**Development of natural gas transmission network by four efficient energy recovery technologies**

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## Abstract

The objective of this research is to focus on developing a model for the improvement of gas transmission network overall efficiency with detailed characteristics of compressor stations and pressure reduction stations. In this study, we suggest three systems where a gas turbine is used. These cycles are: organic rankine cycle (ORC), air bottoming cycle (ABC) and ABC along with steam injection (SI-ABC). In addition, the study is conducted on a turbo-expander, as a good choice in place of expansion valves due to increment energy recovery in pressure reduction stations. In this paper, a real-world case study of a natural gas transmission network is investigated. It is concluded that the highest efficiency is obtained where ORC with n-pentane as the working fluid and turbo-expander are used in the case study. More ever, the improvement in efficiency is estimated to be 22% averagely for the existing network where the flow is in the span of 50 to 90 MMSCMD.

**Key words**: Natural gas transmission network, organic rankine cycle, air bottoming cycle, steam injection air bottoming cycle, turbo-expander, overall efficiency

1. **Introduction**

Natural gas is becoming one of the most widely used sources of energy in the world due to its environmental friendly characteristics and its demand as a primary energy is also increasing. It is predicted that the natural gas demand increases at an average rate of 2.4 percent annually until 2030 in the world [1]. Usually, the location of natural gas resources and the place where the gas is needed for various applications are far apart. As a result, gas has to be moved from deposit and production sites to consumers through the pipeline networks. Despite high demand, the gas transmission network overall efficiency is low. A main reason is that currently, lots of energy is wasted in the transmission network which is not recovered. As a result, the development of gas transmission network is a key issue in order to satisfy the ever growing demand from the various customers [2].

|  |  |  |  |
| --- | --- | --- | --- |
| Pressure (kPa) | P |  | **Nomenclature** |
| volume flow of natural gas (MSCMH) | Q | air bottoming cycle | ABC |
| heat duty (kW) | QHex | air compressor | AC |
| bottoming pressure ratio | rb | combustion chamber | CC |
| speciﬁc-gas constant (kJ /kg.K) | R | ideal gas specific heat (kJ/kgmol.K) | Cpig |
| speciﬁc entropy (kJ /kg.K) | s | Butane | C4 |
| entropy(kJ/K) | S | Pentane | C5 |
| steam injection ABC | SI-ABC | gas compressor | GC |
| ambient temperature (K) | T0 | speciﬁc enthalpy (kJ) | H |
| turbine inlet temperature (˚C) | TIT | enthalpy (kJ /kgmol) | H |
| acentric factor | w | heat recovery steam system | HRSG |
| work (kW) | W | mass ﬂow rate (kg/s) | M |
| compressibility factor  | Z | Million standard cubic meter per day | MSCMD |
| heater effectiveness factor | ε | Million standard cubic meter per hour | MSCMH |
|  Efficiency | η | organic rankine cycle | ORC |

When gas moves by using the transport network, it ﬂows through pipes and various devices such as regulators, valves and compressors. The gas pressure is reduced mainly due to friction with the wall of the pipe and heat transfer between the gas and the surroundings. Hence, compressor stations are usually installed to boost the gas pressure and keep it moving to the required destinations. The compression of gas is usually performed in centrifugal or reciprocating compressors driven by gas turbines or electric motors. Gas turbines are more common to use because of several reasons; first of all, the operating cost of electric motors is more than the cost of gas turbines. Secondly it could be costly to install a new interconnecting electric power transmission line, and finally it may be difficult to obtain the necessary regulatory approvals. However, it is estimated that 3-5% of the transported gas is consumed by the gas turbines as fuel [3, 4] and the exhaust gases from turbines with high temperature are released into the environment which lead to serious environmental pollution [5].

After passing all compression stations, at distribution points, the gas pressure must be reduced. Currently most pressure reduction stations use expansion valves which cause lots of destroyed pressure energy. Hence, energy recovery from pressure compression and reduction stations is considered as one of the basic elements to improve the overall efficiency of gas transmission network.

Martch et al. [6] in 1972, Edgar et al. [7] in 1978, Cobos-Zaleta et al. [8] in 2002, Ríos-Mercado et al. [9] in 2006 and Kabirian et al. [10] in 2007 presented different models or diverse solution procedures to optimize the gas transmission network. The objective of their models is minimization of cost or system energy consumption. The effect of pressure reduction station and the function of energy recovery from current gas transmission network were not considered in their work.

Minimizing the energy consumption and maximizing the ﬂow rate through pipes are the basic issues of this study and the main objective of this research is to maximize energy recovery and excess power by utilizing different technologies on compression and reduction stations. Evaluation of diverse technologies such as turbo-expander, organic rankine cycle, air bottoming cycle and steam air bottoming were studied previously [11-21]; It is worth mention that, this research has investigated all these systems together and represents a comprehensive model to compare them. As a result, the most efficient system is introduced.

In 2008, Maddaloni et al. [11] investigated a turbo expander which could be applied in gas pressure reduction station to produce electricity .the electricity can either be routed back into the electric distribution grid or used to produce small amounts of hydrogen. They showed at their assumed peak efficiencies, electricity can be extracted from the pressure reduction with 75% exegetic efficiency and hydrogen can be produced with 45% energetic efficiency.

In 2008, Cho et al. [12] studied a turbo expander in place of expansion valves in refrigerator or air-conditioner. They tested four different turbo expanders to find the performance characteristics of the turbo expander when they operate at a low partial admission rate. They showed a maximum of 15.8% total efficiency is obtained when the pressure ratio and the partial admission ratio are 2.37 and 1.70%, respectively.

In compression stations, one common solution for increasing the performance of a gas turbine is to combine it with a steam cycle or with an organic rankine cycle, either to generate electricity alone, or to cogenerate both electrical power and heat for industrial processes or district heating [13].

Organic ﬂuids are to be preferred to water when the required power is limited and the heat source temperature is low, as these ﬂuids often have lower heat of vaporization and can better follow the heat source to be cooled, thus reducing temperature differences and therefore irreversibilities at the evaporator. Furthermore, turbines for organic cycles can provide higher efﬁciencies at part loads as well and are usually less complex due to the lower enthalpy drop of the ﬂuid [14, 15].

In 2009, Desai et al. [16] proposed a methodology for accurate optimization of an ORC as a cogeneration process to generate shaft-work, for 16 different organic ﬂuids. In addition, they probed the beneﬁts of integrating ORC with the background process and the applicability of the proposed methodology, through illustrative examples.

In 2010, Roy et al. [17] studied parametric optimization and performance analysis of a waste heat recovery system along with Organic Rankine Cycle, for power generation. This analysis was performed for R-12, R-123 and R-134a as working ﬂuids. They also showed that R-123 has the maximum work output and efﬁciency among all the selected ﬂuids.

Combining the gas turbine cycle with an air bottoming cycle (ABC) is another method that has been introduced to increase the performance of a gas turbine [18]. In 1995, Kambanis [19] and in 1996, Bolland [20], showed that by using the exhaust gas of a simple gas turbine in the air bottoming cycle the efﬁciency of the combined cycle improved about 47% and 46.6% respectively.

In the recent study in 2011, Ghazikhani et al. [21] developed a model for steam injection in the gas turbine with air bottoming cycle. They also suggested two new cycles with ABC, These cycles are: the Evaporating Gas turbine with Air Bottoming Cycle (EGT-ABC), and Steam Injection Gas turbine with Air Bottoming Cycle (STIG-ABC). They showed that EGT-ABC have a lower irreversibility and higher output work when compared to the STIG-ABC.

We have developed a model for the technical analysis of transmission network incorporating pressure compression and reduction stations and the energy recovery technologies characteristics. The developed model estimates extractable net output power, overall efficiency and system energy loss to evaluate the performance of the gas transport network.

It is worth note that, the model utilizes equations of actual gas for estimation of enthalpy causes accurate results and would no longer need Moulier graphs.

1. **Methodology**
	1. **Definition of Transportation efficiency**

Transportation efficiency is a function of the overall system design, the efficiency of individual components, and how the system is operated. Transportation efficiency is measured in terms of fuel or electric power burned per unit of throughput (i.e., British thermal unit (Btu) or kW/Mcf). Within this general definition of transportation efficiency, there are three other pertinent measures.

1. Hydraulic efficiency is a measure of the loss of energy (pressure drop) caused by the friction of the flowing gas in the pipeline facilities.

2. Thermal efficiency is applied to a prime mover (engine, turbine or motor), thermal efficiency measures how much of the potential energy of an input fuel or electric power is converted into useful energy that can be used to drive a compressor. The majority of energy that is not converted into useful energy is considered “waste heat” in the exhaust.

3. Compressor efficiency measures how much energy is expended in compressing the gas compared to how much overall energy is used by the compressor. Inefficient compressors heat the gas instead of raising its pressure and thus have lower efficiency values.

* 1. **Definition of the model**

The model was developed for existing gas pipelines networks from supply nodes to demand nodes. The demand nodes are major locations of consuming natural gas in the study area. The demand nodes are either consumption regions in the study area or export terminals of natural gas from the study area to outside. In contrast, supply nodes are locations of resources of processed natural gas in the study area. These nodes are either reﬁneries or natural gas producing plants or the import terminals of natural gas from outside of the study area [10].In addition the model considers all network units i.e. gas compressor, air compressor, gas turbine, combustion chamber, expansion valves and heater in compressor stations and pressure reduction stations.

The model was comprised of four sub-models that are defined below:

1. Base scenario

2. Reduction stations along with turbo-expander

3. Compression stations affixed to ORC

4. Compression stations affixed to ABC and SI-ABC

The main thermodynamic assumptions that have utilized in the present analysis are reported in Table 1. Iran second gas transmission network is chosen for the case study. The considered network whit 7 major compression stations, is one of the most important network systems in Iran

* 1. **Base scenario**

In this section, gas transmission network is considered without energy recovery and it includes simple pressure compression and reduction stations. In this sub-model the model minimizes the energy used in the gas compressors. This can be written [22]:

 (1)

Where α is the unitary energy price ($/kW), ηtherm is the compressor thermic efficiency and Wi, j is compressor required power. Wi, j is calculated from [23]:

 (2)

Where T1, P1 and Z1 are input temperature, pressure and compressibility factor, T2, P2 and Z2 are output temperature, pressure and compressibility factor, respectively. ηa is the compressor adiabatic efficiency, Q is gas flow rate through the pipeline (MMSCMD) and γ is ratio of specific heats that is constant.

According to Weymouth equation, input conditions, designed maximum and minimum pressure and type of pipe material, the maximum flow rate through the pipeline must be calculated; always the demand should be lower than maximum flow rate to satisfy the restrictions of flow speed.

 (3)

 (4)

 (5)

Also, at each exit point, the demand must be satisﬁed at a minimal pressure guaranteed. On the other hand, the gas transmission company cannot take gas at a pressure higher than the one insured. Mathematically [22]:

 (6)

Fig. 1 shows a schematic of gas compression station.

The inlet air enters the compressor at state1. Considering an isentropic efficiency of ηcomp for the compressor and a constant pressure ratio of rc that can be calculated as:

 (7)

Because of high pressure of inlet natural gas, the reduction valve must be used. At next steps compressed air combusts with medium pressure fuel. Consequently exhaust gases exit from combustion chamber and enters the turbine. At final state, turbine output work is obtained which depends on exhaust temperature (TIT) and mass flow. Turbine output power can be estimated from equation 2 but, the outcome of this equation is approximate, so this model has exploited state equations of actual gas for calculation of output power which they have shown below [24]:

 (8)

Which:

 (9)

Every component has a unique that it is -74920 (kJ/kgmol) for methane. Second part of equation 9, can be simplified to:

 (10)

 (11)

Table 2, shows all methane constants for calculation of specific heat. Last part of equation 9, is called residual enthalpy which appears when natural gas is actual. On the other hand, gas pressure is more than atmospheric.

 (12)

 (13)

 (14)

 (15)

Where Z, Tr and w are compressibility factor, reduced temperature and acentric factor, respectively.

In the same way, the power for other units in compression station is shown below:

 (16)

 (17)

 (18)

The next parameter that is different in actual and ideal conditions is entropy. In this way actual entropy is written as:

 (19)

This character will exert in next section. is the standard entropy of ideal gas that is constant for each component and it is 183.48 (kJ/kgmol.K).for methane.

 (20)

Passing all compression stations, near demand nodes, the gas enters to gas pressure reduction stations. The process that currently performs in reduction stations is shown in Fig. 2. At the first stage gas enters to the heater to make up reducing temperature during expansion process and then it passes under the constant enthalpy process in Joule-Thomson valves.

The model estimates dissipation rate of pressure energy in expansion valves, required heat duty and total loss of energy in reduction stations. The result of this part gives a view that how much energy is being wasted in gas pressure reduction stations. This can be written:

 (21)

 (22)

* 1. **Affix of turbo-expander to reduction stations**

In this research we suggest applying turbo-expander instead of J-T valves and have created a model to calculate output power and heat duty of turbo-expander system. Fig. 3 shows a simple schematic of an improved station.

Equations 9-15, 19 and 20 are the base exploited equations in this part. Inlet conditions of heat exchanger and outlet conditions from turbo expander are key inputs. Considering an isentropic efficiency of ηexp for the expander and pressure drop of 1.46(%) during heater.

The most important operational problem of turbo-expander is that hydrate formation due to the slight amount of water in gas. Two factors that intensify hydrate formation are low temperature and high pressure. So a proper temperature for outlet heat exchanger should be found out. At the first step, expander is considered as isentropic process and then by means of trial and error method for T2 and equation of expander efficiency, the model can represent the correct answer for T2.

 (23)

The output power and required heat can be obtained as follows. Having considered the heat exchanger pressure drop, turbo-expander efficiency, fuel mass ﬂow rate and generator and gearbox efficiency we can have:

 (24)

 (25)

The thermodynamic properties of natural gas are used in the program and the procedure of model solution is given in flow chart 1 in the Appendix.

* + 1. **Validation**

To validate the first sub-model, the results of the program are compared to those of the experiments performed by Pozivil. [25] and also to the simulation results. Fig. 4 and 5 show the comparison of model results with those of the simulation and Fig. 6 shows a comparison of model results with experiments performed by Pozivil. For the output power and heat duty against temperature inlet expander and inlet flow of natural gas, a good agreement is observed. The discrepancy between the two results is less than 6%.

* + 1. **Sensitivity analysis**

A sensitivity analysis is conducted in order to better understand the effect of key-parameters of process performance. In this analysis, the effect of an additional percentage of parameters P1, T1 and flow rate on the output power and heat duty are investigated using the model. The expander outlet pressure and temperature are 1825 Kpa and 18 ºC, respectively.

Fig. 7 and 8 show the details of the sensitivity analysis. Two Figs were plotted for capacities of 37, 139.73 and 371 MSCMH.

As seen in Fig. 7, although output power and required heat have a direct relationship with inlet pressure and flow, when inlet pressure is close to outlet pressure, the expansion system efficiency will be greater due to the difference between two graphs is lower.

 (26)

It is evident in Fig. 8 that expander power and heat duty are more sensitive to inlet temperature, so with using boiler in maximum load more power would be obtained without extra cost. In addition, there is a specific temperature for each inlet pressure where output power and required heat are equal; on the other hand expansion system efficiency is 100%.

* 1. **Affix of ORC to compression stations**
		1. **Thermodynamic analysis of ORC**

The ORC system consists of an evaporator, turbine, condenser and pump. It can be classiﬁed in two groups according to the level of turbine inlet pressure, including supercritical ORCs and sub-critical ORCs [26]. In the present study, the sub-critical ORCs are investigated.

As is shown in Fig.9, the working ﬂuid leaves the condenser as saturated liquid (point 1). Then, it is compressed by the liquid pump to the sub-critical pressure (point 2). The working ﬂuid is heated in the evaporator until it becomes superheated vapor (point 3). In this research, heating process to working fluid was considered indirect. In other words, an inductor fluid such as oil over takes heat transfer to working fluid. So security and management are increased by this way. The superheated vapor ﬂows in to the turbine and expands to the condensing pressure (point 4), and then, the low pressure vapor is led to the condenser and condensed by air. The condensed working ﬂuid ﬂows in to the receiver and is pumped back to the evaporator, and a new cycle begins.

In the mentioned cycle, if the temperature t4 is markedly higher than the temperature t1, it may be rewarding to implement an internal heat exchanger (IHE) in to the cycle as shown in Fig.10 this heat exchanger is also represented in Figs.9 by the additional state points 4a and 2a. The turbine exhausts ﬂow in to the internal heat exchanger and cool in the heat exchanger in the process (4–4a) by transferring heat to the compressed liquid that is heated in the process (2–2a) [27].

Each process in the ORC can be described as follows:

Process 2 to 3: This is the heat absorption process in the evaporator. The pressure drop due to evaporator is considered. The heat transferred from the waste heat to the working ﬂuid is [28]:

 (27)

Or

 (28)

If the internal heat exchanger is added, the amount of heat transfer is presented by:

 (29)

Process 3 to 4: This is a non-isentropic expansion process in the turbine. Ideally, this is an isentropic process 3–4s. However, the efﬁciency of the energy transformation in the turbine never reaches 100%, and the state of the working ﬂuid at the turbine outlet is indicated by state point 4. The isentropic efﬁciency of the turbine can be expressed as:

 (30)

The power generated by the turbine can be given as:

 (31)

Process 4 to 1: This is a constant pressure heat rejection process in the condenser.

Process 1 to 2: This is a non-isentropic compression process in the liquid pump. The isentropic efﬁciency of the pump can be expressed as:

 (32)

The input work by the pump is:

 (33)

The thermal efﬁciency of the ORC is deﬁned on the basis of the ﬁrst law of thermodynamics as the ratio of the net power output to the heat addition.

 (34)

The procedure of model solution is given in Flow chart 2 in the Appendix.

* + 1. **Validation**

To validate the model, the results of the program are compared to those of the experiments performed by Dai.et al [5]. Fig. 11 shows a comparison of model results with experiments performed by Dai for 3 different working fluids. Fig. 11 shows net power output against turbine inlet temperature where a good agreement is observed. The discrepancy between the two results is less than 5%. It is obvious that as the turbine inlet temperature increases, the net output power for ammonia and water increase correspondingly, but for the butane, an increase in turbine inlet temperature leads to a reduction in net output power. For instance when TIT varies from 90 to 135 (°C) the efficiency of ORC-butane is decreased 7.9 % averagely, although it is increased 7.2 and 2.1 % for ammonia and water, respectively. Consequently, for high TIT ammonia is better choice among these three working fluids.

* + 1. **Sensitivity analysis**

A sensitivity analysis is conducted for n-pentane as working fluid to better understand the effect of key-parameters of process performance. Fig. 12 shows the effect of gas turbine outlet temperature on output power of n-pentane turbine and ORCs efficiency. The output work is increased by increasing flu gas temperature, because more amount of n-pentane can be evaporated by this way. But the ORCs efficiency is decreased due to increase of Q in equation 34.

As seen in Fig.13 the extractable work is grown by increment of flu gas rate, because of increase in heat absorption process in the evaporator. The net output power is enhanced too. In addition, Fig.13 shows a changeless trend for ORCs efficiency due to the ratio of increase of net output power and required heat is constant.

The effect of n-pentane turbine inlet pressure on net output power and ORCs efficiency at constant inlet temperature (490 °C) is displayed in Fig.14. The net output power and ORCs efficiency are augmented by accretion of inlet pressure.

* 1. **Affix of ABC and SI-ABC to compression stations**

An Air Bottoming Cycle system consists an air compressor, regenerator and turbine. Fig.15 shows such a combined cycle in which the exhaust of an existing, topping gas turbine is sent to a gas-air heat exchanger that heats the air in the secondary gas turbine cycle.

ABC was proposed in the late 1980s as an alternative for the conventional steam bottoming cycle. Now this cycle is being considered as a compact and simple bottoming cycle in various applications: as an upgrading option for simple-cycle gas turbines in the oﬀshore industry; as a hot-air cogeneration plant; and as a heat recovery installation at high-temperature furnaces.

Fig.16 shows Steam Injection Gas Turbine with Air Bottoming Cycle (SI-ABC) too; the topping exhaust gases have high temperature after passing through the regenerator. Thermal energy of these gases can be used for evaporating water. The steam is then mixed with ABC compressor discharged air in a mixer [21].The evaporating process is done in the HRSG. The temperature of the exhaust gas is decreased in about 120 °C by generating steam. The amount of the injected steam per unit fuel ﬂow is 5-6 (kg/kg-fuel) [29].

The computer model developed in this study includes the calculation of three cycles: simple gas turbine, ABC and SI-ABC. Each process in the ABC can be described as follows:

Process 1 to 2: This is the heat absorption process in the regenerator. The pressure drop due to regenerator is considered. Enthalpy of flu gas is calculated by equations 9-15. Equation 35 is utilized for estimation of air outlet temperature from regenerator.

 (35)

Process 3 to 4: The inlet air enters the compressor at state 3. The compressor inlet power can be calculated as:

 (36)

Air outlet temperature from regenerator is function of outlet enthalpy which is estimated by energy balance equation around regenerator.

 (37)

Process 5 to 6: This is a non-isentropic expansion process in the turbine. The isentropic efﬁciency of the turbine can be expressed as:

 (38)

The power generated by the turbine can be given as:

 (39)

SI-ABC process is similar to ABC; just the differences between the ABC gas turbine and SI-ABC are the HRSG and a mixer which provides steam for the bottoming cycle. The procedure is given in Flowchart 3 and 4 in the Appendix for ABC and SI-ABC.

* + 1. **Validation**

To validate the model, the results of the program of ABC and SI-ABC are compared to those of the experiments performed by Ghazikhani. et al [21]. Fig.17 displays variations of compression station overall efficiency with ABC or SI-ABC against TIT. SI-ABC system has more output power than ABC at the same bottoming cycle pressure ratio and TIT. This is due to more heat recovery in the regenerator in the SI-ABC cycles results in a lower exhaust temperature and more inlet mass to bottoming turbine. In addition, form Fig.17 a good agreement and low error is observable. The discrepancy between the two results is less than 4%.

* + 1. **Sensitivity analysis**

The effect of key parameters of process on performance is evaluated by a sensitivity analysis. In Fig.18, the thermal efficiency of the ABCs and SI-ABCs are varied against bottoming pressure ratio. The thermal efficiencies of steam injection system are higher than ABCs. Fig.19 shows, the efﬁciency reduces as the ambient temperature is increased. In addition, in the ABC, the reduction rate of efﬁciency with ambient temperature is steeper. In ABCs the effectiveness of the heat recovery in the bottom-cycle is also decreased by increasing ambient temperature due to a smaller difference between the two stream temperatures. Fig.19 also shows the advantage of the SI-ABC for having the highest efﬁciency in different ambient temperatures.

1. **Results and discussion**

The model has implemented for Iran second gas pipeline network. Table 3 shows the properties of this network. Because of the demand is different in diverse seasons and it is an important variable that has to be satisfied, the model has performed for different demands. Optimization of the required power for gas compressors in various demands is the first step of the model for minimizing of the total power consumption. Then the model uses the energy recovery technologies within the pressure compression and reduction stations to estimate the improved overall efficiency and net output power. Table 4 displays the optimization results for simple network in the case study. The demand and the pressure ratio for all gas turbines was considered 50 MMSCMD and 14, respectively.

Having a clear comparison between the referred systems, the variation of gas transmission network overall efficiency against the demand, was shown in Fig. 20. The comparison includes six systems:

1. Simple pipeline

2. Simple pipeline with the turbo-expander in reduction stations

3. Simple pipeline with ABCs in compression stations and turbo-expander in reduction stations

4. Simple pipeline with SI-ABCs in compression stations and turbo-expander in reduction stations

5. Simple pipeline with ORC-butane in compression stations and turbo-expander in reduction stations

6. Simple pipeline with ORC-pentane in compression stations and turbo-expander in reduction stations

Fig.20 shows the overall efficiency of the equipped transmission network by ORC-pentane and turbo-expander is higher than other systems at the same flow rate, and these technologies increase the network overall efficiency 13-28 % in span of 50- 90 MMSCMD. As seen in Fig.20, the overall efficiency of SI-ABCs is higher than ORC-butane at lower 65 MMSCMD. The ﬁgure also shows, there is a slight difference among simple and improved network by turbo-expander at low flow rate through the pipelines but, the turbo-expander has a great impact on the increase of the overall efficiency transmission network in cold seasons, when demand is high.

Fig.21 displays the increase of the overall efficiency with bottoming pressure ratio (rb) in the range of 5-15 for all systems. The flow rate was considered 70 MMSCMD. As seen in Fig.21, the overall efﬁciency of the ORC-pentane is more sensitive to bottoming pressure ratio than other systems.

1. **Conclusions**

In the present study, the effect of three technologies on the overall efficiency of gas transmission network has been investigated. Based on an energy analysis a computer program has been developed to survey improving the performance of gas transmission system. The examined technologies are: organic rankine cycle, air bottoming cycle and steam injection air bottoming cycle that are use in the pressure compression station and turbo-expander which is utilized within the pressure reduction stations.

The main conclusions can be summarized as follows:

1) The turbo-expander outlet powers and required heat have a direct relationship with expander inlet pressure, temperature and flow. In addition, when the inlet pressure ranges from 450 to 750 Pisa and gas flow is maximum i.e. 866 MSCMH, the efficiency of the expander system is 82-44%, respectively.

2) The SI-ABC was found to have maximum output power at the same bottoming cycle pressure ratio and turbine inlet temperature (TIT) in comparison with ABS. This is due to more heat recovery in the regenerator in the SI-ABC cycle that results a lower exhaust temperature; and more inlet mass to bottoming turbine causes a higher output work. More ever, the results display that the overall efﬁciency is decreased as the ambient temperature is increased for ABC and SI-ABC.

3) In this study, an organic rankine cycle using working ﬂuid such as ammonia, butane and water has been analyzed and the results were compared together. In addition, when turbine inlet temperature varies from 90 to 135 (°C) the efficiency of ORCs with butane as working fluid is decreased 7.9 % averagely, although it is increased 7.2 and 2.1 % for ammonia and water, respectively.

4) The ORC-pentane is more sensitive to variation of bottoming pressure ratio.

The model has performed for Iran second gas transmission network. The results show the highest efficiency is obtained from implementation of ORC with n-pentane as working fluid and turbo-expander in pressure compression and reduction stations, respectively. The demand or gas flow rate through pipeline is the most variable that needs to satisfy, therefore variation of the transmission system overall efficiency has been investigated based on flow rate.

When the study network is equipped by ORC-pentane and turbo-expander, the overall efficiency is grown 22 % averagely, in the span of 50 to 90 MMSCMD.

**APPENDICES**

T2NEW

T3S

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**Flowchart 1:** Turbo-expander calculation

**Flowchart 2:** ORCs calculation

ηreg=

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**Flowchart 3:** ABCs calculation

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**Flowchart 4:** SI- ABCs calculation

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