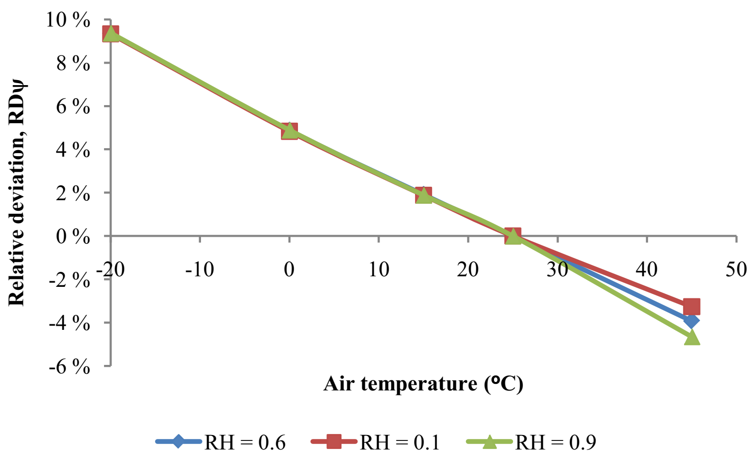

Fig. 6-3 Relative deviation in exergy efficiency for Air Liq at different ambient air temperatures

6.5.2 Heat pump water heater - HPWH

The temperatures, pressures and mass flow rates in the HPWH at T_{Air} = 15°C and 60% RH are presented in Table 6-2. Streams 1–6 represent CO_2, streams 7–8 represent air, while streams 9–10 represent water. This particular case resulted in a compressor input power of 44.35 kW, while the coefficient of performance (COP, taken as the rate of water enthalpy increase divided by the compressor input power, where the efficiency of the electric motor is 100%) was 4.7.

Table 6-2 Temperatures, pressures and mass flow rates in HPWH at T_{Air} = 15°C, RH = 60%

Stream	T [°C]	p [bar]	FR [kg/s]
1	0	32.21	0.82
2	86.27	90.26	0.82
3	17	90.26	0.82
4	15.10	90.26	0.82
5	-3	32.21	0.82
6	-3	32.21	0.82
7	15	1.01	30
8	9.57	1.01	30
9	10	1.01	1
10	60	1.01	1

As expected from the former study, the exergy efficiencies were sensitive to change of ambient air temperature. The relative humidity of air had negligible effect on the perceived performance. The exergy efficiencies dropped from 54% to 5% in a temperature range of -20°C to 45°C.

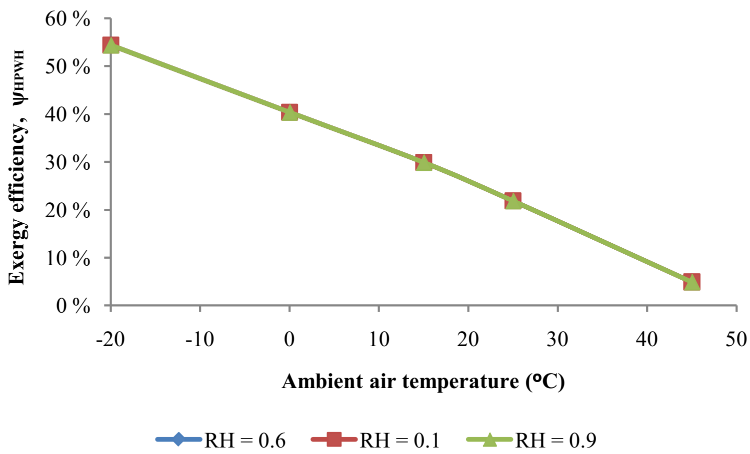

Fig. 6-4 HPWH exergy efficiencies at different ambient air temperatures when $T_0 = T_{Ambient\ air}$

The deviations between performances calculated at the fixed environmental state of 25°C and performances with the ambient air as the environment were extraordinary. The relative humidity had no impact on the results even at 45°C, as opposed to Air Liq. This was due to the chemical exergy of the systems cancelled out in the exergy balance. For the given temperature range the deviation in exergy efficiencies varied from -44.5%-points to 31.3%-points, which gives the corresponding relative deviations of -81.9% to 634.50%. These trends are shown in Fig. 6-5 and Fig. 6-6.

The deviations in %-points appeared to be close to linear, while the relative deviation increased rapidly for air temperatures above 25°C.

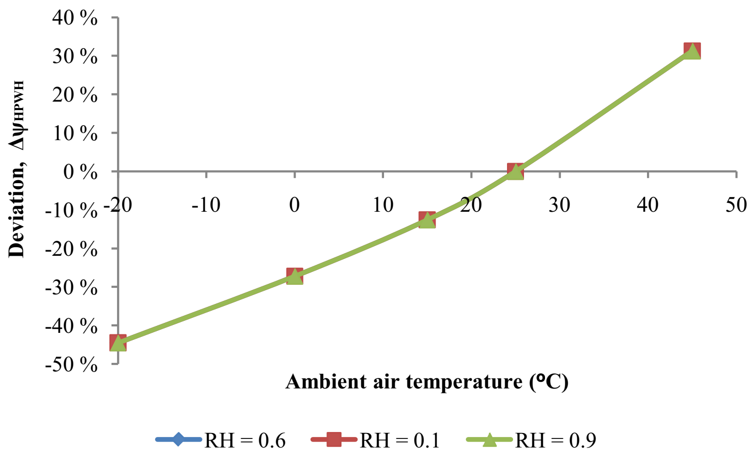

Fig. 6-5 Deviation in exergy efficiency (%-points) for HPWH at different ambient air temperatures

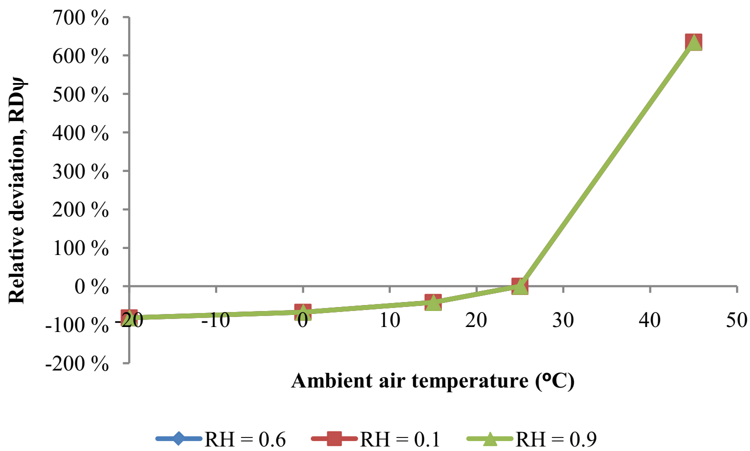

Fig. 6-6 Relative deviation in exergy efficiency for HPWH at different ambient air temperatures

6.5.3 Accuracy

The accuracy was in accordance with the discussion presented in Appendix C, as the same procedure was followed.

6.5.4 General findings

Two different systems were investigated. As expected from the characteristics (cf. Appendix C), the range of exergy efficiency changed for each system. Obviously, the closer the ambient temperature was to 25°C, the smaller the (relative) deviation in perceived performance. The relative humidity had a negligible effect on the results.

Some key results of the investigation are summarized in Table 6-3. The choice between a fixed environmental state and ambient air as the environment had different impacts on the exergy results for Air Liq compared to HPWH. When the actual ambient state deviated from this theoretically fixed state chosen as the environment, the system received exergy from the ambient bodies. When the exergy from the ambient body was considerable in comparison to a typical exergy amount for the system, i.e. power received, the impact on the exergy results was significant.

The results were in accordance with the findings of "Energy analysis of systems interacting with two bodies of different temperatures." The choice of environment affected the exergy results when the exergy received from the ambient body was of magnitude 1–2% of a main exergy flow of the system. This limit would, however, depend on the circumstances and on the required accuracy of the analysis. Note that the thermomechanical exergy from the air was negligible for Air Liq, whereas the contribution from the other ambient body (ambient water at 10°C) was modest.

Table 6-3 Overview of systems: Key results at $T_{Ambient\ air}$ = –20, 0, 15, 45°C, $T_{Ambient\ water}$ = 10°C

System	Exergy efficiency [%][a]	Impact of choice on perceived performance	T_0 = 25°C Thermomech. exergy from ambient air[b]	T_0 = 25°C Thermomech. exergy from ambient water[b]
Air Liq	7 – 9	Modest	0.01 – 0.04%	2.1%
HPWH	5 – 54	Large	12 – 144%	2.1 – 8.0%

[a] $T_0 = T_{0,Air}$
[b] of supplied work

As pointed out in Appendix C, the irreversibility rate of the system could replace the supplied (or net produced) power in the argument above, as it was of the same order of magnitude. However, the power is usually more readily estimated and requires less computation and effort.

6.5.5 Overall discussion

Obviously, there are many systems that are not covered by this study, as the issue investigated does not only apply to systems interacting with two ambient bodies of different temperatures. The results were in accordance with the findings of the former study, which were expected.

This study indicates that the ratio of the exergy received from every ambient body (interacting with the system) to the main exergy flows of the systems ought to be considered before following the approach of fixed environmental conditions as it may affect the exergy results considerably. Even small amounts of ambient exergy could have an impact on the perceived performance.

Are there aspects of the systems which may explain the findings? As the systems remain unchanged from the former study (cf. Appendix C), the characteristics which were discussed are also valid for this analysis. The main (or target) mass flow of the HPWH is taken from the surroundings (ambient water), and moreover, the main energy flow (heat) is also taken from the surroundings (ambient air). The Air Liq system's main material flow (air) is taken from the surroundings, whereas the main energy flow (work) is provided from outside. For both HPWH and Air Liq, the product of the process is a change of state in the material flow from the surroundings. The HPWH had small exergy flows and no change in composition, which in turn caused small absolute deviations, but there were huge relative deviations. In this context, the definition of the performance ratio was particularly important.

As the findings supported the results of Appendix C, an impact on the perceived performance would not be expected for the regenerative steam injection gas turbine system investigated in the former paper. The characteristics of RSTIG included that the main energy flow (fuel) was provided from outside, whereas the main flow was from the surroundings and the desired product was produced power, not a change in state.

6.6 Conclusions

Exergy analyses have been conducted for two different systems: A simple Linde air liquefaction system (Air Liq) and an air-source heat pump water heater (HPWH). It was investigated how much a theoretically fixed environmental state of 25°C and 70% relative humidity would affect the exergy results compared to an environment defined from the natural surroundings. A special feature of the systems was that the systems interacted simultaneously with two ambient bodies of different temperatures: Ambient water and ambient (atmospheric) air. However, the findings also apply to systems with a single ambient body which deviates from this theoretically fixed environmental state. The environmental pressure remained at 1 atm, the ambient water temperature was 10°C. The study was limited to 10%, 60% and 90% RH and ambient air temperatures of -20°C, 0°C, 15°C and 45°C.

For the air liquefaction system the choice of environment had some, yet still a modest impact on the exergy results, whereas the effect of relative humidity was negligible. The relative deviation in exergy efficiency amounted to 1.9% at an ambient air temperature of 15°C.

The impact on the exergy results for the HPWH was large when the temperatures of the ambient bodies interacting with the systems differed from 25°C. A fixed environmental state of 25°C resulted in an exergy efficiency five times larger than that found when choosing ambient air of -20°C as the environment. The relative deviation in exergy efficiency came here to -44.6% On the other hand, the relative humidity showed no effect on the perceived performance.

The ratios of the thermomechanical exergy of the ambient bodies interacting with the system to supplied (or net produced) work can serve as indicators for the significance on the exergy results of a fixed environmental state of 25°C and 70% RH versus the natural surroundings as the environment. However, the relative humidity had a negligible impact on the exergy results. The findings from the former investigation holds, i.e. such ratios above 1% should be given special attention.

The results were expected, though it was important to quantify them in order to visualize the effect on perceived performance for a fixed environmental state which does not correspond with the actual surroundings. Again the importance of assessing both ambient bodies, when the system interacts with more than one ambient body, was confirmed. The findings suggest caution in terms of using a fixed environmental state in calculations and engineering tools when this state deviates from the natural surroundings. The significance depends on the deviation from the actual surroundings, the magnitude of the significance indicator and obviously the required accuracy of the analysis.

7 Conclusions

Three distinct cases have been investigated for different purposes. It is thus convenient to present the conclusions for each study separately before common findings are drawn.

7.1 Exergy evaluation of the Arctic Snøhvit LNG processing plant in northern Norway – significance of ambient temperature

The ratio of the energy of the products to the energy of the feed was 93.3%, while the corresponding exergy ratio was 95.1%. The high figures were due to the dominant, nearly unchanged chemical exergy. Thus, it was adequate to adapt another expression for the exergy efficiency, i.e. the ratio of the desired change in exergy of the products due to separation, cooling and compression to consumed exergy to achieve this. This approach gave an exergy efficiency of a modest 23.2%. The cooling of LNG accounted for the main part of the desired exergy change of 20.6%-points, whereas separation processes and compression of CO_2 amounted to 1.9 %-points and 0.7 %-points of the consumed exergy, respectively. The exergy losses were distributed as 37% in the processing plant, 52% in the gas turbines including the combustors, and 11% in the heat recovery unit including the exhaust to stack. The combined heat and power plant had an exergy efficiency and energy efficiency of 43.5% and 62.2%, respectively. The heat recovery unit is the most obvious candidate for improvement, i.e. minimizing the temperature differences and decreasing the exhaust temperatures (to stack). Estimates indicate that separation processes were less efficient compared to the processes of both compression and refrigeration.

The profit of the Arctic location in terms of fuel consumption was also quantified. It had a considerable positive impact, as reducing the ambient temperature from 36°C to 4°C gave a 19.9% reduction, whereas a decrease of the ambient temperature from 20°C to 4°C resulted in a 10.9% reduction. Hence, a decrease in fuel consumption beyond these figures can be solely attributed to technological improvements. If the reduction of 30–50% expected by the developer holds, the Snøhvit LNG plant represents a substantial improvement.

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

The exergy efficiency of the existing system was 44.3%, which dropped 1.8%-points when one of the HRSGs was inoperative. The efficiencies of the CHPs were up to 15%-points higher than the direct fired boilers. Only 6.2% of the total irreversibilities were attributed to the steam distribution system.

The fuel consumption increased in a range of 2.6–6.6% by implementing the different alternatives. The total irreversibilities also increased, except for the two-stage pressure steam production which decreased. However, the additional power production was considerable and resulted in remarkable 89–104% marginal electric efficiencies. These figures may be compared to the typical figure for electric efficiency of a new conventional power plant, which is approximately 55–58%.

An additional 16.2 MW of electrical power production and 17.6 MW additional fuel exergy consumption resulted from replacing throttling with steam turbine expansion, i.e. marginal exergy efficiency of 92%. Increasing the steam production pressure from 59 bar to 120 bar resulted in an additional 18.5 MW of power, i.e. a marginal exergy efficiency of 89.8% (single pressure), or 15.4 MW of additional power, i.e. a marginal exergy efficiency of 98.7% (59/120 bar dual pressure). Combining additional steam turbines in the steam distribution system and elevated production pressure (120 bar) resulted in the corresponding figures of 35.5 MW/90.8%.

Implementing and using the steam turbines are two relatively simple actions, whereas increasing the pressure in the HRSGs/boilers will most likely require new equipment. Nevertheless, the study demonstrated that there is considerable potential for improvement when new HRSGs/boilers are installed and the pressure is increased from the present level. In fact, the combined alternative of steam turbines and elevated production pressure gave an additional power production larger than that of the original plant.

The study provided an example from an existing, industrial steam system that illustrates both the losses from throttling and low-pressure steam production, as well as the practical potentials for improvement and “mental barriers.”

7.3 Exergy analysis of systems interacting simultaneously with two ambient bodies of different temperatures

Exergy analyses have been performed for three different systems that each interacted with two ambient bodies of different temperatures: A regenerative steam injection gas turbine (RSTIG), a simple Linde air liquefaction system (Air Liq) and an air-source heat pump water heater (HPWH). One of these ambient reservoirs (air/water) was chosen as the environment, and then the impact of this choice on the exergy results was investigated.

There were minor effects on the exergy results for RSTIG, as the relative deviation in exergy efficiency amounted to a negligible 0.2% at an ambient temperature difference of 10°C. This is probably due to the small exergy streams received from the other ambient body compared to the main exergy flows, such as supplied fuel or net produced power. The relative impact on the exergy efficiency was also minor for Air Liq, where the corresponding figure was 2.0%. On the other hand, HPWH was remarkably sensitive to the choice of environment. For some cases, choosing ambient water as the environment resulted in exergy efficiencies two to three times larger compared to ambient air as the environment. The relative deviation in exergy efficiency amounted here to 61% at an ambient temperature difference of 10°C. Moreover, the exergy results were very sensitive to the definition of exergy efficiency.

The relative deviations in exergy efficiency for Air Liq and HPWH remained unaltered, and close to for RSTIG when some cases of less efficient versions of these systems were studied. This means that the choice of environment becomes more important for more effective versions of each system.

A specific feature of HPWH may be that both the main mass and main energy flows were taken from one of the ambient bodies. For Air Liq the main mass flow was also from one of the ambient bodies, while work which represented main energy flow was provided from outside the system. As regards RSTIG, where the desired product was not the change of state, but produced work, the main mass flow was from one of the ambient bodies.

Three systems only were investigated. Suitable indicators to reveal the significance for other systems may be the ratio of the thermomechanical exergy of ambient water to supplied (or net produced) work when ambient air is chosen as the environment and the ratio of thermomechanical exergy of air to supplied (or net produced) work when ambient water is the environment. The study indicated that systems with corresponding ratios above 1% should be given special attention. Moreover, the investigation suggested that the choice of environment will affect the results even for relatively small amounts of ambient exergy, i.e. exergy from the other ambient body not chosen as environment.

7.4 Significance of fixed environment versus natural environment

The objective was to evaluate how a theoretically fixed environmental state of 25°C, 70% RH and 1 atm would affect the exergy results compared to an environment defined from an actual ambient body. The study was limited to the simple Linde air liquefaction system (Air Liq) and the heat pump water heater (HPWH) at the ambient air temperatures of -20°C, 0°C, 15°C and 45°C and 10%, 60% and 90% RH.

The study confirmed the findings from Paper 3 as expected and suggested that neither calculations nor software tools should uncritically be based on a fixed environment when this fixed state deviates from the natural ambient state. The ratios of the thermomechanical exergy of the ambient bodies interacting with the system to the main exergy stream of the system, i.e. supplied work (or net produced work), can serve as indicator for the significance on the exergy results. However, the relative humidity had a negligible impact on the exergy results. The significance depends on the deviation from the actual ambient state, the magnitudes of the significance indicator(s) and also the required accuracy of the analysis.

7.5 Overall review

Three rather distinct cases have been investigated for different purposes. Two Norwegian operational plants have been investigated, one operative for close to 30 years (Kårstø steam production and distribution system), whereas the other has just started its expected 30 year of production (Snøhvit LNG plant).

There are, however, a few common features. The critical presentation of performance data has been emphasized to avoid misleading the reader regarding the actual performance. In theory, and cynically speaking, performance ratios may be used to demonstrate whatever outcome that is wanted. Here, performance ratios were used to illustrate the potential for improvement instead of finding impressive figures. In particular, the study of the Snøhvit LNG plant quantified the significance of large transit exergy for the perceived performance. “Significance indicators” were used to assess the importance, also for the study of the Kårstø steam production and distribution system in terms of marginal efficiencies.

The different findings may be useful in future studies and in the optimization of processing plants. The authorities may use the findings of the operational plants to encourage and demand better performance in future plants. The findings for the systems with two ambient bodies suggest that caution should be exercised when performing exergy analyses where the temperatures of the ambient bodies are not equal. This will, of course, depend on the needed accuracy level. It could be interesting to perform further studies of other such systems, or systems that actually utilize the difference in ambient bodies to produce power, such as hydro power, geothermal, ocean thermal conversion and osmotic power production. In particular, the condensing power plant could be worth investigating. The findings from the investigation of a fixed environmental state versus an actual ambient state as the environment also suggest that caution is needed in terms of applying a fixed environment which deviates from the natural ambient state, i.e. not used uncritically in calculations or by future engineering tools. This study was limited to two systems and a rather coarse selection of ambient temperatures, so further analysis of other systems would be interesting.

References

Ahern, J.E., 1980. The exergy method of energy systems analysis. New York, John Wiley & Sons, Inc.

Ahrendts, J., 1980. Reference states, Energy 5, 667–677.

Aleklett, K., Höök, M., Jakobsson, K., Lardelli, M., Snowden, S., Söderbergh, B., 2010. The peak of the oil age – analyzing the world oil production reference scenario in World Energy Outlook 2008. Energy Policy 38 (3), 1398–1414.

Baehr, H.D., 1968. Zur Definition exergetischer Wirkungsgrade – eine systematische Untersuchung. Brennst-Wärme-Kraft 20 (5), 197–200 (in German).

Boem, R.F., 1997. Developments in design of thermal systems. Cambridge, UK: Cambridge University Press.

Bosnajakovic, F., 1938. Kampf den Nichtumkehrbarkeiten, Arch. F. Wärmewirtshaft und Dampfkesselwesen 19 (1) (in German).

Brodyansky, V.M., Sorin, M.V., Le Goff, P., 1994. The efficiency of industrial processes: exergy analysis and optimization. Amsterdam, The Netherlands, Elsevier.

Campbell, J.C., Laherrere, J.H., 1998. The end of cheap oil, Scientific American, March 1998.

Carnot, N.L.S., 1824. Reflections on the motive power of heat. Translated from French by R.H. Thurston. New York, John Wiley & Sons, 1897.

Demirel, Y., 2004. Thermodynamic analysis of separation systems. Sep. Sci. Technol. 39 (16), 3897–3942.

Ertesvåg, I.S., 2007. Sensitivity of the chemical exergy for atmospheric gases and gaseous fuels to variations in ambient conditions. Energy Convers. Managm. 48 (7), 1983–1995.

Fratzscher, W., Brodjanskij, V.M., Michalek, K., 1986. Exergy – Theory and application (in German), 1st ed. Leipzig, Germany, VEB Deutscher Verlag für Grundstoffindustrie.

Grassmann, P., 1950. Zur allgemeinen Definition des Wirknungsgrades. Chem-Ing.-Technik 22 (4), 77–80 (in German).

Höök, M., Hirsch, R., Aleklett, K., 2009. Giant oil field decline rates and their influence on world oil production. Energy Policy 37 (6), 2262–2272.

ISO2314:2009(E). Gas turbines – acceptance test, international standard, 2009-12-15, 3rd ed.

Kotas, T.J., 1995. The exergy method of thermal plant analysis, 2nd ed. Malabar, Florida: Krieger Publishing Company.

Leites, I.L., Sama, D.A., Lior, N., 2003. The theory and practice of energy savings in the chemical industry: some methods for reducing thermodynamic irreversibility in chemical technology processes, Energy 28 (1), 55–97.

Lior, N., Zhang, N., 2007. Energy, exergy and second law performance criteria. Energy 32 (4), 281–296.

Moran, M.J., Shapiro, H.N., 2004. Fundamentals of engineering thermodynamics. 5th ed. Hoboken, NJ, Wiley.

Nekså, P., Rekstad, H., Reza Zakeri G., Schiefloe, P.A., 1998. CO_2-heat pump water heater: characteristics, system design and experimental results, Int. J. Refrig. 21 (3), 172–179.

Nishida, K., Takagi, T., Kinoshita, S., 2005. Regenerative steam-injection gas-turbine systems, Applied Energy, 81, 231–246.

Norsk Varmepumpeforening. 2010. Fakta om varmepumper (in Norwegain).
See also:
http://www.novap.no/fakta_om_varmepumper (Last accessed January 2011)

Norwegian Petroleum Directorate, 2010. Norwegian continental shelf – a journal from the Norwegian Petroleum Directorate 2.

Rant, Z., 1956. Exergie, ein neues Wort für Technische Arbeitsfähigkeit, Forsch. Ing. Wes. 22, 36–37 (In German).

Sciubba, E., Wall, G., 2007. A brief commented history of exergy from the beginnings to 2004, Int. J. of Thermodynamics 10 (1), 1–26.

Simsci Inc., Pro/II v.8.0. See also:
http://www.simsci.com

Statistics Norway (Statistisk Sentralbyrå), 2008. Energy consumption per household, 2006. See also:
http://www.ssb.no/english/subjects/01/03/10/husenergi_en/ (last accessed Dec 2010)

Statistics Norway (Statistisk Sentralbyrå), 1.10.2010. Focus on population. See also:
http://www.ssb.no/english/subjects/02/befolkning_en/ (last accessed Dec 2010).

Statoil, 2001a. Impact assessment for Snøhvit (in Norwegian) Konsekvensutredning Snøhvit LNG, April 2, 2001.
http://www.statoil.com/no/EnvironmentSociety/Environment/impactassessments/fielddevelopments/Downloads/Sn%C3%B8hvit%20LNG%20-%20Konsekvensutredning%20-%20April%202001.pdf (last accessed Nov 2010).

Szargut, J., Morris, D.R., Steward, F.R., 1988. Exergy analysis of thermal, chemical and metallurgical processes. New York, Hemisphere.

Winterbone, D.E., 1996. Advanced thermodynamics for engineers, 1st ed. Butterworth-Heinemann.

Wølneberg, P.V., Ertesvåg, I.S., 2008. Alternatives for power supply to natural-gas export compressors combined with heat production evaluated with respect to exergy utilization and CO_2 emissions, Energy Convers. Managm., 49 (12), 3531–3540.

Appendix A – Paper 1

Exergy evaluation of the Arctic Snøhvit LNG processing plant in northern Norway – significance of ambient temperature

Anne Berit Rian[1], Ivar S. Ertesvåg[2]

Department of Energy and Process Engineering
Norwegian University of Science and Technology
Kolbjørn Hejes v. 1b
NO-7491 Trondheim, Norway

[1] anne.b.rian@ntnu.no, anne.berit.rian@gmail.com, tel: +47 73 59 35 97, fax: +47 73 59 35 80

[2] Corresponding author: ivar.s.ertesvag@ntnu.no , tel: +47 73 59 38 39, fax: +47 73 59 35 80

Is not included due to copyright

Appendix B – Paper 2

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

Anne Berit Rian[1] and Ivar S. Ertesvåg[2]

Department of Energy and Process Engineering
Norwegian University of Science and Technology
Kolbjørn Hejes vei 1b, NO-7491 Trondheim, Norway

[1] Corresponding author: anne.b.rian@ntnu.no, tel: +47 73 59 35 97, fax: +47 73 59 35 80
[2] ivar.s.ertesvag@ntnu.no

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

Anne Berit Rian *, Ivar S. Ertesvåg

Department of Energy and Process Engineering, Norwegian University of Science and Technology, Kolbjørn Hejes vei 1b, NO-7491 Trondheim, Norway

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

* Corresponding author. Tel.: +47 73 59 35 97; fax: +47 73 59 35 80.
E-mail addresses: anne.b.rian@ntnu.no (A.B. Rian), ivar.s.ertesvag@ntnu.no (I.S. Ertesvåg).

0196-8904/$ - see front matter © 2010 Elsevier Ltd. All rights reserved.
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-

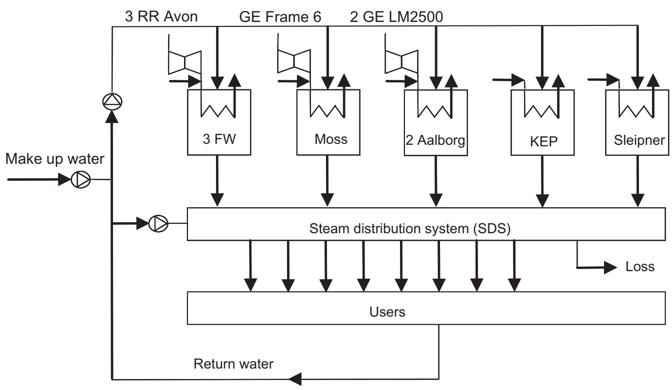

Fig. 1. The existing steam production system (CHP/boilers) and the accompanying steam flows.

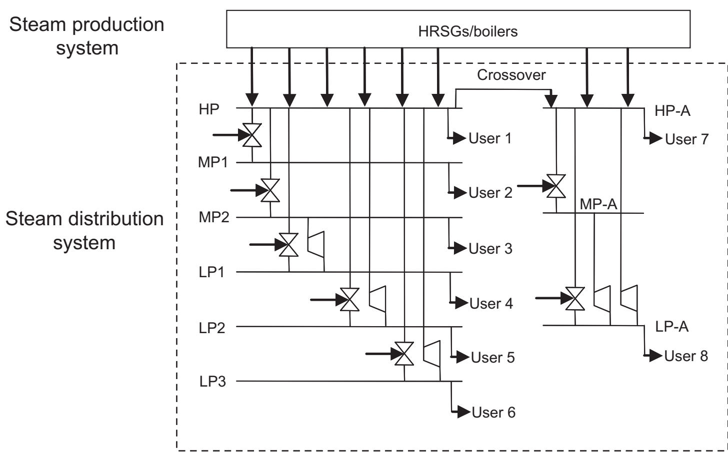

Fig. 2. The existing steam distribution system (SDS). The user flows go out of the sub-system.

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	∑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

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

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, $\varepsilon = \varepsilon_{th} + \varepsilon_0$. The thermomechanical exergy is determined from

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

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

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

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 v_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

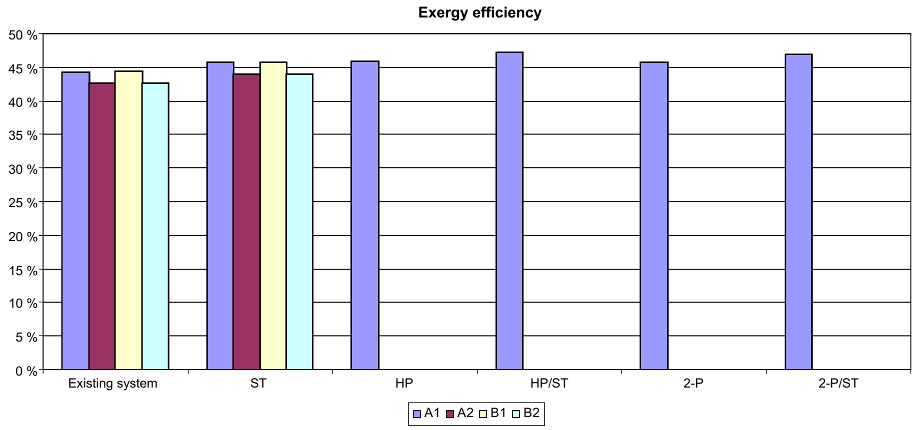

Fig. 3. Exergy efficiency in terms of ratio of useful exergy to fuel exergy.

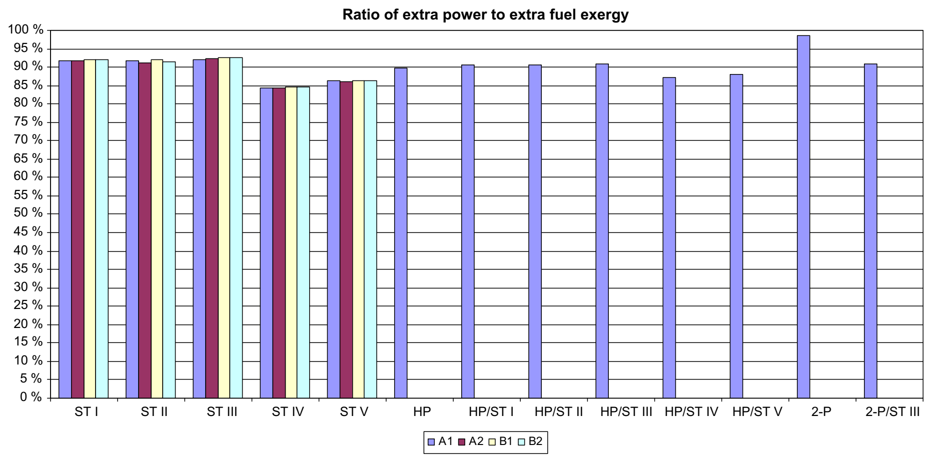

Fig. 4. Marginal exergy efficiency in terms of ratio of extra power to extra fuel exergy.

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

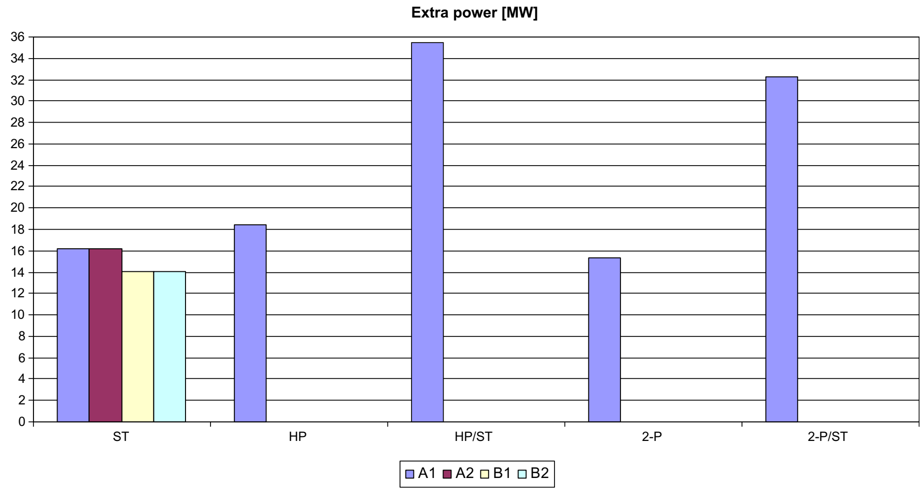

Fig. 5. Extra power (MW) compared to the existing system.

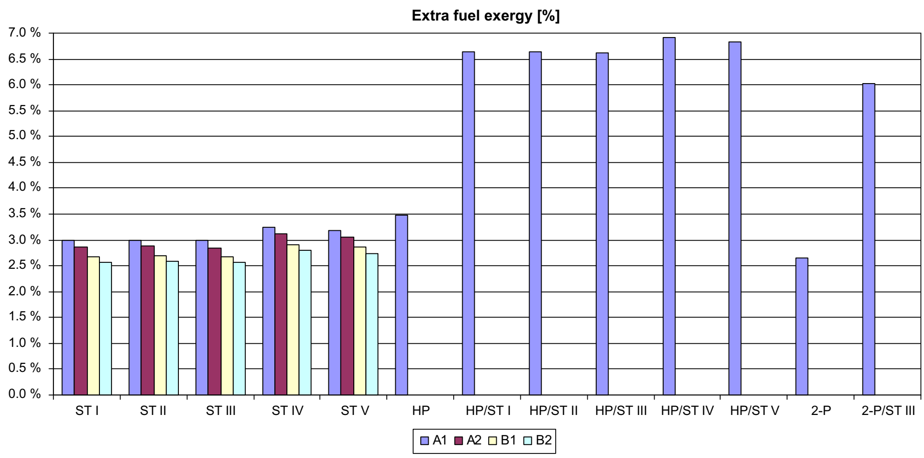

Fig. 6. Extra fuel (%) compared to the exisiting system.

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

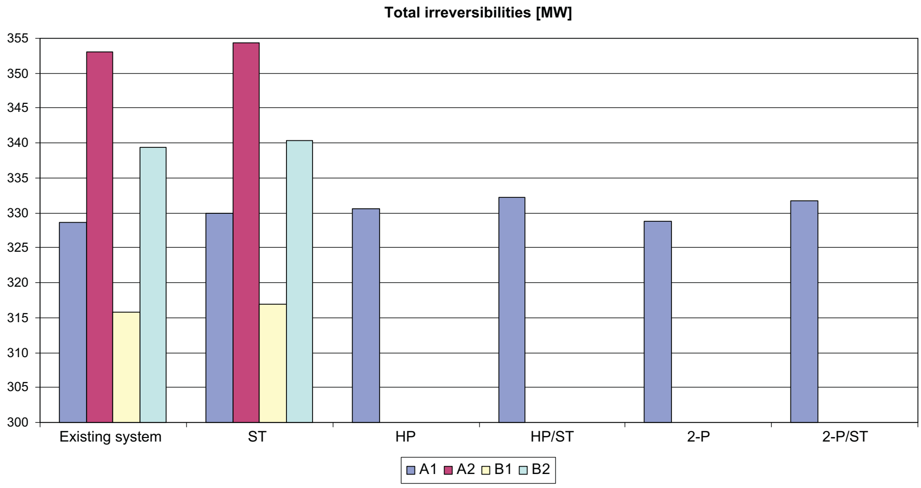

Fig. 7. Total irreversibilities (MW).

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.

References

[1] Norwegian Petroleum Directorate. Facts 2009 – The Norwegian petroleum sector. Oslo, Norway; 2009. <http://www.npd.no/English/Produkter+og+tjenester/Publikasjoner/Faktaheftet/Faktaheftet+2009/Fakta+2009.htm>.
[2] Boem RF. Developments in design of thermal systems. Cambridge, UK: Cambridge University Press; 1997.
[3] Leites IL, Sama DA, Lior N. The theory and practice of energy savings in the chemical industry: some methods for reducing thermodynamic irreversibility in chemical technology processes. Energy 2003;28(1):55–97.
[4] Moran MJ, Shapiro HN. Fundamentals of engineering thermodynamics. 5th ed. John Wiley & Sons; 2004.
[5] Wølneberg PW. Energy use and energy utilization at Kårstø gas processing plant (Energibruk og energiutnyttelse ved Kårstø gassprosesseringsanlegg), Master thesis. Department of Energy and Process Engineering, Norwegian University of Science and Technology. Trondheim, Norway; June 2007 [In Norwegian].
[6] Rian AB, Wølneberg PW, Ertesvåg IS. Steam production scenarios for a natural gas processing plant in Norway with respect to exergy utilization, Paper presented at ECOS2008.
[7] Wølneberg PV, Ertesvåg IS. Alternatives for power supply to natural-gas export compressors combined with heat production evaluated with respect to exergy utilization and CO_2-emissions. Energy Convers Manage 2008;49(12):3531–40.
[8] Kotas TJ. The exergy method of thermal plant analysis. 2nd ed. Malabar, Florida: Krieger Publishing Company; 1995.
[9] Szargut J, Morris DR, Steward FR. Exergy analysis of thermal, chemical and metallurgical processes. New York: Hemisphere; 1988.
[10] Ertesvåg IS. Sensitivity of the chemical exergy for atmospheric gases and gaseous fuels to variations in ambient conditions. Energy Convers Manage 2007;48(7):983–1995.
[11] Simsci Inc. Pro/II v.8.0. <http://www.simsci.com>.
[12] Soave G. Equilibrium constants from a modified Redlich–Kwong equation of state. Chem Eng Sci 1972;27:1197–203.
[13] Kabadi VN, Danner RP. A modified Soave–Redlich–Kwong equation of state for water-hydrocarbon phase equilibria. Ind Eng Chem Proc Des Dev 1985;24(3):537–41.
[14] Norwegian Meteorological Institute. eKlima – climatological database. <http://eklima.met.no> [Visited in August 2009].
[15] Haar L, Gallagher JS, Kell GS. NBS/NRC steam tables: thermodynamic and transport properties and computer programs for vapour and liquid states og water in SI units. Washington: Hemisphere Publ. Corp.; 1984.
[16] EES, Engineering Equation Solver. In: Student resource DVD enclosed with cengel and boles, thermodynamics: an engineering approach, 6th ed., Berlin: McGraw Hill; 2008.

Appendix C – Paper 3

Exergy analysis of systems interacting simultaneously with two ambient bodies of different temperatures

Anne Berit Rian[1] and Ivar S. Ertesvåg[2]

Department of Energy and Process Engineering
Norwegian University of Science and Technology
Kolbjørn Hejes vei 1b, NO-7491 Trondheim, Norway

[1] Corresponding author: anne.b.rian@ntnu.no, anne.berit.rian@gmail.com, tel: +47 73 59 35 97, fax: +47 73 59 35 80
[2] ivar.s.ertesvag@ntnu.no

Is not included due to copyright