

Development and performance evaluation of high speed cryogenic turboexpanders at BARC, India

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Abstract. Turboexpanders are a key focus area for Bhabha Atomic Research Centre (BARC), Mumbai, India in the program for development of helium refrigerators and liquefiers for intra departmental requirements. To start with, a turbine impeller with major diameter 16 mm and design speed of 264,000 RPM, suited for use in the 1st stage of a modified Claude cycle/ reverse Brayton cycle based standard helium liquefier/refrigerator, is developed. Later on, a second series of turboexpander with the same major diameter (16 mm) and design speed of 260,000 RPM is developed with “splitter” blades at the major diameter end. Yet another turboexpander series, size 16.5 mm and design speed 168,000 RPM, is also developed suited for use in the 2nd stage of a standard helium liquefier/refrigerator. The present article describes these turboexpander development efforts at BARC, including results obtained during field trials with the BARC helium refrigerator and liquefier.

Table 1. Nomenclature

U	Turbine impeller tip speed	η	Turbine isentropic efficiency
D	Turbine impeller major diameter	θ	Turbine characteristic flow
P_{in}	Pressure at turbine nozzle inlet	ω	Turbine impeller angular speed
P_{out}	Pressure at turbine diffuser exit	Δh_{os}	Turbine isentropic enthalpy drop
T_{in}	Temperature at turbine nozzle inlet	\dot{m}	Mass flow rate through turbine
T_{out}	Temperature at turbine diffuser exit	a_1	Speed of sound at nozzle inlet
h_{in}	Enthalpy at turbine nozzle inlet	P_R	Pressure ratio across the turbine
h_{out}	Enthalpy at turbine diffuser exit	M_u	Ratio of impeller tip speed to sound speed at nozzle inlet
s_{in}	Entropy at turbine nozzle inlet	$h_{out,isen}$	Enthalpy (isentropic) at turbine diffuser exit
ρ_1	Density of helium gas at nozzle inlet	C_s	Isentropic velocity

1. Introduction

The coefficient of performance (COP)/liquid yield of a modern helium refrigeration/liquefaction system is largely dependent on the performance of the cryogenic turboexpanders employed in the thermodynamic process cycle. The effect of isentropic efficiency improvement of turboexpanders on Carnot efficiency of helium liquefier/refrigerator process is documented in literature [1]. Bhabha Atomic Research Centre (BARC), Mumbai has been involved in the development of cryogenic turboexpanders to cater to the development program of helium liquefiers and refrigerators of different capacities for



departmental usage [2, 3]. Three different series of turboexpanders, Series A, B and C are developed and subjected to field trials. The series A and B correspond to the first expansion stage of a standard helium refrigerator/liquefier while series C caters to the second expansion stage. Liquefaction of helium as well as 4.5 K refrigeration has already been demonstrated (figure 1) and reported [4] using Series B and C turboexpanders in a modified Claude cycle based process.

2. Major features of the turboexpander system

The BARC turboexpander system consists of a shaft with a turbine and a brake compressor impeller mounted at the two ends (figure 2). The vertically oriented shaft consists of a collar, which together with the thrust bearings on its either side forms the axial bearing system. This bearing system takes up the thrust load due to difference of pressure between the expansion turbine and brake compressor ends. The radial bearing system is responsible for rotor alignment and stable operation at high speeds. The turbine expansion system consists of an inlet nozzle for flow guidance, a rotating expansion turbine impeller and a diffuser for pressure recovery. The brake compressor system consists of a compressor wheel, an inducer, an inlet throttle valve and a water-cooled heat exchanger unit (figure 2). The process gas helium expands through the turbine system, performs work on the turbine and thus cools down. This work is utilized by the brake compressor to increase pressure (and temperature, as a consequence) of flowing helium gas through its circuit. This circulating gas is throttled back to the inlet of the brake compressor inducer through the brake valve (figure 2). The heat generated through this compression process is dissipated via the water-cooled heat exchanger unit.

After initial investigations and studies based on literature on the subject [5, 6, 7, 8], a series of turbine impellers belonging to the typical 90° Inlet Flow Radial (IFR) turbine class, which suited very well the BARC specifications (table 2), are decided to be developed. The overall design of the IFR turbines are based on Balje [5] and Kun [6]. For the blade design, methods prescribed by Hasselgruber [7] with inputs from Balje [8] are adopted.

The Series A turbines (figure 3), nozzles (figure 4) and diffusers are designed catering to the first expansion stage of a reverse Brayton cycle based 20 K helium refrigerator serviced by a standard helium process compressor. Based on the process flow – head requirements, impeller size and design speed are computed using design charts [5]. Considering the small impeller size (major diameter 16 mm) and coupled by the fact that standard milling cutters lower than 0.5 mm in diameter are not readily available, the number of blades are limited to 13. This is much lower than the optimum reported in literature [5, 9], prompting the development of a second series (Series B) of turbine impellers with “splitter” blades (figure 5). A total number of 8 full blades are designed and 8 more “splitter” blades are interspaced at the inlet section (major diameter end) to eliminate the eddy and vortex losses [9, 10] within the blade passage. Moreover, a design feature in the form of negative incidence of -30° is introduced in the nozzle following reports [9, 10] that maximum efficiency is obtained at negative incidence values ranging from -20° to -30° at the inlet to IFR turbines. As another design enhancement, the nozzle and diffuser in Series B are combined together to reduce misalignment effects. In addition to the design aspects, the parameters of Series B turboexpanders are slightly altered (table 2) to cater to the first stage of a modified Claude cycle helium refrigerator/liquefier developed by BARC [4].

A third series of turbine (Series C) with a large pressure ratio and computed major diameter 16.5 mm, catering to the 2nd (and final) expansion stage of the same Claude cycle liquefier/refrigerator, is also developed. The Series C turbines (figure 6), of standard construction without splitter blades, may also be used for the second stage of the Brayton cycle refrigerator. Following the same design methodology as that of the turbines, brake compressor impellers are also developed with designed major diameter of 28 mm for Series A & B (figure 7) and 35.5 mm for Series C (figure 8). The larger size of the Series C brake compressor impellers owes to the fact that the rotor design speed is lower and the compressor operates in a lower pressure domain. The material of construction of all the turbine and brake compressor impellers is high strength aluminium alloy (7075T6).



Figure 1. The 4.5 K BARC helium liquefier/refrigerator [4].

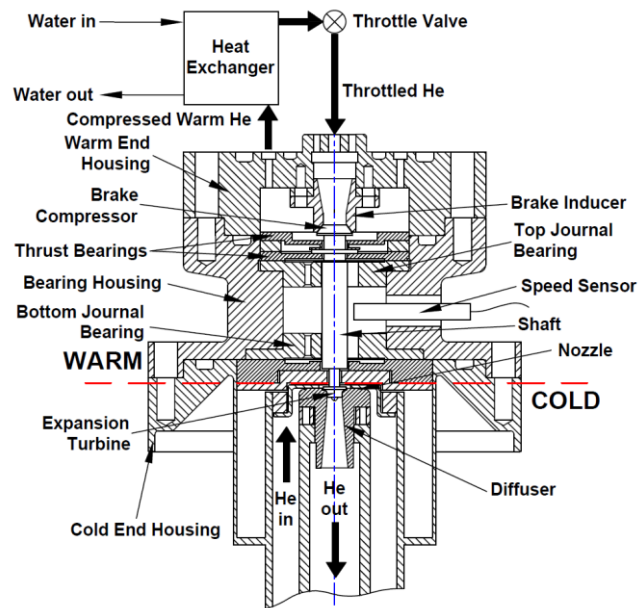


Figure 2. Schematic of the BARC cryogenic turboexpander system.

Table 2. Comparison of major design and operational parameters of expansion turbine series.

Parameter	Series A		Series B		Series C	
	Design	Operation*	Design	Operation*	Design	Operation*
P_{in} (MPa)	1.65	1.285	1.2	1.013	0.649	0.576
P_{out} (MPa)	1.1	0.759	0.65	0.492	0.195	0.174
T_{in} (K)	70	67.81	50.09	46.00	13.56	14.45
T_{out} (K)	63.3	59.74	42.5	37.76	9.58	10.47
Rotational Speed (Hz)	4400	4295	4500	4447	2833	2826
Mass flow rate (g/s)	50	48.1	45	46.7	45	41.5
Power developed (W)	1820	2084	1824	2044	779	744
Velocity ratio, U/Cs	0.65	0.58	0.67	0.64	0.66	0.63
Isentropic efficiency	0.65	0.63	0.7	0.72	0.7	0.67
Characteristic flow	0.044	0.054	0.046	0.055	0.039	0.043

* Best Efficiency Point (BEP).

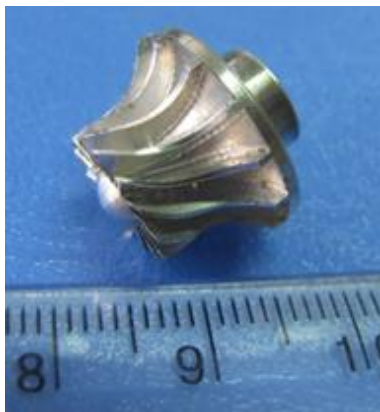


Figure 3. BARC Series A turbine impeller.



Figure 4. Inlet nozzle for BARC Series A turboexpander.



Figure 5. BARC Series B turbine impeller (with splitter blades).



Figure 6. BARC Series C turbine impeller.

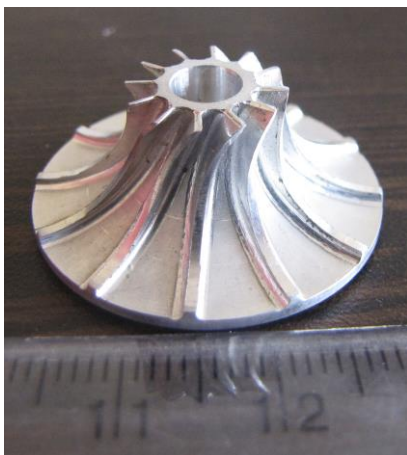


Figure 7. BARC Series A & B brake compressor impeller.



Figure 8. BARC Series C brake compressor impeller.

3. Field trials of turboexpanders

3.1. Series A

The field trials for Series A turboexpanders are carried out in a reverse Brayton cycle based 20 K helium refrigerator developed by BARC (figure 9). This helium refrigerator consists of a large cold box with two turbines in series interspaced with two heat exchangers to form a two stage Brayton cycle (figure 10). Although, two similar turboexpanders are used in both the stages, the first expansion stage turboexpander is regarded as the test point because of the similarity of the process parameters with the design condition of the turboexpander under consideration. The cold box is operated by a helium screw compressor (0.2 MPa to 1.7 MPa, 52 g/s). The mass flow into the cold box is measured using an orifice meter located at the process compressor suction end in proximity of the cold box exit. A manually operated control valve is provided at the process compressor discharge end at the entry to the cold box to control the flow of gas into it. Temperature (silicon diode) and pressure sensors (inductance based) are installed at nozzle inlet and at diffuser exit of the turboexpanders.

For measurement of rotor vibration, accelerometers are mounted on the body of the turboexpanders. These accelerometers, along with speed sensors assembled into each of the turboexpander housings, are

connected to the vibration measuring instrumentation for online monitoring and recording. To start with, the process compressor is switched on and runs in the cold box by-pass mode until the steady suction and discharge pressures of 0.2 MPa and 1.7 MPa are reached. Next, gas is slowly fed into the cold box using the manually operated control valve in the compressor discharge line. As the lowest process temperature drops, so does the turbine speed, and more gas is fed into the cold box to maintain turboexpander speeds in excess of 4 kHz. The entire process is done in quasi-steady mode so that the vibration and speed values, as are the output from temperature and pressure sensors, get registered correctly at each step. During the trials, minimum temperatures (state point 7, figure 10) of 14.9 K and 16.5 K are registered without refrigeration load and with a load of 200 W respectively. About 470 W of refrigeration load capacity is achieved at 20 K.

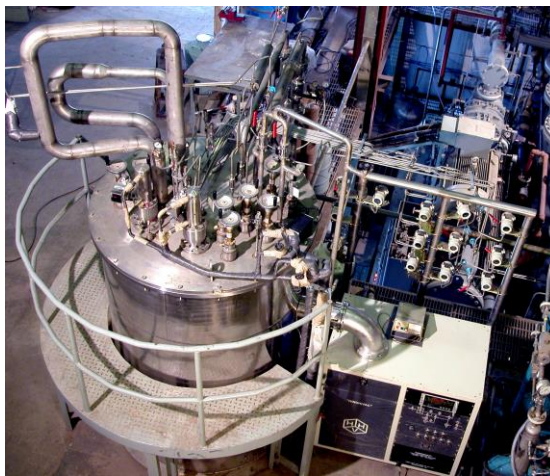


Figure 9. BARC 20 K helium refrigerator cold box used for the field test.

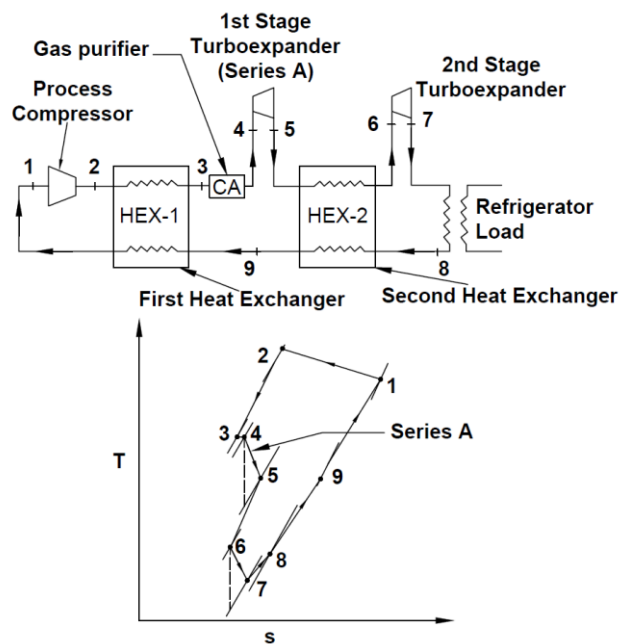


Figure 10. Process schematic of the 2-stage reverse Brayton cycle helium refrigerator.

3.2. Series B and C

The field trials for Series B and C turboexpanders are carried out in a 4.5 K helium liquefaction/refrigeration system developed by BARC (figure 1). The modified Claude cycle for the liquefaction/refrigeration process is shown in figure 11. This helium liquefier/refrigerator consists of a pre-cooler and two process turboexpanders interspaced by a multi-stream heat exchanger. The Series B and C turboexpanders correspond to the first (high pressure) and the second (low pressure) expansion stages of the process respectively. The pre-cooler turboexpander serves the purpose of augmenting liquefaction/refrigeration capacities, by utilizing the higher process compressor pressures, when available. For the experimental results presented in this article, the pre-cooler turboexpander is not in operation. Detailed description of the BARC helium liquefaction/refrigeration system may be found in Ansari et al [4].

From the process screw compressor, a maximum flow rate of about 67 g/s, measured at 1.05 bar suction pressure, is available to the cold box. The discharge pressure ranges from 13 – 17 bar(g). For the purpose of measuring flow rates through the turboexpander and JT circuits, orifice plates are installed in the piping [4]. Temperature sensors (silicon diode) with redundancy and pressure sensors (inductance based) are provided at nozzle inlet and diffuser exit, as also at other process points of significance. Similar to field trials with Series A turboexpanders, rotor speed and vibration are measured and monitored in real time and signals (especially during transients) recorded for off-line analysis.

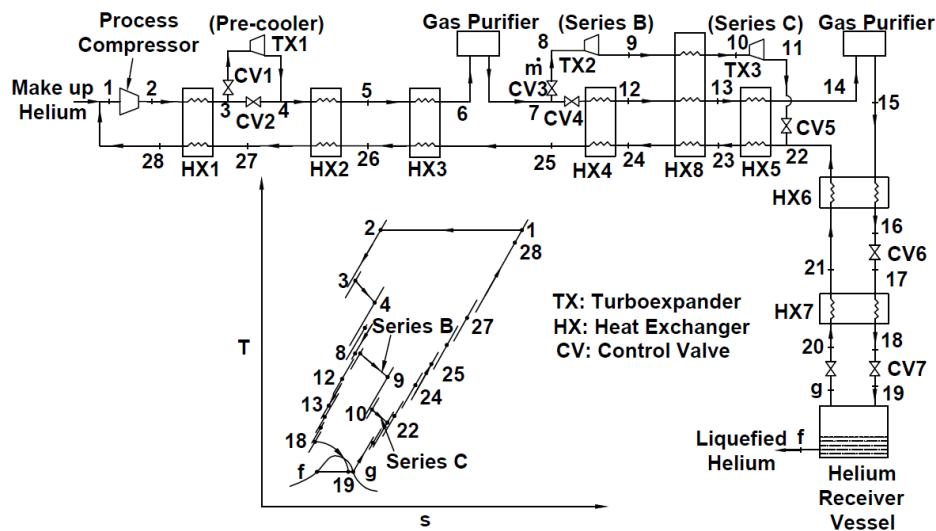


Figure 11. Process schematic of the modified Claude cycle based helium liquefaction/refrigeration system.

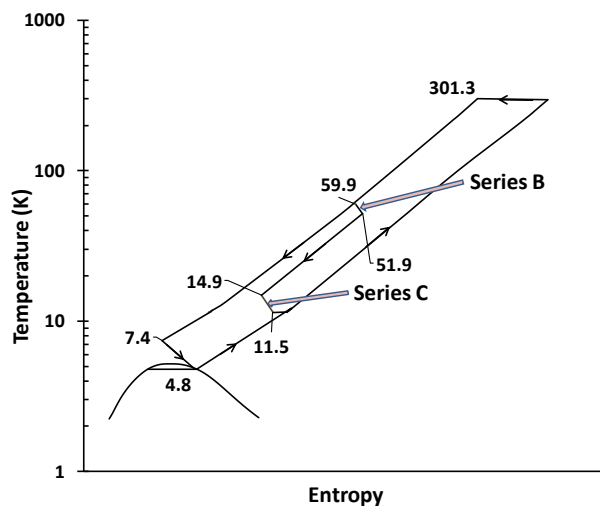


Figure 12. The T-s diagram corresponding to maximum helium liquefaction (32 l/hr).

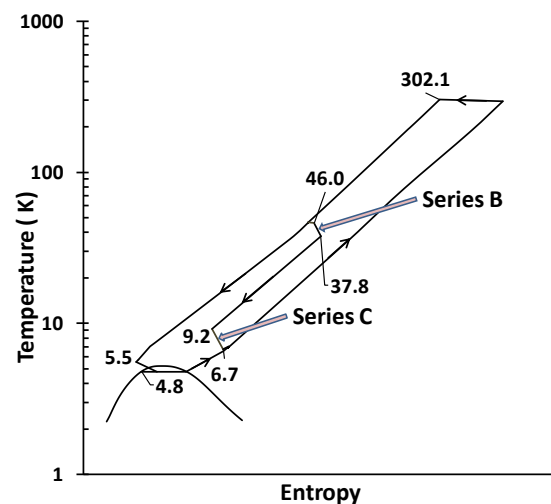


Figure 13. The T-s diagram corresponding to maximum refrigeration of 190 W at 4.8 K.

During the start-up of the system, the valve CV – 5 (figure 11) is throttled and the turboexpanders are started at full pressure available from the process compressor. A higher pressure leads to larger load carrying capacity for the gas bearings which is assumed to be beneficial by helping the rotors get over the start-up transients. As the system cools down, the pressure-drop across CV – 5 reduces, larger pressure head is available across the turbines and more gas flows into the circuit. However, the turboexpander speeds do not shoot up since the gas velocities do not change much owing to an increased density of colder gas. Slight adjustment of the inlet valve is required during different modes of operation to bring the turboexpander speed, circuit pressure and flow parameters to optimum levels. The process parameters registered during the cooldown are regarded as quasi-steady, since the process is sufficiently slow. Figures 12 and 13 presents the temperature-entropy (T-s) diagrams of the helium liquefaction and refrigeration processes respectively. Major process-point temperatures, especially, those at the inlet and exit to the Series B and C turboexpanders are marked in the figures. Maximum liquefaction and refrigeration capacities of 32 l/hr and 190 W respectively, at 4.8 K, are realized during the experimental runs [4].

4. Performance evaluation of turboexpander series

For applications restricted to a single working fluid and for high flow Reynolds numbers, the full non-dimensional performance characteristics of a turbomachine may be described by the following relationship [9]:

$$f(P_R, \eta, \theta, M_u) = 0 \quad (1)$$

It is a common practice [9, 10] to present the isentropic efficiency, η , in relation to the isentropic velocity ratio, U/C_s , which is a combination of stage pressure ratio, P_R and rotor non-dimensional speed. The rotor tip speed, U and the isentropic velocity, C_s are defined as following:

$$U = \omega \cdot \frac{D}{2} \quad (2)$$

$$C_s = (2 \cdot \Delta h_{0s})^{1/2} \quad (3)$$

The isentropic enthalpy drop across the turboexpander, Δh_{0s} , is expressed as:

$$\Delta h_{0s} = h_{in} - h_{out_isen}$$

The turbine isentropic efficiency, η , is computed as follows:

$$\eta = \frac{h_{in} - h_{out}}{h_{in} - h_{out_isen}} \quad (4)$$

$$h_{in} = h(P_{in}, T_{in}) \quad (5)$$

$$h_{out} = h(P_{out}, T_{out}) \quad (6)$$

$$h_{out_isen} = h(P_{out}, s_{in}) \quad (7)$$

The non-dimensional mass flow parameter (characteristic flow) [9], θ , is defined as:

$$\theta = \frac{\dot{m}}{\rho_1 a_1 \pi D^2 / 4} \quad (8)$$

For thermal performance evaluation of turboexpanders, the enthalpies, corresponding to the inlet and exit temperatures and pressures, are computed using HEPAK® software. The discrete data points of computed isentropic efficiencies are presented (figures 14, 15) against the computed isentropic velocity ratios and characteristic flows at different (quasi) steady state operating conditions. A comparison between design and the Best Efficiency Point (BEP) operational parameters of all the series is presented in table 2. To get an idea of the trend, data points corresponding to lower efficiencies are also plotted, however, the data cluster at near maximum efficiency are better representations of the quasi steady state operating conditions.

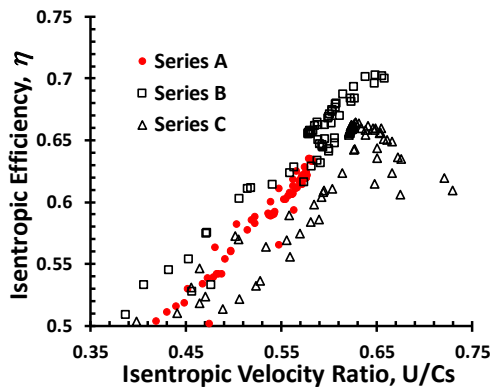


Figure 14. Variation of isentropic efficiency with isentropic velocity ratio during field trials of turboexpanders.

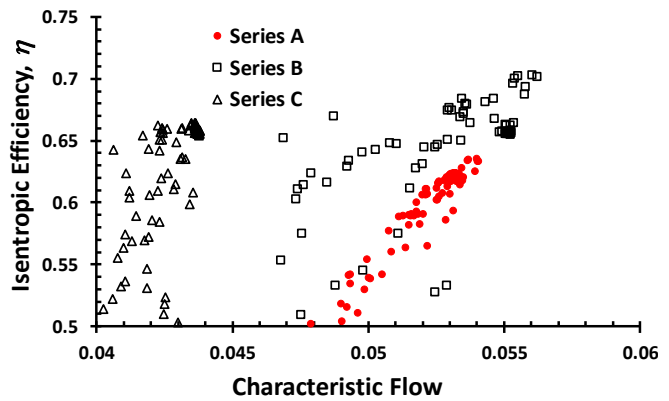


Figure 15. Variation of isentropic efficiency with characteristic flow during field trials of turboexpanders.

It is evident from figure 14 that the isentropic efficiency increases with U/C_s ratio for all the turbine series. In general, the efficiencies computed for Series B are a notch higher than those of Series A for same U/C_s ratios, hence, it appears that Series B is more efficient than Series A, which may be attributed to the described design modifications undertaken for Series B. The Series C turbine efficiency is found to be the lowest, which may be attributed to the fact that it operates (and is designed for) a much larger pressure ratio than what is normal for IFR turbines. The efficiency values of Series B and C peaks out at around U/C_s ratio of 0.63 – 0.65. On the other hand, since it is not possible to reach the design U/C_s ratio (table 2) for Series A during operation, its efficiency shows an upward trend without any sign of peaking.

What is interesting is that all the turbine series exhibit best efficiencies at maximum characteristic mass flows much higher than design (table 2, figure 15). In fact, the plot in figure 15 seems to indicate the possibility of even larger flow capacity of the turbines A and B, than that achieved during field trials, without affecting the turbine efficiency. If only the data corresponding to higher turbine efficiencies (when the plant is truly in steady state) are looked into, it seems that the efficiency of Series C turbine is also quite unaffected by the characteristic flow. This behavior of seemingly larger flow capacities of turbines than what these are designed for, without affecting the performance, is something not understood yet and needs to be investigated further.

5. Conclusion

Successful field trials of BARC turboexpander series, exhibiting isentropic efficiency of around 70%, is described in this article. The aim of the experiments described in this article is to investigate gross performance trends of developed turboexpanders during actual in-plant trials. From the preliminary analysis of experimental data, it is evident that more runs with detailed error analysis on measurements of process parameters are required to understand the off-design characteristics, especially, for the high-pressure ratio Series C turbines. The higher flow capacity of the turbines also needs to be investigated thoroughly. In this context, it may be mentioned that in order to develop turbines with even higher efficiencies, it may be necessary in future to review and improve upon the currently employed design methodology at BARC.

References

- [1] Creteigny D, Schönfeld H, Decker L and Löhlein K 2004 Adv. in Cryo. Engg. **49** 272 – 78
- [2] Chakravarty A and Singh T 2011 *Indian J. of Cryo.* **36** 1 – 9
- [3] Menon R et al. 2012 *Indian J. of Cryo.* **37** 40 – 45
- [4] Ansari N A et al. 2017 *IOP Conf. Series: Mat. Sci. and Engg.* **171/1/012095**
- [5] Balje O E 1981 *Turbomachines* (USA: John Wiley and Sons)
- [6] Kun N C and Sentz R N 1985 *Proc. Int. Conf. of Production and Purification of Coal Gas and Separation of Air (Beijing)* 1 – 21
- [7] Hasselgrüber H 1958 *Konstruktion* **10** 22
- [8] Balje O E 1970 *Trans. ASME J. Eng. Power* **70** 287 – 300
- [9] Whitfield A and Baines N C 1990 *Design of radial turbomachines* (England: Longman Scientific & Technical)
- [10] Baines N C and Sieverding C H 1992 *Radial Turbines* (Belgium: von Karman Institute for Fluid Dynamics)

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