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Benchmarking of Energy Saving Potential and CO_2 Reduction in Iranian Compressor Stations

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Conventional benchmarking of different process industries, such as compressor stations does not consider energy saving potentials that exists within the process. On the other hand, full Pinch Analysis of these units is both costly and time consuming. For energy planning at macro economy level, we need a rapid and precise procedure that enables the planners to consider almost every potential for energy saving. This research investigates the use of an Organic Rankine Cycle (ORC) to recover waste heat for different types of compressor stations in Iran. In fact, applying ORC to produce electricity is a practical way to recover heat through gas pipeline compressor stations. The Pinch concepts are used in this work to maximise the heat recovery in heat exchanger network for producing maximum power in the expander of ORC. The developed model in this work is applicable for simply calculating the potential for energy saving and CO_2 reduction in different gas compressor stations at different load operation and minimum approach temperature in heat exchanger network. The verification of the proposed model was evaluated and showed a very small error less than 2 %. Having developed the above-mentioned model, there is no need to carry out a full retrofit study for each existing compressor station as the model can simply be applied to similar processes and opportunity of energy saving can be recognized.

1. Introduction

Natural gas compressor stations are the most important parts of gas transmission pipeline networks. These facilities consume a significant amount of the transmission gas to supply the compressor energy demand. A large portion of this energy is discharged to the atmosphere by hot exhaust gasses of gas turbines. Many works have been done to use organic Rankine cycle for recovering the waste heat.

Hedman (2008) investigated three waste energy recovery opportunities applicable to the gas compressor stations and reported that the heat recovery systems are economical only when the compressor capacity, load factor and power purchase price are above certain minimums. Kostowski et al. (2015) performed energy and exergy balance to find opportunities for improving the thermodynamic performance of a natural gas compressor station and reported that the intermediate oil loop for heat recovery plus ORC for electricity generation have the highest energy savings; however, its economic feasibility is not achieved if it works part-load. Muñoz de Escalona et al. (2012) analyzed the rated and part-load performances of combined cycles of gas turbines and ORC and reported that keeping the live vapor conditions in the ORC as constant as possible result in maximum power at any operating scenario, compensating for the inefficiency due to the working of gas turbine at part-load.

Handayani et al. (2012) reported a systematic technique to identify an optimal working fluid for ORC. Yilmazoglu et al. (2014) studied ten different working fluids for the ORC and reported that the n-Pentane has the highest possible efficiency and lowest possible environmental impacts. Saavedra et al. (2010) performed the thermodynamic optimization of ORC using several working fluids and reported that ORC's thermal efficiency can increase to 24 % depending on the working fluid and condensing temperature.

Ahmadi et al. (2012) performed energy and exergy analysis for combined heat recovery system of ORC, single-effect absorption chiller and domestic water heater and reported that the trigeneration system reduces the CO_2 emission more than the conventional ORC. Liu et al. (2013) performed the life cycle assessment to investigate the environmental impact of ORC for heat recovery in power plants and reported that the payback time for CO_2

discharge is 3 - 5 y in ORC. In this work, application of ORC for heat recovery in four different compressor gas stations is investigated using Pinch Technology. Then, after economic analysis, mathematical models are developed to benchmark the generated power of ORC and reduction of CO_2 emissions.

2. Organic Rankine cycle layout

The power plants in this study consist of two closed loop cycles and a heat recovery exchanger, which recover the heat from the gas turbine exhaust. A closed loop of oil is used as an intermediate loop, which transfers heat from the hot gasses of the turbine to the organic working fluid of ORC and reduces the risk of organic fluid explosion. In the second loop, organic working fluid is heated in the economizer, evaporator and superheater and then enters to the expander for generating power; the exhaust gas from the expander is cooled in the recuperator and air-cooled condenser and pressurized with the pump. The condensed fluid is then preheated in the recuperator. ORC layout used in this work is illustrated in Figure 1.

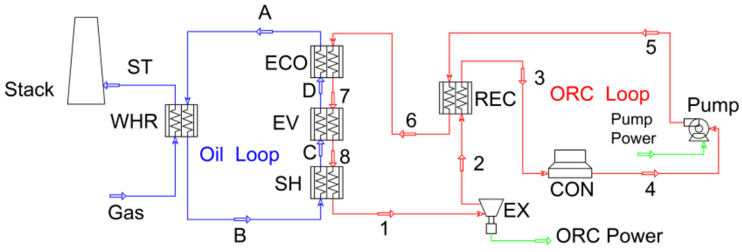


Figure 1: Organic Rankine cycle layout

3. Thermodynamic and design approach

The recovered heat from the exhaust gasses of the gas turbine will be maximized if the temperature of exhaust gasses reduced to the ambient temperature, but this will need an infinitive exchange area. In this work, the outlet temperature of exhaust gas (T_{ST}) from the waste heat recovery (WHR) is assumed to be more than 145 °C due to the dew point temperature. Based on the direct relation between the superheated temperature of working fluid and cycle power and also considering safety issues due to the auto-ignition temperature of normal pentane (260 °C), it is assumed that T_B is 260 °C. It is also assumed that T_{ST}-T_A= ΔT_{min} in order to maximize the heat recovery (Figure 2). The energy balance for WHR is:

$$\dot{m}_{gas}C_{P,gas}(T_{gas}-T_{ST})= \dot{m}_{oil}C_{P,oil}(T_B-T_A) \quad (1)$$

T_8 and T_7 are saturated temperatures of normal pentane and T_D-T_8= ΔT_{min} (Figure 2), so energy balance of superheater (SH) and evaporator (EV) is:

$$\dot{m}_{oil}C_{p,oil}\left(T_B-(T_8+\Delta T_{min})\right)= \dot{m}_{wf}(C_{p,wf}(T_B-\Delta T_{min}-T_8)+\lambda_{wf}) \quad (2)$$

Energy balance of economizer (ECO) is:

$$\dot{m}_{oil}C_{p,oil}(T_8+\Delta T_{min}-T_A)= \dot{m}_{wf}C_{p,wf}(T_8-T_6) \quad (3)$$

Energy balance of Recuperator (REC) is:

$$\dot{m}_{wf}C_{p,wf}(T_6-T_5)= \dot{m}_{wf}C_{p,wf}(T_2-T_5-\Delta T_{min}) \quad (4)$$

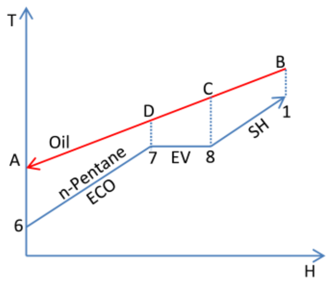


Figure 2: T-H diagram of the heat recovery

Although cooling tower can reach a lower temperature in the condenser (T_4) but in this study, an air-cooled condenser (CON) is selected because usually there is a lack of water in most of the gas station sites and also the capital cost of cooling tower is high. In order to consider the high ambient temperature in extreme situations, T_4 is assumed to be 50 °C and the pressure of working fluid in the outlet stream from the expander (EX) is equal to the saturated pressure of working fluid at condenser outlet temperature (1.583 bar). T_A, T_{ST}, T_6, $\dot{m}_{oil}$ and $\dot{m}_{wf}$ are defined by the algorithm presented in Figure 3.

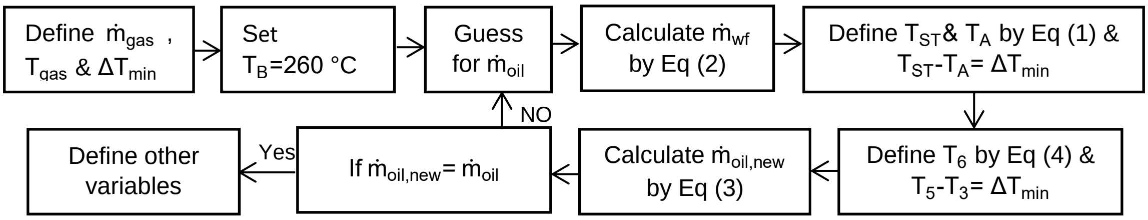


Figure 3: Trial and error algorithm for solving the ORC's thermodynamic equations

4. Simulation of ORC

The mentioned ORC is simulated by commercial simulation software using the Peng–Robinson equation of state and the assumptions that the isentropic efficiencies of the pumps and turbine are 85 % and pressure drops are negligible. The flow diagram of simulation is illustrated in Figure 4.

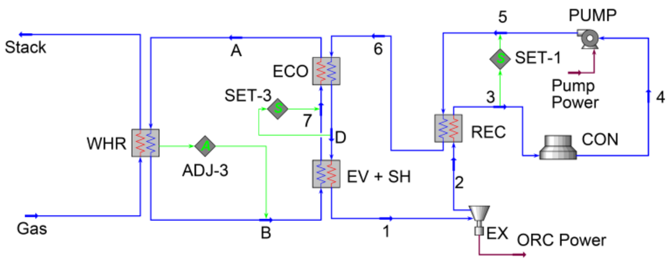


Figure 4: Simulation's flow diagram of Organic Rankine cycle

5. Economic analysis

The total investment for the ORC is calculated by Eq (5) (Smith, 2016).

$$C_{ORC} = \sum_i (f_{M,i} f_{p,i} f_{T,i} + f_I) C_{E,i} \quad (5)$$

Cost correlations ($C_{E,i}$) for the different components (i) in the year 2013 are presented in Table 1. All correction factors in Eq(5) are dimensionless. The installation cost (f_I) factor is assumed to be 0.6 (Smith, 2016). The material correction factor ($f_{M,i}$) is assumed to be 1 for carbon steel and 1.7 for stainless steel (Smith, 2016). The pressure correction factor is assumed to be 1.5 for the pressure between 20 - 30 bar in pumps and the temperature correction factor is assumed to be 1.6 for the temperatures higher than 300 °C in heat exchangers (Smith, 2016). The costs are updated by the cost index of year 2015.

Table 1: Cost correlations for the different components in the ORC (Smith, 2016)

Component	Capacity measure	Size Range	Cost Correlation ($)
Shell & Tube heat exchanger	A (m^2)	80-4,000 m^2	$4.725\times10^4\times(\frac{A}{80})^{0.68}$
Centrifugal pump	$\dot{W}$ (kW)	4-700 kW	$14.1885\times10^3\left(\frac{\dot{W}_{pump}}{4}\right)^{0.55}$
Air-cooled condenser	A (m^2)	200-2,000 m^2	$2.2545\times10^5\times(\frac{A}{200})^{0.89}$
Condenser's fan	$\dot{W}$ (kW)	50-200 kW	$1.7685\times10^4\left(\frac{\dot{W}_{fan}}{50}\right)^{0.76}$
Expansion turbine	$\dot{W}$ (kW)	0.1-20 MW	$-2.241\times10^4+966.6\times\dot{W}_{turbine}^{0.8}$

6. Case Studies

In this work, four different Iranian compressor gas stations have been studied. Based on the size and technology of each turbine and environmental conditions, the flow and temperature of exhaust gases are varied. The specifications of these cases are presented in Table 2.

Table 2: Specifications of selected case studies

	Case 1	Case 2	Case 3	Case 4
Model	GTK-10-3	UGT16000	MS-5002C	SGT600
Turbine power (MW)	10	16.7	28.3	24.4
Fuel mass flow (kg/s)	0.86	1.12	2.1	1.6
Gas turbine thermal eff. (%)	25	32	29	34
Turbine exhaust mass flow (kg/s)	50	98	125	80.7
Turbine exhaust gas (˚C)	500	360	516	543
Ambient average temperature (˚C)	17	10	15	13
Average relative humidity (%)	78	68	74	63
Elevation (m)	1,373	1,796	1,373	1,000

7. Results

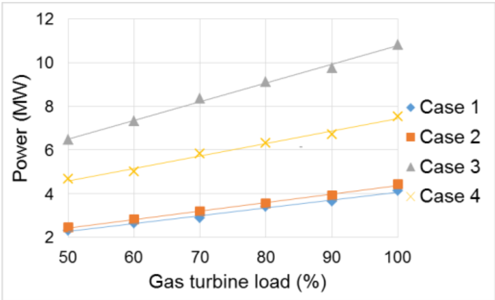
The results for designing of ORC for four cases including mass flow of oil and other thermodynamic specifications of the cycle are presented in Table 3. The technical and economical results are presented in Table 4 and show that simple payback periods for ORC are between 2.8 to 5.4 y; the payback period reduces as the ΔT_{min} increases. One of the factors changing the flow and temperature of exhaust gasses is turbine load. In this part, a base situation with a turbine load = 100 % and ΔT_{min} = 20 °C is selected in ISO bar conditions, then the effect of ΔT_{min} and the turbine load is investigated by fixing one of them equal to the base condition value and changing the other one and comparing the ORC generated power with the base condition (Figure 5).

Table 3: Thermodynamic specification of ORC designed for heat recovery from gas compressor stations

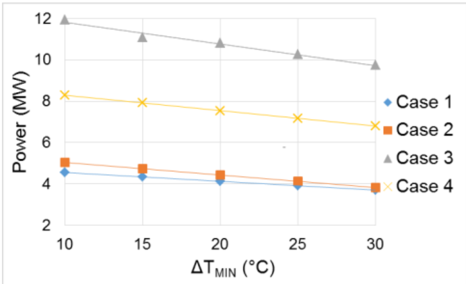
	ΔT_{min}	10 °C	15 °C	20 °C	25 °C	30 °C
Case 1	Mass flow of oil (kg/s)	73.00	72.60	72.26	72.14	71.85
	Temperature of stack (°C)	171.2	177.5	184.1	191.1	198.7
	Mass flow of working fluid (kg/s)	38.00	36.72	35.43	34.12	32.80
	Power of ORC (MW)	4.54	4.32	4.11	3.90	3.70
	Power of pump (kW)	176	170	164	158	152
	ORC's thermal efficiency (%)	24.6	23.9	23.3	21.6	20.1
Case 2	Mass flow of oil (kg/s)	81.00	79.30	77.80	76.20	74.76
	Temperature of stack (°C)	171.2	177.5	184.1	191.1	198.7
	Mass flow of working fluid (kg/s)	42.15	40.15	38.12	36.00	33.90
	Power of ORC (MW)	5.00	4.70	4.40	4.10	3.80
	Power of pump (kW)	195	186	176	167	157
	ORC's thermal efficiency (%)	23.5	22.8	22.2	21.5	20.7
Case 3	Mass flow of oil (kg/s)	192.00	190.80	190.10	190.00	189.00
	Temperature of stack (°C)	171.2	177.5	184.1	191.2	197.7
	Mass flow of working fluid (kg/s)	192.0	190.8	190.1	190.0	189.0
	Power of ORC (MW)	11.90	11.37	10.80	10.28	9.73
	Power of pump (kW)	462	447	432	416	400
	ORC's thermal efficiency (%)	23.7	22.9	22.2	21.6	20.9
Case 4	Mass flow of oil (kg/s)	133.5	132.9	132.6	132.0	131.8
	Temperature of stack (°C)	171.2	177.5	184.1	191.2	198.7
	Mass flow of working fluid (kg/s)	69.4	67.2	65.0	62.77	60.53
	Power of ORC (MW)	8.30	7.90	7.54	7.10	6.80
	Power of pump (kW)	321	311	301	290	280
	ORC's thermal efficiency (%)	23.7	22.9	22.2	21.3	20.9

Table 4: Results of economic analysis of ORC for heat recovery from gas compressor stations

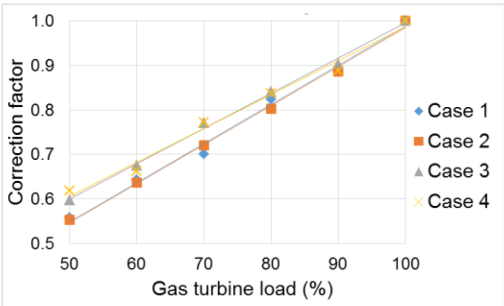
	ΔT_{min}	10 °C	15 °C	20 °C	25 °C	30 °C
Case 1	Total cost ($)	7,398,090	6,683,250	6,210,750	5,839,050	5,507,250
	Annual income ($/y)	1,644,000	1,554,000	1,552,000	1,536,000	1,508,000
	Payback time (y)	4.50	4.30	4.00	3.80	3.65
Case 2	Total cost ($)	10,522,050	9,507,750	8,753,850	8,362,200	7,595,700
	Annual income ($/y)	1,954,000	1,818,000	1,750,000	1,742,000	1,616,000
	Payback time (y)	5.38	5.23	5.00	4.80	4.70
Case 3	Total cost ($)	15,342,600	13,976,550	13,198,500	12,409,950	11,667,600
	Annual income ($/y)	4,614,000	4,508,000	4,399,500	4,279,000	4,167,000
	Payback time (y)	3.33	3.10	3.00	2.90	2.80
Case 4	Total cost ($)	11,608,000	10,503,000	9,780,000	9,232,000	8,741,000
	Annual income ($/y)	2,831,000	2,693,000	2,716,000	2,637,000	2,648,000
	Payback time (y)	4.10	3.90	3.60	3.50	3.30



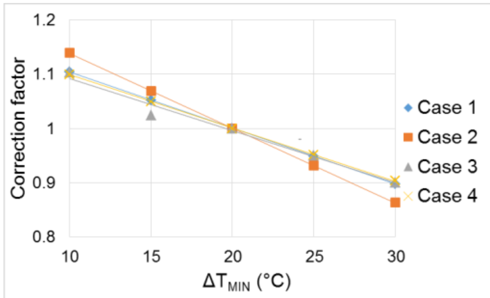
a) ORC's power in ΔT_{min}= 20 °C and different turbine load



b) ORC's power in turbine load =100 % and different ΔT_{min}



c) Correction factor of ORC's power in ΔT_{min}= 20 °C and different turbine load



d) Correction factor of ORC's power in turbine load =100 % and different ΔT_{min}

Figure 5: Investigating the influence of turbine load and ΔT_{min} on the ORC's power

Table 5: Correction factors' correlation of ORC's power

	$F_{load,w}$	$F_{\Delta T_{min},w}$	$\dot{W}_{ORC,base}$
Case 1	$F_{load,w}$ = 0.0092×Load(%)+0.0573	$F_{\Delta T_{min},w}$ = -0.0103×ΔT_{min}+1.207	4.29 MW
Case 2	$F_{load,w}$ = 0.0083×Load(%)+0.1389	$F_{\Delta T_{min},w}$ = -0.0138×ΔT_{min}+1.2761	4.64 MW
Case 3	$F_{load,w}$ = 0.0078×Load(%)+0.2074	$F_{\Delta T_{min},w}$ = -0.0096×ΔT_{min}+1.1872	11.28 MW
Case 4	$F_{load,w}$ = 0.0073×Load(%)+0.253	$F_{\Delta T_{min},w}$ = -0.0098×ΔT_{min}+1.1969	7.86 MW

Table 6: Results of verifying Eq(6) under the condition of turbine load = 70 % and ΔT_{min} = 30 °C

	$\dot{W}_{ORC,Simulated}$(MW)	$\dot{W}_{ORC,model}$(MW)	Error (%)
Case 1	2.66	2.7	1.5
Case 2	2.82	2.87	1.7
Case 3	7.53	7.64	1.46
Case 4	5.34	5.42	1.49

The ORC's power in different operation loads and ΔT_{min} can be calculated by the Eq (6) and the correlations for determining the correction factors and $\dot{W}_{ORC,base}$ are presented in Table 5.

$$\dot{W}_{ORC} = F_{load,w} \times F_{\Delta T_{min},w} \times \dot{W}_{ORC,base} \tag{6}$$

Using the ORC for heat recovery in gas compressor stations will generate power without emitting extra amount of carbon dioxide and other exhaust gases. The reduction of carbon dioxide is calculated by the Eq(7). The average benchmark for emitting CO_2 by generating electricity by small and medium-sized enterprises and power plants is 0.716 $\frac{kg_{CO2}}{kW.h}$ in Iran.

$$E_{CO2}\left(\frac{kg_{CO2}}{y}\right) = \dot{W}_{ORC}(kW) \times 8{,}000 \left(\frac{h}{y}\right) \times 700 \left(\frac{kg_{CO2}}{kW.h}\right) \tag{7}$$

To verify the models shown in Table 5, ORC is simulated under the sample condition of turbine load = 70 % and ΔT_{min}= 30 °C, then the results are compared with the results calculated by developed equation to determine the error of Eq(6). The results are presented in Table 6 and shows that Eq (6) and correction factors in Table 5 are precise and acceptable. The errors for other turbine loads and ΔT_{min} are also insignificant.

8. Conclusion

The results of investigating the application of heat recovery system for four different types of natural gas compressor stations by optimum ORC shows that total capital costs are in the range of 1,300 to 2,300 $/kW and the payback periods are in the range of 3 to 5 y. The sensitivity analysis has been carried out on the parameters affecting the ORC performance in compressor stations and a conceptual-mathematical based model has been developed for benchmarking of ORC performance and CO_2 emission in different operation conditions. The verifications show that the model is precise enough to apply for benchmarking of any other similar compressor stations. By the developed model, the engineers can simply calculate the energy savings and CO_2 reductions for any other cases.

Nomenclature

A	Area (m^2)	Ex	Expander	T	Temperature (˚C)
C_P	Heat capacity (kJ/kg.K)	f	Correction factor	wf	Working fluid
CON	Condenser	$\dot{m}$	Mass flow rate (kg/s)	$\dot{W}$	Power (kW)
E	Emission (kg/y)	ORC	Organic Rankine Cycle	WHR	Waste heat recovery
ECO	Economizer	REC	Recuperator	λ	Heat of vaporization (kW/kg)
EV	Evaporator	SH	Super heater	ΔT	Temperature approach

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