Experimental and numerical analysis to predict the performance of a turboexpander at cryogenic temperature

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Funding information
Board of Research in Nuclear Sciences; Ministry of Human Resource Development; National Institute of Technology Rourkela

Abstract
The emerging trend for using cryogenic fluids in scientific and research applications requires a Claude cycle-based efficient turboexpander system. This paper presents the mean-line design procedure, numerical, and experimental investigation to distinguish the fluid flow and thermodynamic analysis of a turboexpander operating at cryogenic temperature. Initially, the mean-line design of turboexpander components has been carried out. Thereafter, the three-dimensional model and computational mesh of nozzle, turbine, and diffuser have been generated using ANSYS BladeGen® and ICEM®. The three-dimensional computational fluid dynamics analysis has been carried out to characterize the effect of fluid flow properties such as pressure, velocity, and thermal properties such as temperature, static enthalpy, and entropy at different axial locations. Finally, an experimental test rig has been developed to examine the performance of the turboexpander at cryogenic temperature and validating the numerical findings for a case study. The methodology provides insight into the realistic fluid flow phenomenon and thermal performance of a turboexpander under rotating conditions, as it is the most essential and expensive part of a gas liquefaction system.

KEYWORDS
experimental and numerical analysis, fluid flow, thermal characteristics, turboexpander

1 | INTRODUCTION

The generation of ultra-low temperature using a radial expansion turbine was a significant milestone achieved in cryogenic history. Primarily, the cryogenic temperature or liquefaction of gases was produced through step-by-step development, starting from Siemens and Kirk’s engine acting as an expander to throttle valve, which is based on the Joule–Thomson effect. In 1896, K. Onnes suggested the usage of a radial expansion turbine for the liquefaction of hydrogen gas. Based on this concept, Claude used a radial turbine that replaces the expansion device (J–T valve) for the air liquefaction system. It was a modified form of Linde–Hampson cycle, often referred to as the Claude cycle, which has a comparatively higher system efficiency. Later on, Linde Pvt. Ltd. started commercial production of turboexpander based...
gas liquefaction system in 1930. After considering these developments, the era of cryogenic turboexpander has been started for gas liquefaction cycles.

In the present scenario, researchers are interested in the optimal design of a cryogenic turboexpander as a replacement of Joule–Thomson valves to save energy. Turboexpander can fulfill the huge requirements of cryogenic fluids which are used in chemical, biomedical, superconductivity, space applications, and other scientific and research industries. Radial turbine, nozzle, diffuser, brake compressor, journal and thrust bearings, cryogenic heat exchangers (CHX), and so on are the major components of a liquefaction system. Out of which, the radial expansion turbine, nozzle, and diffuser are the major components. Therefore, the optimum design and computational fluid dynamics (CFD) analysis of these components have a significant contribution in increasing the system efficiency.

Whitfield and Aungier proposed the most celebrated design methodology of radial inflow turbine considering different loss correlation. However, the design approach considered an ideal gas equation of state which has to be modified due to random change in thermophysical properties of the fluid at cryogenic operating conditions.

Yang et al. explained the one-dimensional design procedure of a radial turbine. The analysis was further extended to develop an experimental test rig for investigating the thermodynamic performance of gas bearings used in a two-stage radial expansion turbine. Kumar et al. performed the analytical, numerical, and experimental analysis of helium and nitrogen turboexpander for performance modeling at a range of operating conditions. The design procedure includes various loss correlations and Sobol sensitivity analyses to optimize the non-dimensional design variables to obtain an optimum turbine blade profile. The study also suggests the usage of artificial intelligence networks for optimizing the design variables for better performance of a radial inflow turboexpander.

Harinck et al. conducted the three-dimensional CFD analysis to understand the fluid flow physics of a turboexpander used in an organic Rankine cycle (ORC). Pasquale et al. focused on optimizing the blade profile of a radial inflow turbine using the real gas equation of state. Chang et al. proposed the thermodynamic design and numerical analysis of hydrogen liquefaction system using a reversed Brayton cycle-based helium refrigerator. The numerical analysis revealed the thermal performance of the liquefaction cycle at a different mass flow rate and pressure ratio. Kumar et al. developed an experimental test set-up to study the thermal performance of a modified Claude cycle-based nitrogen turboexpander that operates at various operating conditions (mass flow rate, pressure ration, inlet temperature, and so on).

Zhao et al. carried out the experimental analysis to measure the performance of the radial turbine used in the reversed Brayton cycle-based air refrigerator. Yang et al. studied the dynamic behavior of a reverse Brayton cycle-based turboexpander system. A transient cooling model has been proposed to investigate the performance of the refrigerator at different operating conditions. Sam et al. conducted a numerical study to understand the flow physics of a cryogenic turboexpander under rotating conditions using helium as a working fluid. Ke et al. performed a numerical and experimental investigation to study the effect of vaneless space on the performance of the radial inflow turbine used in the cryogenic application.

The literature survey suggests that the experimental and CFD analysis of a turboexpander is necessary for medium pressure and ultra-low temperature nitrogen fluid. Also, these operating conditions (mass flow rate and blade speed ratio) are not considered in our previous studies. The present work attempts to fulfill this literature gap.

The present study is an extended work that is presented by one of the authors in conference proceedings. In this study, the mean-line design, numerical and experimental analysis have been carried out for the performance measurement of a turboexpander using nitrogen as a working fluid. The mean-line (1-D) design procedure and ANSYS Bladegen® have been used to develop a three-dimensional model of a turboexpander. Thereafter, the numerical analysis has been carried out to understand the insights of flow behavior such as pressure, Mach number, velocity, and thermal characteristics such as temperature, Prandtl number, static enthalpy, and entropy at different cross-sections of the turboexpander. Finally, an experimental test set-up has been developed to characterize the thermodynamic performance of the turboexpander at high rotational speed and a range of mass flow rates. The numerical results are also validated with the experimental results for a case study. The present investigation is beneficial in understanding the thermal performance of a cryogenic turboexpander and for developing a prototype at a laboratory scale.

2 | MODEL DEVELOPMENT AND NUMERICAL SET-UP

The present mean-line design methodology is based on the design procedure suggested by Aungier and Whitfield. Figure 1 represents a schematic diagram of the rotor geometry, inlet, and outlet velocity triangles obtained from a one-dimensional design approach, which is discussed in our previous study. In the design procedure, the optimized range of design variables and loss correlations are considered to obtain an efficient turbine and nozzle blade profile. The
Figure 1  (A) Schematic of the rotor geometry, (B) inlet velocity triangle, (C) outlet velocity triangle

The thermodynamic properties (at cryogenic operating conditions) affect the blade profile of the turbine and nozzle. Therefore, the precise design of these components requires a real gas model in the design procedure. The thermodynamic datasets of the real gas model have been obtained from REFPROP® software. The specifications of the designed turboexpander are mentioned in Table 1.

The coordinates obtained from the mean-line design procedure are imported in ANSYS Bladegen® to develop a three-dimensional (3-D) model of blade profile, hub, shroud, and flow passage. After that, the high-quality unstructured tetrahedral computational mesh has been created in ANSYS ICEM®. The node distribution at near-wall regions is higher to capture the flow separation and boundary layer effects. The y⁺ values at different sections of the turboexpander are mentioned in Table 2. Figure 2 shows the flow chart of mean-line design procedure, three-dimensional model of turboexpander, and computational mesh.

Table 1  Major specification of the designed turboexpander

| Parameter                  | Value       |
|----------------------------|-------------|
| Inlet pressure             | 8 bar       |
| Inlet temperature          | 150 K       |
| Pressure ratio             | 1.80–4.80   |
| Turbine inlet diameter     | 24.86 mm    |
| Blade height at turbine inlet | 1.36 mm   |
| Blade height at turbine outlet | 4.54 mm  |
| Turbine axial length       | 10.52 mm    |
| Number of turbine blades   | 13          |
| Number of nozzle           | 17          |
| Tip clearance              | 0.10 mm     |
TABLE 2  Non-dimensional wall distance at various sections

| Sections                      | y* Value |
|-------------------------------|----------|
| Nozzle inlet                  | 2.10     |
| Near nozzle blade             | 1.20     |
| Center of the nozzle passage  | 2.64     |
| Turbine inlet                 | 1.38     |
| Near nozzle blade             | 0.94     |
| Center of the turbine passage | 1.26     |
| Turbine outlet                | 1.85     |
| Diffuser                      | 2.60     |

3 | MODELING AND SIMULATION

The finite volume based three-dimensional, viscous, and compressible Reynolds-averaged Navier–Stokes (RANS) equation has been used to solve the continuity, momentum, and energy equations using a CFD tool ANSYS CFX®. The convection terms have been discretized using a high-resolution scheme (second-order). The diffusion terms (spatial derivatives) are computed by shape functions using a finite element approach. The real gas Peng–Robinson model has been used during the numerical simulation. The discretization schemes for different governing equations have been mentioned in Table 3A. The governing equations, numerical approach, and additional numerical set-up are explained in our previous studies.18

### TABLE 3A  Discretization schemes for different governing equations

| Equation                      | Advection scheme | Transient scheme                |
|-------------------------------|------------------|---------------------------------|
| Continuity                    | Upwind           | Second-order backward Euler     |
| Momentum                      | Upwind           | Second-order backward Euler     |
| Energy                        | Upwind           | Second-order backward Euler     |
| Turbulent kinetic energy      | High-resolution  | High-resolution                 |
| Turbulent eddy frequency      | High-resolution  | High-resolution                 |
**4 | BOUNDARY CONDITIONS**

Total inlet pressure and temperature are taken at the nozzle inlet boundary. The mass flow rate has been opted at the outlet of the diffuser. The additional information regarding boundary conditions are shown in Table 3B.

**5 | GRID INDEPENDENCE ANALYSIS**

The grid independence test has been performed for the reliability of the numerical simulation. In this regard, four different grid resolutions have been used, as mentioned in Table 4. It is noticed that the variation of isentropic efficiency between the third and fourth row is very less whereas the time taken for a converged solution is higher. Therefore, the third-row grid resolution has been considered for the present study. All the numerical simulations are conducted at Dell Workstation 64 GB RAM.

**6 | EXPERIMENTAL SET-UP DEVELOPMENT**

Figure 3 shows the schematic diagram, experimental test-rig, and different components of a turboexpander. The test-rig consists of a screw compressor (KASER SIGMA BSD 72), an air filter, a Freon cooler, a cryogenic plate-fin heat exchanger (PFHX), a reservoir for turboexpander and bearing supply line, and a turboexpander assembly.

The high-pressure compressed air obtained from the screw compressor passes through a three-layer filtration system to absorb the air impurities, oil particles, and so on. After that, a Freon cooler is placed for dehumidifying the air and to decrease its temperature. Then, the high-pressure fluid supplies to PFHX in which liquid nitrogen is circulated. Finally, high-pressure (8 bar) and low-temperature (150 K) fluids are available for experimentation.

The turbine inlet and nozzle pitch circle diameter are 24.86 and 40.00 mm respectively. The shaft diameter is 16.00 mm. The turboexpander assembly is placed in the bearing housing, having an external diameter and height of 160.03 mm and 157.50 mm respectively. The brake compressor is placed on the same shaft. The main work of the brake compressor is to dissipate the power generated by the turbine. Aluminum alloy (Al 6082) is used to manufacture the radial turbine. The other stationary components are made of chromium–vanadium stainless steel.

Initially, the compressed air (5 bar) is supplied to the aerodynamic bearing units. Once the bearing units are in floating condition, the high-pressure and low-temperature air is supplied to the turboexpander unit through the main supply line.

**TABLE 3B**  Boundary conditions

| Parameter              | Value                                      |
|------------------------|--------------------------------------------|
| Inlet pressure         | 8 bar                                      |
| Inlet temperature      | 150 K                                      |
| Rotational speed       | 119,196 rpm                                |
| Mass flow rate         | 0.02–0.09 kg/s                             |
| Walls                  | Adiabatic, no-slip, hydraulically smooth   |
| turbulence intensity   | 5%                                         |
| Convergence criteria   | $10^{-6}$                                  |
| Working fluid          | Nitrogen                                   |

**TABLE 4**  Grid independence test

| No. of nodes (millions) | Isentropic efficiency | CPU time (h) |
|-------------------------|-----------------------|--------------|
| 0.84                    | 0.73                  | 43           |
| 1.61                    | 0.76                  | 73           |
| 2.81                    | 0.79                  | 137          |
| 3.19                    | 0.796                 | 156          |
The turboexpander and heat exchanger supply lines are made of flexible stainless steel pipes (Grade SS304), which are covered with perlite powder (Grade 45) and nitrile rubber to achieve the adiabatic conditions. The control valves and measuring instruments are placed at different locations to control the mass flow rate of the fluid and for obtaining the thermophysical datasets which are obtained during the experiments. The specifications of various measuring components and their ranges are mentioned in Table 5.

### RESULTS AND DISCUSSIONS

The experimental investigation has been carried out to understand the variation of isentropic efficiency, pressure ratio, blade speed ratio, and power output with a mass flow rate. The blade speed ratio is defined as the ratio of turbine inlet blade speed to spouting velocity.

### TABLE 5 Specifications of measuring instruments

| Instrument          | Company  | Range          |
|---------------------|----------|----------------|
| Pressure gauge      | Swagelok | 0–25 bar       |
| Pressure transducer | Endevco  | 0–35 bar       |
| Temperature sensor  | ADAM     | 70–800 K       |
| Flow meter          | Alflow   | 0–30 m³/h      |
| Speed sensor        | Emerson  | 0–200,000 rpm  |
| Oscilloscope        | Tektronix| 20–100 MHz     |
| Digital caliper     | Mitutoyo | 0–150 mm       |
7.1 Experimental results

Figure 4 illustrates a comparison between numerical and experimental results. It is observed that the isentropic efficiency increases by increasing the blade speed ratio (up to 0.69); after that, it decreases. Therefore, for the present turboexpander, the optimum blade speed ratio is 0.69 to achieve the maximum isentropic efficiency. The deviation between numerical and experimental results is in the range of 5% which shows that the present numerical results are reliable.

Figure 5 represents the effect of mass flow rate on the pressure ratio and power output of the turboexpander. It is noticed that the pressure ratio of the turboexpander has been increased by increasing the mass flow rate of the working fluid. The increase in mass flow rate also increases the rotational speed of the turboexpander. During this process, the pressure energy of the fluid is converted into the kinetic energy of the turboexpander. As a result, the enthalpy drop takes place that ultimately provides the power output of the turboexpander. It is interesting to notice that the pressure ratio increases after the optimum mass flow rate (0.085 kg/s). However, the power output decreases. It shows the independence of the mass flow rate on pressure ratio. This condition may arise due to optimum mass flow rate has been achieved and
the flow becomes choked. Therefore, we can say that for this particular turboexpander design, the optimum mass flow rate is approximately 0.085 kg/s. It is observed that the maximum power output has been obtained for the mass flow rate and pressure ratio of 0.075 kg/s and 2.0 respectively.

Figure 6 represents a comparison of numerical and experimental results for isentropic efficiency at different pressure ratios. It shows that the highest isentropic efficiency occurs at a pressure ratio of 2.10. After that, the efficiency decreases by increasing the pressure ratio.

7.2 Numerical results

The flow field and thermal behavior inside the turboexpander are very critical at cryogenic temperature due to random variations in density, specific heat, and molecular viscosity of the fluid. Figure 7 represents the pressure and temperature contours at different cross-sections (along the axial direction) of the nozzle and turbine. The uniform pressure distribution inside the nozzle and turbine sections indicates that the design of these components is adequate.

From Figure 7(B), it is observed that 26 K (approximately) temperature drop takes place for 8 bar and 150 K inlet pressure and temperature operating conditions. The temperature drop is obtained because of the work done by the fluid for rotating the turboexpander. As a result, the pressure energy of the fluid is converted into kinetic energy. During the energy transformation process, the fluid enthalpy decreases (Figure 8(A)). It is interesting to observe that the enthalpy reduces at a relatively higher rate inside the turbine. It may happen because of work done by the process fluid on the turbine blade.
Figure 8(B) represents the static entropy variation inside the turboexpander. It shows that the entropy increases at the outlet of the nozzle because of higher losses in these regions. However, it increases inside the turbine because of the sudden expansion (non-equilibrium) process and viscous friction in the boundary layer between the blade passages. The increase in entropy inside the turbine may also take place due to secondary losses and vortices formation which finally increases the irreversibility of the expansion process.

Figure 9 represents the velocity and Mach number contours at different axial locations. The main aim of the convergent nozzle is to deliver the highest velocity at the nozzle outlet and the flow must be in the subsonic regime. The maximum velocity at the nozzle outlet is observed to be 168 m/s for which the Mach number is approximately 0.72.

Figure 10 represents the area-averaged density and turbulence kinetic energy (TKE) variation along the axial distance of the turboexpander. It shows that the density is continuously decreasing from the nozzle inlet to the turbine outlet, whereas it increases slightly inside the diffuser due to the increase in static pressure. The increase in static pressure inside the diffuser is important for decreasing the exit energy losses of the turbine.

Figure 10(B) represents the TKE variation inside the turboexpander. The variation in TKE depends on the flow behavior, the cross-sectional area of the flow domain, and turbulence intensity inside the flow field. The results show that the highest TKE has been obtained at the inlet of the turbine. This may happen due to a relatively higher turbulence intensity is obtained in these regions (x = 15 mm) because of the high rotational speed of the turboexpander. After that, the intensity decreases as one moves toward the outlet.

Figure 11 represents the Prandtl number variation along the axial distance. It defines the thermal condition of the fluid. It is noticed that the Prandtl number decreases up to the nozzle outlet (x = 5 mm). After that, it increases at the interface of the nozzle and turbine. This may happen due to the ramming of the fluid at the interface. In these regions, the momentum diffusivity increases and that is the main reason for the increase in Prandtl number. Finally, it decreases inside the turbine because of a relatively thicker thermal boundary layer as compared to that of the velocity boundary layer. As a result, the temperature and enthalpy of the fluid decreases. Finally, a slight increase in Prandtl number has been noticed in the diffuser section (x = 16–29 mm) due to an increase in thickness of the velocity boundary layer.
The pressure and static enthalpy variation at cryogenic temperature are different as compared to normal conditions because of the difference in the thermophysical properties of the fluid. Although, it is governed by the law of isentropic expansion, it must be remembered that its exact variation inside the turboexpander is a result of flow through both stationary and rotating components. Figure 12 represents that enthalpy decreases at a higher rate inside the turbine as compared to the nozzle. It happens due to work done by the fluid on the turbine blade. However, the pressure drop is responsible for the decrease in enthalpy inside the nozzle.

8 | CONCLUSIONS

This study reports the mean-line design methodology, numerical analysis, and some aspects of the experimental analysis of a cryogenic turboexpander. The numerical analysis shows detailed flow physics and thermal variation such as pressure, temperature, Mach number, velocity, TKE, Prandtl number, and so on, at different cross-sections of the turboexpander. It is observed that the highest temperature drop (26 K) has been obtained inside the turboexpander at a pressure ratio of
2.10. Additionally, the Mach number throughout the turboexpander is in the subsonic regime, which is desirable for the stability of the fluid flow. Finally, the experimental analysis has been carried out for the performance measurement of a turboexpander at different operating conditions. The numerical results are also validated with the experimental results for a case study. It shows that the optimum blade speed ratio and mass flow rates are 0.69 and 0.075 kg/s for the maximum isentropic efficiency of the turboexpander. The study shows some valuable information for the design, numerical, and experimental performance analysis of a turboexpander at cryogenic temperature.

ACKNOWLEDGMENTS
The authors are thankful to the Board of Research in Nuclear Sciences (BRNS), the Ministry of Human Resource Development (MHRD), Government of India, and the National Institute of Technology, Rourkela for providing financial support. Additionally, the authors are thankful to Mr. Debashis Panda for his critical suggestions.

PEER REVIEW INFORMATION
Engineering Reports thanks the anonymous reviewers for their contribution to the peer review of this work.

DATA AVAILABILITY STATEMENT
Research data are not shared.

CONFLICT OF INTEREST
The authors have no conflict of interest.

AUTHOR CONTRIBUTIONS
Manoj Kumar: Methodology (Lead); Software (Lead); Writing-original draft (Lead). Rasmikanti Biswal: Investigation (Equal); Validation (Equal). Suraj K. Behera: Resources (Lead); writing-review and editing (Lead). Amitesh Kumar: Conceptualization (Supporting); Software (Supporting). Ranjit Sahoo: Conceptualization (Lead); project administration (Supporting); resources (Supporting).

NOMENCLATURE
b blade height (m)
C absolute velocity (m/s)
D diameter ratio
Lt total loss
m mass flow rate (kg/s)
P power output (kW)
rP pressure ratio
T temperature (K)
U  blade speed (m/s)
ν  blade speed ratio
W  axial velocity (m/s)
Z  number of blades
Z_r rotor axial length (m)

GREEK SYMBOLS
α  absolute velocity angle (degree)
β  absolute flow angle (degree)
ψ  stage head coefficient
φ  flow coefficient
ρ  density (kg/ m³)
ε_x  axial clearance (m)
ε_r radial clearance (m)
Ω  degree of reaction
ζ_b back friction loss
ζ_l leakage loss
η_{is} isentropic efficiency

ACRONYMS
ORC  organic Rankine cycle
PFHX plate-fin heat exchanger
RANS  Reynolds-averaged Navier–Stokes

SUBSCRIPTS
h  hub
m  meridional
r  radial
s  shroud
x  axial
0  stagnation
1  nozzle inlet
2  tip clearance
3  turbine inlet
4  turbine outlet
θ  tangential

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**How to cite this article:** Kumar M, Biswal R, Behera SK, Kumar A, Sahoo RK. Experimental and numerical analysis to predict the performance of a turboexpander at cryogenic temperature. *Engineering Reports*. 2021;3:e12346. [https://doi.org/10.1002/eng2.12346](https://doi.org/10.1002/eng2.12346)