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Evaluation of Natural Gas Energy Recovery Powering Distribute with Turbo-Expander

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Abstract: A study is conducted on a turbo-expander as a good choice in place of expansion valves due to improvement of the cycle efficiency by recovering energy in gas pressure reduction stations (GPRS) on transmission pipelines. A model has been developed to calculate the output power and required heat for different inlet conditions based on Thermodynamics parameters. It is worth mention that, the model utilizes equations of actual gas for estimation of enthalpy causes accurate results and would no longer need Moulier graphs. The model predicts 476 MW excess power production by installing turbo-expander (with a pressure ratio 2.5) in all of Iran`s GPRS. The required heat is a function of hydrate formation temperature in gas; if the expander outlet temperature has been selected close to its limit temperature i.e. hydrate formation temperature, maximum power efficiency can be achieved for each flow rate entering the facility and also minimum heat duty will be needed. In addition, since the gas flow rate is maximum (i.e. 866 MSCMH) and inlet pressure is varied from 3800 to 5200 kPa, the efficiency of the expander system varies from 82 to 44%, respectively.
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Keywords: turbo-expander; energy recovery; gas pressure reduction station; modeling

1. Introduction

Natural gas is one of the hydrocarbon compounds which has been recognized as a superior energy in many countries. Distribution of natural gas in diverse areas has led to use of different transportation and distribution systems, that network systems and liquefaction are two common methods. In Iran, gas pipeline is the most economical method for transmission, in this way first gas is achieved to high pressure in compression stations and then dispenses. Gas pressure is increased about 8273 kPa but it has to decrease to suit consumer needs, the reduction occurs in gas pressure reduction stations (GPRSs). Currently, this action is accomplished by expansion valves which cause wasting large amount of energy [1].

Todays power recovery applications are increasing due to a change in market conditions driven by growing environmental awareness and an increase in power costs [2 and 3]. Also lots of energy recovery in the GPRSs is possible by applying of turbo-expander instead of reduction valves.

In 2007, Maddaloni et al. [4] quantified the energy that can be extracted from various pressure reduction facilities using an expander coupled to an electric generator. Produced electricity can either be routed back into the electric distribution grid or used to produce small amounts of hydrogen. A problem with this process is the variable nature of the gas flow rate entering the facility so they represented a model to analyze the seasonal variations and produce functions that allow the hydrogen production potential of any pressure reduction facility. They showed if the coupled technologies operate at their assumed peak efficiencies, then 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. [5] investigated a small turbo expander which could be applied to the expansion process instead of expansion valves in refrigerator or air-conditioner to improve the cycle efficiency by recovering energy from the throttling process. 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 at the partial admission rate of 1.70% or 2.37%, expanders are operated in the supersonic flow. In addition, a maximum of 15.8% total to-static efficiency is obtained when the pressure ratio and the partial admission ratio are 2.37 and 1.70%, respectively.

The process that currently performs in reduction stations is shown in Figure 1.a. First 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.

In this research, applying of turbo-expander instead of J-T valves has been investigated and a model introduced to calculate output power and heat duty of turbo-expander system. Figure 1.b shows a simple schematic of the developed station.

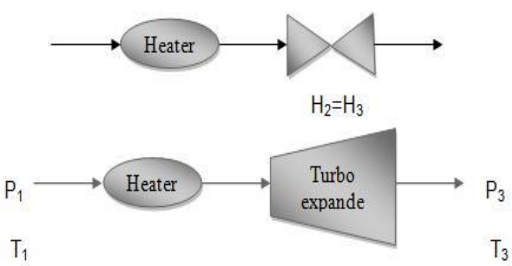

Figure 1: a. The schematic of conventional station.
b. The schematic of developed station

Nomenclature

Cp^{ig}	ideal gas specific heat (kJ/kgmol.K)
GPRS	gas pressure reduction station
H_1	inlet enthalpy of heater (kJ/kgmol)
H_3	outlet enthalpy of expansion (kJ/kgmol)
H_{3C}	expansion outlet computed enthalpy (kJ/kgmol)
H_{3S}	expansion outlet isentropic enthalpy (kJ/kgmol)
H_0^{ig}	ideal gas standard enthalpy (kJ/kgmol)
H^R	extra enthalpy (kJ/kgmol)
P_1	inlet pressure of heater (kPa)
P_2	outlet pressure of heater (kPa)
P_3	outlet pressure of expansion (kPa)
P_c	critical pressure (kPa)
Q	volume flow of natural gas (MSCMH)
Q_{Hex}	heat duty (kW)
S_2	expansion inlet entropy (kJ/kgmol.K)
S_3	expansion outlet entropy (kJ/kgmol.K)
S_0^{ig}	ideal gas standard entropy (kJ/kgmol.K)
S^R	extra entropy (kJ/kgmol.K)
T_0	Ambient temperature (K)
T_1	heater inlet temperature (K)
T_2	heater outlet temperature (K)
T_3	expansion outlet temperature (K)
T_c	critical temperature (K)
T_r	reduction temperature (K)
W_{exp}	turbo-expander power (kW)
m_{NG}	natural gas flow (kg/s)
r_{exp}	turbo-expander pressure ratio
w	acentric factor
z	compressibility factor
η_{exp}	expansion system efficiency (%)
η_{GB}	gearbox efficiency (%)
η_{Gen}	generator efficiency (%)

2. Computer model

The main assumption that has been used in the computer model is natural gas includes only methane. This is a good assumption since methane forms approximately 90% of natural gas. Other thermodynamic assumptions that have been utilized in the present analysis are reported in Table 1.

Table 1. Thermodynamics assumptions

Variable		Value
Expander efficiency	η_{exp}	0.75
Gearbox efficiency	η_{GB}	0.92
Generator efficiency	η_{Gen}	0.95
Ambient temperature	T_0	298 K

2.1. Calculation

The computer model developed in this study includes the calculation of expander output power and required heat for heat exchanger. The calculation procedure is as follows. Inlet conditions of heat exchanger and turbo expander outlet conditions are key inputs. Considering an isentropic efficiency of ηexp for the expander and pressure drop of 1.46% during heater, the expander outlet can be calculated as [6]:

$$H = H_0^{ig} + \int_{T_0}^{T} Cp^{ig} dT + H^R$$

Each component has a unique . For methane this value is -74920 kJ/kgmol. Second part of equation 1, can be simplified to:

$$\int_{T_0}^{T} Cp^{ig} dT = (Cp^{ig})_H \times (T - T_0)$$

$$(Cp^{ig})_H = R(A + \left(B/2 \times T_0 \times \left(T/T_0 + 1\right)\right)$$
$$+ \left(\left(T/T_0\right)^2 + T/T_0 + 1\right) \times C/3 \times T_0^2$$
$$+ (D/T_0 \times T))$$

Table 2, shows all methane constants for calculation of specific heat.

Table 1. Methane constants

constants	Value
A	1.702
B	9.081e-3
C	-2.164e-6
D	0
T_C (K)	190.6
P_C (kPa)	4599000
w	0.012

Last part of equation 1, is called residual enthalpy which appears when natural gas is actual. On the other hand, gas pressure is more than atmospheric.

$$\frac{H^R}{RT} = Z - 1 + \left(\frac{dLna(T_r)}{dLnT_r} - 1\right) \times q \times I$$

$$q = \frac{a(T)}{bRT}$$

$$I = \frac{\beta(T,P)}{Z + \varepsilon\beta}$$

$$a(T) = c \times (1 + a_1 \times a_{2)})^2$$

$$a_1 = 0.37464 + 1.54226\, w - 0.26992\, w^2$$

In this way actual entropy is written as:

$$S = S_0^{ig} + \int_{T_0}^{T} Cp^{ig} \frac{dT}{T} + S^R - RLn\frac{P}{P_0}$$

is the standard entropy of ideal gas that is constant for each component and for methane is, 183.48 kJ/kgmol.K.

$$\frac{S^R}{R} = Ln(Z - \beta) + \frac{dLna(T_r)}{dLnT_r} \times q \times I$$

The most important operational problem of turbo-expander is that hydrate formation is possible 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. In the first step, expander is considered as isentropic process and then the model represents the correct answer for T_2 by equation of expander efficiency and trial and error method.

$$\eta_{exp} = \frac{H_2 - H_3}{H_2 - H_{3s}}$$

The calculation was performed with different inlet conditions. The output power and heat duty can be obtained as follows. Having considered the heat exchanger pressure drop, turbo-expander efficiency, mass flow rate of the fuel and generator and gearbox efficiency we can have:

$$\dot{W}_{exp} = \dot{m}_{NG}(H_2 - H_3).\eta_{GB}.\eta_{Gen}$$

$$\dot{Q}_{Hex} = \dot{m}_{NG}(H_2 - H_1)$$

In this work, Visual FORTRAN 6 software was employed to perform the calculations. The thermodynamic properties of natural gas are used in the program. The procedure of model solution is given in flow chart1 in the Appendix.

2.1. Model validation

To validate the model, the results of the program are compared to those of the experiments performed by Pozivil [7]. Figure 2 shows a comparison of model results with experiments performed by Pozivil. For the output power and heat duty against expander inlet temperature a good agreement is observed. As the temperature increases the obtainable power and required heat are also increased. The discrepancy between the two results is less than 4%.

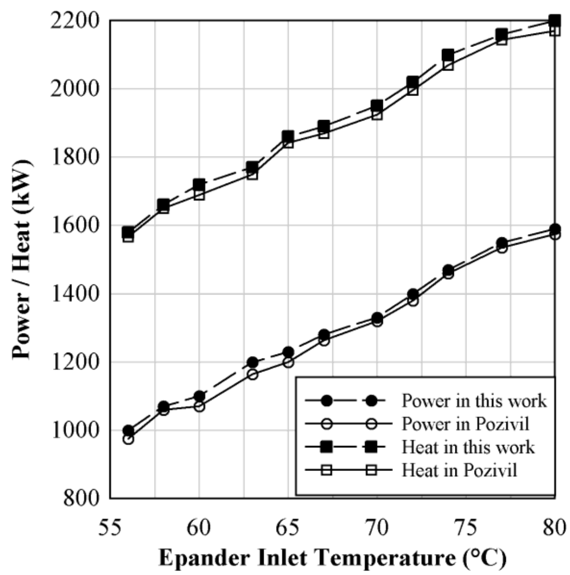

Figure 2: Comparison of model results with those of Pozivil. [7]

3. Results and discussion

Table 3 shows the necessary information of all Iran's GPRSs [8]. The highest capacity is in the range of 10-50 MSCMH that was considered 37 MSCMH in calculations as its average capacity.

Table 2. Information of Iran's GPRSs

Capacity (MSCMH)	Number	Average capacity
< 10	78	9
10-50	218	37
50-100	23	72.381
100-200	53	139.73
200-500	12	371
500-1000	6	866

Currently, expansion valve is used in all Iran's GPRSs which has caused lots of energy to squander. If turbo-expander is installed in place of expansion valve, it is possible to utilize the pressure potential to obtain power and prevent waste of energy.

The results of the model for Iran's GPRSs, based on average capacity are given in Table 4. The P1, rexp and T3 were considered 4466 kPa, 2.5 and 18 °C, respectively, and other assumptions were given in Table 1.

As seen in the table 4, about 476 MW power is extractable in the case study. In addition, the generated power is grown by increasing of pressure ratio or inlet flow.

4. 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 P_1, T_1 and Q on the output power and heat duty are investigated using Visual FORTRAN 6 software. The outlet pressure and temperature of expander are 1825 kPa and 18 °C, respectively for the sensitivity analysis.

Figures 3 and 4 show the details of the sensitivity analysis. Two figures were plotted for capacities of 37, 139.73 and 371 MSCMH.

As seen from Figure 3, energy has a direct relationship with inlet pressure and flow, moreover, since the inlet pressure is close to outlet pressure, the expansion system efficiency which is defined as the ratio of output power to heat duty, will be greater.

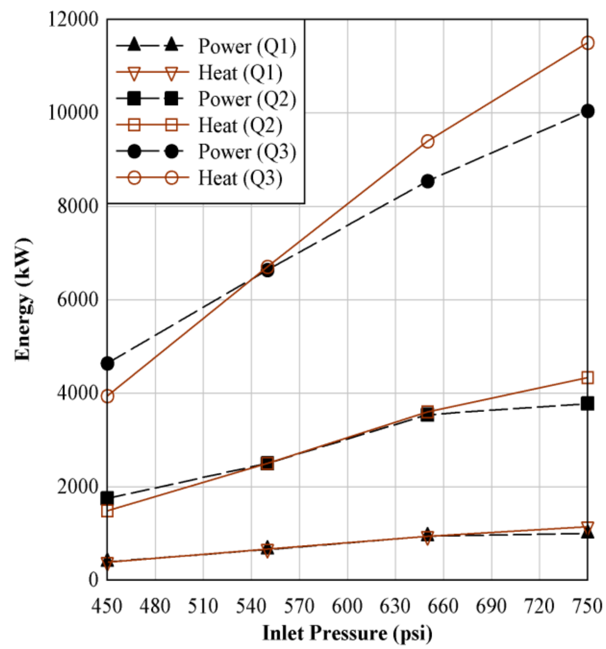

Figure 3: Effect of inlet pressure on the energy (Q_1=37, Q_2=139.73, Q_3=371 MSCMH, T_1=30 $^{\circ}C$, P_3=1825 kPa, T_3=18 $^{\circ}C$)

It is evident from Figure 4 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; so at this point the expansion system efficiency is 100%.

Capacity (MSCMH)	Number	Average capacity	Average power (MW)	Required heat (MW)	Stations total power (MW)	Country total power (MW)
< 10	78	9	0.15295	0.157	11.9301	
10-50	218	37	0.631028	0.647	137.5641	
50-100	23	72.38	1.262056	1.304	29.02729	476.025
100-200	53	139.73	2.437586	2.518	129.1921	
200-500	12	371	6.47197	6.687	77.66364	
500-1000	6	866	15.107964	15.609	90.64778	

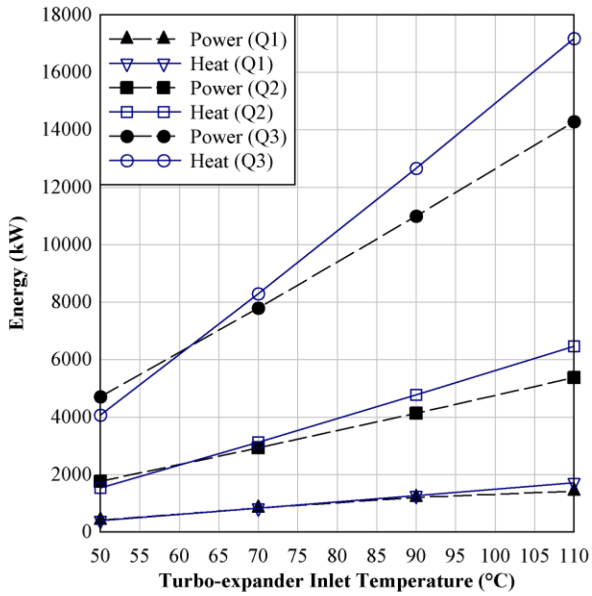

Figure 4: Effect of inlet temperature on the energy (Q_1=37, Q_2=139.73, Q_3=371 MSCMH, P_1=4466 kPa, T_1=30 °C, P_3=1825 kPa, T_3=18 °C)

There is little water in natural gas, so freezing is possible. Small ice crystals can make corrosion and other problems in pipelines. Prediction of hydrate formation is based on dew temperature of pure water. Figure 5 shows the relationship between dew temperature of hydrocarbon and water [9]. This curve is for a sample of natural gas whose mole fractions are: 88% methane, 5% ethane, 2% propane, 1% butane, 2% nitrogen and 0.005% water.

As seen from Figure 5, at pressure less than 70 bar, dew temperature of water is lower than that of natural gas.

In this research dew temperature of natural gas in gas pipeline is considered about -11 °C. So the maximum outlet temperature of turbo-expander can be -5 °C since the temperature of hydrate formation is -24.9 ˚C.

Figure 6 shows the effect of the lowest possible outlet temperature against expander power and heat duty for maximum average capacity. Despite the reduction of output power, the expansion system efficiency is higher than other conditions; therefore cost of production power is lowest and it is more efficient.

5. Conclusions

In this study, the effect of turbo-expander system instead of expansion valve has been investigated in all Iran's gas pressure reduction stations. Based on thermodynamic equations of actual gas, a computer program has been developed to investigate the improvement of the GPRS performance. Several conditions are examined: various inlet pressures, temperature and flow of natural gas and also the effect of hydrate formation against energy has been investigated. The results of the model show that if turbo-expander with a pressure ratio of 2.5 is installed in place of expansion valve in all Iran's GPRS, it can be produced about 476 MW power. This is due to pressure potential utilization.

The output powers of turbo-expander and heat duty have a direct relationship with inlet pressure, temperature and flow of expander. The required heat is a function of hydrate formation temperature in gas, if the expander outlet gas temperature equals the maximum temperature of hydrate formation which is -5 °C for considered gas. The maximum power efficiency can be achieved for each flow rate entering the facility and also minimum heat duty will be needed. In addition, when the inlet pressure ranges from 3800 to 5200 kPa and gas flow is maximum i.e. 866 MSCMH, the efficiency of the expander system is 82-44%, respectively.

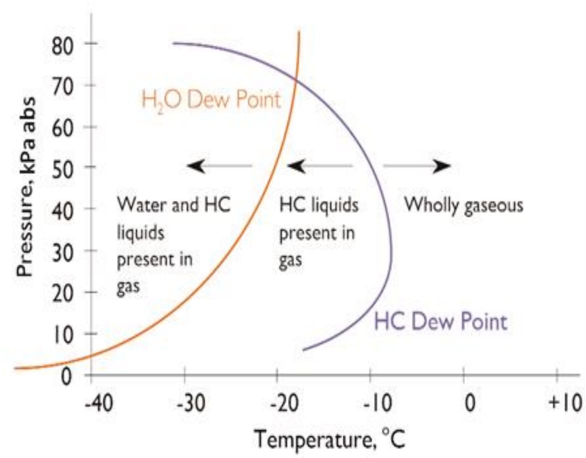

Figure 5: Relationship between dew temperature of natural gas and water. (88% methane, 5% ethane, 2% propane, 1% butane, 2% nitrogen and 0.005% water).

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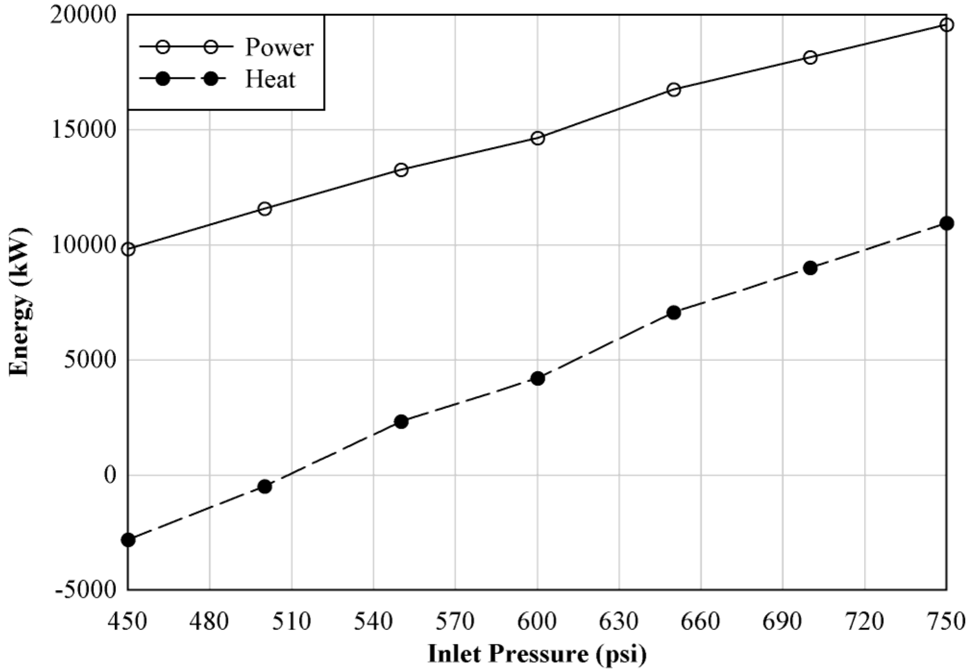

Figure 6: The effect of inlet pressure on energy at the lowest possible outlet temperature (Q=866 MSCMH, T_1=30 °C, P_3=1825 Kpa, T_3= -5 °C)

References

[1]. Ghasempour M, Yaenouni S. Energy recovery from gas reduction stations by turbo expander for production of electricity and cooling. Third gas national convention of Iran, 2007.

[2]. Bloch HP. Become familiar with turboexpander applications. Reliability. Hydrocarbon Processing, 2001; 80(2): 17-18.

[3]. Greeff IL, Visser JA, Ptasinski KA, Janssen FJJG. Using turbine expanders to recover exothermic reaction heat-flow sheet development for typical chemical processes. Energy, 2004; 29: 2045-2060.

[4]. Maddaloni JD, Rowe AM. Natural gas exergy recovery powering distributed hydrogen production. Hydrogen Energy, 2007; 32: 557-566.

[5]. Cho SY, Cho ChH, Kim Ch. Performance characteristics of a turbo expander substituted for expansion valve on air-conditioner. Experimental Thermal and Fluid Science, 2008; 32: 1655-1665.

[6]. Van Ness JM, Smith HC, Abbott MM. Introduction to chemical engineering thermodynamics. 6rd ed. New York, McGraw-Hill, 2005; 186-224.

[7]. Pozivil J. Use of expansion turbine in natural gas pressure reduction stations. Acta Montanistica Slovaco, 2004; 9: 258-260.

[8]. Saboohi Y, Mirahmadi M. Opportunities in power production in gas reduction stations with turbo expander. Sharif energy research institute. Tehran, Iran, 2010.

[9]. Benton A. Michell Instruments, Inc, American school of gas measurement technologies white papers, Determination of Hydrocarbon Dew Point in Natural Gas, 2006.

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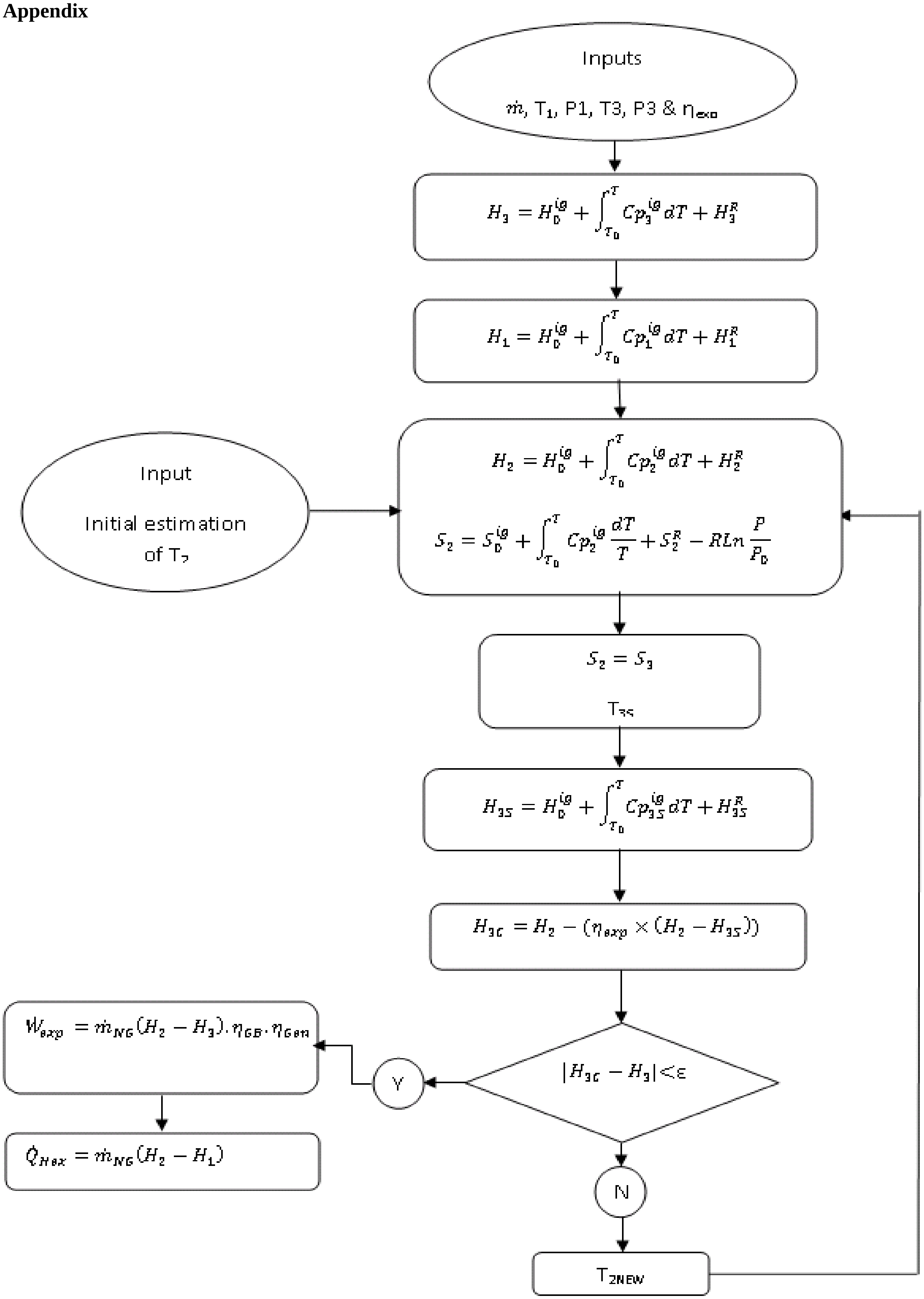
Appendix



Flowchart 1: Turbo-expander calculations algorithm