Solar low-pressure turbo-ejector Maisotsenko cycle-based power system for electricity, heating, cooling and distillation

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Abstract
The article describes the innovative solutions of power, heating and cooling generation utilizing low- or medium-grade heat sources. The proposed technology based on the well-known irreversible Brayton cycle and the revolutionary Maisotsenko cycle (M-cycle) operates at atmospheric or sub-atmospheric pressures. Such energetic systems are simple and reliable and utilize moisture-saturated air as a working fluid. The ejector replacing the mechanical compressor in the Brayton cycle system allows increasing the cycle work by three to five times at the constant airflow. At the same time, the utilized heat serves for simultaneous heating and cooling production that makes the system economically viable and environmentally friendly with the increased integral performance. For system’s performance improvement, the schematic and the cycle were upgraded allowing the off-the-shelf components to be employed and replace the electrically driven fan with fluidic jet-fan that served for energy saving of the innovative turbo-ejector system operation.

Keywords: ejector; Maisotsenko cycle; Brayton cycle; power system

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1 INTRODUCTION
The global increase in energy consumption calls urgent actions for simple and effective technical solutions to reduce environmental loads by utilizing affordable energy sources, especially renewable and secondary like, solar and geothermal energy, waste heat, other fuels (natural gas, green wastes etc.). Among well-known perspective power technologies, utilizing low- or middle-grade heat there are organic Rankine cycle (ORC), which operates as geothermal power plants mainly [1, 2], Brayton cycle and humid air turbine (HAT). It is fitting that the COP of ORC equals to 10–15% and depends on the heat source and working fluids employed, while HAT’s maximal efficiency is close to 50% and COP of the Brayton turbo-compression cycle with air humidification through Maisotsenko cycle (M-cycle) reaches 70% at certain parameters [3–7]. One of the main advantages of the HAT and Brayton technologies is an application of the humidified air as a working fluid. However, the compressor as a driving unit of the system consumes the major part of the cycle’s work, i.e. specific power output of the system decreases significantly. This paper describes an improved technology based on Brayton and M-cycles, where the steam-air ejector that does not consume any mechanical work produced in the turbine, replaces the conventional mechanical compressor.

2 HIGH-PRESSURE HEAT UTILIZING POWER CYCLES
The traditional methods of power generation are related to the most aggressive environmental pollutions. In spite of the high-grade heat required for its activation, the maximal achieved COP of such systems can hardly reach 50%. Due to the thermal stability of the materials employed, the temperature level is limited by 1650 K (Figure 1).

All those technologies are fossil fuels activated and related with toxic substances and greenhouse gasses emissions. Nuclear fuel has severe restriction, thus nuclear plants are considered as
'high-consequence facilities'. Hydropower occupies insignificant share (15–20% only) in the world’s energy mix due to the high depletion level of the rivers’ potential, including runoff control leading to irreversible changes of ecosystem, i.e. blue-green algae and wood breakdown contamination, erosion of fertile lands etc. Therefore, further spontaneous expansion of the traditional energy supply technologies can be disastrous and requires less harmful alternatives.

The cogeneration power plants become very popular nowadays, since they add a value of 25–50% to the single-effect system at the same fuel volume combustion. The water vapour Rankine cycle as well as the ORC is purposed as a bottom stage. The ORC becomes very promising for small-scale power generation with the advent of special thermopump [8]; however, its weak sides remain the same: ozone depletion potential (ODP) and global warming potential (GWP) restrictions for the working fluids, toxicity, risk of fire and explosion. Therefore, searching of the most efficient solutions is continued.

3 POWER SYSTEMS BASED ON THE M-CYCLE AND EJECTOR STEAM-AIR COMPRESSORS

The Brayton cycle combined with the M-cycle may become such an alternative to the ORC (Figure 2). Its special features are utilization of moisture-saturated heated air for power generation [3, 7] along with low-pressure in the system close to atmospheric, while turbine outlet pressure equals to 20–30 kPa. The system's working parameters make feasible to use less capacious and cheaper components, such as polymers. The combined cycle has two- to three-fold higher COP compared with the ORC; however, increasing of the airflow to generate 1 kWh of the useful energy requires a proportional increase of the electricity spent to run the air fans. The jet air fans are suggested instead of electrical fans to avoid additional electricity consumption by the system.

The steam-air ejector used in the Brayton cycle utilizes mid-grade heat instead of mechanical work that increases the cycle’s work by two to four times and lowers the mass-dimensional characteristics of the whole power system. Figures 3 and 4 shows schematics and cycle of the power system based on the Brayton cycle, the M-cycle and the steam-air ejector replacing mechanical compressor.

The additional heat is spent for water vapour generation required to run the steam-air ejector. The moisture is dropped out after the ejector, while water vapour condenses at 100°C temperature and the atmospheric pressure in the vapour generator of the ejector refrigerating system (ERS) attached to the power system. The ERS produces cold for air conditioning and refrigeration. The combustion chamber can be replaced by the gas-turbine
Where COP\textsubscript{integral} is overall systems COP; \( W_{\text{gas engine}} \) \( W_{\text{turbo ejector}} \) work produced by gas engine and turbo-ejector power system respectively, kW; \( Q_{\text{total}} \) total heat consumed, kW; \( \eta_{\text{Carnot}} \) COP of Carnot cycle; \( E_{\text{fuel}} \) \( E_{\text{distil}} \) \( E_{\text{ref}} \) \( E_{\text{heat}} \) heat exergy of fuel, distillation, refrigeration and heating; \( \epsilon_{\text{distil}} \) \( \epsilon_{\text{heat}} \) distillation and heating heat factor, \( T_{\text{cond}} \) \( T_{\text{eva}} \) condensation and evaporation temperature, K, respectively.

The primary energy flow is generated in the gas engine with efficiency of 45–55%. The exhaust heat goes to the low-pressure turbo-ejector M-cycle power system with efficiency of 30–35%. The heat from water condensation is utilized by the distillation system [9], with exergetic efficiency of 17%. The same heat flux serves for cooling production in the ERS with efficiency of 8% at \( T_{\text{eva}} = 285 \text{ K} \) and \( T_{\text{cond}} = 308 \text{ K} \). The remained heat at temperature of \( T = 353 \text{ K} \) is utilized for heating purposes with an efficiency of 13% (ambient temperature is taken as \( T_{\text{amb}} = 308 \text{ K} \)). When considered that exergy of the fresh water’s distilled vapour equals to exergy of the condensation heat at the ambient pressure and this heat is finally utilized to produce cooling, the exergy COP value can exceed 1. The maximal fuel cycle temperature is limited by the real cycle parameters and the temperature value of 1650 K [10]. The exergy COP of the heat is 81%, while an integral exergy COP of the whole system equals to 93–100%.

### 4 ENERGY PERFORMANCE COMPARISON OF TURBINE ACTUATED POWER SYSTEMS WITH MECHANICAL COMPRESSOR AND STEAM-AIR EJECTOR

For a comparative analysis of various heat utilizing power systems the calculation of the power cycle proposed by Prof. V. S. Maisotsenko was performed. All calculations using variable turbine and compressor COP from 0.75 to 0.95 were carried out. Two separate ways of calculation were considered: calculation using engineering method and using computer simulation of the system. All thermophysical properties of humid air were represented on additive basis using partial pressure of components:

\[
P = \sum_{i=1}^{n} p_i = p_1 + p_2 + \cdots + p_n
\]

where \( P \) is a full pressure of gas mixture, kPa, and \( p_1 - p_n \) is a pressure of a component, kPa.

Enthalpies of humid air were calculated using

\[
h_{\text{humid air}} = \frac{h_{\text{steam}}d + h_{\text{dry air}}}{d + 1}
\]
where $h_{\text{humid air}}$ is an enthalpy of humid air, kJ kg$^{-1}$; $h_{\text{steam}}$ is water steam enthalpy, kJ kg$^{-1}$; $d$ is absolute air humidity, kg kg$^{-1}$; $h_{\text{dry air}}$ is a dry air enthalpy, kJ kg$^{-1}$.

The COP of the system was calculated using the following equation:

$$
\text{COP}_{SLPTC} = \frac{W_{\text{turb}} \eta_{\text{turb}} - W_{\text{comp}} \eta_{\text{comp}}}{Q_{\text{comb}}}
$$

(4)

where $W_{\text{turb}}$ is work produced in the turbine, kW; $\eta_{\text{turb}}$ is COP of the turbine; $W_{\text{comp}}$ is a work consumed by the compressor, kW; $\eta_{\text{comp}}$ is COP of the compressor; $Q_{\text{comb}}$ is a heat consumed in the combustion chamber, kW.

Similar calculations were made for the heat utilizing power system with the steam—air ejector as the compressor, where the entire turbine work is considered as a useful cycle work. However, the ejector operation requires a significant amount of low-grade heat, which is then returned to the system. In this case, efficiency mapping is performed correctly, since heat flows are adjusted to a single temperature level through the heat exergy.

The COP of combined power cycle:

$$
\text{COP}_{SLPTE} = \frac{W_{\text{turb}} \eta_{\text{turb}} + E_{\text{heat}}}{Q_{\text{comb}} + Q_{\text{steam}} (\eta_{\text{Carnot, steam}}/\eta_{\text{Carnot, comb}})}
$$

(5)

where

$$
\eta_{\text{Carnot, steam}} = \frac{T_{\text{gen}} - T_{\text{amb}}}{T_{\text{gen}}}
$$

$$
\eta_{\text{Carnot, comb}} = \frac{T_{\text{comb}} - T_{\text{amb}}}{T_{\text{comb}}}
$$

(6)

where $W_{\text{turb}}$ is work produced in the turbine, kW; $\eta_{\text{turb}}$ is a COP of the turbine; $E_{\text{heat}}$ exergy of utilized heat, kW; $Q_{\text{comb}}$ heat consumed in the combustion chamber, kW; $Q_{\text{steam}}$ heat consumed for steam generation, kW; $\eta_{\text{Carnot, steam}}$ COP of Carnot cycle at steam generation and ambient temperature, $\eta_{\text{Carnot, comb}}$ COP of Carnot cycle at temperature output of combustion chamber and ambient temperature, $T_{\text{gen}}$ steam generation temperature, K; $T_{\text{comb}}$ temperature of humid air out of combustion chamber, K; $T_{\text{amb}}$ is ambient air temperature, K.

Enthalpy after the ejector:

$$
h_{\text{out}} = h_{\text{steam}} X_c + h_{\text{air}} (1 - X_c)
$$

(6)

Entrainment ratio of the ejector was calculated using empirical data and verified using CFD modelling of the ejector flow part.

The empirical formula (Eq. 7) for the entrainment ratio is described in [11]:

$$
U = \frac{K_1 (a_{\text{gen, crit}}/a_{\text{cond, crit}}) \lambda_{\text{gen, A}} - K_3 \lambda_{\text{cond, C}}}{K_4 \lambda_{\text{cond, C}} - K_2 (a_{\text{air, eva, crit}}/a_{\text{cond, crit}}) \lambda_{\text{eva, B}}}
$$

(7)

where $K_1, K_2, K_3, K_4$ are integrated velocity coefficients; $a$ local sound speed, m s$^{-1}$; $\lambda$ reduced isentropic speed; crit critical parameter; A,B,C nozzle outlet, cylindrical mixing chamber inlet and cylindrical mixing chamber outlet cross-section area.

The basic diagram of the power system with mechanical compressor was described in [3]. This diagram was calculated using the process simulation tools. The results of the process simulation are shown in Figure 5. Simulation parameters: dry air flow $G_{\text{air, dry}} = 0.036$ kg/s, ambient air parameters: temperature $t_{\text{amb}} = 23^\circ \text{C}$, pressure $P_{\text{amb}} = 101.325$ kPa, humidity $AH = 0.009$ kg/G$\text{air, dry}$; air temperature out of HMX: $t_{\text{hmx}} = 71^\circ \text{C}$, $AH_{\text{hmx}} = 0.0126$ kg/G$\text{air, dry}$; air temperature out of combustion chamber $t_{\text{comb}} = 240^\circ \text{C}$, expansion pressure $P_{\text{exp}} = 30$ kPa, COP$_{\text{turbine}} = 0.85$, COP$_{\text{comp}} = 0.85$. Optimal air temperature out of combustion chamber and its humidity requires additional study. Based on simulation and engineering calculation data at various expansion pressures $P_{\text{exp}} = 20–65$ kPa, and turbine efficiencies COP$_{\text{turbine}} = 0.75–0.95$ and COP$_{\text{comp}} = 0.75–0.95$, a schematic chart of system’s COP was presented in Figure 6. An optimal turbine expansion pressure at the lower turbine and compressor COP lies in a range of 40–50 kPa, while at the higher turbine and compressor COP an optimal turbine expansion pressure is 20–30 kPa. The system’s COP equals to 0.2 at $P_{\text{exp}} = 30$ kPa, COP$_{\text{turbine}} = 0.85$, COP$_{\text{comp}} = 0.85$. The effective

![Figure 5. The process simulation of the low-pressure turbo-compression M-cycle power system.](https://academic.oup.com/ijlct/article-abstract/10/2/157/785457/160)
cycle work values are presented in Table 1. The difference between theoretical calculation and process simulation results is 5–30%.

The similar calculations were made for low-pressure turbo-ejector M-Cycle power system (Figure 7). The system’s COP is 0.3 at the same parameters that is 1.5 times higher compared
with the turbo-compression system efficiency. The results shown in Figures 7 and 8 and Table 2 prove the optimal expansion pressure in the range of 20–35 kPa and do not depend on the turbine’s efficiency.

Double-stage ejector improves systems COP, even though ejector’s efficiency goes down (Figure 9). It allows decreasing expansion pressure to a level of 14–16 kPa that increases the turbine’s work (Table 3).

The M-cycle-based power systems’ functioning faces encounters several specific challenges related to the Maisotsenko heat-and-mass exchanger (HMX) design. The major are deformation of the splitting baffles between the adjacent channels of HMX up to its breakup and significant electricity consumption for electrical fans operation required for air circulation inside HMX channels. These issues could be eliminated by changes to the system’s design and also with the replacement of the main fans for fluidic jet fans, which utilizes high-pressure water vapour instead of electrical power.

Figure 10 presents practical schematics, where the main ejector is placed between the turbine and HMX in order to equalize the vapour-air mixture’s pressures inside the HMX. Such optimization allows employing the off-the-shelf HMXs, produced by Coolerado Inc., which have soft splitting baffles and do not splay out provided the pressure in channels is equalized. Application of the supplemental ejector as a jet-fan device serves for air-circulation avoiding electrical fans thus making the whole device completely thermally activated.

CFD modelling presented in Figure 11 was performed for steam-air ejector to optimize its geometry for highest efficiency.
in the low-pressure turbo-ejector M-cycle power systems. According to CFD data, the ejector device is designed and manufactured.

Obviously, an economic effect of the heat activated power systems depends on the price of heat available in a certain region or certain industry. The areas of maximal profitability of every
system application are described in Figures 12 and 13. In Figure 12 heat prices were taken equal to electricity prices, while in Figure 13 heat prices compose half of the electricity price. The same diagrams can be calculated for any heat and electricity prices. All calculations were performed for dry air flow $G_{\text{air,dry}} = 0.036 \text{ kg/s}$.

4 CONCLUSIONS

The replacement of the mechanical compressor with the steam-air ejector results in 2–4 times increase of the power generation at the same air-flow rate. Along with that, the system’s capital costs decreases by 15–20% at the expense of the system’s enhanced reliability and simplicity. The electricity used for running the fans for M-cycle decreases two times per power produced unit. The multipurpose energy systems utilize almost all the heat or fuel exergy for power, fresh water, heating and cooling supply. It makes sense to use turbo-compressor system design if electricity price is low and heat price is high. Otherwise, turbo-ejector design shall be preferred. If simultaneously with power generation, low-temperature cooling is required, the low-pressure turbo-ejector M-Cycle power system is more beneficial despite of the cost of the heat available. Displacement of the main ejector between the turbine exit and HMX entrance avoids a splay out of the standard HMX’s air channels, while application of the additional fluidic jet-fan serves for complete heat activation of the power system in the turbo-ejector M-cycle based power system.

REFERENCES

[1] Quoilin S, Lemort V. Technological and economical survey of organic Rankine cycle systems. In: 5th European Conference Economics and Management of Energy in Industry, 14–17 April 2009, Vilamoura, Portugal.
[2] Hettiarachchi HDM, Golubovic M, Worek WM, et al. Optimum design criteria for an organic Rankine cycle using low-temperature geothermal heat sources. Energy 2007;32:1698–706.
[3] Wicker K. Life below the wet bulb: the Maisotsenko cycle. Power Magazine, Platts/McGraw Hill, 2003. http://www.coolerado.com (14 June 2014, date last accessed).
[4] Gillan L, Maisotsenko V. Maisotsenko open cycle used for gas turbine power generation. In: International Joint Power Generation Conference 2003, ASME Turbo Expo 2003, Atlanta, Georgia, USA, 16–19 June 2003.
[5] Khalatov AA, Karp IN, Isakov BV. Maisotsenko thermodynamic cycle and prospects of its application in Ukraine. Messenger Natl Acad Sci Ukraine 2013;2:1–10.
[6] Gadalla M, Alsharif A, Dincer I. Energy and exergy analyses of Maisotsenko cycle for sustainable development. In: The 5th International Conference on Energy Sustainability ES2011, 7–10 August 2011, Washington, DC, USA.
[7] Reyzin I. Evaluation of the Maisotsenko power cycle thermodynamic efficiency. Int J Energy A Clean Environ 2011;12:129–39.
[8] Buyadgie D, Buyadgie O, Drakhnia O, et al. Prototype ejector refrigeration system with thermopump testing in air-conditioning mode. In: The 5th International Conference on Cryogenics and Refrigeration, ICCR2013, 6–9 April 2013, Hangzhou, China.
[9] Buyadgie D, Buyadgie O, Drakhnia O, et al. Combined heat, cold and fresh water supply using binary fluid ejector refrigerating system. In: The 23rd International Congress of Refrigeration, 21–26 August 2011, Prague, Czech Republic.
[10] Szargut J, Petela R. Exergy. Energia Publisher, 1968.
[11] Sokolov EYa, Zinger NM. Jet Devices, 3rd edn. Energoatomizdat Publishing house, 1989.