Applied Thermal Engineering 63 (2014) 297–303

Contents lists available at ScienceDirect

Applied Thermal Engineering

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

Design and performance measurements of an organic vapour turbine

Piotr Klonowicz [a,d,*], Aleksandra Borsukiewicz-Gozdur [b], Paweł Hanausek [c], Władysław Kryłłowicz [d], Dieter Brüggemann [a]

[a] Center of Energy Technology, University of Bayreuth, Germany
[b] West Pomeranian University of Technology, Power Plants Research and Development Centre, Poland
[c] Turboservice Sp. z o.o., Poland
[d] Lodz University of Technology, Poland

ARTICLE INFO

Article history:
Received 24 May 2013
Accepted 9 November 2013
Available online 19 November 2013

Keywords:
ORC
Turbine
CFD
Organic working fluid
R227ea

ABSTRACT

The paper presents design assumptions and process with numerical analysis and preliminary experimental results of a small scale turbine. The turbine is an integral part of a hermetic turbogenerator operating in a low temperature, sub-critical ORC system. The working fluid is R227ea. First the selection of the turbine type is discussed along with the design assumptions. The set of applied empirical loss correlations as well as the geometry design methodology are outlined. Next, the results of the stage numerical simulations are briefly described. The last part of the paper contains the description of the experiment with a presentation of the test rig, measuring equipment and comparison of the predicted and measured turbogenerator efficiency. Theoretical and measured values of the efficiency reveal a very good mutual agreement.

1. Introduction

For many years one can observe a significant increase in the use of working fluids other than water in power plants and CHP systems. Because of their thermodynamic properties, various types of organic fluids are often considered and utilized. For this reason the abbreviation "ORC" (Organic Rankine Cycle) is common now in the literature of the subject. The power plants with ORC are used for distributed generation, especially if the source temperature is low or medium [1]. The mentioned subject literature concerning various aspects of organic cycle optimization is currently quite wide especially in the area of fluid selection and system operational parameters with respect to the source parameters [2–4]. In the paper of Karellas et al. [5] the influence of the fluid on the heat exchanger design was discussed. In the work of Borsukiewicz-Gozdur [6] the influence of the fluid type on the pumping work was investigated. Papers devoted to the aspect of the turbine for the cycle [7–15] are increasing. Harinck et al. [7] presented a RANS simulation of a radial turbine designated to work in a system based on a toluene cycle. The toluene was regarded as a real gas described by a proper EOS. The turbine efficiency for the design point is equal to about 70% according to the numerical model. The nozzle blade row for this radial turbine was optimized [8] by means of a genetic algorithm and off-line trained metamodels with respect to 3 criteria (total pressure loss, mean angle deviation and flow uniformity). The computations were based on solving two-dimensional equations for inviscid flow. Major improvements in all of the three criteria were achieved. Qiu et al. [9] summarized the findings of the marked research for the expanders, including turbines, and discussed the selection and choices of the expanders for the ORC-based micro-CHP systems. Uusitalo et al. [10] investigated the suitability of siloxans for small ORC turbogenerators based on high-speed technology. The MDM and D4 were selected as the most suitable fluids in that fluid class due to relatively high cycle efficiency and acceptable condensing pressure. The whole enthalpy drop of the mentioned fluids for the parameters given by the authors can be processed in a single turbine stage. Turunen-Saaresti et al. [11] assessed the influence of the turbine efficiency on the profitability of the electric power production. The authors show that introducing complex design solution in order to increase efficiency of small units is not economically justified. The work of Klonowicz et al. [12] presents a numerical study concerning the influence of the stator angle on the impulse turbine stage efficiency. The study shows that there exists an optimal value of this angle at about $10°–11°$ but at lower angles a dramatic efficiency drop occurs. Thus, it is recommended to apply the stator angles of above $12°$. Klonowicz and Hanausek [13] used a commercial CFD code in

* Corresponding author. Center of Energy Technology, University of Bayreuth, Germany.
E-mail addresses: pklonowicz@gmail.com, piotr.klonowicz@uni-bayreuth.de (P. Klonowicz).

1359-4311/$ – see front matter
http://dx.doi.org/10.1016/j.applthermaleng.2013.11.018

order to study the flow field in a few ORC turbine stages. They concluded that a major improvement can be achieved by proper designing of the turbine blades. Kang [14] presented a design and an experimental study of a radial reaction turbine operating with the fluid R245fa. The design rotational speed of the machine was 20,000 rpm and the predicted nominal efficiency was 75%. The turbine was tested at three operating points corresponding to measured turbine efficiencies of 76.0%, 77.5% and 82.2%. Pini et al. [15] presented design and analysis methodology for a multistage centrifugal turbine. Initially the turbine was designed with a 1D mean line method and loss correlations. The 1D code has been coupled with a 2D, axisymmetric and inviscid throughflow solver which is a bridge between 1D and 3D approach. It is fast comparing to the 3D solver but more accurate then 1D calculations which make it suitable for optimization of multistage machines.

In the present paper the whole process including the assumptions, design process, numerical simulations and experimental validation for the turbine operating with the fluid R227ea is presented.

2. Assumptions

The turbine presented herein was designated to work as the expander for a test rig simulating a geothermal power plant powered by water of temperature 95 °C. As a result of the cycle fluid selection process the substance R227ea was chosen. Its thermodynamic properties result in relatively high output power of the system [16,17] accompanied by good operating properties [18]. This fluid is neither explosive nor poisonous, has limited penetration properties and enables the whole power plant to operate above atmospheric pressure to prevent ambient air leaks into the system. The description of the detailed fluid selection process is out of the scope of this work.

The main assumption for the expander was simple design and low cost. The first model was a high rotational speed turbocompressor (about 30,000 rpm) which provided an easy solution for the power output as the compressor side was working as the break. The detailed description of this unit can be found in the publication of Magiera et al. [19].

In order to make more precise measurements it was necessary to apply a unit with an electric generator. Additionally, to avoid any leaks of the fluid, a concept of a hermetic turbogenerator was developed [20]. For vapours other than steam this solution yields a significant advantage over the traditional sealing system between the turbine part and the generator part (Fig. 1). In this case it prevents any losses of the expensive fluid. For explosive and toxic fluids the advantages are even more obvious. The generator itself is a three-phase permanent magnet unit. Its nominal power P_{NG} and efficiency η_g are 9.0 kW and 91% respectively which corresponds to the rotational speed of 3000 rpm.

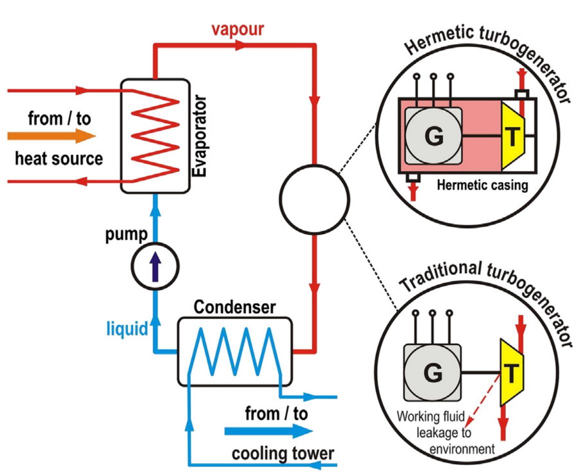

Fig. 1. Hermetic and traditional turbogenerators operating in a cycle.

3. Turbine design and analysis

3.1. One-dimensional design

To sustain the condition of simplicity and low cost it was decided to apply a one stage, low reaction turbine. This approach is common for low to medium power outputs [7,8,12,13,21–24]. Adding more stages would considerably increase the overall cost and would require more demanding design and manufacturing. Moreover, a significant turbine rotor overhang would occur which would cause additional problems with the rotor dynamics. The vapour leaving the rotor is used to cool down the generator. For this goal, the axial architecture is the most suitable.

The partial admission (11% of the circumference) had to be applied in order to increase the blade lengths and to reduce the secondary loss. The small partial admission is another reason for applying a single stage solution. In multistage machines it is necessary to make sealing between the consecutive stages. In case of a small partial admission the ratio of the mass flow rate flowing through the sealing and through the blades would become significant. This would diminish, at least partially, the benefit of the application of an additional stage. The unsupplied rotor part was covered on both sides. The flow direction in the stage is purely axial. The input parameters for the design are gathered in Table 1.

Additionally, the stage was designed in a way to provide a pure axial outflow with no fluid angular momentum at the outlet. This set is not the optimal one from the point of view of efficiency in a partial admission stage [25] but if the stage is expanded to full admission it shows the desired kinematics. The nozzle outflow angle was set to 14°. This value could be potentially optimized in future design taking into account, for instance, the conclusions from the work of Klonowicz et al. [12]. The whole stage kinematics is presented in Fig. 2.

The rotor wheel was not shrouded. The design size of the rotor clearance g_2 was equal to 0.5 mm. The stator blade row was not welded but slid into the casing so there was a very small gap g_1 between the stator blade tips and the casing (which could not be precisely measured). Its value was assumed to be 0.05 mm and was also introduced into the model.

To estimate the losses in the blade rows the Abramov–Filipov–Frolov total pressure loss model (Fig. 3) was applied [26,27]. The estimation of the disc friction loss P_{DF} was performed according to Eq. from (1) to (3) [28].

$$P_{DF} = B_{DF}\rho_2 u_{p2}^3 D_{p2}^2 \tag{1}$$

Table 1
The set of turbine input parameters.

Fluid	R227ea
Mass flow rate	1.64 kg/s
Inlet total pressure (absolute)	10.92 bar(a)
Inlet total temperature	58 °C
Outlet static pressure (absolute)	4.53 bar(a)
Rotational speed	3000 rpm
Reaction (pure impulse)	0
Admission size	0.11