Design and Performance of butane mixed-flow turbine used in combined power –refrigeration system

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***Abstract*— Power production and refrigeration are two problems of human communities which contribute in environmental degradation. Clean power can be produced by means of solar energy using organic Rankin cycle where refrigeration systems are power consumption and must be associated to clean energy source to conserve environment. In this paper a combined cycle for power production and refrigeration is presented. The system produces 30 Kw of power by a mixed flow turbine and 10 Kw of cold using solar energy and butane us working fluid. A CFD model is used to test the performance of the mixed flow turbine for a wide range of working conditions.**

***Keywords* — Power; solar energy; mixed flow turbine; ejector; butane; CFD.**

I. INTRODUCTION

Ozone depletion and global warming are major environmental concerns with serious implications for the future development of refrigeration-based industries and power production. The challenge is to use the renewable energy sources in power plant to eliminate pollutant emissions and non-toxic working fluid in refrigeration plant to conserve the ozone layer.

Organic Rankin cycles (ORC) are one of the available solutions for converting low grade heat source into electrical power. The benefit of ORC systems is the recovery of useful energy, often as electrical output, from low-energy sources such as the low-pressure steam associated with steam-driven turbines used for electricity generation [1–5].

Ejector refrigeration systems, which were very popular in the early 1930s, are also receiving renewed interest since they can be activated by low temperature thermal energy from renewable sources or thermal wastes thus reducing the use of fossil fuels or improving the efficiency of their usage. An ejector is a simple apparatus using the low pressure created by the accelerated stream of a primary (or motive) fluid to aspirate and compress a secondary (or suction) fluid [6].

Some researcher proposed ejector-refrigeration cycle and Rankin cycle (ORC) based combine power and ejector-refrigeration cycles to utilize the low temperature heat. In this type of combined cycles, the ejector refrigeration cycle and the power cycle share the same heat source.

The cycles could be divided into two kinds according to the configuration. In the first type, the ejector and power turbine are arranged in parallel. It was first proposed by Oliveira et al. [7 and 8].

The other type is that the ejector and the power turbine are arranged in series. The vapor form the power system acts as the primary flow and there is no generator in the refrigeration cycle. Dai et al. [9] proposed this kind of cycle, in which the high temperature and pressure organic gas expands in the turbine. Then the exhaust gas enters the ejector as primary flow (Fig. 1). This cycle is considered in this study, where a mixed flow turbine is designed and tested numerically.

II. CYCLE DESCRIPTION

In the combined cycle (Fig. 1), the liquid flow from the exit of the pump (state 1) is heated to super heat gas (state 2). Then the vapor enters the turbine and expands. The turbine exhaust gas (state 3) enters the ejector. In the ejector, the exhaust gas induces a low pressure region at the exit of the ejector nozzle and accelerates to the supersonic condition as it passes through the converging– diverging nozzle of the ejector. Then the secondary flow is entrained into the ejector from the evaporator (state 7) as secondary flow. Subsequently, these two streams are mixed in the mixing section. The mixed flow then undergoes a transverse shock and a pressure rise. And then it is compressed to state 4. In condenser, the compressed gas is cooled to saturated liquid (state 5). A part of the saturated liquid is pumped to high pressure in the pump. The other part of the saturated liquid expands in the valve (state 8) and then evaporates to saturated vapor (state 7). The solar energy with cylindro-parabolic concentrator can be used as high temperature heat source.

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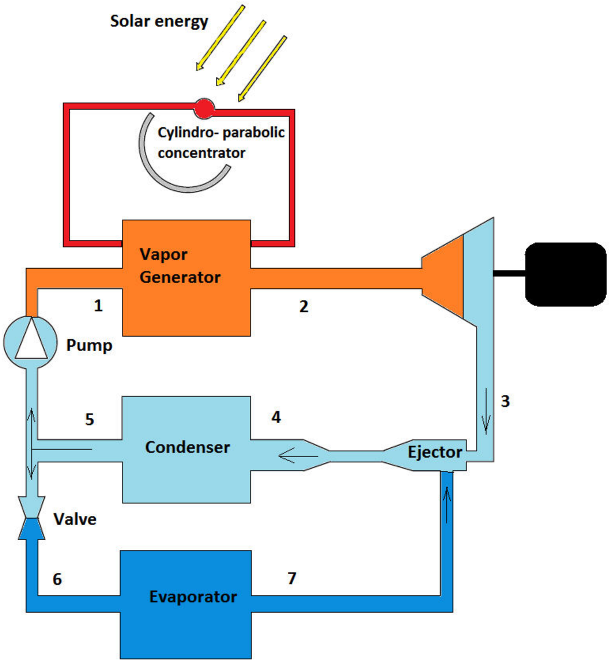

Fig. 1 Schematic diagram of the combined power and ejector-refrigeration cycle

III. Working Fluid

The only fluids of substitution without additional inconvenience both regarding the ozone layer and the greenhouse effect are non-halogenated fluids like ammonia (R717), propane (R290), butane (R600), isobutane (R600a), carbon dioxide (R744) and water (R818).

The main concern regarding the adoption of hydrocarbons as a refrigerant is their flammability. It should be remembered that millions of tonnes of hydrocarbons are used safely every year worldwide for cooking, heating, vehicle fuelling and aerosol propellants. In these industries, procedures and standards have been developed and adopted to ensure the safe use of the product.

Figure 2 shows the changes in saturated steam pressure and temperature for R134a, R22, R12, and mixtures of propane (R290)- butane R600, As shown in this figure, the saturated steam pressure curves for various proportions of mixtures are close to the refrigerant steam pressure curve R134a and R12, and the R290 curve is very close to the R22 curve. This indicates that these mixtures may have similar properties and could be used as a substitute for R134a and R22 refrigeration system. The butane curve indicate that it can be used when a law-grade energy sources is considered with the generation temperature ranging from 70 C to 95 C and low compression ratios.

IV. Radial Turbine

Radial inflow turbines have established their place in industrial applications, especially in the field of small turbomachinery because of their simplicity, reliability, low manufacturing cost, relatively high performance, easy installation and maintenance and fast response.

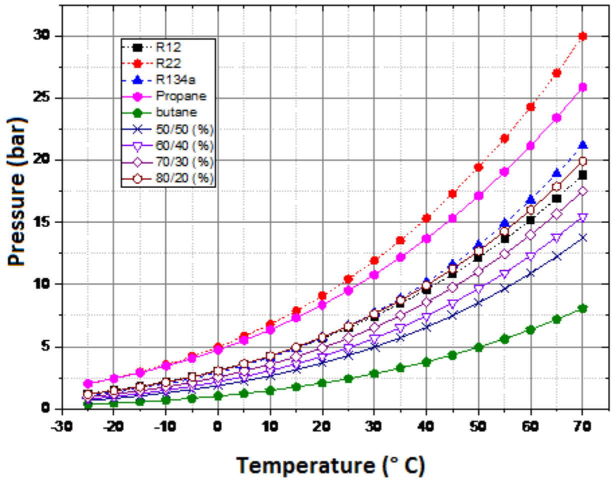

Fig. 2 Pressure and saturation temperature of Propane-Butane mixtures

When a low power output is required, the radial gas turbine has many advantages, both economically and aerodynamically, over the axial turbine. Where small sizes are concerned the relatively large clearances at the rotor tips have a smaller effect on turbine performance than in the axial flow machine. From the economic point of view the simpler fabrication of the radial rotor and nozzle assembly compared with the blade assembly for the axial turbine renders the turbine at a particular advantage when the machine is used as an accessory, for example, in turbochargers or starters. Furthermore, since it is essentially a low specific speed machine, it can cope with large overall pressure ratios in a single stage and this has advantages as an expander in liquid gas plant. Among the works that give a detailed approach to optimal sizing of a radial turbine one can quote that of Benson [10], and that of Ebaid et al. [11]. When large mass flow is required a mixed flow rotor with positive blade angle must be used as opposed to the usual value of zero for radial rotor. This means that the rotor inlet cannot be radial, but must be mixed, so that inlet streamlines in the meridional plane have radial and axial components [12].

V. Mixed Flow Turbine Design

A 30 KW mixed flow turbine with butane as working fluid is designed and tested numerically by the turbo-system tool of ANSYS Academic R19.2 software. A schematic sketch of the turbine is presented in figure 3. The main dimensions of the rotor are calculated by VISTA RTD according to the design conditions mentioned in table 1. The results are transferred to BLADEGEN to generate the turbine blade geometry and then transmitted to CFD MESH to create a tetrahedral mesh. The numerical simulations for design conditions and off design conditions are executed within CFX and the results are explored by the CFD POST tool.

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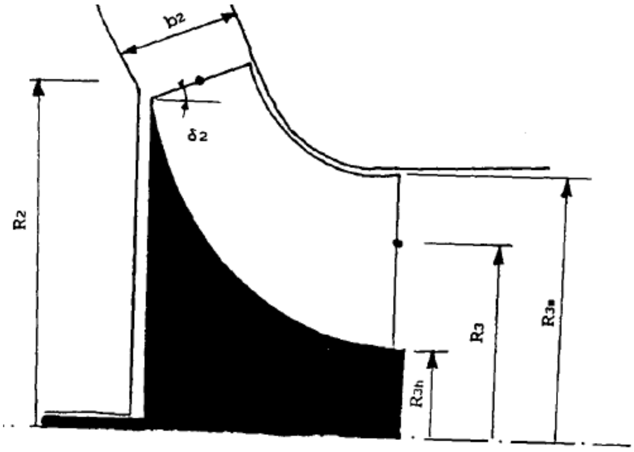

Fig. 3 Mixed flow rotor overall dimensions

TABLE 1
TURBINE DESIGN CONDITIONS

Rotational speed	80000 RPM
Total inlet temperature	430 K
Total inlet Pressure	14 bar
Pressure ratio	2.8
Mass flow rate	0.65 kg/s

TABLE 2
TURBINE OVERALL GEOMETRICAL CHARACTERISTICS

Rotor inlet mean diameter D_2	50.74 mm
Rotor inlet blade height b_2	5.54 mm
Rotor inlet blade angle β_{2b}	20.0 deg.
Rotor inlet cone angle δ	30.0 deg.
Exducer hub diameter D_{3H}	11.2 mm
Exducer shroud diameter D_{3S}	40.6 mm
Rotor exit blade angle β_{3b}	-50 deg.
Rotor axial length X_R	18.0 mm
Number of blades Z	14
Tip clearance e_x	0.40 mm

VI. RESULTS AND DISCUSSIONS

A three dimensional view of the resulted rotor is shown in figure 4. For simplicity and time reduction, only a blade passage is considered in the numerical simulation with the boundaries conditions shown in figure 5. The flow characteristics inside the blade passage are presented in figures 6 to 9. The pressure and temperature decrease smoothly from inlet to outlet and the absolute flow Mach number are high at the blade leading edge but don't exceed the sonic value. The blade loading is illustrated by the static pressure contours around the pressure and suction sides of the blade. The blade geometry generates the pressure gradient between the suction and pressure sides, resulting in the torque transmitted to the blade.The pressure distribution around the blade is interpreted in figure 8, presenting a positive pressure gradient at the blade suction side, which produces boundary layer separation and the flow recirculation near the leading edge as shown in figure 9.

The total to static pressure ratio influence on the turbine performance is shown in figure 10. A decrease in pressure ratio induces drop off in swallowing capacity of the turbine and the power. The turbine efficiency remains acceptable at the tested conditions. At the design conditions (Pr=2.8) the calculated mass flow rate and power are little higher than those assumed at the design stage. The turbine inlet temperature can be affected by the change in the cylindro-parabolic concentrator temperature, when this variation is in around of the design temperature; the turbine power and efficiency are not affected (fig. 11).

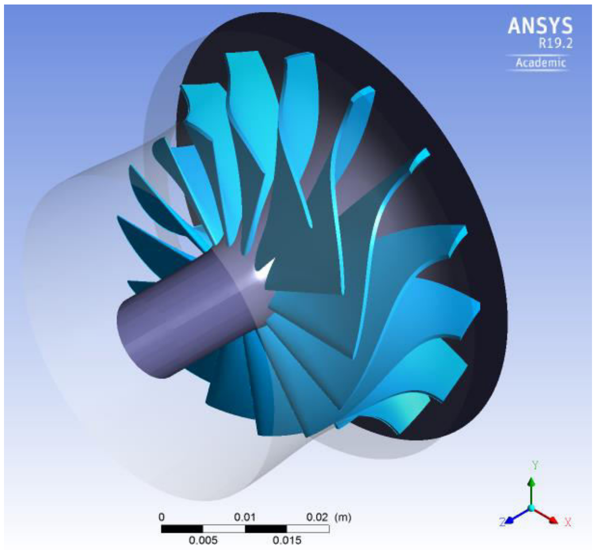

Fig. 4 Mixed flow turbine rotor

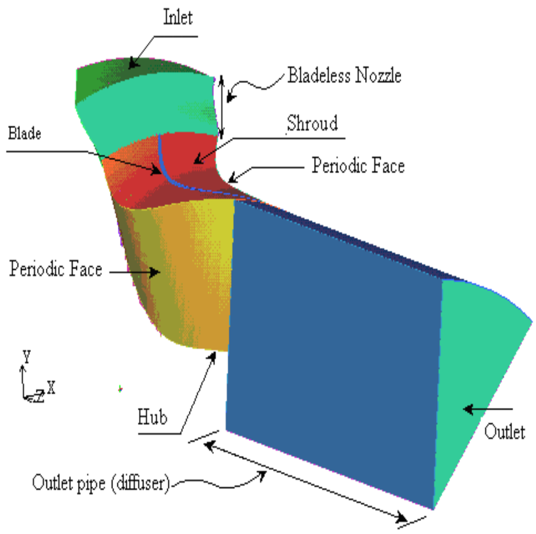

Fig. 5 Computational Flow Domain

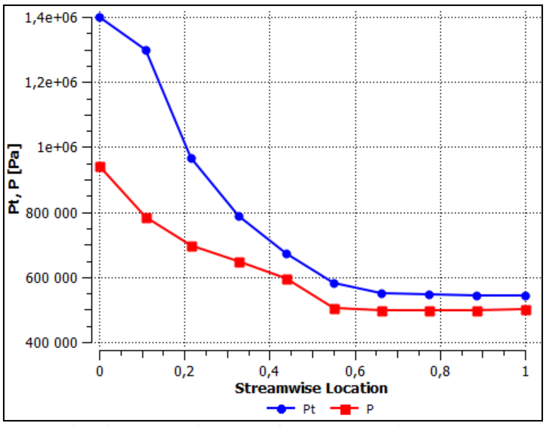

Fig. 6 Streamwise plot of total and static pressure

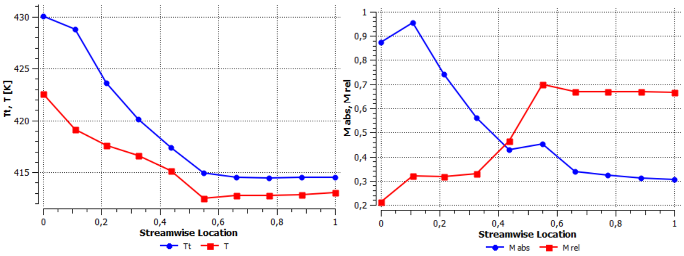

Fig. 7 Streamwise plot of total and static temperature (left) and absolute and relative Mach number right

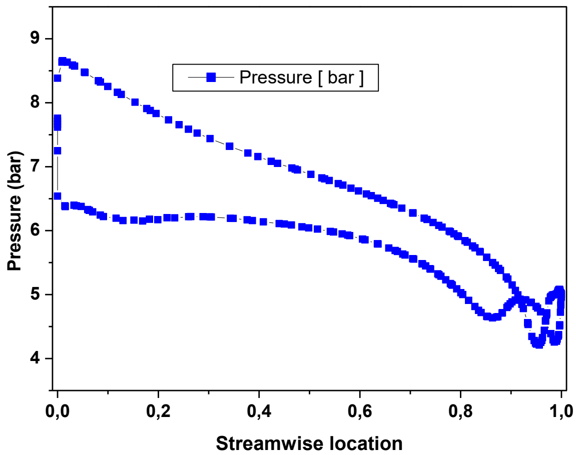

Fig. 8 Pressure variation at blade mid-span

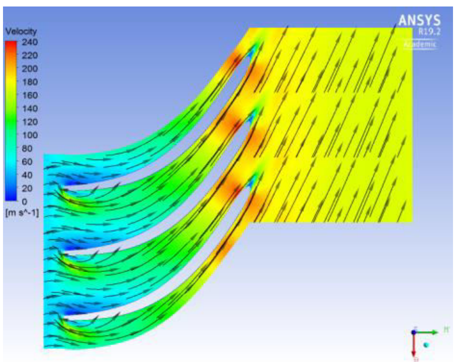

Fig. 9 Velocity Vectors at 50% Span

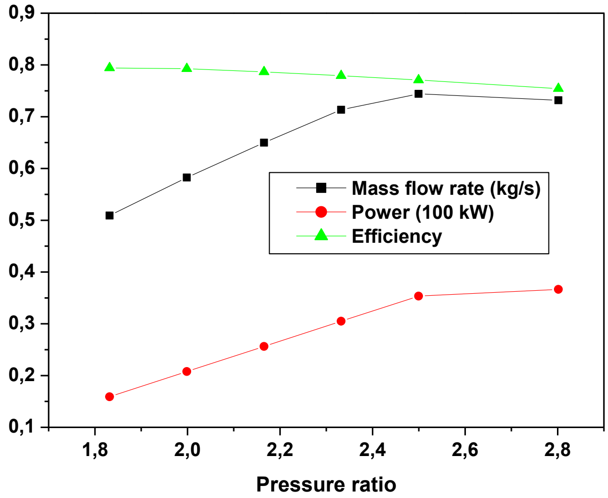

Fig. 10 Pressure ratio effects on turbine performance

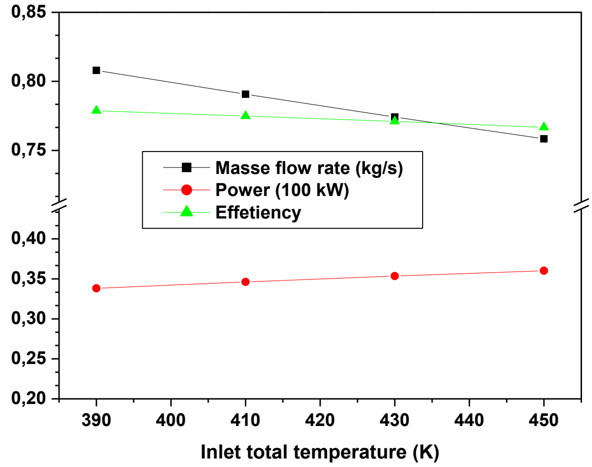

Fig. 11 Inlet total temperature effects on turbine performance

VII. CONCLUSION

A mixed flow turbine is designed and tested by using the turbo-system tool of ANSYS Academic R19.2 software. At the design, the butane is considered as working fluid and the solar energy as the heat source to conserve environment. The numerical results show that the turbine power can exceed 30 Kw with high efficiency. The high gas pressure at the turbine exit (5 bar) can be used in an ejector to produce cold after condensation and expansion.

REFERENCES

[1] Capata, R.; Toro, C. Feasibility analysis of a small-scale ORC energy recovery system for vehicular application. Energy Convers. Manag. 2014, 86, 1078–1090.
[2] Hyunjin, K.; Jeseung, Y. Design of a Scroll Expander for an ORC Applicable to a Passenger Car for Fuel Consumption Improvement; Department of Mechanical Engineering, University of Incheon: Incheon, Korea, 2013.
[3] Johnson, I.; Choate, W.T. Waste Heat Recovery: Technology and Opportunities in U.S. Industry; BCS, Incorporated, US Department of Energy: Washington, DC, USA, 2008.
[4] Qiu, G.; Liu, H.; Riffat, S. Expanders for micro-CHP systems with organic Rankine cycle. Appl. Therm. Eng. 2011, 31, 3301–3307.

ISSN 2356-5608

[5] Rettig, A.; Lagler, M.; Lamare, T.; Li, S.; Mahadea, V.; McCallion, S.; Chernushevich, J. Application of Organic Rankine Cycles (ORC). In Proceedings of the World Engineer's Convention, Geneva, Switzerland, 4–8 September 2011.

[6] Nicolas Galanis*, Mikhail Sorin; Ejector design and performance prediction; International Journal of Thermal Sciences 104 (2016) 315e329.

[7] Oliveira A, Afonso C, Matos J, Riffat S, Nguyen M, Doherty P. A combined heat and power system for buildings driven by solar energy and gas. Appl Therm Eng 2002;22:587–93.

[8] Wang J, Dai Y, Sun Z. A theoretical study on a novel combined power and ejector refrigeration cycle. Int J Refrig 2009;32:1186–94.

[9] Dai Y, Wang J, Gao L. Exergy analysis, parametric analysis and optimization for a novel combined power and ejector refrigeration cycle. Appl Therm Eng 2009;29:1983–90.

[10] R. S. Benson, "Computer aided design of radial gas turbines: a method for determining the overall rotor dimensions," *Computers and Fluids,* vol. 5, pp. 219-240, 1977.

[11] M. S. Y. Ebaid and Q. Z. Al-Hamdan, "Optimization techniques for designing an inward flow radial turbine rotor," Proceedings of the Institution of Mechanical Engineers Part a-Journal of Power and Energy, vol. 218, pp. 655-668, Dec 2004..

[12] M. Abidat, H. Chen, N.C. Baines, M.R. Firth, Design of a highly loaded mixed flow turbine, Proc. IMechE, Part A: J. Power and Energy 206 (1992) 95–107.

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