

Evaluation of the Turboexpander Control System for Optimizing LPG Production Using Unisim Design

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| Submitted: April 07, 2026 | Revised: April 09, 2026 | Accepted: May 24, 2026 |

| Published: October 23, 2026 |

ABSTRACT

The study analyses operational guidelines for optimising energy efficiency in the production of LPG and condensate at gas extraction facilities within cryogenic systems, utilising turboexpander equipment to reduce gas pressure during the cooling process in cryogenic applications involving natural gas. This study employs a quantitative method involving the collection of parameter data through turboexpander analysis calculations and Unisim Design simulations as a guide for performance analysis. Subsequently, tests were conducted on selected modelling variations regarding changes in Differential Pressure (psid), Pressure (psig), and Control Constant (K-Factor) to assess the influence of process parameters on the thermal efficiency of the Turboexpander. The parameters analysed include gas flow rate and composition, covering specific heat (C_p) and enthalpy (H) data both design and actual to obtain the total ΔH and resulting efficiency. The results of the selected variations show that a reduction in gas flow rate causes a decrease in the efficiency of the processed gas at the turboexpander outlet flow rate ($K=0.730$), averaging 1.03 MMSCFD with a maximum difference of 1.58 MMSCFD. Conversely, efforts were made to mitigate the reduction in flow rate and gas composition by optimising via Unisim Design simulation and performing calculation analyses for the ideal control constant ($K=0.736$), resulting in an increase in efficiency with a reduction in processed gas to 0.055 MMSCFD, with the largest difference of 0.194 MMSCFD. Changing the control constant from $K=0.730$ to $K=0.736$ increases the flow rate and efficiency of the turboexpander, particularly under low-flow conditions, with an average increase of 83.10%.

Keywords: energy efficiency, turboexpander, thermal efficiency, control constant, unisim design

INTRODUCTION

Energy conservation efforts in Indonesia, driven by energy constraints, reveal a steadily rising level of energy consumption that is outpacing the availability of existing energy sources. Industrial companies worldwide are required to continuously optimize and improve energy efficiency in their production processes [1]. In addition to accelerated industrialization and technological development, economic growth and population growth are the most significant factors driving the rising demand for energy. To meet this energy demand, many industrial companies have invested in seeking more environmentally friendly alternative energy sources [2].

The environmental impact on energy consumption and use has led to sustainable solutions for the major challenges of our generation, such as poverty, fuel shortages, and global warming [3]. In recent decades, global awareness of climate change has driven rapid advancements in renewable energy technologies to address these issues [4].

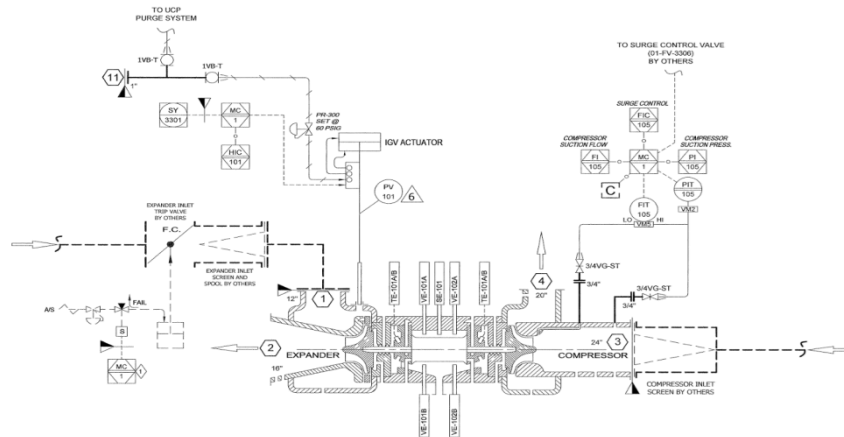
Companies and government agencies are required to implement energy management systems to optimize energy use and establish efficiency standards for various energy-consuming equipment [5]. According to the Minister of Energy and Mineral Resources of the Republic of Indonesia, to implement energy conversion through improved energy consumption efficiency in energy-using equipment, it is necessary to comprehensively regulate the application of minimum energy performance standards for energy-using equipment traded within the territory of the Republic of Indonesia [6]. The objective of energy conservation through systematic, planned, and integrated efforts is to preserve domestic energy resources (energy diversification) and improve the efficiency of energy resource utilization (energy efficiency) [7], whereby energy conservation is

implemented at all stages of energy management, including energy supply, energy utilization, energy consumption, and the conservation of energy resources. The development and implementation of advanced energy technologies to address global challenges related to clean energy, climate change, and sustainable development

Indonesia has set a target for renewable energy to account for 23% of the national primary energy mix by 2025 and 31% by 2050, as well as a target of 17% energy savings by 2025. Among the factors underpinning these targets are the country's commitment to reducing carbon emissions and the need to transition toward Green and Clean Energy, in line with Sustainable Development Goal (SDG) No. 7 "Affordable and Clean Energy" [8] (Progress in Implementing Indonesia's SDGs, 2020). Pertamina's contribution to energy conservation spans all business lines, particularly through one of the company's five strategic priority pillars: achieving operational cost efficiency through operational strategies. At both the holding and sub holding levels, the implementation of cost-efficiency programs within the Pertamina Group has reached USD 535.56 million (approximately Rp7.83 trillion), a figure that continues to be applied across all Pertamina business lines [9].

LPG consumption in Indonesia has risen to 8.71 million tons, an increase of 44.66% from 2012 to 2023, while LPG imports rose by 168.48% during the same period [10]. Additionally, there was a reduction in the nominal volume of gas supply contracts under the 2013 Gas Sales and Purchase Agreement (GSPA) with PHR (Pertamina Hulu Rokan) from 250 MMSCFD to 208 MMSCFD by 2024 [11].

An important component of the production system is the turboexpander-compressor its performance has a significant impact on the overall efficiency of the system. Turbine efficiency is the most critical parameter for system efficiency, as the turbine-compressor is a device that consumes a great deal of power even a moderate increase in turbine efficiency will result in a significant improvement in system efficiency [12]. The operating system and key parameters of a turboexpander-compressor are shown in Figure 1.



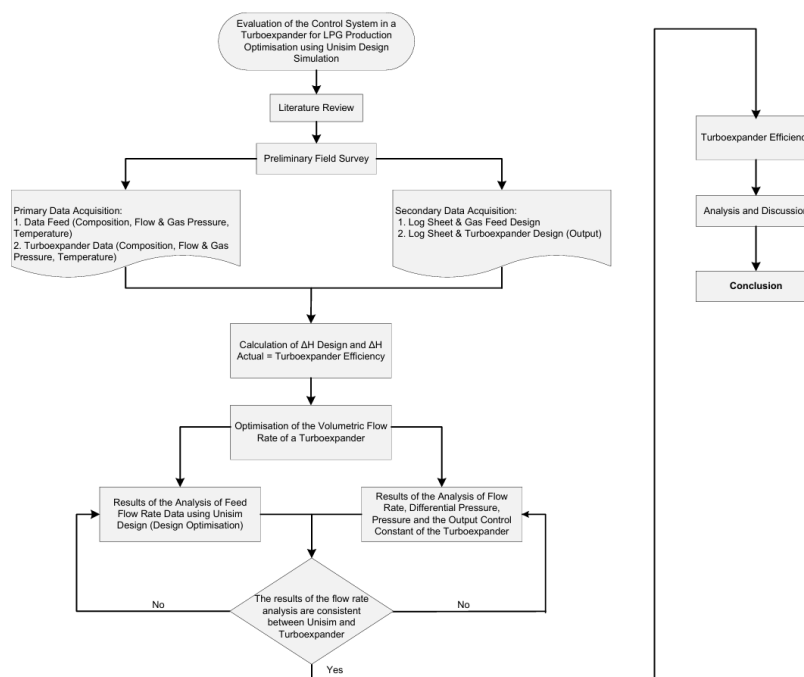


Figure 2. Research Flow chart

Methods

The analysis carried out includes calculations of the total flow rate of the Turboexpander and the overall operational total flow rate, as well as visualization of the resulting data. Software tools such as Microsoft Excel, Unisim Designer Simulator, PLC Programmer and Turboexpander calculation analysis were utilized.

A. Expander Performance Analysis (Before)

At this stage, performance values are calculated using feed gas parameters, based on data design and actual data (Data 1 – Data 6) to determine the design ΔH , actual ΔH and actual thermal efficiency. The scenarios applied in the study are based on operational parameters or trend data prior to evaluation to obtain the expander's performance values.

B. Baseline Performance Analysis of Actual Data Using Unisim Design

A flow sheet modeling simulation was carried out as a system analysis using Unisim Design to determine the ideal volumetric flow rate that can be generated from the actual compound, pressure and temperature data at the inlet and outlet of the Turboexpander process, by inputting parameters in the form of actual production data (data 1–6) into the output parameters of the Turboexpander, where the results of the analysis will be compared with the initial analysis and used as the ideal volumetric baseline for the Turboexpander system output

C. Expander Performance Analysis (Static Test)

This analysis involves identifying the expander's performance values through experiments on expander parameters, specifically by varying the values of Differential Pressure (psid), Pressure (psig) and the Turboexpander control constant (K-Factor), thereby determining the optimal configuration for the compressor expander.

D. Energy Efficiency Analysis of the Turboexpander

Testing the Expander performance values as input for the LNG operation process (processed flow rate) to determine the flow rate efficiency ($\Delta\eta$) of the Turboexpander process.

Data Analysis

A. Plant Efficiency Analysis

The analysis of heat release efficiency begins with calculating the heat capacity (C_p/R) or specific heat capacity of each compound, followed by calculating the enthalpy (H) using both

design data [13] and actual data (Data 1–Data 6) Turboexpander outlet [14] subsequently, calculations of ΔH Design and ΔH Actual are performed to obtain the total heat transfer or enthalpy (H) value, in order to determine the actual process efficiency η . The calculation formula Cp/R in Inlet Design and Cp/R outlet Turboexpander of is as follows:

$$Cp = A + BT + CT^2 + DT^3 + ET^4 \quad (1)$$

$$\frac{Cp}{R} = \frac{A+BT+CT^2+DT^3+ET^4}{8.314} \quad (2)$$

The Calculation of the inlet (H) of the Turboexpander of is as follows:

$$H = \sum x_i \frac{Cp}{R} \times T_{in} \times R \quad (3)$$

The Calculation of the outlet (H) of the Turboexpander of is as follows:

$$Cp_{Liquid} = C_1 + C_2T + C_3T^2 + C_4T^3 + C_4T^4 \quad (4)$$

$$Cp_{Liquid} = \frac{C_1^2}{t} + C_2 - 2C_1C_3t - C_1C_4t^2 - \frac{C_3^2t^3}{3} - \frac{C_3C_4t^4}{2} - \frac{C_4^2t^5}{5} \quad (5)$$

$$Cp_{Liquid} = (\sum x_i Cp)EQN2 + (\sum x_i Cp)EQN1 \quad (6)$$

$$H_{Liquid} = Cp_{Liquid} \times T_{out} \quad (7)$$

The Calculation of the ΔH and Efficiency η of the Turboexpander of is as follows:

$$\Delta H = (H_{Vapor} + H_{Liquid}) - H \quad (8)$$

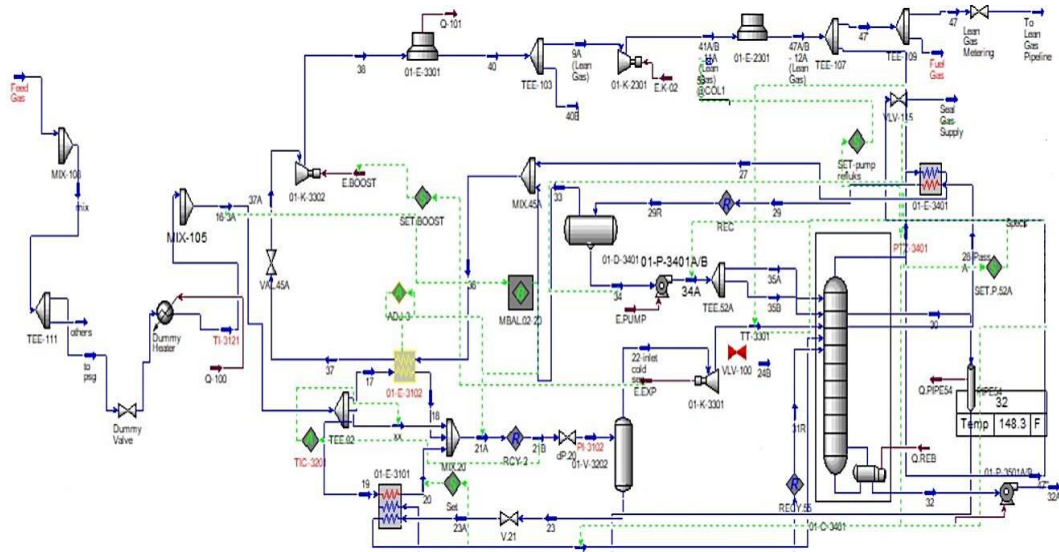
$$\eta = \frac{\Delta H_{aktual}}{\Delta H_{desain}} \quad (9)$$

Where :

A – E =	Regression Coefficient Chemical Compound	R =	Gas Constant(J Mol ⁻¹ K ⁻¹)
C ₁ – (	Regression Coefficient Chemical Compound	T _{In} =	Temperture Desain (Deg K)
C _p	Heat capacity (J/Mol K)	H _{Liquid} =	entalpi $\left(\frac{J}{Mol}\right)$
X _i =	Composition of the inlet compound(%)	T _{Out} =	Temperture Out (Deg K)
T =	Temperture Desain (Deg K)	η =	Efficiency (%)

B. Unisim Design

The design of the project database must follow the processes and flow sheets specific to the extraction plant so that calculations yield results consistent with the process system in this simulation, a flow sheet model as shown in Figure 3. is required as a system analysis using Unisim Design to determine the ideal volumetric output that can be produced from the compound, pressure and temperature of the actual inlet and outlet data of the Turboexpander process, by inputting parameters in the form of actual production data and Turboexpander output parameters, where the analysis results will later serve as the baseline for the ideal volumetric output of the Turboexpander system. This simulation calculation is performed on 6 models, which are: Data 1, Data 2, Data 3, Data 4, Data 5, and Data 6—production process data.



C. Optimization of Turboexpander Flow Rate

Optimization analysis of the Turboexpander flow rate using simulation modeling of several variations to determine the ideal maximum volumetric value from the selected modeling results for the Turboexpander in the compound process, based on actual inlet and outlet pressure and temperature data for the Turboexpander. Formula for the anti-surge output volumetric flow rate of the Turboexpander, which will subsequently be used by the compressor to control the anti-surge valve set point [15]. The formula for calculating the output volumetric flow rate is as follows:

$$Qp = 13,383 \text{ K} \sqrt{\frac{d_p Z T_{1R}}{(P_1 + 14,7) \text{ mw}}} \quad (10)$$

$Q_p =$ Volumetric Flowrate (ACFM)
 $d_p =$ Flow dP Pressure (psid)
 $T =$ Comp. Inlet Temp (Deg R)
 $P_1 =$ Compressor Inlet Pressure (psig)
 $mw =$ Molecular weight $\left(\frac{lb}{lbmol}\right)$
 $K =$ Flow Element Konstan

H (Enthalpy) Inlet Design Calculation

The results of the Cp/R calculations based on the design data heat and material balance for the inlet and outlet of the Expander. Table 1 show that the initial calculations, based on the inlet design data for each compound (xi) and the heat capacity (Cp), were used to obtain $\sum x_i$. Cp/R involves the heat capacity (Cp) and gas constant (R) for each compound (xi) [13]. In this analysis, the highest heat capacity value xi.Cp/R is found in compound C1 (methane) at 3.6775, with a composition of 88.572 %mol, and the lowest in compound C7 (heptane) at 0.0005, with a composition of 0.003 %mol. The high increase in heat capacity values is due to the proportion of the composition [16], with a total value of $\sum x_i$.Cp/R amounting to 4.195959103.

Table 1. Calculations Cp/R Data Inlet Desain

Compounds	Cp/R (j/mol.k)	xi (fraksi mol)	xi.Cp/R
C ₁ (methane)	4.15E+00	0.88572	3.68E+00
C ₂ (ethane)	5.68E+00	0.04783	2.72E-01
C ₃ (Propane)	9.22E-01	0.02210	2.04E-02
C ₄ (Butane)	8.79E+00	0.00284	2.50E-02
i-C ₄ (iso-butane)	9.19E+00	0.00271	2.49E-02
i-C ₅ (iso-Pentane)	1.27E+01	0.00054	6.84E-03
C ₆ (Hexane)	1.50E+01	0.00009	1.35E-03
CO ₂ (Carbon Dioxide)	4.40E+00	0.03564	1.57E-01
N ₂ (Nitrogen)	3.49E+00	0.00227	7.92E-03
C ₅ (Pentane)	1.27E+01	0.00022	2.79E-03
N ₇ (Heptane)	1.74E+01	0.00003	5.21E-04
		Σ	4.195959103

A calculation was performed to determine the enthalpy (H) at the inlet of the turboexpander (vapor) using equation (3), where the design inlet temperature is 242.261 K, this resulted in an inlet enthalpy (H) of 8451.32440 J/mol

H (Enthalpy) Outlet Design Calculation

Using the outlet design composition and the Turboexpander outlet temperature of 194.59 K, the highest value of heat capacity xi was obtained. The Cp/R ratio for compound C₁ (methane) is 3.5561, corresponding to a C₁ composition of 88.572 % mol, whilst the smallest value is for compound C₇ (heptane) with a heat capacity of 0.0004 and a composition of 0.003 % mol, resulting in a total Σxi . Cp/R of 4.031067695. The thermodynamic model analysis explains that specific heat capacity is calculated based on the gas and liquid phases, which further supports the proportional relationship between heat capacity and phase composition the greater the heat capacity, the greater the composition [17] . After obtaining the sum of xi.Cp/R, proceed to calculate the vapour enthalpy (H) at the outlet just as with the previous calculation of the inlet enthalpy (H) using equation (4), the outlet enthalpy (H) is calculated to be 6521.68107 J/mol

Table 2. Calculations Cp/R Data Outlet Design

Compounds	Cp/R (j/mol.k)	xi (fraksi mol)	xi.Cp/R
C ₁ (methane)	4.01E+00	0.88572	3.56E+00
C ₂ (ethane)	5.11E+00	0.04783	2.44E-01
C ₃ (Propane)	7.79E-01	0.0221	1.72E-02
C ₄ (Butane)	8.86E+00	0.00284	2.52E-02
i-C ₄ (iso-butane)	7.68E+00	0.00271	2.08E-02
i-C ₅ (iso-Pentane)	1.08E+01	0.00054	5.85E-03
C ₆ (Hexane)	1.27E+01	0.00009	1.15E-03
CO ₂ (Carbon Dioxide)	4.20E+00	0.03564	1.50E-01
N ₂ (Nitrogen)	3.49E+00	0.00227	7.92E-03
C ₅ (Pentane)	1.08E+01	0.00022	2.38E-03
N ₇ (Heptane)	1.47E+01	0.00003	4.41E-04
		Σ	4.031067695

Having obtained the sum of $\sum x_i C_p/R$, next calculate the vapor enthalpy (H) at the outlet. Using equation (3), the outlet enthalpy (H) is calculated to be 6521.68107 J/mol. The next step in determining the total ΔH using the design data is to calculate H for the liquid fraction produced at the outlet of the Turboexpander. As the heat capacity of the liquid fraction is given by equations (4) and equation (5), the calculation results based on equation (4) and equation (5) are as follows

Table 3. Calculations liquid fraction C_p/R Data Outlet Design (equation 4)

Compounds	C_p (J/mol.k)	x_i (fraksi mol)	$x_i C_p$
i-C ₄ (iso-butane)	1.19E+05	0.00271	3.23E+02
i-C ₅ (iso-Pentane)	1.37E+05	0.00054	7.39E+01
N-C ₅ (Neo-Pentane)	1.10E+05	0.00022	2.42E+01
C ₆ (Hexane)	1.70E+05	0.00009	1.53E+01
CO ₂ (Carbon Dioxide)	2.17E+04	0.00356	7.72E+02
N ₂ (Nitrogen)	1.68E+06	0.00227	3.81E+03
N-C ₄ (Butane)	1.19E+05	0.00284	3.38E+02
		Σ	5356.09507

Table 4. Calculations liquid fraction C_p/R Data Outlet Design (equation 5)

Compounds	C_p (J/mol.k)	x_i (fraksi mol)	$x_i C_p$
C ₂ (Ethane)	1.16E+05	0.04783	5.57E+03
C ₃ (Propane)	1.51E+05	0.0221	3.35E+03
C ₇ (Heptane)	4.40E+05	0.00003	1.32E+01
		Σ	8924.23239

Table 3 shows the process described by equation 4 regarding changes in heat capacity for the compounds iso-butane and butane, where the results for each compound fraction are used to determine $\sum x_i C_p$. The graph shows that the highest heat capacity is found in nitrogen at 3809.40, with a composition of 0.00227 %mol, whilst the lowest is in the compound hexane at 15.30, with a composition of 0.00009 %mol. The total heat capacity produced in equation 4 $\sum x_i C_p$ is 5356.10.

In Table 4, the heat capacities of the liquid fractions of ethane, propane and heptane were calculated using equation 5, with the specific heat capacity (C_p) of each compound applied to obtain $\sum x_i C_p$. The resulting heat capacity was 8924.23

Once the results for the liquid C_p have been obtained using both equations (Equation 4-5) and each compound's C_p has been multiplied by the molar fraction of the liquid outlet, the total liquid C_p is found to be 14280.32746 J/mol·K using equation (6). To obtain the total C_p of the liquid from the Turboexpander outlet, the calculation of H for the liquid using equation (7) results in a value of 27,748,866.04 J/mol

Calculation of ΔH Design

Once the three key data points (inlet H, vapor outlet H, and liquid outlet H) have been obtained, ΔH can be calculated to determine the total heat transfer or enthalpy (H). Using equation (8), the total heat transfer ΔH for the design data is found to be 2,776,936.398 J/mol.

Calculation of ΔH Actual and Efficiency

Using the actual data from No. 1 to No. 6, calculations were performed to determine the efficiency percentage, by comparing the design delta H (design heat transfer) with the actual delta H (actual heat transfer). Calculations were performed for the inlet ΔH , followed by the liquid outlet ΔH and the vapor outlet ΔH using actual data. Once the actual ΔH and design ΔH values had been obtained using equation (9), a comparison was made with the outlet flow rate of the

Turboexpander to determine the resulting efficiency. Based on design data and actual data obtained for the period from June 2024 to March 2025, the following calculations were obtained:

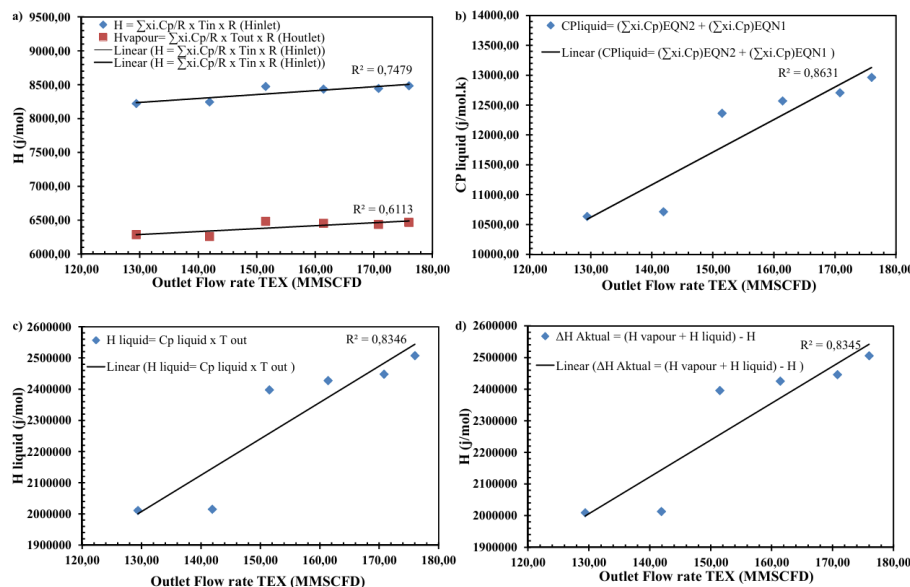


Figure 4. a. Calculation of inlet head (H) and outlet head (H) data b. Liquid CP c. Outlet liquid head data d. Design head difference (ΔH) versus actual head difference (ΔH)

Figure 4.a shows a graph of the relationship between the calculated inlet H data and outlet H data generated at each outlet flow rate of the Turboexpander. Where, during high flow rate processes, heat transfer at the inlet H and outlet H increases, this increase in heat transfer at the inlet H and outlet H is due to the influence of process temperature, with higher outlet temperatures typically indicating greater heat transfer, and is influenced by the design of the inlet and outlet channels [18] as well as changes in the specific heat capacity (Cp) resulting from the contribution of each component of the compound itself.

Figure 4.b shows a graph of the relationship between the Turboexpander outlet flow rate and liquid Cp. There is a clear positive correlation between the Turboexpander outlet flow rate and liquid Cp. This means that as the outlet flow rate increases, the liquid Cp value tends to increase. The analysis of the liquid Cp uses two different equations depending on the type of composition being analyzed. The level of the resulting liquid Cp is influenced by the composition of the gas. In the initial calculation, the R-squared coefficient of determination for the resulting outlet flow rate was 86.31%, influenced by the liquid Cp variable.

Figure 4.c shows a graph of the relationship between the outlet flow rate of the Turboexpander and H liquid. As the outlet flow rate of the Turboexpander increases, H liquid increases. The increase in H liquid, which occurs in tandem with an increase in Cp liquid, may be attributed to the composition of the processed gas actual conditions based on flow rate analysis indicate a similar relationship with the increase in the composition of the gas. An increase in the gas flow rate will naturally affect the H liquid of the Cp liquid process in relation to the gas composition. In processes involving H liquid (heat transfer) and Cp liquid (heat capacity) alongside gas composition, changes in the gas flow rate can influence mass transfer efficiency and reaction rates. Different gas compositions affect mass transfer characteristics and the rate of gas dissolution into the liquid generally, the higher the gas flow rate, the easier it is for gas to evolve from the solution because more gas nuclei are formed in the liquid [19].

Figure 4.d shows a graph of the relationship between the outlet flow rate of the Turboexpander and the actual ΔH. The linear line indicates an upward trend, reflecting a positive relationship between the outlet flow rate of the Turboexpander and the actual ΔH. The actual ΔH value is derived from

the sum of the vapour head (outlet data) and the liquid head (outlet data), minus the design head (outlet design) the higher the outlet flow rate, the higher the actual ΔH value. The increase in outlet flow rate is influenced by the values of H vapour and H liquid, whilst H design remains constant (stable). An increase in flow rate is associated with an increase in pressure difference or head (ΔH), although the exact relationship may be influenced by system design, outlet diameter, and other factors. However, the basic principle remains: to achieve a higher flow rate, a higher ΔH is required [20].

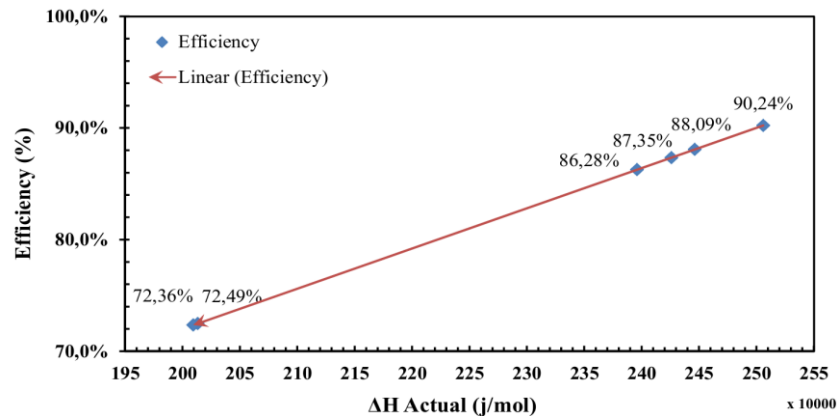


Figure 5. Actual ΔH and Turboexpander Efficiency

Figure 5. shows the results of the analysis of data 1–6 for Design ΔH and actual ΔH , where the final result is thermal efficiency. A decrease in this process indicates that as actual ΔH decreases, the resulting thermal efficiency also decreases this is caused by several factors influencing the decrease in H vapor (outlet data), H liquid (outlet data) and H (design outlet). The correlation between thermal efficiency and ΔH is more complex and depends on the context of the thermal system in question.

Results of the analysis of actual performance data using Unisim Design

Volumetric calculations were performed to determine the maximum volumetric flow rate using Unisim Design modeling, taking into account the conditions and process stages at the inlet and outlet of the Turboexpander. The results of the volumetric flow rate analysis in the modeling were subsequently compared with the actual volumetric flow rate data to determine the percentage difference between the actual data and the Unisim Design simulation the results of the comparative analysis of the volumetric flow rate from the Unisim Design simulation are shown in Figure 6.

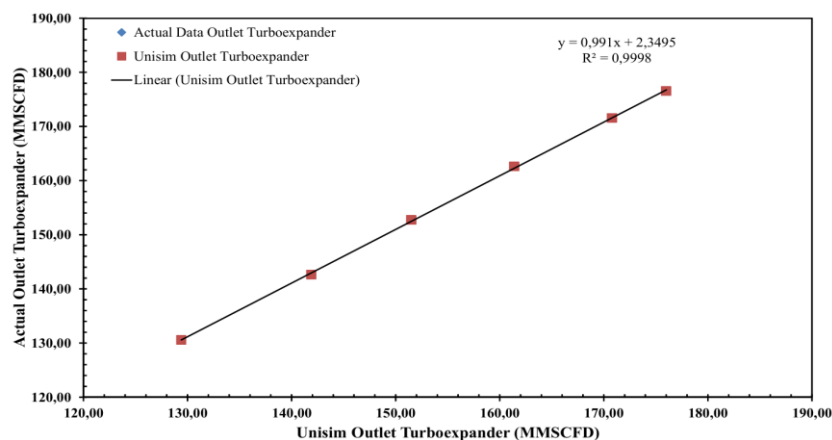


Figure 6. Flow rate analysis simulation using Unisim Design

Figure 6. shows a graph of the relationship between the Unisim Design flow rate and the actual outlet flow rate, where the average difference in flow rate was 0.96 MMSCFD, equivalent to 0.55%, The R-squared value of the coefficient of determination for the unisim outlet flow rate of the Turboexpander is 99.98% compared to the actual outlet flow rate of the Turboexpander, where the Unisim Design simulation results show a difference of $\leq 2\%$ thus, this value is used to validate the optimal flow rate or ideal volumetric flow rate, which is employed as a limit in determining the control calculations for the Turboexpander.

Results of the analysis of variance for volumetric flow rate optimization

Using the process data from actual data, simulation calculations were performed for changes in differential pressure, temperature and control constants to obtain a Turboexpander flow rate that approximates the ideal flow rate derived from the results processed using Unisim Design.

The results of this simulation analysis indicate that there is a proportional decrease in flow rate between changes in differential pressure (psid) and volumetric flow rate (MMSCFD), in the pressure simulation, the lower the flow rate (MMSCFD), the lower the processed pressure (psig). Meanwhile, in the simulation of the control constant, the higher the value of the control constant, the higher the resulting flow rate, thereby influencing the sensitivity and characteristics of the equipment.

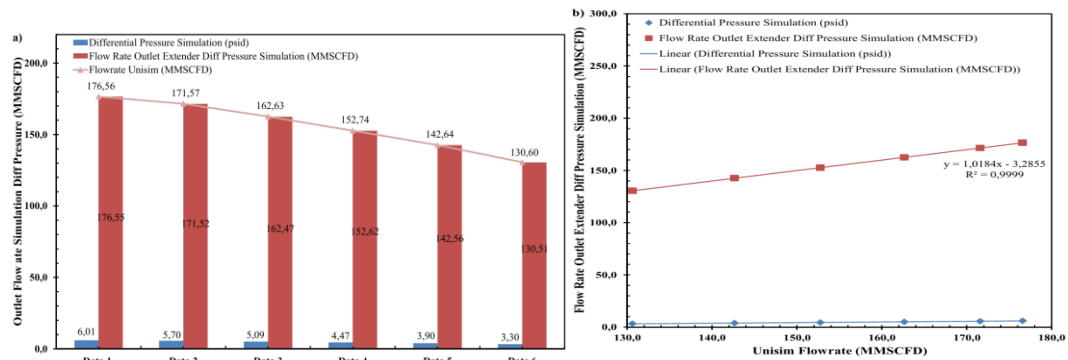


Figure 7. a. Pressure Difference Diagram, Outlet Flow Rate from the Pressure Difference Simulation and Unisim Flow Rate b. Graph of Outlet Flow Rate from the Pressure Difference Simulation versus Unisim Flow Rate

Using the volumetric flow rate equation (10), simulation results were obtained for the relationship between differential pressure dp (psid) and the decrease in flow rate Q_p (MMSCFD), as shown in Figure 7.a. This illustrates that the flow rate Q_p (MMSCFD) is proportional to the decrease in differential pressure dp (psid). From Data 1 to Data 6, when examining the ratio of the decrease, every 0.54 psid decrease in dp results in an average decrease in flow rate of Q_p 9.21 MMSCFD. If the differential pressure dp (psid) is low, this indicates that the driving pressure of the Turboexpander is beginning to weaken or that the density of the gas leaving the Turboexpander is decreasing. The power that can be extracted by the Turboexpander depends primarily on the pressure ratio (p_{in}/p_{out}) and the mass flow rate, the smaller the pressure ratio, the smaller the work expansion that can be extracted [21]. Figure 7.b. shows a graph of the relationship between the Unisim Design flow rate and the expander outlet flow rate in the differential pressure simulation, where there is a difference between the dp flow rate (MMSCFD) and the Unisim flow rate (MMSCFD) in data points 1–6 ranging from 0.01 to 0.16 MMSCFD, with an average difference of 0.086 MMSCFD, indicating that the simulated dp flow rate approaches the Unisim flow rate value when viewed through the R-squared coefficient of determination against the Unisim dp flow rate, the resulting value is 99.99% relative to the Unisim flow rate.

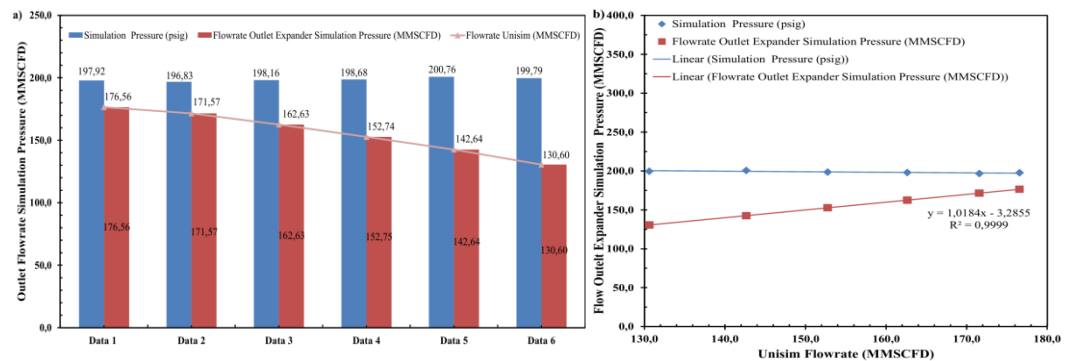


Figure 8. a. Pressure Diagram, Outlet Flow Rate from the Pressure Simulation and Unisim Flow Rate b. Graph of Outlet Flow Rate from the Pressure Simulation versus Unisim Flow Rate

Figure 8.a. shows an analysis of the variation in pressure P (psig) against the decrease in flow rate of the Turboexpander, according to the ideal gas law, pressure is directly proportional to the number of moles (flow). A change in downstream conditions (increased backpressure) reduces the operating pressure ratio, thereby decreasing the mass flow rate on the same characteristic map [22]. Technically, if the inlet pressure of a Turboexpander remains stable whilst the flow rate decreases, this indicates that the system is experiencing a reduction in transport capacity (mass flow) from the Turboexpander's inlet feed gas. From Data 1 to Data 2, the pressure dropped slightly (1.09 psig), however the flow rate Qp decreased significantly by 4.99 MMSCFD. This indicates high flow sensitivity to initial pressure. From Data 3 to Data 5, there was a pressure increase from 198.16 psig to 200.76 psig, but the flow rate Qp actually dropped sharply from 162.63 to 142.64 MMSCFD. The rise in pressure accompanied by a decrease in flow indicates the occurrence of back pressure. As the gas source (feed gas) becomes weaker, it is unable to push the same volume of gas past the 200.76 psig pressure barrier. Figure 8.b. shows the variation in pressure (psig) required to obtain the final result of the Turboexpander outlet flow rate (MMSCFD). In Data 1–Data 6, pressure variations ranging from 196.83 to 200.76 psig were applied, yielding an output flow rate of the Turboexpander (Qp) with a minimum difference of 0.0002 MMSCFD and a maximum of 0.0027 MMSCFD. Overall, the system demonstrated linear and stable performance. The expander's output capacity is directly proportional to the input flow rate without compromising the system's pressure stability. An R^2 value approaching 1 indicates that there is no significant deviation in the modeling of gas flow within the Unisim simulation for this range of flow rates.

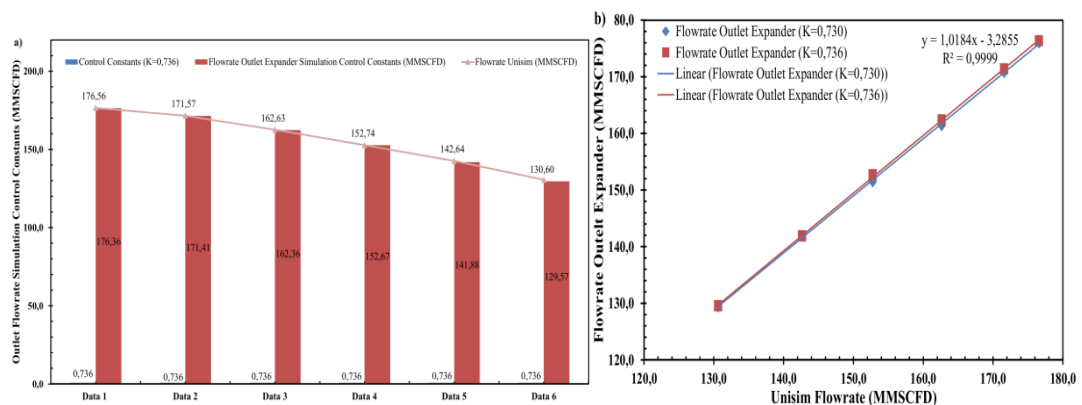


Figure 9. a. Control Constant Diagram, Outlet Flow Rate: Simulation of control constants and Unisim flow rate b. Graph of simulated control constant versus Unisim flow rate

Experimental variations were carried out on the initial control constant 0.730 to obtain a flow rate (MMSCFD) consistent with the results of the Unisim Design simulation in the appendix. Using the available data 1–6, calculations were then performed to obtain the volumetric flow rate

(MMSCFD) using equation (10). In contrast to the simulation variations of differential pressure (psid) and pressure p (psig), the simulation control constant was varied by applying the same value changes to the input data from Data 1 to Data 6.

Figures 9.a. and 9.b. show the relationship between the control constant, the output flow rate from the control constant simulation, and a comparison of the flow rate from the Unisim Design analysis. A control constant value of $K=0.736$ yields a Turboexpander output flow rate that closely approximates the results of the Unisim Design simulation flow rate analysis. The results show an average difference in the Turboexpander output for $K=0.736$ of 0.4181 MMSCFD, with the smallest difference being 0.08 MMSCFD and the largest 1.03 MMSCFD. Figure 9.b. shows a decrease in the flow rate output of the Turboexpander ($K=0.736$), approaching the flow rate output of the Unisim model. This is because the R-squared value of the coefficient of determination for the Turboexpander flow rate output generated by Unisim is close to 100% compared to the flow rate output of the Turboexpander ($K=0.736$). This indicates a strong linear relationship or, at the very least, minimal deviation (residual error).

From the three simulation results Differential Pressure (psid), Pressure (psig) and the control constant ($K=0.736$) which produced the outlet flow rate of the expander, a final technical analysis was conducted regarding the impact of these variables on the plant's operational processes. It can be concluded that varying the control constant ($K=0.736$) is superior and offers the highest accuracy for the Unisim flow rate value compared to variations in Differential Pressure (psid) or Pressure (psig). Furthermore, the use of $K=0.736$ is the best approach because, with this parameter, variations can integrate pressure and equipment geometry variables into a single fixed coefficient value, ensuring that the calculated flow rate is always synchronized with the plant's operational processes and consistent with the mathematical modeling system used by Unisim Design. the impact of compressor efficiency on the combined coefficient of performance (COP) is such that the higher the compressor efficiency, the lower the compressor power consumption, thereby increasing the ratio of cooling output to input energy [23]

Result of efficiency Turboexpander

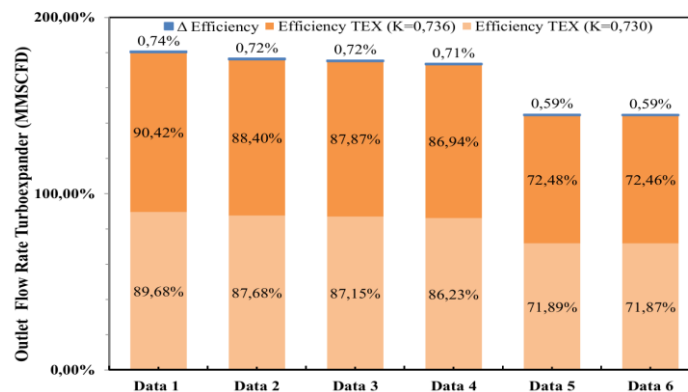


Figure 10. Efficiency Graphs for TEX ($K=0.730$), TEX ($K=0.736$) and Δ Efficiency

Figure 10. shows the relationship between the optimization results and efficiency for the actual flow rate output and the Turboexpander flow rate output. The simulation and calculation results for each data point, representing the actual flow rate output with a control constant $K = 0.730$ and the efficiency for each data point, were subsequently compared with the simulation and calculation results obtained by changing the control constant to $K = 0.736$. The results show a reduction in the flow rate deviation for the processed gas output flow rate of the Turboexpander to 0.135 MMSCFD the highest increase is observed at the lowest flow rate in data set 1, where $Q = 174.93$ MMSCFD increases to $Q = 176.36$ MMSCFD, representing an increase of 1.43 MMSCFD and = 89.68% rising to = 90.42% or an increase of 0.74 %. Meanwhile, the lowest increase was observed in data points 5 and 6, where the actual flow rate changed from $Q = 141.27$ MMSCFD to $Q =$

142.44 MMSCFD, representing an increase of 0.145 MMSCFD and = 72.17 % to = 72.77 %, an increase of 0.59 %. Changes to the value of the constant K successfully improved efficiency across all data points (1–6). This indicates that the control system has become more responsive in maintaining the engine's operating point optimally in response to load fluctuations. The reduction in flow rate deviation to 0.135 MMSCFD indicates that the system has a higher level of accuracy at the highest flow rate increase of 1.43 MMSCFD (under low load conditions), this directly maximizes production and ensures that the processed gas output becomes more stable and closer to the design target.

CONCLUSION

Based on the results of the design data calculations, which were derived by calculating the heat capacity of each required phase (vapor phase and liquid phase), a design ΔH value of 2,776,936.398 J/mol was obtained. Analysis of the thermal efficiency of the actual data shows that the calculated decrease in ΔH Actual for each set of actual data results in a reduction in Turboexpander efficiency of 90.24%, 88.09%, 87.35%, 72.51%, 72.37% this is caused by a decrease in H vapor, H liquid and H design outlet, as well as a decrease in the flow rate of the system's feed gas. The results of the simulation analysis indicate that the use of the control constant ($K = 0.736$) yields the most accurate and stable simulation results as it consistently represents the physical characteristics of the Turboexpander. Variations in Differential Pressure (ΔP) have the drawback of moderate accuracy as they are sensitive to changes in gas density, whilst variations in Pressure (P) yield less accurate results as they cannot directly describe the dynamics of gas flow. Therefore, the use of the control constant (K) is the most effective method for improving the accuracy of Turboexpander performance simulations. The results of the analysis show that increasing the control constant from $K=0.730$ to $K=0.736$ can improve the flow rate and efficiency of the Turboexpander, particularly under low flow conditions, with an average efficiency of 83.10%. This improvement is due to the system's ability to optimize the volume of gas discharged and enhance the performance of the Turboexpander. These results indicate that adjusting the control constant can be an effective strategy for improving the performance of the Turboexpander system in gas processing.

ACKNOWLEDGEMENT

Authors would like to thanks PT Pertasamtan Gas and Master Program of Energy, Diponegoro University for assisting and supporting with this research.

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