Photovoltaic Pumped Storage Sytem for Small and Large Scale Applications

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Abstract—**The photovoltaic market has been growing at a fast rate in the last two decades, and in 2014 reached a global capacity of about 180GW. Due to the increasing penetration level, large-scale energy storage systems that can compensate for its fluctuations become increasingly important. In case of off-grid systems, suitable energy storage is naturally a necessity. In areas with favorable geographical properties, pumped storage has an excellent potential to be applied. The behavior and the performance of the PV pumped storage systems for small and large scale applications are analyzed in this paper. The stochastic character of PV energy is into account. The system is tested and validated on a RTDS (Real Time Digital Simulator) platform.**

Keywords-photovoltaic energy; pumped storage; efficiency; control

I. INTRODUCTION

The photovoltaic sector experienced a strong growth of the installed power in the last years. In 2014, at least 40 GWp of PV systems were installed globally and is reported that the installed cumulative worldwide capacity was 178 GWp by the end of the year [1].

The price of the PV systems is decreasing. Solar Power Europe estimates a decrease of around 75% in the next 10 years [1]. Moreover, new PV technologies have been developed, and even energy conversion efficiency of 40% have been reported. The PV integration level will increase even more considering these premises.

In grid-connected applications, stability problems, equipment overloading, power quality degradation, increased power losses, and even blackouts can take place due to grid overloading. Appropriate energy storage technologies can overcome the problem of an ineffective grid penetration. Spinning reserve, frequency and voltage regulation, peak curtailment and levelling are just a few important functions that can be met by storage [2]-[3]. The same phenomena can take place in the case of microgrids, where the effects can be more pronounced.

Among different storage technologies available, pumped storage can significantly contribute to renewable energy integration, including PV. This storage technology has gained attention especially in the countries with high potential of hydropower. High capacity, along with long period of storage, are the most attractive features of this technology [4]-[8].

The analysis of PV pumped storage systems is important, as this kind of systems can facilitate the increase of PV energy integration. Studies regarding the wind integration increase by means of pumped storage can be found in the literature, but photovoltaic-based pumped storage system were found only for very small applications [9]-[13].

The behavior and the performance of a PV - pumped storage system for small and large scale applications is analyzed in this paper. The variable operating conditions of such a system due to the fluctuating solar irradiance, and consequently the power provided by the PV system are taken into account. The efficiency of the PV pumped storage system has been established for both small and large scale applications.

II. PV-PUMPED STORAGE SYSTEM DESCRIPTION

This paper considers a standalone system, which consists of a PV plant, and a pumped storage system. During the daytime, the energy from the PV plant is used to satisfy the load, while the extra energy will be used by the pumped storage system. During the night, the pumped storage system assures the energy for the consumers. In addition, the pumped storage system can provide the energy for the peak periods of the daytime or in case of low irradiance conditions.

The block scheme of the system is presented in Fig. 1, and a short discussion considering the most important aspects of each of the system's elements is given in the following.

A. PV system

The considered installed power is 16.5 kWp for the small scale the PV system, while for the large scale system 2.5 MWp were considered. Panels with 250 Wp were chosen for both systems. Their technical characteristics are given in Table 1. In order to reach the 16.5 kWp power, 65 panels were needed, while 10575 panels were needed for the large scale system.

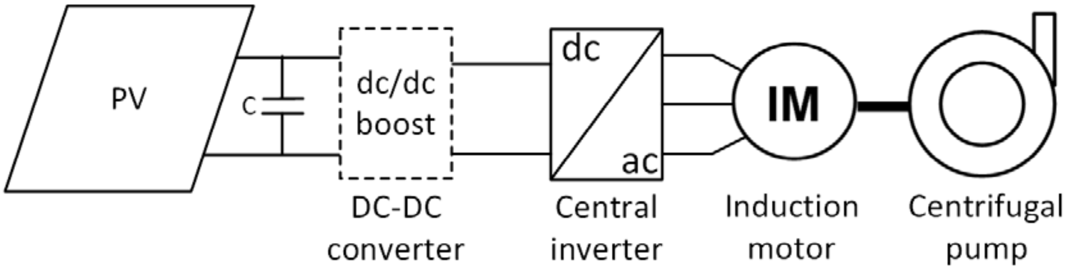

Fig. 1. Standalone PV Pumped Storage system

The losses in the PV systems were neglected, as the focus of this paper is on the performance of the pumped storage system.

B. Power electronics interface

A PV plant layout with central inverter is considered for both small and large scale systems. A dc-dc boost converter is included for assuring the DC voltage for small irradiation conditions in the case of the small scale PV system. At MW the boost is no longer necessary. This is the reason for which the dc-dc boost converter was represented with dash line.

C. Motor-pump group

The efficiency of the pumped storage system is around 80% (for MW power order). The motor-pump group have a great impact on the system efficiency. The induction motors have an efficiency of about 90% at MW power level, and lower at kW power level [14], but their efficiency decreases as the load increase.

TABLE I. PV PANEL CHARACTERISTICS

Parameter	Value	SI
Watt peak	250	W
V_{oc}	37.2	V
I_{sc}	8.87	A
V_{mpp}	30.1	V
I_{mpp}	8.3	A

TABLE II. INDUCTION MACHINES PARAMETERS

Parameters	Power order		SI
	MW	kW	
Nominal power	2.300	15	kW
Nominal voltage	690	400	V
Nominal current	2174	28	A
Mechanical speed	1410	1460	rpm
Frequency	50	50	Hz
Power factor	0.887	0.8	-
No of pole pairs	2	2	-

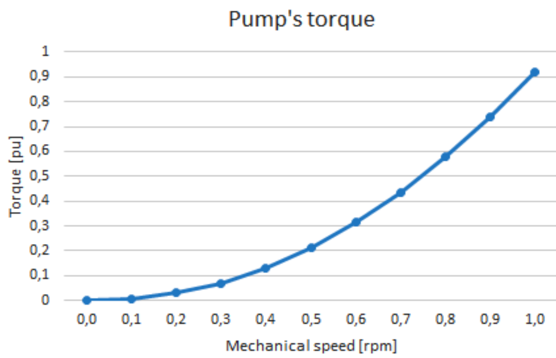

Fig. 2. Pump's torque

On the other hand, in pumped storage applications, their load torque is dependent on the square of the speed, while the pump's power is dependent on the third power of the speed [15]. This means that the loading of the motor is lighter at lower speeds than at nominal speed. Therefore, variable speed operation with different loading levels can lead to an efficiency increase of the system. The pump's torque is presented in Fig. 2.

In [16] the authors established the motor's most efficient operating points in pumped storage systems, and the analysis concluded that the most efficient operating zone of the induction motors turns to be between 25 and 35 Hz of the supply voltage for MW power order (Fig. 3), and between 35 Hz and 45 Hz for kW power order (Fig. 4). The correlation between the motor-pump group's mechanical speed and the frequency of the supply voltage by means of motor's number of poles has been considered.

The impact of the motor efficiency on the pumped storage system can also be seen for the system presented in this paper, because the motor has to operate under 50 Hz for reduced levels of solar irradiance. This will be presented in the System evaluation and results section of this paper. It is noted that the same type of system as in [16] is used in the current study.

The classical control method, V/Hz, has been implemented by means of power electronics for both small and large scale PV pumped storage systems. This facilitates the variable speed operation, which is a very important aspect for pumped storage systems. In addition, both the stator voltage and frequency of the motor are simultaneously controlled.

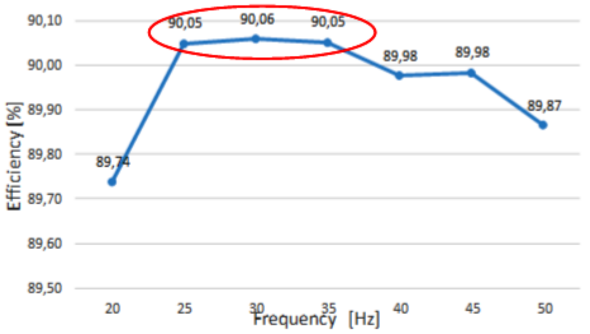

Fig. 3. Induction machine most efficinet operating points in pumped storage applications for MW power order

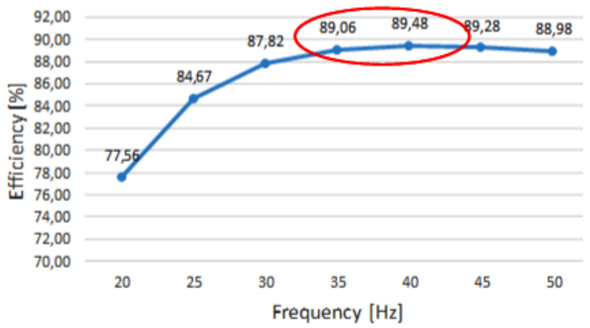

Fig. 4. Induction machine most efficinet operating points in pumped storage applications for kW power order

The induction machines parameters used for the pumped storage systems are presented in Table II.

A specific strategy for the efficiency increase has not been considered in this paper. In [17] the authors proposed a motor control based on a cubic polynomial which can lead to up to 10% efficiency improvement, but in this paper the focus is on the natural behaviour of the system for small and large scale.

III. System Evaluation and Results

The PV pumped storage system for both small and large scale is tested on a RTDS (Real Time Digital Simulator) platform. This facilitates the MW power study at laboratory level. The RTDS is capable of simulating complex systems including power converters in real time and it is a well-established tool for power system studies. It accomplishes the necessities for the calculation effort and presents high accuracy.

The RSCAD software was used for interfacing the RTDS Simulator hardware. The standalone PV-pumped storage system has been modeled in the RSCAD/Draft module. The system monitor and the variation of the operating conditions in real time were performed by means of the RSCAD/RunTime module. The RTDS platform can be seen in Fig. 5.

The PV-pumped storage system is analyzed in the case of different levels of solar irradiance, from 200 W/m^2 to 1000 W/m^2, to take into account the stochastic nature of PV energy.

The DC available power of the PV system is lower as the solar irradiance decreases. Therefore, the system's frequency needs to be decreased below 50 Hz, thus the motor for pump's driving will operate on different points on the mechanical characteristic, Fig. 6. The mechanical speed and the load torque of the motor for each solar irradiance level are presented in Fig. 7, respectively in Fig. 8.

The pumping power depending on the solar irradiance is presented in Fig. 9. The system's efficiency for both the small and the high power order can be seen in Fig. 10. The efficiency is 89.84% for the large scale system, and 81.28 % for the small scale system. It has to be noted that these efficiency values do not include the pump's efficiency, which is very sensitive at low speeds, and the PV panels energy conversion factor.

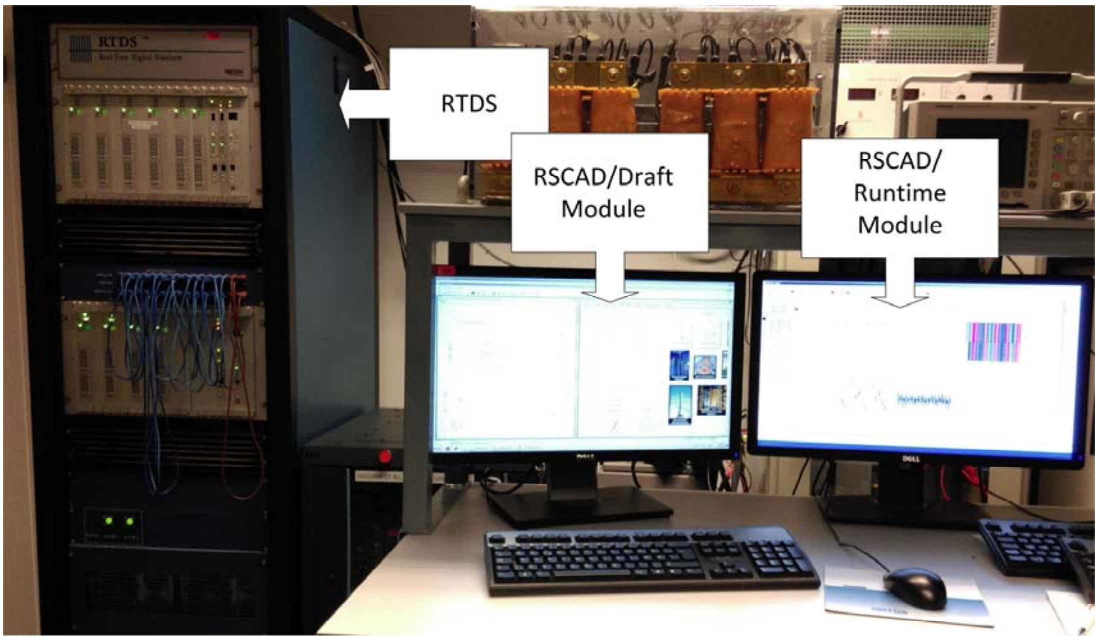

Fig. 5. Real time digital simulator (RTDS)

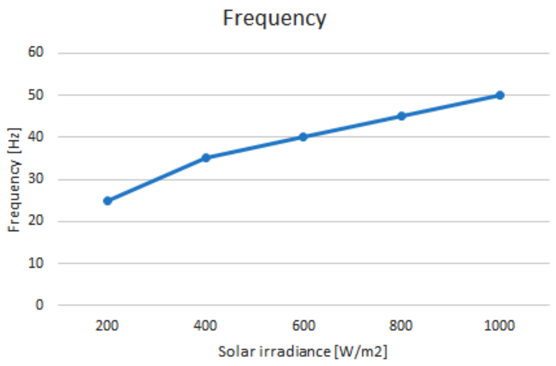

Fig. 6. Pump’s motor operating frequencies

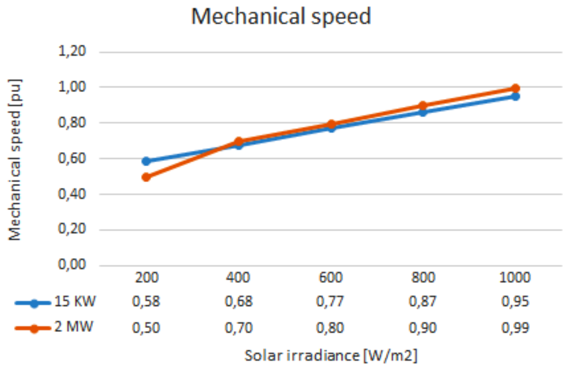

Fig. 7. Motor’s mechanical speed

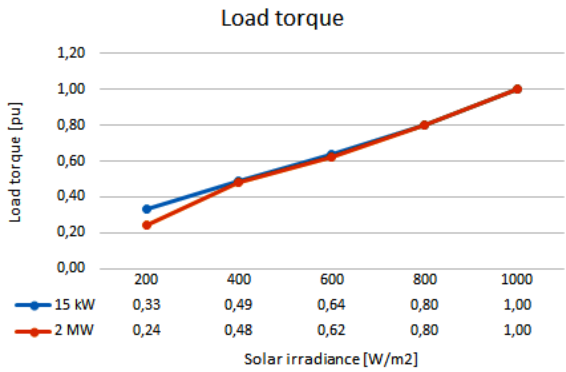

Fig. 8. Motor’s load torque

The maximum efficiency reached by the pumped storage system is at 600 W/m^2 irradiance.

The increased efficiency for 400, 600, and 800 W/m^2 obtained with both control methods considered is influenced by the motor’s efficiency. As it was previously stated, the efficiency of the motor is affected by its loading. In pumped storage applications, the load of the motor is lighter at lower speeds and frequencies than at nominal speed.

Therefore, the motor has a better performance at lower speeds. This happens also in the PV-pumped storage system, where for lower level of irradiance, the system has to operate at lower frequencies.

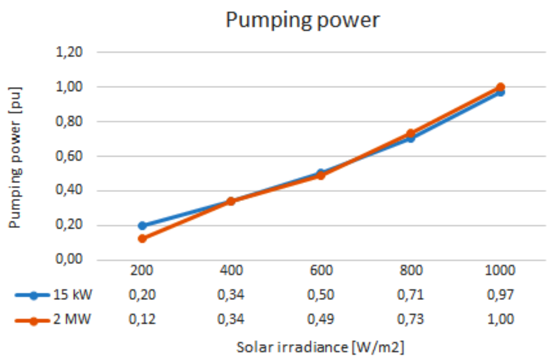

Fig. 9. The available power for pumping for the PV pumped storage system in the case of 15 kW and 2 MW power order

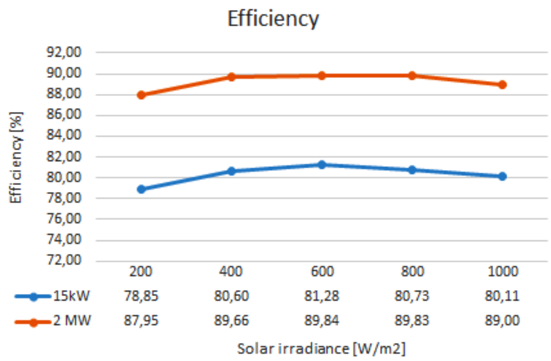

Fig. 10. The efficiency of the PV pumped storage system in the case of 15 kW and 2 MW power order

IV. Conclusion

Appropriate energy storage technologies can overcome the problem of an ineffective grid penetration. A potential candidate for PV integration is pumped storage considering its valuable characteristics (high efficiency, high quantities of energy which can be stored).

This paper demonstrates the good operability of a PV pumped storage system. The performance of the pumped storage system for various levels of solar irradiance has been established. The pumped storage system’s efficiency is up to 90% for the large scale system, and up to 82% for the small scale system. The PV energy conversion factor and the pump’s efficiency have not been taken into account at this point.

However, further research considering a specific site situation based on hourly energy profile of a PV system based on the solar irradiance is necessary. In addition, realistic load profiles can be included.

Acknowledgment

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Electrical Power Generation with Low-Temperature Organic Rankine Cycle Machines

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***Abstract* — By 2050, global energy consumption is projected to grow by 71%. At the same time, energy-related carbon dioxide emissions are expected to rise by more than 40%. In this context, waste and renewable energy sources may represent alternatives to help reduce fossil primary energy consumption. This paper focuses on the theoretical design and performances of a 50 kW laboratory beta-prototype Organic Rankine Cycle (ORC) machine using low-enthalpy heat as industrial waste or deep geothermal renewable energy at temperatures varying between 85 and 125°C. The experimental study shows that the electrical power generated and the overall net conversion efficiency of the ORC machine depend on parameters as the inlet temperatures of the waste (or renewable) heat and cooling fluids, as well as on the control strategy under variable operating conditions.**

*Keywords—**Electrical power generation; Industrial waste heat recovery; Energy efficiency***

I. INTRODUCTION

The use of fossil fuels (coal, oil, natural gas) in industry, as iron and steel, pulp and paper, and chemical, contributes to the emission of pollutants that are damaging to the environment. On the other hand, up to 50% of the consumed energy is lost into the environment at relatively low temperatures (< 370°C) in the form of stack gases, vapor or liquid effluents. At such temperatures, the wasted heat cannot efficiently be used to produce electricity using steam-based power generation such as conventional Diesel, Stirling or Rankine cycles [1]. Consequently, different energy conversion techniques are required to efficiently use low-grade "free" waste heat resources for power generation. Among these alternatives, Organic Rankine Cycle-based (ORC) machines, similar to basic Rankine power plants, do not use water, but rather vaporize high-molecular-mass organic fluids with boiling points below that of water [2]. ORC machines can use various types of low-grade industrial waste heat or renewable (solar, geothermal, biomass,) energy sources available at temperatures between 85°C and 350°C.

II. EXPERIMENTAL SET-UP

A pre-commercial small footprint, skid-mounted prototype based on the ORC cycle was designed [2], built [3] and installed on a test bench (Fig. 1) [4]. It can generate up to 50 kW of electrical power by recovering heat from waste heat in liquid form at temperatures ranging from 85°C to 125°C.

As shown in Fig. 1, the prototype includes a preheater (P-H) and an evaporator (EV) intended for preheating and vaporizing the organic fluid, respectively, a single-stage twin screw expander directly coupled (i.e. without gearbox) to an asynchronous alternator (generator), and a stainless steel plate condenser (CD). The induction alternator generates electric power at 460 Volts, which is transferred to the local electrical grid via a 25 kV/600V electrical transformer. The liquid mixture (glycol/water 50%) pump (P-C) circulates the cooling fluid within a closed-loop to be cooled by an ambient air-cooled liquid cooler. The organic fluid pump P-OF returns the organic fluid liquid accumulated in the liquid receiver LR into the preheater. A 700 kW electric boiler simulates the thermal waste heat source, and the pump (P-W) circulates the waste heat thermal carrier (water) in a closed-loop.

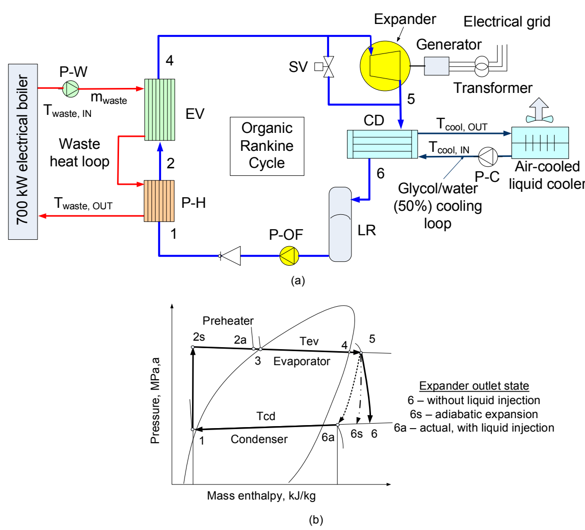

Fig.1. (a) Schematic of the experimental set-up; (b) typical thermodynamic cycle in the p-h diagram; SV - solenoid valve

The working fluid is HFC-245fa, a low-pressure, high-temperature refrigerant, with relatively high enthalpy of vaporization and density, high chemical stability, safe in regards to explosion, flammability, and health, and available at relatively low prices. Moreover, HFC-245fa becomes superheated during the expansion, thus avoiding condensation of the organic fluid as it passes through the expander [5]. Its Ozone Depletion Potential is zero, but its Global Warming Potential is relatively high (1900).

The experimental bench was comprehensively instrumented with thermocouples, power and pressure transducers, flow meters for the working fluid, and both heat source and sink thermal carriers. A data transmission system and associated analysis software were set up to monitor/control system operation. All parameters were scanned at 15 second intervals, then averaged and saved every minute, to help determine the cycle's instantaneous and overall thermodynamic performance.

III. DESIGN OF THERMODYNAMIC CYCLE

The low-pressure organic fluid leaving the condenser as a saturated or sub-cooled liquid (state 1) accumulates inside the receiver in equilibrium with its vapour phase (Fig.1). Then it enters the working fluid feed pump (P-OF) where its pressure is adiabatically raised to the saturation (evaporating) pressure (state 2s), prior to entering the preheater/evaporator. The multistage feed pump is driven by a variable frequency drive in order to supply the flow rate required (11-12 kg/s) to achieve a superheating amount below 5°C at the evaporator outlet. Also, up to between 5% of the organic fluid flow rate is injected into the expander for cooling and lubrication purposes.

The waste heat fluid carrier (water) enters the preheater/evaporator where it transfers heat to the organic fluid during the preheating, evaporation and superheating processes. At the evaporator outlet, 20% of the water total flow rate (7 to 12 kg/s) enters the preheater to preheat the organic fluid from 2s up to (near) the saturated state 2a prior entering the evaporator where it will vaporise and superheat at a constant pressure (process 2a-3-4-5). The preheater/evaporator thermal power recovered from the waste heat source is given by:

$$\dot{Q}_{preheat+evap} = \dot{m}_{org}(h_5 - h_{2s}) \tag{1}$$

Where $\dot{m}_{org}$ is the organic fluid mass flow rate (kg/s); h_5 and h_{2s} - organic fluid specific enthalpy leaving and entering the preheater/evaporator, respectively (kJ/kg).

The high-pressure saturated (state 4) or superheated vapour (state 5) leaving the evaporator enters the twin screw expander. It accelerates the 50 kW (460 Volts) asynchronous induction generator (alternator) connected to the local electrical grid. Inside the screw expander, the working fluid expands creating pressure and temperature drops at the exit port, thus converting the thermal energy of the high pressure vapor into mechanical work. Without any liquid injection, the organic fluid would leave the expander at state 6 but, because of this process, it actually leaves the expander at state 6a, closer to the vapor saturated wet curve.

The power generated by the twin screw expander is calculated as follows:

$$\dot{W}_{exp} = \dot{m}_{org}(h_5 - h_{6s})\eta_s\eta_m \tag{2}$$

where $\eta_s = \frac{h_5 - h_6}{h_5 - h_{6s}}$ is the expander isentropic efficiency (≈ 70%), η_m - mechanical efficiency, h_5 and h_{6s} - working fluid specific enthalpies at the expander inlet and outlet in the ideal expansion process, respectively (kJ/kg).

The working fluid enters the condenser at state 6a, close to its saturated state, where it is condensed at constant pressure and temperature (process 6a-1) to become a saturated or sub-cooled liquid (state 1). The condensing (latent) heat (enthalpy) is transferred from the vapor to the cooling fluid (10 to 15 kg/s) circulating within a water/glycol (50% by weight) closed loop linked to an air-cooled liquid cooler. The condenser thermal power is expressed by the following equation:

$$\dot{Q}_{cond} = \dot{m}_{org}(h_{6a} - h_1) \tag{3}$$

where h_{6a} and h_1 are the organic fluid specific enthalpies entering and leaving the condenser (kJ/kg). The condensed organic fluid at state 1 is stored inside the liquid receiver, pumped back to the preheater/evaporator assembly and then a new cycle begins.

Since the exergy destruction rate in the feed pump is relatively small, the pumping process 1-2s is considered as isentropic (adiabatic), and the pump power input can be expressed as:

$$\dot{W}_{pump} = \frac{\dot{m}_{org}(h_{2s} - h_1)}{\eta_{pump}} \tag{4}$$

where η_{pump} is the feed pump isentropic efficiency (≈ 90%), h_1 and h_{2s} - organic fluid specific enthalpies at the fed pump inlet and outlet in the ideal case, respectively (kJ/kg).

IV. RESULTS

The organic fluid feed pump ran at full range variable speed with the current frequency varying between 0 and 60 Hz in order to get small amounts of superheat at the evaporator outlet and adjust the power generated to the system actual temperature lifts.

A. Temperature lifts

Experimental tests were conducted with the waste heat thermal carrier (water) and cooling fluid entering the ORC-50 machine at temperatures varying between 85°C and 115°C, and 15°C to 30°C, respectively. Under these thermal boundary conditions, the temperature lifts (i.e. the difference between the waste heat and cooling fluid inlet temperatures) varied between 55°C and 100°C (Fig.2).

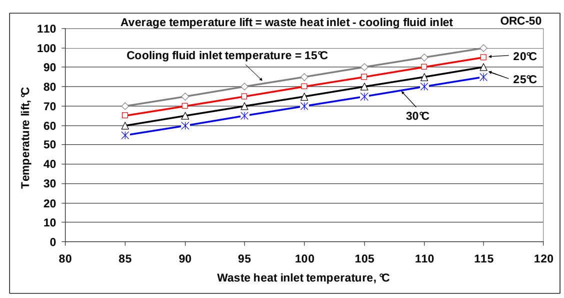

Fig. 2. Experimental temperature lifts

B. Net power output

Fig. 3 shows that the net power output increased linearly with the waste heat inlet temperatures when the cooling fluid inlet temperatures were kept constant. It can be seen that a maximum net power of 47 kW was generated with waste heat entering the ORC machine at 115°C and cooling fluid at 15°C, i.e. 94% of the maximum design output power at a temperature lift of 100°C.

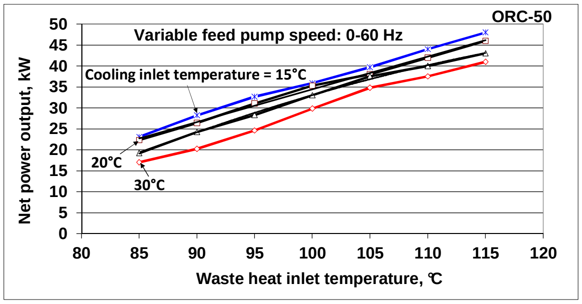

Fig. 3. Net electric power output as a function of waste heat and cooling fluid inlet temperatures

Fig. 4a shows the thermodynamic cycle of a representative run (test AD-14) with waste heat and cooling fluids entering the evaporator and condenser at constant temperatures of 105°C and 20°C, respectively. Under these operating conditions, the expander net power output was 39.9 kW, i.e. about 79% of the maximum design power output of the ORC-50 machine.

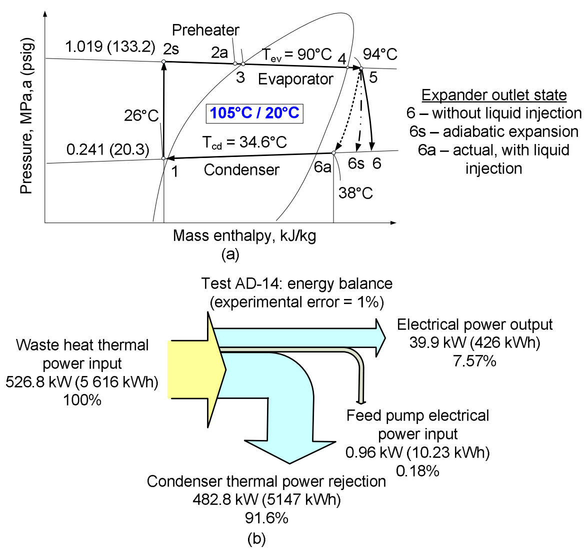

Fig. 4. (a) Thermodynamic cycle of test AD-14 using waste heat and cooling fluid inlet temperatures of 105°C and 20°C, respectively; (b) experimental energetic balance; T_{cd} - condensing temperature; T_{ev} - evaporating temperature

The energetic balance of test AD-14 was achieved with a net heat-to-electricity conversion energetic efficiency of about 7.6% (Fig. 4b).

C. Energetic conversion efficiency

The net heat-to-electricity energetic conversion efficiency rate (η_{net}) of the ORC machine is a dimensionless number defined as the ratio of the net electrical power output ($\dot{W}_{gross,exp} - \dot{W}_{pump}$) to the sum of the preheater plus evaporator ($\dot{Q}_{preheat+evap}$) thermal power inputs. As the electrical power required by the feed pump is about 1% of the expander power output, it has been neglected, and, thus:

$$\eta_{energ} = \frac{\dot{W}_{net}}{\dot{Q}_{preheat+evap}} \approx \frac{\dot{W}_{gross,exp}}{\dot{Q}_{preheat+evap}} \qquad (5)$$

where $\dot{W}_{net} \approx \dot{W}_{gross,exp}$ is the expander net electrical power output, i.e. the gross power generated by the expander less the (neglected) feed pump electrical power input. The preheater plus evaporator thermal power inputs are expressed as follows (see Fig.1):

$$\dot{Q}_{preheat+evap} = \dot{m}_{waste}\bar{c}_{p,waste}(T_{waste}^{IN} - T_{waste}^{OUT}) \qquad (6)$$

It can be seen from eq. 5 that any parasitic electrical energy consumption (or power input), such as that of the waste heat (water) and cooling fluid (water/glycol) circulating pumps and of the air-cooled cooler fans, was taken in consideration. However, in actual industrial applications, such parasitic power has to be analysed carefully and, if possible, eliminated or substantially reduced. This approach is sometimes possible in practice because many industrial sites are already equipped with circulating pumps for waste heat (hot water) and cooling fluids, as well as with cooling towers or other similar cooling devices.

Fig. 5 shows that, if the cooling fluid inlet temperature remains constant, the heat-to-electricity net conversion efficiency, as defined by eq. 5, increases along with the waste heat inlet temperature. It can be seen, for example, that with cooling fluid inlet temperature of 15°C, the net conversion efficiency was 6.5% with waste heat inlet temperature of 85°C, and 9% when it attained 115°C. By linearly extrapolating the measured values, the net conversion efficiency may increase to about 10% by using waste heat entering the ORC machine at 125°C. Table 1

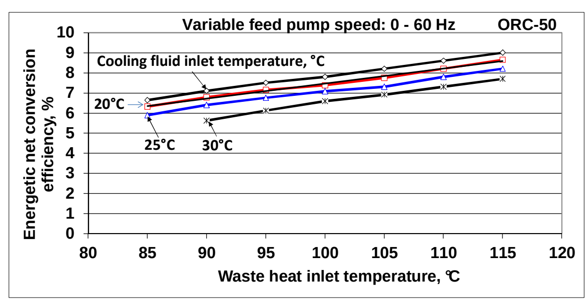

Fig. 5. Energetic net conversion efficiency as a function of waste heat inlet temperatures and cooling fluid inlet temperatures constant at between 15 and 30°C

C. Exergetic conversion efficiency

Similar to heat-to-electricity net conversion efficiency, the exergetic net conversion efficiency can be defined as:

$$\eta_{exerg} = \frac{\dot{W}_{net}}{\dot{E}_{available}} = \frac{\dot{W}_{net}}{\dot{Q}_{preheat+evap}\left(1-\frac{T_a}{T_{waste}^{IN}}\right)} \qquad (7)$$

where $\dot{E}_{available}$ is the exergy flux available in the waste heat source at the preheater/evaporator inlet (kW); $\left(1-\frac{T_a}{T_{waste}^{IN}}\right)$ - Carnot factor, i.e. the maximum amount of waste heat input which can be transformed into mechanical work; T_a - ambient absolute temperature (K); T_{waste}^{IN} - waste heat inlet temperature (K).

By ignoring the potential and kinetic energies, the maximum reversible work per unit mass flow, equal to the decrease in flow availability plus the reversible work that can be extracted from an ORC cycle operating between the waste heat inlet absolute temperature (T_{waste}^{IN}) and the ambient absolute temperature (T_a), is defined as:

$$e = (h - h_a) - T_a(s - s_a) \qquad (8)$$

where a subscript refers to the dead-state, usually the environment temperature, but here, it is the state of the cooling fluid entering the condenser.

Table 1 summarizes the energetic and exergetic net conversion efficiencies of five representative tests as functions of the waste heat inlet and outlet temperatures (at a constant flow rate of 11.6 kg/s) for a reference ambient temperature of 20°C representing the cooling fluid temperature entering the condenser.

TABLE 1

Energetic and exergetic net conversion efficiencies

Test	T_{waste}^{IN}	T_{lift}	$\dot{W}_{net}$	η_{energ}	η_{exerg}
	°C	**°C**	**kW**	**%**	**%**
AD-1	85	65	22.3	6.62	3.95
AD-3	90	70	26.4	6.98	4.11
AD-8	95	75	31	7.2	4.29
AD-9	100	80	35	7.38	4.38
AD-14	105	85	39.9	7.57	4.43

Note: cooling fluid inlet temperature constant at 20°C.

V. POTENTIAL APPLICATIONS

The increase in fossil fuel prices, the growing availability of industrial waste heat and of renewable energy (solar, geothermal, biomass, etc.) open a huge market for heat recovery machines based on the ORC cycle, if they could be manufactured and installed at competitive costs [2]. The competitiveness of ORC machines also depends on the price of conventional fuels (e.g. natural gas, oil, electricity), as well as on their utilization factor that must be at least of 5,000 hours per year.

Applications of ORC machines include heat recovery from industrial processes, internal combustion and reciprocating engines (from both jacket cooling water and combustion gas exhaust), medium size gas turbines, deep geothermal, oil and gas coproduced fluid wells, solar energy sources, biomass boilers, combined heat and power plants, and gas compression stations.

Some large-scale, high-temperature plants as those for aluminum processing lose up to 50% of the energy used for the electrolytic process, about 2/3 of which is lost through the walls of hundreds of furnaces in each plant and 1/3 from the combustion gases. The energy lost through the walls is difficult to recover, but the heat from electrolysis gases at about 100°C – 130°C is easier to recover. After filtering and purifying the exhaust gases, the waste heat could be recovered and used for power generation, thus enhancing process efficiency and reducing greenhouse emissions. The power generated may avoid the need to purchase extra electricity to increase aluminum production, while the energy intensity of the process can be reduced, and the overall competitiveness, improved.

In most of internal combustion engines, only 30-40% of the total fossil thermal heat input is turned into mechanical work, while the remaining 60-70% leaves the engine as waste heat, mainly through the jacket water cooling system and the exhaust pipe. By recovering the waste heat from engine liquid coolants and/or from the exhaust gases of the internal combustion engines in order to convert it into electrical power, the overall efficiency of these devices could exceed the present limits.When fuel costs are high and internal combustion engines are used for electrical production, ORC machines will save fuel costs by allowing the engine to operate at a lower fuel input rate for the same electrical output. In such systems, the waste heat from the engine jacket water cooling process (at about 90°C) is combined with the waste heat recovered from the exhaust gases at temperatures above 300°C.

Biomass is mainly used locally because of its low energy density and bio-gases are another renewable energy sources available in many off-grid combined heat and power plants. By recovering heat from biomass boilers at temperatures up to 300°C, the heat-to-electricity net conversion efficiency of ORC machines could increase up to 13% and even more.

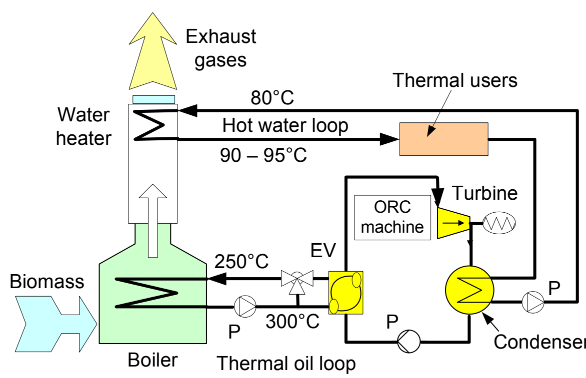

Fig. 6. Principle of a power and heat generation plant using biomass renewable energy; ORC – Organic Rankine Cycle machine; P - circulating pump

In the single-stage system shown in Fig. 6, the biomass boiler is fed by virgin wood chips. Thermal oil (mineral or synthetic) at 300°C flowing through a closed loop is cooled-down to 250°C inside the evaporator EV of the ORC machine. At full load, the ORC machine's net heat-to-electricity efficiency could attain 18%, and at 50% load, it may drop at 16.5%.

To increase the overall energy efficiency of ORC machines, the heat rejected by the condenser at around 80°C could be further heated up to 90-95°C by the aid of boiler's exhaust gases and, then, used for industrial (e.g. wood drying, greenhouses and fish farms heating, etc.) or district heating.

Gas compression stations also generate a lot of waste heat mainly from compressors' inter-cooling heat exchangers. Recovering such a waste heat presents a double benefit in both producing electrical power from the heat generated and reducing the cooling power loads on the existing systems.

Finally, the ORC machines are perfectly suited to exploit low-enthalpy deep-geothermal heat sources at temperatures ranging between 85-350°C, and also for converting solar energy into electricity.

VI. CONCLUSIONS

This paper succinctly presents a number of experimental results obtained with a small-scale beta-prototype 50 kWe ORC machine using HFC-245fa as a working fluid. It converts low-grade waste heat or renewable energy at inlet temperatures ranging between 85°C and 125°C into electricity, using a cooling fluid at inlet temperatures varying from 15°C to 30°C. To achieve superheating amounts lower than 4-5°C at the evaporator outlet, the speed of the organic fluid feed pump has been set to theoretically vary between 0 and 60 Hz. Under these thermal operating conditions, the net power output and net heat-to-electricity energetic conversion efficiency rate varied between 22.3 and 39.9 (electric) kW, and from 6.62% to 7.57%, respectively. With waste heat inlet temperatures higher than 115 °C, the net conversion factor may reach 10% or more at 125°C with cooling fluid inlet temperature of 15°C.

The results obtained show that both net power output and energetic conversion efficiency depend on the cooling fluid inlet temperatures, i.e. on ambient thermal conditions. For example, at the same waste heat source inlet temperature (ex: 90 °C), the expander electrical power output increased by about 28.3% when the cooling fluid inlet temperature dropped from 30°C to 15°C.

In the future, R&D programs are required to identify technological options to recover energy from industrial plants to evaluate their efficiency and costs, and find the most efficient application fields. ORC machines could become energy efficient and environmentally friendly alternatives allowing the industrial plants to reduce their energy consumption, and, thus, become less dependent on the fluctuating prices of primary energy sources.

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The thermodynamic benefits of the integration of cogeneration installations in bakery ovens

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Abstract — **The present work is motivated by the need to reduce the irreversibility of the process with combustion gas in bakery ovens, especially it was analyzed the process of the mixing of the obtained gases in the combustion chamber with the recycled gases in order to achieve the necessary temperature of the flue-gas in the oven channels. There are also exposed the arguments in favor of using gas tunnel ovens in the bakery compared with the electrical ovens. There are presented the essence and benefits of the implementation of two solutions of ovens with integrated cogeneration installation.**

Key words – bakery oven, cogeneration installation, exergy losses, efficiency of the thermodynamic perfection, irreversibility, energy efficiency.

I. INTRODUCERE

The bread industry is one of the main branches of the food industry in Moldova, which offer the population the vital food – the bread. The food industry has a considerable share in the entirely industrial sector - about 51,6%, according to statistics, [1].

Baking industry share in total industrial production is approx. 4% [2], and in the total food production – 8,8%, but that does not diminish its importance in the development of industrial sector. Bakery branch has been, it is and will be one of the most needed, because this branch assure the vital human needs.

Recently, prices of bakery products were increased by approx. 15%. An important argument was put forward by the bakers - to increase of tariffs in august by 39,3% for electricity and 15% for natural gas. This has increased energy share in the cost price of bread from 10% to 11,5% [3].

In the mentionate context, the increasing energy efficiency of processes in bakery ovens is a significant concern and necessary.

The continuous increase of the price of the used fuel demonstrates the need to rationalize the energy consumption.

The researches devoted to developing a method and optimal schemes of fuel utilization are very important, not only in the bakery processes but in all technological industrial processes.

In the technological process of baking, in bakeries are used practically all forms of energy: warm water – for the preparation of the dough and for cleaning of the equipments; steam - for steaming bread in ovens and for drying of pasta; natural gas – for ovens; electricity – for ovens and for different electrical equipments; compressed air and cold - in auxiliary processes.

In general, there is a detailed analysis of the bread-making process by steps made in order to highlight the energy-intensive operations. The analysis results are shown in tab. I [4].

TABLE I.

THE ANALYSIS OF SHARE OF THE ENERGY CONSUMPTION IN THE BREAD BAKERY PROCESS

Nr.	The step of technological process	Share, %	
		electricity	heat
1	Reception and storage of raw materials	1,5	0,3
2	Reception and dosing of raw and auxiliary materials	4,5	0,8
3	Preparation of dough	13,8	1,4
4	Pocessing of dough	23	2,4
5	Final rising	22,6	1
6	Baking	33,7	93,8
7	Receive product	0,5	0,2
8	Storage	0,4	0,1

As it is shown in the tab. I, the highest consumption of energy takes place in the cooking process, practically the entire heat introduced in the process of production of bread (approx. 93,8%) is used in the cooking chamber. Therefore, the present work will approach especially the issue of the increasing energy efficiency of the process of baking the bread.

II. COGENERATION – THE MEASURE TO INCREASE ENERGY EFFICIENCY OF PANIFICATION

At the moment, most bakeries in Moldova use electric bakery ovens. The main argument is simplicity of installation and their exploitation, compared with natural gas ovens, especially if it is the lack of access to a gas pipe. At the same time, in the use of electric ovens, there is no problem with the evacuation of combustion gases, because they aren't.

Actually, the country at the moment, deals with essential increase of the price of electricity, and it is absolutely necessary to pass the ovens from the electrical supply to the natural gas.

In [5] there were presented the detailed essence and benefits of implementing the measures to improve energy efficiency in the process of baking bread. A particular attention is paid to method of transition of ovens from the electric to natural gas supply and the cogeneration application in bakeries.

An electrical oven whose surface of baking is 50 m^2, has the average working power of 200 kW, and with the same surface and the same productivity, in the use of natural gas - the average consumption of natural gas– 23 m^3/h. Considering the operating schedule – 16 hours per day and 330 days per year, the ovens consume respectively 1056 MWh/year and 121,4 thousands m^3 of natural gas per year. For generating the indicated quantity of electrical energy, at the power plant are consumed 313 thousands m^3 of natural gas. The transition from electrical supply to natural gas supply of ovens will be reduced the expenses for energy resources, at the current fares [6], by over one million lei per year, or 20 thousand lei per m^2 per year.

The bakeries operates entire year at a practically constant productivity. It slightly varies the electrical and thermal load. Therefore, an installation with cogeneration, based on piston engine on natural gas, at the same bakery can operate with a high coefficient of the use of the installed power, which would reduce the cost of the produced energy.

In the case of using the cogeneration instalations result a fuel economy, in comparison with separate generation of electricity and heat, from 25% to 40 %.

III. THE BENEFITS OF TUNNEL OVENS WITH GASES

Lately, the tunnel ovens with gases became widespread in bakeries. The benefits of the tunnel ovens, in comparison with other types of gas ovens, are:

- mechanization of the processes of loading and unloading the products;
- continuous production process;
- uniform distribution of the heat in 4-5 areas of baking;
- more efficient automation of the baking and steaming areas;
- disappearance of the "gas" flavor of bread, because the flue gases flowing through the gas channels and don't enter in the baking chamber of the oven;
- viewing the baking processes through special viewfinders.

These are the arguments favoring the choice of tunnel oven. But the arguments in favor of the supply option to natural gas of the ovens, comparing with the electrical supply, are:

- the electric ovens have a higher thermal inertia and an expensive function, due to higher tariff for electricity;
- the taste of the bread baked in the gas ovens is better than that obtained in electric ovens;
- the probability of the interruption of electricity supply at the factory (in case of damage or repairs to electrical networks) is much higher than for natural gas.

In addition, some researches [7], demonstrated that total equivalent emission of greenhouse gases, generated by ovens, to produce 1 kg of bakery products, is greater than 2 times in electrical ovens than in the case of gas ovens. But environmental problem is very acute today, at national and global level.

In the above context, the purpose of this paper is to analyze the efficiency of the installation of oven by integrated internal combustion engine.

Thus, this instalation will produce electricity and the flue gases evacuated from the cogeneration installation, will be used to perform the technological processes baking in the oven.

IV. EXERGETIC ANALYSIS – RELEVANT METHOD TO ASSESSMENT THE ENERGY EFFICIENCY

In baking ovens take place baking processes of bread. For this purpose, it is consumed a certain amount of energy. It is important to understand how effectively is this process from the energy point of view.

A method of analysis would be one based on balance and energy efficiency of the oven.

Energetic analysis is the classic method of assessing the energy efficiency of an installation or process. But this method does not take into account the following important factors: thermodynamic state of the system, form of energy consumption, the degree of perfection of the process (the degree of irreversibility), the state of environment. That's why, this method leads to the difficulty of interpreting the energy efficiency level.

This has caused the scientists to search the new methods, more complex, for technological processes analysis, in sequance with the second law of thermodynamics, which would allow qualitative assessment of different forms of energy. So, the exergetic analysis was developed.

In concordance with the second law of thermodynamics, energies with limited capacity of transformation, can be converted partially into mechanical work, ie in exergy, the rest of these energies is anergy.

The economic value of energy is so great then the suitable energy is greater, therefore the exergy can be used to assess circulating energies, and the quality of processes which take place in the installation.

In concordance with the second law of thermodynamics, the sum of the input exergy into the ovens is equal to the sum of the output exergy plus exergy losses.

The thermodynamic perfection of the oven is expressed through the efficiency of the thermodynamic perfection of this installation γ_{ex}^{cupt} and constitutes the report between the amount of exergy output of the oven $\sum_{i=1}^{n} E_{xi}^{e}$ and the amount of exergy input the oven $\sum_{i=1}^{n} E_{xi}^{i}$ [8]:

$$\gamma_{ex}^{cupt} = \frac{\sum_{i=1}^{n} E_{xi}^{e}}{\sum_{i=1}^{n} E_{xi}^{i}} = 1 - \frac{\sum_{i=1}^{n} P_i}{\sum_{1=1}^{n} E_{xi}^{i}}, \tag{1}$$

in wich $\sum_{i=1}^{n} P_i$ are the exergy losses of the oven .

That's why, knowing and the calculation need of the losses of exergy have considerable importance to determine the methods to reduce the irreversibility of the processes which take place in the studied oven.

To calculate these losses it can be used the entropy method, which requires the calculation of exergy losses in each process separately using Guy-Stodola theorem::

$$P_{ex} = T_o \cdot \Delta S, \tag{2}$$

where: T_0 is the thermodynamic temperature of the environment; ΔS - increase of entropy in examined process because of the irreversibility.

From the last relationship it follows that the question of exergetic losses calculation in any process is reduced to calculation of variation of entropy.

V. WHY DO WE NEED OF THE INSTALLATION OF OVENS WITH INTEGRATED COGENERATION?

That's why the real processes from the oven are irreversible; in this processes take place destruction of exergy. The improvement of the processes, ie reducing their irreversibility, can be achieved by improving of ovens in order to reduce losses of exergy, which may lead to a decrease of operating costs (due to the reduction of primary energy consumption).

One of the most irreversible processes resulting in the oven is the process of mixing of the gases to the combustion chamber with the recirculated gases.

For example, the oven PPP 3 54 211ST is equipped with combustion chamber with the gas burner. Gas temperature in the combustion chamber is approx. 1630°C, while necessary temperature of the gases in the channels of oven is about 450 ºC (fig. 1).

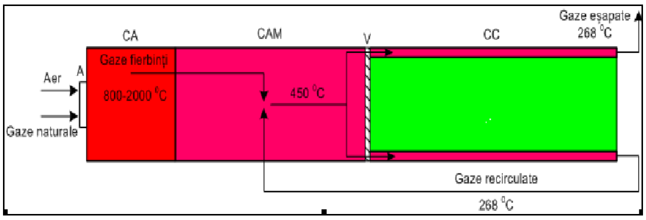

Fig. 1. Heat flow diagram of the oven:
A – burner; **CA** – combustion chamber; **CAM** – mixing chamber; **V** – fan; **CC** – baking chamber.

There is a huge difference between the gas temperature in the combustion chamber and the required temperature in the channels of baking chamber. The reduction of the gas temperature in the combustion chamber is achieved after mixing of hot gases with the cold gases recycled from the channels of the baking chamber, whose temperature is 268 ºC.

In case when currents, which are mixed, are the same ideal gas, with the same constant of gas *R,* and the same specific heat capacity c_p, is valid the following relationship for calculating entropy variation (were the term 1 refer to the parametres of the hot combustion gas in the combustion chamber, 2 – the parametres of the recirculated exhaust gases in the oven and 3 – the parameters of the gas mixture resulting from the mixing of the first two), [9]:

$$\frac{\Delta\dot{S}}{\dot{m}_3} \approx x(1-x)\left(\frac{T_1-T_2}{T_1}\right)^2 + x\frac{R}{c_p}\left(\frac{p_1-p_3}{p_3}\right) + (1-x)Rc_p\left(\frac{p_2-p_3}{p_3}\right) \geq 0 \quad (3)$$

where $x=\frac{\dot{m}_1}{\dot{m}_3}$ is the ratio between the flow rate of the hot flue gas in the combustion chamber and the flow rate of the gas mixture.

As it is shown in the quation (3), thermal ireversibility, increase of entropy and losses associated of mixing processes increse with the square of difference of the two gases mix of mixing temperature difference of the two mixed gases.

A measure to increase the energy efficiency of the process of baking bread would be the reducing the irreversibility of the process by replacing the process of mixing by a heat transfer process, which is less irreversible.

The essence of this tehnology involves „integrating" a cogeneration installation (based on the internal combustion engine, or a gas turbine installations) in an ovens installation.

The flue gases after expansion in internal combustion engine will be debited in a heat exchanger for the heating of the air taken from the environment and its subsequent circulation in baking channels of the oven (fig. 2).

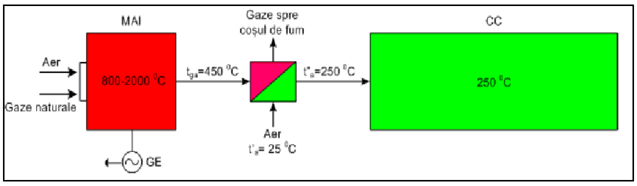

Fig 2. Schematic diagram of CHP plant with heat exchanger:
MAI - internal combustion engine; **GE** – electric generator; **CC** – cooking chamber.

Exergy losses after irreversibility of heat transfer will be determined by the relation, [9]:

$$P_{ex} = E_x^g - E_x^a = Q\frac{T_o}{T_{med}^a}\cdot\frac{1}{\left(1+\frac{T_{med}^a}{\Delta T}\right)}, \quad (4)$$

where: T_{med}^a is the avearge thermodynamic temperature of the air in the heat exchanger;

ΔT - mean difference of the ambient temperature wich environment are cooled and heated;

Q – the flow of heat exchanged in the heat exchanger, between the two fluids.

As it is shown in equation (6), the loss of exergy is increased by the increase of the difference ΔT, but not square as in the case of mixing processes.

Applying this technology, the working agent for heating the baking chamber will be the air heated in the heat exchanger. This solution is welcome for ovens in which termal agent is debited directly in the baking compartment, as the exhaust gas from the engine can contain drops of oil used to lubricate the engine and can not be debited in the baking chamber because it can contaminate the bread.

In the exposed technology, are diminished the irreversibility of the process for the preparation of the heat by replacing the mixing process to heat transfer process, with a lower degree of irreversibility. The disadvantage of this solution lies in the fact that the heat exchanger can be larger, leading to an incrase in the massiveness in the entire installation.

In case of the ovens tunnel, the thermal agent is not charged directly to the baking chamber, but circulates through the channels made in the walls of the chambre (so they have no contact with the bread), the flue gases discharged from internal combustion engine can be handled the cannels. Thus, are totally exhosted the losses of exergy associate with the made process of the heat (fig.3).

At the same time, it will significantly simplifies the installation by excluding the heat exchanger from its composition, in comparation with the solution described above.

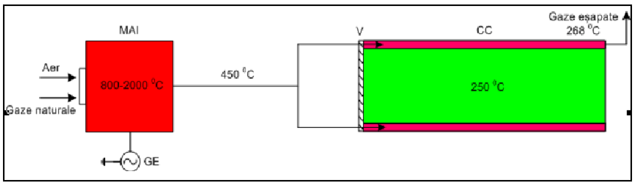

Fig 3. Schematic diagram of CHP plant without heat exchanger

Even if the combustion of fuel in the combustion chamber of the internal combustion engine, anyway there will be irreversibility and respective loss of exergy, they will be associated to the process of producing electricity (as the main product) and the exhaust gas evacuated from the internal combustion engine and subsequently used for the process of baking, they will result, in effect, as a waste.

The internal combustion engine is chosen depending on the thermal capacity of the baking chamber of the oven so that the oven heat load – *Q*, must corespond to the flow of the exhausted gases of internal combustion engine. Also, the combustion gas flow out of the internal combustion engine must provide the necessary flue gas-V_{gt} in baking channels of the oven.

In tab. II, are presented the basic parameters of the ovens PPP required for selecting the type of internal combustion engine.

TABLE II.

THE TOTAL VOLUME OF FLOW GASES AND HEAT LOAD OF THE OVENS PPP

Oven type [10]	Productivity, kg/h	V_{gt}, m^3/s	Q, kW
PPP 2,1 18,9	342	0,56	163
PPP 2,1 25,2	450	0,74	216
PPP 2,1 31,5	558	0,84	245
PPP 2,1 37,8	684	1,02	297
PPP 2,1 44,1	792	1,21	353
PPP 2,1 50,4	900	1,40	408
PPP 2,1 56,7	1008	1,58	461
PPP 2,5 30,0	540	0,84	245
PPP 2,5 37,5	684	1,02	297
PPP 2,5 45,0	810	1,30	379
PPP 2,5 52,5	954	1,49	435
PPP 2,5 60.0	1080	1,67	487
PPP 3,0 54,0	972	1,49	435
PPP 3,0 63,0	1134	1,76	513
PPP 3,0 72,0	1296	2,05	598
PPP 3,0 81,0	1458	2,23	650
PPP 3,0 99,0	1782	2,79	814
PPP 3,0 108,0	1944	3,07	895

CONCLUSIONS

In the above article, we can conclude that the tehnology of integrated cogeneration installation in ovens is very relevant from the thermodinamic point of view, respectively, in terms of energy efficency.

This measure allows the minimization of the exergetic losses associated to the process of securing the necessary temperature of thermal agent (by mixing), by replacing it with a heat transfer process, which is less irreversible than the mixing processe (solution shown in Fig.2) or total evoidance of these exergy losses (the solution shown in Fig. 3).

The production of two forms of energy, after retrofitting of installation, will contribute to the increase of the amount of the flow exergy from the modernized installation (because the exergy work is equal to the the work done and exergy of electricity also is equal to the value of produced electricity), by increasing the thermodynamic effciency of the installation calculated the equation (1).

It should be noted that the proposed technology can be implemented in any enterprise equipped with natural gas oven, it doesn't matter with the type of products cooked, especially in cases where the required temperature of the flue gas for performing the processes is much lower compared with the temperature of combustion natural gas.

However, the result of implementing the cogeneration installations at the enterprise, it will be ensured with the electricity needed to carry out other processes or utilities, and coolants agents can be then used to require thermal energy consumption for preparing blanks or cleaning the equipments.

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Estimation of the theoretical cogeneration potential in Republic of Moldova

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Abstract — **The article represents authors' attempt to assess the potential for cogeneration in Republic of Moldova. Based on statistical data and using generalized indexes was calculated theoretical cogeneration potential in residential and tertiary buildings, in industry and in agriculture. The total thermal potential represents 6222 MW, the electrical potential 4636 MW, from which 5040 MW and 3700 MW, respectively, are attributed to building sector. Republic's electrical potential surpasses its power demand. Taking in consideration that some EU countries have an estimated economic potential for cogeneration of 5% from the theoretic potential, the electrical economical potential of 200…300 MW could be easily adopted by Republic's electrical system.**

Keywords— cogeneration, potential, buildings, industry, agriculture

I. COGENERATION POTENTIAL

EU Parliament actively incorporated cogeneration as a well-dev„eloped and successful tool of energy efficiency into its energy policy in the opening paragraphs of the European Union's Cogeneration Directive 2004/08/EC, repealed by Directive 27/2012/EU on energy efficiency. Cogeneration, also referred to as combined heat and power, is a mature and efficient approach to the simultaneous production of electrical and thermal energy from a single fuel. By making use of the heat rejected from one process in the production of the other, it creates an extremely efficient approach to electrical energy generation and thermal energy distribution that creates a substantial increase in electrical energy production from fossil fuels [1]. Use of cogeneration as a way of energy efficiency currently is not sufficiently applied across European Community and even less in Republic of Moldova. Adapting and evaluating the cogeneration potential is an important step in effective cogeneration promotion.

It is distinguished theoretical cogeneration potential, technical potential and economic potential for cogeneration.

Theoretical cogeneration potential is the top potential amount of cogeneration that could be installed in our country. It can be estimated from the maximal normative demand of thermal energy of the country which is defined by the macro - economic data and by generalized indexes or by specific standards of consumption.

Theoretical cogeneration potential in industry can be determined for the branches with low temperature technological processes that can be supplied with hot water or steam. These processes are most common for processing industry enterprises.

Greenhouses and zootechnical sector show potential for cogeneration in agriculture. In agriculture, for greenhouses it is necessary to take in consideration the specific destination including: type of the cultivated plants (tomatoes, cucumbers, strawberries flowers, etc.), roofing type (glass, polyethylene, and polycarbonate), building type and dimensions.

However in transportation use of the heat, available for cogeneration, is practically absent. Also consumption for buildings heating as a part of the branch is non-significant and can be attributed to tertiary sector.

The real technical potential for the building sector is evaluated for every particular residential area or every single thermal energy consumer using their individual data and characteristics.

For the technical objects technical potential can be established from the thermal energy and natural gas meter or sub-meter data, according to the measurements of control apparatus and parameters of the systems with thermal energy use.

For the zoo technical objects it is of value to know the final product: bovine meat, milk, poultry- chickens for eggs, broiler, turkey, ducks, etc.

The economic viability of cogeneration projects can be determined for systems from the prefeasibility studies for every particular suggested cogeneration installation.

Economic potential is much inferior to the technical potential, not to mention the theoretical potential as a consequence of that from the total thermal energy demand is subtracted the quantity feasibility produced by the heat pumps and solar panels, and for the rest of excess high efficient cogeneration is feasible only for the base, with the installation being in use over 4500 hours per year. Thus, according to [2], in some states EU members, in average only 5% of the evaluated technical potential is perceived as economic potential.

Theoretical potential for cogeneration in the country is drafted from the residential buildings and retrial potential also industry and agriculture potentials.

Cogeneration potential is calculated according to the thermal energy demands.

II. THEORETICAL POTENTIAL FOR COGENERATION IN BUILDINGS

In Republic of Moldova 50% of the total electrical energy and up to 70% of the heat [3] are consumed by the building sector. Therefore this is the ground for the highest theoretical cogeneration potential.

Buildings use thermal energy to meet their heating, cooling and hot water requirements. Thermal energy demands were calculated based on the numbers of urban and rural population separately, on approximate heat demand for heating, cooling and on average hot water consumption per capita.

Calculation results are displayed in Table I. Population number were based on [4], specific heat and cooling consumption per person based on data from [5]. The specific heat consumption for heating water was calculated based on use of 60 l. of water with a temperature of 65 0C per day per person and an non-uniformity coefficient of 3.4

TABLE I.

THEORETICAL COGENERATION POTENTIAL IN BUILDING SECTOR

Sector	Urban	Rural	Total
Population, thous.pers.	995,2	1918	2913
Specific consumption heating & ventilation (kW/capita)	0,73	1,0	-
Hot water specific consumption (kW/capita)	0,54	0,54	-
Tertiary consumption share (%)	30	15	-
Heating network loss, %	10	5	-
Thermal cogeneration potential (MW)	1643	3397	5040
Cogeneration index	0,9	0,6	-
Electrical cogeneration potential (MW)	1627	2079	3706

Tertiary consumption share was established based on the recommendations [5] and actual building conditions in the Republic's cities and villages. Heating network loss and thermofication index (heat to power ratio) data were used assuming that unitary power of the installations is greater in the cities than in rural areas.

Consequently theoretical potential in rural areas is considerably higher compare to the urban areas , and considering the economy and the structure of Moldavian villages (with small houses placed at relatively long distance from each-other) the perspective of cogeneration development in this sector is low.

III. THEORETICAL COGENERATION POTENTIAL IN THE INDUSTRY

Industry enterprises are a significant consumer of thermal energy. In addition to heating, cooling and hot water needs, this facilities use heat within their technologic processes. Technologic processes use heat of two deferent level of temperature:

- Low potential: with a temperature between 50…300 ^{0}C, transported by way of water or steam, representing an actual suitable source of cogeneration , especially for enterprises with annual production cycle, and not seasonal;
- High potential: with temperature above 350 0C. Industrial furnaces are using heat directly from the combustion installation and cannot be included in cogeneration system, and in many cases emission gases can serve only as secondary energy resources for heating water or steam production, consequently that reduces economical cogeneration potential of the enterprises.

Total cogeneration potential of the enterprises can be calculated from the thermal energy consumption with hot water for technological processes, heating, and utilities and with technological steam. The annual technological consumption for different industrial branches was established from the branches production and specific consumption data.

The calculations results for the main branches of the industrial sector are presented below in Table 2. The annual average for 2010, 2011 and 2012 [6] was used for annual production, specific consumption data were gathered from [7 and 8] and audits reports of several enterprises from Republic of Moldova. Also the mentioned above audits were a data source for utilities consumption share and load factor.

TABLE II. THEORETICAL COGENERATION POTENTIAL IN INDUSTRIAL SECTOR

Potental	Annual production	Heat specific consum	Annual technological consumption (TJ)	Utilities and heating share	Load factor	Thermal cogeneration potențial (kW)
Dairy industry	98 kt	0,95 GJ/t	92,7	0,35	0,8	4960
Meat industry	27 kt	0,9 GJ/t	24,7	0,50	0,7	1676
Planning	129 kt	0,56 GJ/t	72,4	0,20	0,9	3062
Wine	14,8 mil.dal	3,2 MJ/dal	47,2	0,50	0,4	5617
Sugar	92 kt	5,3 GJ/t	485,8	0,05	0,4	40440
Confectionery	13 kt	10,7 GJ/t	136,2	0,10	0,7	6789
Oils	88 kt	2,7 GJ/t	237,4	0,15	0,7	12368
Canned food , juices	61 kt	1,5 GJ/t	91,7	0,20	0,5	6979
Tobacco	7 kt	3,5 GJ/t	24,5	0,15	0,5	1787
Bear	1,1 mil dal	7,7 MJ/dal	8,5	0,15	0,8	387
Chemical industry	16 kt	1,5 GJ/t	24,0	0,20	0,6	1522
Leader, textile industry			45	0,25	0,7	2548
Total industry			1290,2			172198

Given cogeneration index of 0,8 was calculated electrical potential for the industry -138 MW.

IV. THEORETICAL POTENTIAL FOR COGENERATION IN AGRICULTURE

In agriculture heat consumers are greenhouses, cattle and poultry farms, and dryers. Unfortunately the vast majority of dryers are using heated air and combustion gases as thermal agents, which are available seasonally roughly of 2…3 months per year and due to that were not studied as a potential for cogeneration implementation, limiting our research to greenhouses and farms only

TABLE III. THEORETICAL POTENTIAL FOR COGENERATION IN GREEN HOUSES

Article	Value
Greenhouses surface (ha)	100
Specific heat loss (kW/m^2)	0,35
Thermal potential for cogeneration (MW)	350
Cogeneration index	0,8
Electrical potential for cogeneration (MW)	280

According to [9] in 2008 in Republic were available 45 ha of greenhouses with artificial heating systems and it was expected the construction of additional 17 ha. Since the significant difference to the 80s when the total areas of heated greenhouses was 180 ha, for potential calculation purposes we considered a surface of 100 ha, which is 0.003% of the total agricultural surface of our Republic. The results of calculation are related in Table III.

TABLE IV. THEORETIC POTENTIAL FOR COGENERATION IN ZOOTECHNICAL SECTOR

Animals	Bovines	Pigs	Poultry
The number of animals (thous.cap.)	200	400	3000
Specific heat consumption for heating &ventilation purposes (W/capita)	433	245	85
Specific heat consumption for technological pourposes (W/capita)	330	270	15
Total specific consumption (W/capita)	763	515	100
Thermal potential (MW)	153	206	301

Among zootechnical facilities evaluated as a potential for cogeneration were cattle farms, pig farms and poultry farms. The number of animal and poultry was taken from [6]. Calculation were made according to specific consumption data from [10, 11]. The results were shown in Table IV.

Theoretical thermal potential is estimate at 660MW. Given the cogeneration coefficient of 0.8 was calculated electrical cogeneration potential for zoothecnical facilities of 528 MW and in entire agriculture sector thermal potential- 1010 MW and electrical potential - 808 MW.

V. TOTAL THEORETICAL POTENTIAL FOR COGENERATION IN REPUBLIC

The calculations for total theoretical potential are summarized in Table V.

TABLE V. THEORETICAL POTENTIAL FOR COGENERATION IN REPUBLIC OF MOLDOVA

Sector	Thermal potential (MW)	Electrical potential (MW)
Buildings	5040	3700
Industry	172	128
Agriculture	1010	808
Total	6222	4636

The significant difference among the results of Table 5 reveals the relativity of theoretical potential. Buildings potential that includes residential, commercial and public spaces is extremely high when the industry potential estimated from the industrial actual production level is low. Considering the fact that the industry is much more predictable for cogeneration implementation, thus the technical and economic potentials will be higher in the rest of the sectors. Overall an economic potential of approximately is 5% from the theoretical one, with a value between 200…300 MW. Variation in the powers of the load curve being between 300 and 1000 MW cogeneration units would be able to fit smoothly into the power system of the Republic.

CONCLUSIONS

1. Taking into account the approval of the Law No. 92 as of 29.05.2014 on thermal energy and promotion of cogeneration, current research approaches an actual subject for the Republic of Moldova. Covering 90% of its own energy resources consumption by importing and using at least three times more energy for producing the same good or service, than European countries, Moldova established combined heat and power production as a state policy priority.

2. The total thermal potential for cogeneration in the country was estimated at 6622 MW, the electrical potential at 4636 MW, from which 5040 MW_{th} and 3700 MW_{el}, respectively, correspond to the building sector. Understanding the fact that only densely populated localities can provide feasibility for the CHP technologies, the economic potential, especially for the buildings sector, will be reduced.

3. The economic potential for cogeneration was assessed at between 200…300 MW, which can be integrated without difficulties in the electro-energetic system of our Republic, where the variation of the load curve being between 300 MW_{el} and 1000 $MW_{el.}$ In the context of new developing renewable projects – intermittent electricity generation by wind and PV solar plants, CHP plants will offer more reliability to the grid.

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