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DESIGN OF TURBOEXPANDER FOR CRYOGENIC APPLICATIONS

Subrata Kr. Ghosh *, N. Seshaiah, R. K. Sahoo, S. K. Sarangi

Department Of Mechanical Engineering, NIT, Rourkela, Orissa.

Corresponding author mail ID: subratarec@yahoo.co.in

The indigenous design and development of turboexpander have been started at NIT, Rourkela. This paper briefly discuses the design methodology and the fabrication drawings for the whole system, which includes the turbine wheel, nozzle, diffuser, shaft, brake compressor, two types of bearing, and appropriate housing. With this method, it is possible to design a turboexpander for any other fluid since the fluid properties are properly taken care of in the relevant equations of the design procedure.

Nomenclature

b	blade height
C	absolute velocity
C_r	mean radial clearance
C_n	chord length of nozzle
d_s	specific diameter
D	diameter
d_0	feed hole or orifice diameter of bearing
h	enthalpy (J / kg)
h_{bg}	bearing clearance
K_{bg}	bearing radial stiffness
L	axial length of the journal bearing
$\dot{m}$	mass flow rate
n_s	specific speed
N	rotational speed (rev/ min)
n_h	number of holes in the bearing
p	pressure
P	power produced
Q	volumetric flow rate (m^3 / s)
R_j	radius of journal
r_{t0}	feed hole pitch circle radius of thrust bearing
r_{t1}	outer radius of thrust bearing
r_{t2}	inner radius of thrust bearing
$T_{0,in}$	turbine inlet temperature
t	thickness of blades
U	circumferential velocity
W_L	load capacity of bearing
w	width of flow passage
z	number of blades

Greek symbols

ω	rotational speed (rad/s)
ε	ratio of tip diameter to turbine wheel diameter
ε	eccentricity ratio of the bearing
λ	ratio of hub diameter to tip diameter
η	isentropic efficiency
φ	power input factor
σ	slip factor
β	relative velocity angle
γ	specific heat ratio of bearing gas
ρ	density of gas

Subscripts

0	stagnation condition
1	inlet to nozzle
1	inlet to brake compressor
2	inlet to turbine wheel
2	outlet to brake compressor
3	inlet to the diffuser (exit to wheel)
ex	discharge from diffuser
hub	hub of turbine wheel
tip	tip of turbine wheel
m	meridional component
s	isentropic condition
s	supply
tr	turbine wheel
b	brake compressor
n	nozzle
t	throat

bg bearing
c choking
d discharge

Introduction

The expansion turbine constitutes the most critical component of a large number of cryogenic process plants like air separation units, helium liquefiers and low temperature refrigerators. The primary function of expansion turbine is to produce necessary refrigeration. This is obtained by expanding the process gas through the turboexpander where power is also extracted from the fluid and enthalpy of the gas decreased. The use of expansion turbine offers greater economy, safety and flexibility. We can eliminate problems like high maintenances cost, large size, difficult valve operation and improper sealing. In this study a turboexpander has been designed having the following specifications.

Working fluid	:Nitrogen
Turbine inlet temperature ($T_{0,1}$)	:130K
Turbine inlet pressure ($p_{0,1}$)	: 6 bar
Exit pressure (p_{ex})	:1.5 bar
Throughput $(\dot{m})$	:180nm^3/hr (0.0624 kg/s)
Expected efficiency ($\eta_{T\text{-}st}$)	: 75%

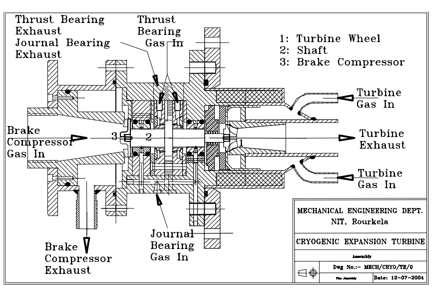

Figure 1 Longitudinal section of the expansion turbine displaying the layout of the components.

This turbine is comparable in characteristics to that developed earlier at IIT Kharagpur [1,2] and drawn heavily from that experience.

Design Methodology

A turboexpander assembly consists of the following basic units:

- the turbine wheel, nozzles and diffuser,
- the shaft,
- the brake compressor,
- two journal bearings and two thrust bearings,
- appropriate housing.

Turbine Wheel Design

Design of turbine wheel has been done following the method outlined by Balje [3] and Kun & Sentz [4], which is based on " similarity principle". The similarity laws state that for given Reynolds number, Mach number and Specific heat ratio of the working fluid, to achieve optimized geometry, two dimensionless parameters specific speed and specific diameter uniquely determine the major dimensions of the wheel and its inlet and exit velocity triangles. Specific speed (n_s) and specific diameter (d_s) are defined as:

Specific speed $$n_s = \frac{\omega \times \sqrt{Q_3}}{\left(\Delta h_{1-3s}\right)^{3/4}} \qquad (1)$$

Specific diameter $$d_s = \frac{D_{tr} \times \left(\Delta h_{1-3s}\right)^{1/4}}{\sqrt{Q_3}} \qquad (2)$$

The true values of Q3 and h_{3s}, which define n_s and d_s are not known a priori. Kun and Sentz [4], however suggest two empirical factors k_1 and k_2 for finding out these parameters.

$$Q_3 = k_1 Q_{ex} \quad \text{and} \qquad (3)$$

$$\Delta h_{1-3s} = k_2(h_{0,1} - h_{ex,s}) \qquad (4)$$

The factors k_1 and k_2 account for the difference between the states '3' and 'ex' caused by pressure recovery and consequent rise in temperatures and density in the diffuser. Following the suggestion of Kun and Sentz, we have taken k_2 = 1.03. While Kun and Sentz have taken k_1 = 1.02 in their system, we have chosen k_1 = 1.07. It is because in our design the velocity at exit from the wheel comes out to be higher, leading to a larger difference between the static and the stagnation conditions.

From Balje [3] the peak efficiency of a radial inflow turbine corresponds to the values of:

$n_s = 0.54$ and $d_s = 3.4$ (5)

Rohlik [5] prescribes that the ratio of inlet diameter to exit tip diameter should be limited to a minimum value of 1.42 to avoid excessive shroud curvature. Corresponding to the peak efficiency point:

$$\varepsilon = D_{tip} / D_{tr} = 1.45, \qquad (6)$$

From Reference [5], the exit hub to tip diameter ratio should limited to a minimum value of 0.4 to avoid excessive hub blade blockage and energy loss. Kun and Sentz [4] have taken a hub ratio of 0.35 citing mechanical considerations.

$$\lambda = D_{hub} / D_{tip} = 0.35 \qquad (7)$$

Power produced:

$$P = \eta \dot{m} \left(h_{0,1} - h_{ex,s} \right) \qquad (8)$$

From continuity equation, the ratio of blade height at entrance to the wheel is computed as:

$$b_{2tr} = \frac{\dot{m}}{(\pi D_{tr} - Z_{tr} t_{tr}) \rho_{2tr} C_{m2tr}} \qquad (9)$$

For small turbines, the hub circumference at exit and diameter of milling cutters available determine the number of blades. In summary, the major dimensions for our prototype turbine, have been computed as follows:

Rotational speed: N = 138,800 r/min
= 14537.29 rad/s
Wheel diameter: D_{tr} = 26.23 mm
Eye tip diameter: D_{tip} = 18.09 mm
Eye hub diameter: D_{hub} = 6.33 mm
Number of blades: Z_{tr} = 7
Power produced: P_{tr} = 1.95 kW.
Thickness of blades t_{tr} = 1 mm
Blade height at entrance b_{2tr} = 1.10 mm.

The blade profile has been worked out using the technique of Hasselgruber [6], which is based on the assumption of a pressure balanced flow path.

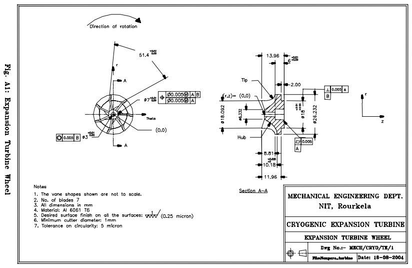

Figure 2 Expansion Turbine Wheel

Nozzle Design

An important forcing mechanism leading to fatigue of the wheel is the nozzle excitation frequency. As the wheel blades pass under the jets emanating from the stationary nozzles, there is periodic excitation of the wheel. The number of blades in the nozzle and that in the wheel should be mutually prime [7] in order to raise this excitation frequency well beyond the operating speed and to reduce the overall magnitude of the peak force. We have chosen the number of vanes as 17 in the nozzle and 7 in the wheel and a nozzle cascade height is 1.00 mm to maintain nearly equal radial velocities between the nozzle exit and the wheel entry. Size of the nozzle vanes is calculated.

Using the Continuity Equation, we calculate the throat width wt and the throat angle α_t.

$$w_{tn} = \frac{\dot{m}}{Z_n b_{tn} \rho_{tn} C_{tn}} = 1.81 \text{ mm} \qquad (10)$$

From conservation of angular momentum over the vane-less space,

$$D_{tn} = \frac{U_{2tr} D_{tr}}{C_{tn} \cos \alpha_{tn}} = 28.33 \text{ mm} \qquad (11)$$

$$D_{1n} = \sqrt{D_{tn}^{\ 2} + w_{tn}^{\ 2} - \frac{2 w_{tn} D_{tr} U_{2tr}}{C_{tn}}} \qquad (12)$$

$= 26.82$ mm

$$C_n = \frac{2 \delta_{un} S_n}{\psi_{Zn} \left[1 + \left(\cot \beta_{\infty n} + \frac{\delta_{un}}{2} \right)^2 \right] \sin \beta_{Sn}} \qquad (13)$$

$= 6.58$ mm

where

S_n = tangential vane spacing

$= \pi D_{1n}/Z_n$ = 4.95 mm.

ψ_{Zn} = Zwifel number
= 0.89 for minimum losses [3]
D_{tn} = throat diameter of nozzle
β_{sn}, $\beta_{\infty n}$ and δ_{un} are blade loading parameters defined as follows.
We have got α_{tn} = throat angle = 20.3^o
α_{0n} = inlet flow angle, which lies normally between 70^o and 90^o. Following Balje [3], we have taken α_{0n} = 78^o,

δ_{un} = cot α_{tn} - cot α_{0n}
$\beta_{\infty n} = \cot^{-1}\left[\dfrac{\cot\alpha_{tn} + \cot\alpha_{0n}}{2}\right]$ = Mean velocity angle
$\beta_{sn} = \beta_{\infty n} - 4^o$

Diffuser Design

For design purposes, the diffuser can be seen as an assembly of three separate sections operating in series – a converging section or shroud, a short parallel section and finally the diverging section where the pressure recovery takes place. The converging portion of the diffuser acts as a casing to the turbine. The recommended clearance is 2% of the exit radius, which is approximately 0.2 mm for our wheel. The straight portion of the diffuser helps in reducing the non-uniformity of flow in the wake of the wheel blades caused by the finite trailing edge thickness. The exit velocity of the diffuser should be near about 20 m/s and half cone angle is 5^0 [3]. The geometrical specifications of the diverging section are given below:

Diameter of throat of diffuser = 18.4 mm.
Diameter of diffuser exhaust = 27 mm.
Length of the diverging section = 44.65 mm

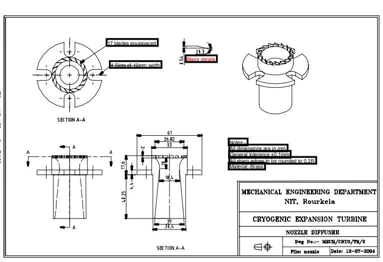

Figure 3 Nozzle & Diffuser

Shaft

Sixsmith and Swift [8] suggest that the shaft should be designed on the basis of safe critical speed, maximum stress and checked for heat conduction. Our approach, however, has been to choose the dimension based on data from comparable installations by other workers and to check for maximum stress, critical speed and heat conduction. Ino et. al [9] have chosen a shaft diameter of 16 mm for their helium turbine rotating at 2,30,000 r/min, while Yang et al [8] have chosen 18 mm for their air turbine rotating at 180,000 r/min. We have taken a shaft of diameter 18 mm and length 104 mm with a thrust collar of diameter 36 mm and thickness7.96 mm.

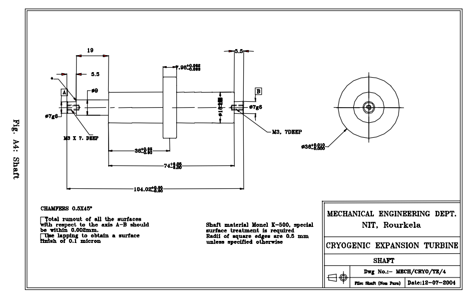

Figure 4 shaft

Brake Compressor Design

The shaft power generated by a turbine must be transferred to a braking device mounted on the shaft. We have chosen a brake compressor of purely centrifugal configuration instead of the mixed flow arrangement adopted for the turbine wheel, in order to reduce the fabrication cost. The design inputs to the brake compressor are the following:

Process gas	: Nitrogen/ air
Power to be dissipated (P)	:1.95 kW
Angular speed	:14537.29 rad/s
Inlet total pressure (p_{01})	:1.32 bar
Inlet total temperature (T_{01})	: 300 K
Expected efficiency (η_{comp})	: 60 %

From Balje [3], we have chosen the operating point corresponding to achieve proper velocity triangles within the constraints of available power and rotational speed. Under these operating conditions,

n_s = 1.87 and d_s = 2.7 (14)

$$P = \varphi\sigma\, \dot{m}\, U_{2b}{}^2 \tag{15}$$

where, φ = power input factor = 1.04 [11]
σ = slip factor = 0.82 [11]

$$U_{2b} = \omega D_{2b}/2 \tag{16}$$

There are several empirical relations for determining the optimum number of blades. We have taken $z_b = \frac{1}{3}\beta_2$ with β_2 given in degrees [12]. By following Yahya [12] we have taken the exit to inlet diameter ratio as 1.7, blade height to diameter ratio as 0.15 at inlet. The required blade height at exit

$$b_{2b} = \frac{\dot{m}_b}{(\pi D_{2b} - Z_b t_b)\rho_{2b} C_{r2b}} \tag{17}$$

Based on these data the following geometrical specifications have been computed:

D_{1b} = 22.27 mm,	D_{2b} = 37.86 mm,
b_{1b} = 4.45 mm,	b_{2b}=2.51mm,
z_b =12	t_b = 1.5 mm,

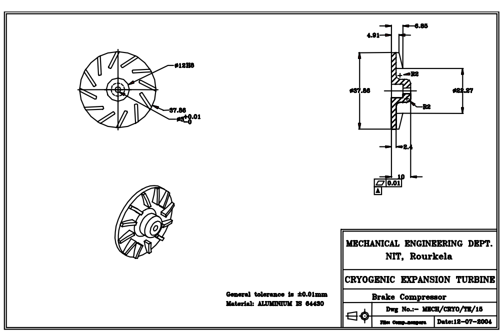

Figure 5 Brake Compressor

Bearing Design

Successful development of a turboexpander strongly depends on the performance of the bearings and their protection systems. We have used [13] gas lubricated aerostatic bearings, the journal bearings being rubber stabilized and the thrust bearings being of inherently compensated orifice type. The main advantages of these bearings are high stability to self-excitation and external dynamic load and fewer constraints on fabrication, albeit at the cost of some process gas consumption.
The dimensionless load capacity:

$$\overline{W} = \frac{W_L}{p_{dbg}\left(2R_j L_j\right)} \text{ (for journal bearing)} \tag{18}$$

$$\overline{W} = \frac{W_L}{p_{dbg} r_{t1}{}^2} \text{ (for thrust bearing)} \tag{19}$$

The bearing radial stiffness K_{bg}, under such conditions, can be expressed as:

$$K_{bg} = \frac{W_L}{\varepsilon_{bg} C_r} \tag{20}$$

The choking pressure of a feed orifice is given by the formula [14]:

$$P_{dc} = P_{sbg}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \tag{21}$$

The bearings have been designed assuming an eccentricity ratio of 0.1, supply pressure of 6.0 bar and discharged pressure of 1.5 bar. Based on the relevant data from literature, we have computed the following parameters. The chocking pressure of a feed orifice is 3.17 bar.

For journal bearing:	For thrust bearing:
R_j = 9 mm	d_0 = 0.4 mm
L_j = 18 mm	r_{t1} = 36 mm
n_h = 8	r_{t2} = 20 mm
ε_{bg} = 0.1	r_{t0} = 27 mm
d_0 = 0.3 mm	n_h = 18
C_r = 20 μm	h_{bg} =12 μm

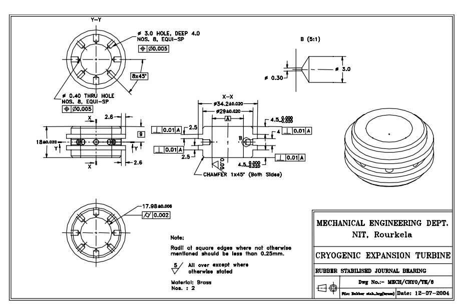

Figure 6 Journal Bearing

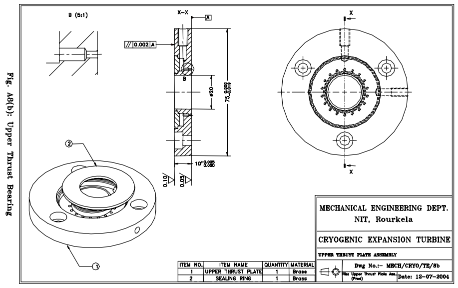

Figure 7 Thrust Bearing

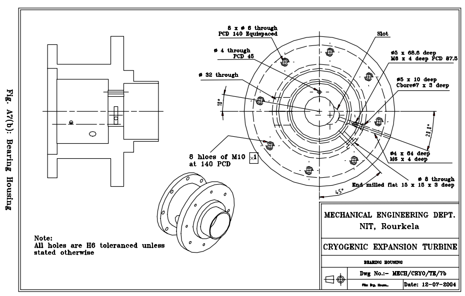

Figure 8 Bearing Housing

Conclusions

The turboexpander is an important mechanical device in the cryogenic plant, which has very wide applicability. The detailed design methodology for such turboexpander has been developed at NIT, Rourkela. This is in the process of fabrication. Suitability of the said design methodology would be confirmed by conducting experiments.

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