Development of design software for Cryogenic Turbo Expander

A Thesis Submitted for Award of the Degree of B.Tech

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CERTIFICATE

This is to certify that the project work entitled "**Development of Design software for Cryogenic Turbo Expander**" by **Asutosh Nayak** has been carried out under my supervision in partial fulfillment of the requirements for the degree of **Bachelor of Technology** during session 2009-10 in the Department of **Mechanical Engineering**, **National Institute of Technology, Rourkela** and this work has not been submitted elsewhere for a degree.

Place: Rourkela
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Abstract

This thesis provides the complete designing procedures encapsulated in an object oriented programming. The software is written in C++codes and gives the detail design of each component of the cryogenic turbo expander. The design procedure is complied an a very systematic manner due to the work of various person in this typical region. The thesis begins with the introduction of a Turbo expander. It contains the literature review which states the work done by various person with passage of time. The anatomy of turbo expander provides the complete picture and the understanding basics of each parts which helps in knowing and analyzing various parameters associated. A systematic approach of calculation is mentioned with the draw of flowcharts and step wise algorithm.

The various chapters helps in designing codes for the design of cryogenic turbo-expander.The codes are effective and isvery userfriendly.

Contents

List of figures(fig.)

I. INTRODUCTION

1.1 Expansion Turbines in cryogenic process.

Iindustrial gases such as oxygen, nitrogen and argon plays an important part of our economy. The production and its proper utilisation is considered to be an index of technological advancement of a society. Though nature has provided an abundant supply of gaseous raw materials in the atmosphere (oxygen, nitrogen) and beneath the earth's crust (natural gas, helium), we should harness and store them for meaningful use. Oxygen is a basic input to many industrial processes - steel making, ferrous non-ferrous metallurgy, welding sewage treatment, rocket propulsion and medical applications etc. Nitrogen is used as a blanket gas in most chemical processes and serves as basic raw material in production of fertilizers and ammonia based chemicals. Nitrogen of high purity finds extensive use as carrier gas in the semiconductor industry and the Liquid nitrogen provides the most effective media for many low temperature processes from shrink fitting to cryosurgery. The main application of argon as an inert gas in high temperature furnaces and TIG welding. Demand for these gases has been increasing dramatically.

The only viable source of oxygen, nitrogen and argon is the atmosphere. For producing atmospheric gases like oxygen, nitrogen and argon in large scale, low temperature distillation provides the most economical route from many point of view. In addition, many industrially important physical processes – from superconducting magnets and SQUID magnetometers to treatment of cutting tools and preservation of blood cells, require very low temperature. As discussed earlier the gases oxygen, nitrogen and argon can be separated by Air Separation method. While room temperature separation, processes based on adsorption and membrane separation are finding increasing application, particularly for production of low purity products, cryogenic distillation remains the predominant method of producing major industrial gases. The cryogenic distillation process, operating at temperature close to 100K provides the following advantages over its room temperature counterparts:

- It is Economical in large scale,
- It delivers both gaseous and liquid products,

- It helps in Production argon and other rare gases (in larger plants),
- It Produces flexible product mix.

The low temperature can be achieved in many ways. Earlier Helium and hydrogen liquefiers were used using the linde and heyland cycles . Recently cryogenic process plants are mostly preferred which are exclusively based on the low-pressure cycles. They use an expansion turbine to generate refrigeration. These plants have the advantage of high thermodynamic efficiency, high reliability and easier integration with other systems. The expansion turbine is the vital component of a modern cryogenic refrigeration or separation system. Cryogenic process plants may also use reciprocating expanders in place of turbines.But due to lesser in efficiency and other factors it is not popular.

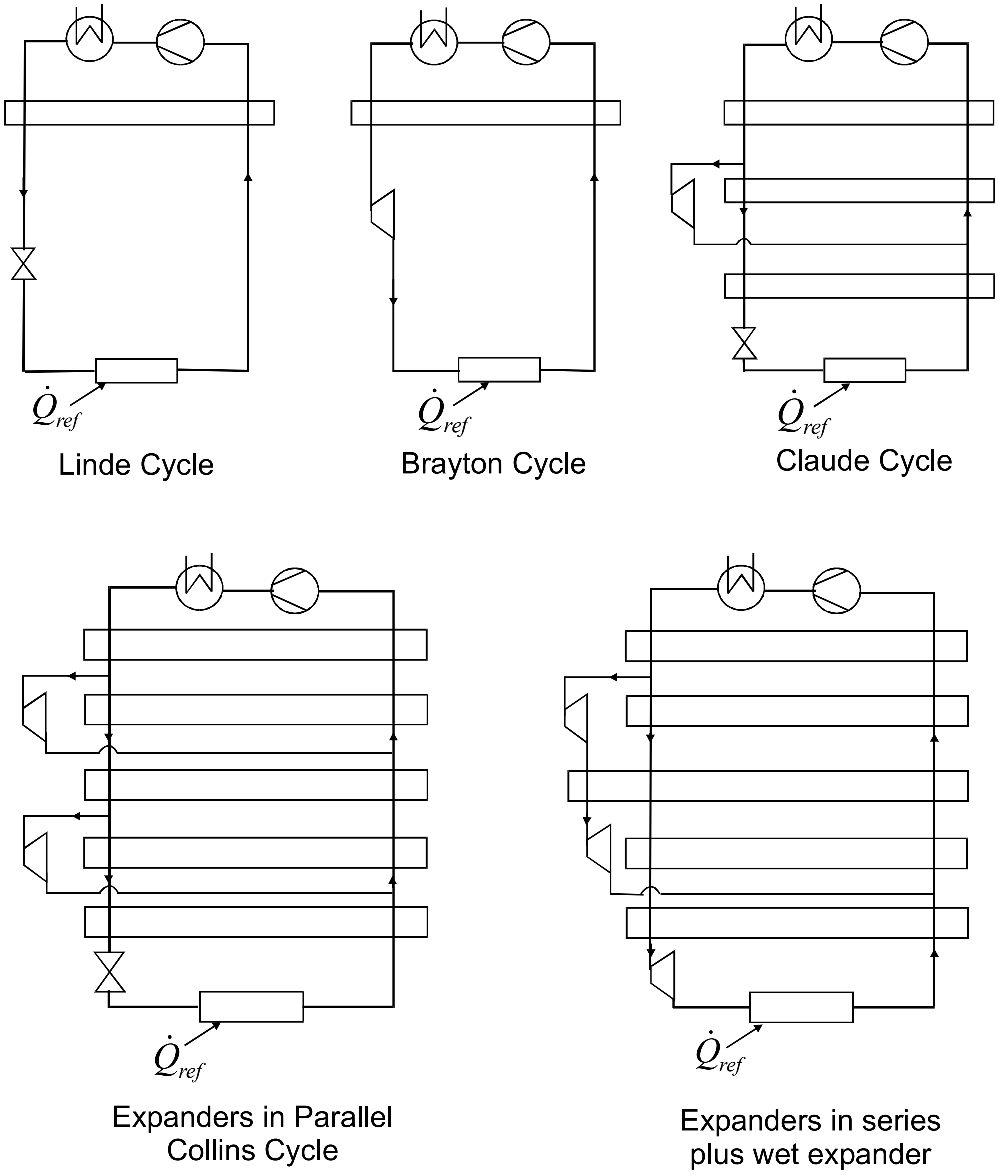
In addition to their role in producing liquid cryogens, turboexpanders provide refrigeration in a variety of other applications, such as generating refrigeration to provide air conditioning in aeroplanes. In petrochemical industries, expansion turbine is used in order to separate propane and heavier hydrocarbons from natural gas streams. The low temperature generated necessary for the recovery of ethane and does it with less expense than any other method. The plant cost is less, and maintenance, downtime, and power services are low, particularly at small and medium scales. Many LNG peak shaving plants use turbo expanders located at available pressure release points in pipelines.

Cryogenic liquefaction cycles can be grouped under three broad categories :

(a) throttle expansion cycles without an active device, e.g. Linde and Mixed Refrigerant Cycles,

(b) expander cycles, e.g., Claude, Brayton, Collins and Kapitza cycles, and

(c) regenerative refrigeration cycles, e.g. Stirling, Gifford McMahon and Pulse Tube systems.



Steady flow cryogenic refrigeration cycles with and without active expansion devices.

Some other utilities of turbine expander are:

i. Energy extraction applications such as refrigeration.
ii. Power is recovered from high-pressure wellhead natural gas.
iii. In power cycles using geothermal heat.
iv. In Organic Rankine cycle (ORC) used in cryogenic process plants in order to achieve overall utility consumption.

v. In paper and other industries for waste gas energy recovery.
vi. Freezing or condensing of impurities in gas streams.

The importance of the expansion turbine as an industrial product is well established. Unlike their counterparts in aircraft propulsion or power generation, cryogenic turboexpanders are generallyy small in size and need to operate continuously for years. This is made possible by use of gas lubricated bearings, having process gas as the lubricant. While larger machines use axial flow geometry;mixed flow, radial inlet and axial discharge, configuration is adopted by universal cryogenic system. Multistaging is difficult to achieve with radial or mixed flow geometry. Therefore, cryogenic turbines always adopt single stage expansion, irrespective of the expansion ratio.

1.2 Development of turo-expander in India

In a modern cryogenic plants a turbo expander is one of the most vital components- be it an air separation plant or a small cryocooler. Industrially advanced countries like the USA, European countries, Japan, Russia etc. have already been advanced with this technology and attained state of the art.

In India, the existing air separation plants do not have the capacity to meet ever-growing demands of pure cryogenic gases. While many of the plants are equipped with facility for both liquid and gas withdrawal, liquid withdrawal leads to severe drop in gas production and/or loss of purity. At the root of this problem is limited refrigeration capacity of the basic Linde-Hampson system. A Plant based on expansion engine operates at higher pressure and needs regular maintenance. The above difficulties can be overcome by using turbo expander based plants which offers reduced energy cost, flexible product mix, higher purity and more reliable operations. In the realm of higher technology such as in nuclear science, space, defense, superconductivity-liquid helium and hydrogen are very essential commodities. An expansion turbine is a key element for these plants.While most of the components of a cryogenic plant can either be indigenously fabricated or can be bought from the open markets, an expansion turbine can not be procured unless: we build our own technology.

2.LITERATURE REVIEW

The expansion turbine or the turboexpander is one of the important component of most cryogenic system. Since the turboexpander plays the role of the main cold generator, its properties – reliability and working efficiency, to a great extent, affect the cost effectiveness parameters of the entire cryogenic plant. The concept that a turbine can be used as a refrigerating machine was first introduced by Lord Rayleigh. In his letter of 26 June 1898 to *Nature,* he suggested the use of turbine instead of a piston expander for air liquefaction because of practical difficulties being encountered with the low temperature reciprocating machines. In this letter, Rayleigh described the most important function of any cryogenic expander, which is to production of the cold, rather than the power produced. This followed a series of early patents on cryogenic expansion turbine.

Therefore the turboexpander has attracted the attention of a large number of researchers over the years. Investigations involving applied as well as fundamental research, experimental as well as theoretical studies, have been reported in this literature. Critical reviews and specialised technical articles on the subject have been published in journals such as C*ryogenics* and *Turbomachinery*, and in major conference proceedings such as *Advances in Cryogenic Engineering* and *Proceedings of the International Cryogenic Engineering Conference*.

2.1 A Historical Perspective

Collins and Cannaday and Sixsmith have presented detailed reviews of the history of turboexpander development. A brief summary of their accounts is given below to help the reader appreciate the full dimension of the subject.

The concept that a turbine can be used as a refrigerating machine was first introduced by Lord Rayleigh in 1898 . He emphasized the most important function of a cryogenic expander, i.e. the production of low temperature rather than mechanical power. Followed by this suggestion, a series of early patents came out on cryogenic expansion turbine. In 1898, a British engineer named Edgar C. Thrupp patented a simple liquefying machine using an expansion turbine. Thrupp’s expander was a double-flow device with cold air entering the centre and dividing into two oppositely flowing streams. At about the same time, Joseph E. Johnson in USA patented an apparatus for liquefying gases. His expander was a De Laval or single stage impulse turbine. A fraction of the in-flowing air condensed in the turbine nozzle and then fell to the bottom of the liquefaction chamber for collection and run off. Other early patents include expansion turbines by Charles and Commett (1894) and Davis (1922).

Successful commercial application of an expansion turbine for gas liquefaction was done at the Linde Works in Germany in early 1930s . The device was an axial flow single stage impulse turbine, which was later replaced by an inward radial flow machine of impulse cantilever type by an Italian inventor named Guido Zerkowitz. One feature of this new design was a reversing chamber fitted inside the turbine wheel to give a second admission of gas to the moving blades. In this way velocity compounding could be achieved with a consequent reduction in the wheel speed. Zerkowitz's patent gave many details of turbine construction to reduce refrigerative and piping losses. For example, the shaft bearings were to be entirely outside the turbine housing, well removed from the cold zone.

Following Kapitza's recommendations, all subsequent developments in the field of cryogenic turbines have used the radially inward flow arrangement. One of the first well documented air liquefaction turbines was that designed by the Elliot company and constructed by the Sharples company in USA. The turbine, described by Swearingen, was a radial inflow, reaction type machine, having a design speed of 22,000 r/min. The turbine was supported on ball bearings.The radial inflow geometry thus became the standard configuration for small and medium sized cryogenic turbines.

Working on the small gas bearing turboexpander commenced in the early fifties by Sixsmith at Reading University on a machine for a small air liquefaction plant [3]"as quoted by Ghosh S.K. In 1958, the United Kingdom Atomic Energy Authority developed a radial inward flow turbine for a nitrogen production plant [2] as quoted by Ghosh S.K. During 1958 to 1961 Stratos Division of Fairchild Aircraft Co. built blower loaded turboexpanders, mostly for air separation service [1] as quoted by Ghosh S.K. Voth et. al developed a high speed turbine expander as a part of a cold moderator refrigerator for the Argonne National Laboratory (ANL) [4]. The first commercial turbine using helium was operated in 1964 in a refrigerator that produced 73 W at 3 K for the Rutherford helium bubble chamber [2] as quoted by Ghosh S.K.

A high speed turboalternator was developed by General Electric Company, New York in 1968, which ran on a practical gas bearing system capable of operating at cryogenic temperature with low loss [5-6]. National Bureau of Standards at Boulder, Colorado [7] developed a turbine of shaft diameter of 8 mm. The turbine operated at a speed of 600,000 rpm at 30 K inlet temperature. In 1974, Sulzer Brothers, Switzerland developed a turboexpander for cryogenic plants with self acting gas bearings [8]. In 1981, Cryostar, Switzerland started a development program together with a magnetic bearing manufacturer to develop a cryogenic turboexpander incorporating active magnetic bearing in both radial and axial direction [9]. In 1984, the prototype turboexpander of medium size underwent extensive experimental testing in a nitrogen liquefier. Izumi et. al [10] at Hitachi, Ltd., Japan developed a micro turboexpander for a small helium refrigerator based on Claude cycle. The turboexpander consisted of a radial inward flow reaction turbine and a centrifugal brake fan on the lower and upper ends of a shaft supported by self acting gas bearings. The diameter of the turbine wheel was 6mm and the shaft diameter was 4 mm. The rotational speeds of the 1st and 2nd stage turboexpander were 816,000 and 519,000 rpm respectively.

A simple method sufficient for the design of a high efficiency expansion turbine is outlined by Kun et. al [11-13]. A study was initiated in 1979 to survey operating plants and generate the cost factors relating to turbine by Kun & Sentz [12]. Sixsmith et. al. [14] in collaboration with Goddard Space Flight Centre of NASA, developed miniature turbines for Brayton Cycle cryocoolers. They have developed of a turbine, 1.5 mm in diameter rotating at a speed of approximately one million rpm [15].

Yang et. al [16] developed a two stage miniature expansion turbine made for an 1.5 L/hr helium liquefier at the Cryogenic Engineering Laboratory of the Chinese Academy of Sciences. The turbines rotated at more than 500,000 rpm. The design of a small, high speed turboexpander was taken up by the National Bureau of Standards (NBS) USA. The first expander operated at 600,000 rpm in externally pressurized gas bearings [17]. The turboexpander developed by Kate et. al [18] was with variable flow capacity mechanism (an adjustable turbine), which had the capacity of controlling the refrigerating power by using the variable nozzle vane height.

A wet type helium turboexpander with expected adiabatic efficiency of 70% was developed by the Naka Fusion Research Centre affiliated to the Japan Atomic Energy Institute [19–20]. The turboexpander consists of a 40 mm shaft, 59 mm impeller diameter and self acting gas journal and thrust bearings [19]. Ino et. al [21-22] developed a high expansion ratio radial inflow turbine for a helium liquefier of 100 L/hr capacity for use with a 70 MW superconductive generator.

Davydenkov et. al [23] developed a new turboexpander with foil bearings for a cryogenic helium plants in Moscow, Russia. The maximum rotational speed of the rotor was 240,000 rpm with the shaft diameter of 16 mm. The turboexpander third stage was designed and manufactured in 1991, for the gas expansion machine regime, by “Cryogenmash” [24]. Each stage of the turboexpander design was similar, differing from each other by dimensions only produced by “Heliummash” [24].

The ACD company incorporated gas lubricated hydrodynamic foil bearings into a TC–3000 turboexpander [25]. Detailed specifications of the different modules of turboexpander developed by the company have been given in tabular format in Reference [26]. Several Cryogenic Industries has been involved with this technology for many years including Mafi-Trench.

Agahi et. al. [27-28] have explained the design process of the turboexpander utilizing modern technology, such as Computational Fluid Dynamic software, Computer Numerical Control Technology and Holographic Techniques to further improve an already impressive turboexpander efficiency performance. Improvements in analytical techniques, bearing technology and design features have made turboexpanders to be designed and operated at more favourable conditions such as higher rotational speeds. A Sulzer dry turboexpander, Creare wet turboexpander and IHI centrifugal cold compressor were installed and operated for about 8000 hrs in the Fermi National Accelerator Laboratory, USA [29]. This Accelerator Division/Cryogenics department is responsible for the maintenance and operation of both the Central Helium Liquefier (CHL) and the system of 24 satellite refrigerators which provide 4.5 K refrigeration to the magnets of the Tevatron Synchrotron. Theses expanders have achieved 70% efficiency and are well integrated with the existing system.

Sixsmith et. al. [30] at Creare Inc., USA developed a small wet turbine for a helium liquefier set up at the particle accelerator of Fermi National laboratory. The expander shaft was supported in pressurized gas bearings and had a 4.76 mm turbine rotor at the cold end and a 12.7 mm brake compressor at the warm end. The expander had a design speed of 384,000 rpm and a design cooling capacity of 444 Watts. Xiong et. al. [31] at the institute of cryogenic Engineering, China developed a cryogenic turboexpander with a rotor of 103 mm long and weighing 0.9 N, which had a working speed up to 230,000 rpm. The turboexpander was experimented with two types of gas lubricated foil journal bearings. The L'Air liquid company of France has been manufacturing cryogenic expansion turbines for 30 years and more than 350 turboexpanders are operating worldwide, installed on both industrial plants and research institutes [32-33]. These turbines are characterized by the use of hydrostatic gas bearings, providing unique reliability with a measured Mean Time between failures of 45,000 hours. Atlas Copco [34] has manufactured turboexpanders with active magnetic bearings as an alternative to conventional oil bearing system for many applications.

One of the more recent developments in the field of cryogenic turbines is the wet turboexpander, where the expanded gas leaves the turbine in a mixed phase. This device, when employed in a liquefier, increases liquid yield and improves plant efficiency by replacing the JT valve.

Wet expanders used in hydrocarbon and petrochemical industries have been discussed by Linhardt [37]. He explains that, in a wet expander, the presence of non-equilibrium expansion (supersaturation) moves the condensation process downstream of the turbine. With large expansion velocities, the resulting condensate droplets are of sub-micron size. These small droplets follow the gas streamlines without slip and thus do not impinge on or erode the turbine wheel. He further suggests that the radial turbine is not acceptable for high liquid content, because the dense liquid droplets are centrifuged outwards resulting in significant flow distortions and unacceptably low efficiency. However, Swearingen [36], an early exponent of the wet expander, has shown that a cryogenic turboexpander of radial inflow configuration discharging the fluid at its dew point has significantly higher efficiency than one discharging the gas few degrees warmer. He further points out that if the blade lean angle is within 10~15° from the radial direction, a radial turbine can act efficiently as a wet expander. Aghai et. al. [38-39] have discussed the manufacturing steps for this expander. Obata et. al. [40] have recently presented a theoretical study on the performance of a wet helium turbine. They have concluded that the performance of a wet turbine is determined more by the outlet temperature than by the presence of mixed phase inside the rotor. Timmerhaus and Flynn [35] have pointed out that the use of wet expanders is generally restricted to systems using helium as the working fluid because the latent heat of the liquid phase is less than the thermal capacity of the compressed gas.

Sixsmith et. al. [41] at Creare Inc., USA developed a small wet turbine for a helium liquefier set up at the particle accelerator of Fermi National laboratory. The gas bearing based turboexpander was designed to reliably withstand the pressure transients and resulting thrust variations occurring in wet expanders. Kato et. al.developed a wet turbine for a large helium liquefier. Their turboexpander was equipped with a relief device at the outlet of the turbine which served as a JT valve to maintain the designed exit pressure. A wet turboexpander for helium liquefier application, capable of producing 7% liquid in the outlet stream, has been developed by the Rotoflow Corporation, USA.

2.2Why development of Turbo-expander in India.

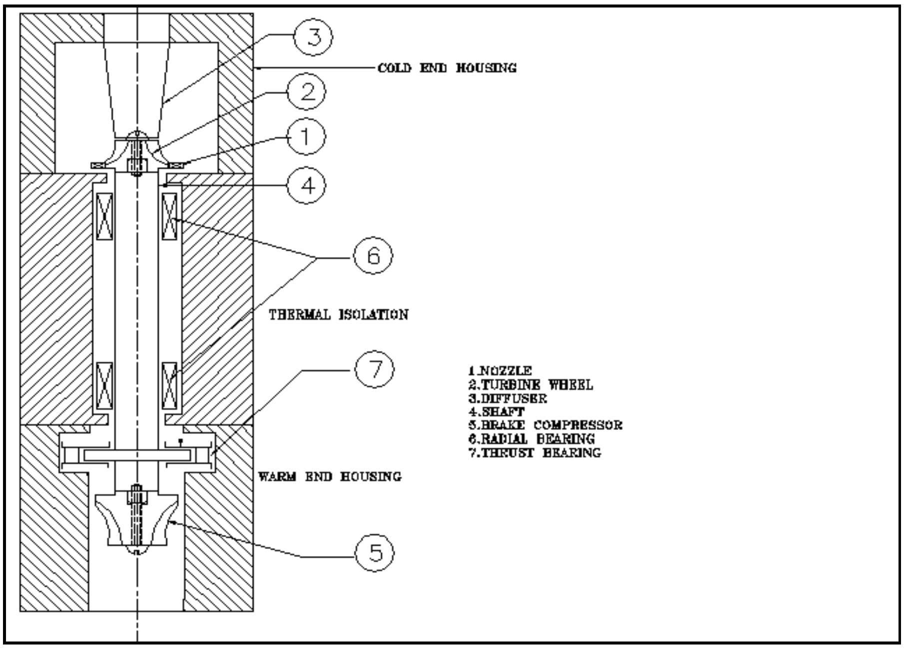
In modern cryogenic plants a turbo-expander is one of the vital components : be it separation plant or small cryocooler.Industrially advanced countries like USA , European countries ,Japan ,Russia etc have already perfected this technology and attained this art.In India the existing air separation plant do not have the capacity to meet the everincreasing demands.

A plant based on expansion engine operates at higher pressure and needs regular maintainence.The above difficulties can be overcome by using turbo-expander based plants which involves reduced energy cost, increase in efficiency and more reliable operations .

3. ANATOMY OF EXPANSION TURBINE

The expansion turbine is a radial inflow configuration and axially outflow configuration.These are suitable for moderately mass flow rate,high head and low power application.The advantage of this type lie in extracting larger work in single step due to loss free centrifugal heads as the gas leaves from larger to smaller radii.They are simple in construction and are very cmpact.The main components are:

1. Nozzle ring.
2. Turbine wheel.
3. Diffuser.
4. Shaft.
5. Brake compressor.
6. Thrust bearing.
7. Journal Bearing.



The high-pressure process gas enters the turbine through piping into the plenum of the cold end housing and enters radially into the nozzle ring. The flow accelerates through the converging passage which abides continuity laws and half the adiabatic expansion takes place through transformation of pressure energy into kinetic energy thereby reducing static temperature and pressure. The nozzle exit angle is such that the flow is directed at the correct angle to the rotating wheel to avoid the losses due to incidence, thus reducing incidence loss.

The turbine wheel is radial-axial type - i.e. the flow enters the wheel radially and exits axially. The expansion occurs due to both momentum diffusion and acceleration. Work is extracted and the process gas undergoes expansion with corresponding drop in the temperature through decrease in kinetic energy as well as centrifugal and Coriolis force.

The diffuser is a diverging passage and acts as a compressor that converts most of the kinetic energy of the gas leaving the rotor to potential energy in the form of a gain in pressure. This appears as a reduction in pressure at the outlet from the rotor. The expansion ratio in the rotor is thereby increased with a corresponding gain in efficiency and this enables greater rate of cold production to be achieved. A small temperature rise in an efficient diffuser does not offset the increased cooling achieved by the wheel.

A loading device is necessary to extract the work output of the turbine. The rotor is generally mounted in a vertical orientation to eliminate radial load on the bearings. A pair of journal bearings, apart from serving the purpose of rotor alignment, takes up the load due to residual imbalance. For horizontally oriented rotors, the journal bearings are assigned with the additional duty of supporting the rotor weight. The shaft collar, along with the thrust plates, form a pair of thrust bearings that take up the load due to the difference of pressure between the turbine and the compressor ends.

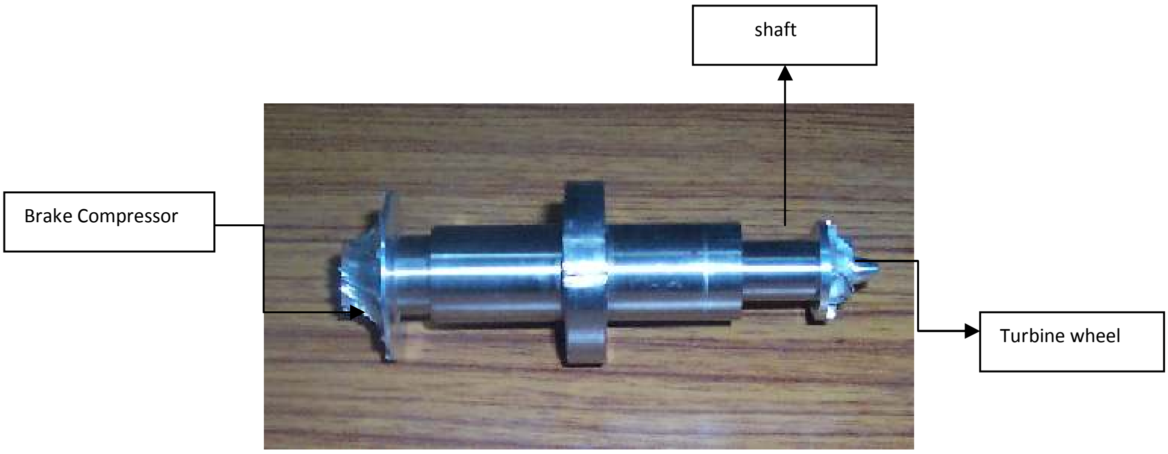
The supporting structures mainly consist of the cold and the warm end housings with an intermediate thermal isolation section. They support the static parts of the turbine assembly, such as the bearings, the inlet and exit ducts and the speed and vibration probes. The cold end housing is insulated to preserve the cold produced by the turbine.

The process of designing turbomachines is very seldom straightforward. The final design is usually the result of several engineering disciplines: fluid dynamics, stress analysis, mechanical vibration, tribology, controls, mechanical design and fabrication. The process design parameters which specify a selection are the flow rate, gas compositions, inlet pressure, inlet temperature and outlet pressure . This section on design and development of turboexpander intends to explore the basic components of a turboexpander.

A brief review of basic turbomachinery concepts is discussed in this section. These concepts are useful in understanding the analytical studies presented in later chapters. They have been taken from standard text books and reviews on turbomachinery and works of various persons[42-43].

3.1 Turbine wheel

The performance chart has become commonly accepted mode of presenting characteristics of turbomachines . Several characteristic values are used for defining significant performance criteria of turbomachines, such as turbine velocity ratio U/C_0 , pressure ratio, flow coefficient factor and specific speed . Balje has presented a simplified method for computing the efficiency of radial turbomachines and for calculating their characteristics The specific speed and the specific diameter completely define dynamic similarity. The physical meaning of the parameter pair n_s, d_s which is taken with consideration with the efficiency is that, fixed values of specific speed n_s and specific diameter d_s define that combination of operating parameters which permit similar flow conditions to exist in geometrically similar turbomachines .



Aluminum is the most ideal material for turbine impellers or blades because of its excellent low temperature properties, high strength to weight ratio and adaptability to various fabrication techniques. This material is widely used in expanders either in cast form or machined from forgings. Expander and compressor wheels are usually constructed of high strength aluminum alloy which provides better design. Low density and relatively high strength aluminum alloys are ideally suited to these wheels as they operate at moderate temperature with relatively clean gas. The weight of the wheels is reduced using low alloy which is desirable to avoid critical speed problems [49] and centrifugal stresses.

A high strength aluminium alloy is used for manufacture of Rotors. A rotor integral with the shaft would be simpler, but it was found difficult to end mill the rotor channels in high tensile titanium alloy . With tip speeds up to 500 m/sec, titanium compressor wheels machined out of solid forgings are standard industry practice [52]. Duralumin is ideal material for use in the rotor disk, it has a high strength to weight ratio and is adaptable to various vibrating techniques.

The basic objective of a cryogenic turbine is to achieve the highest possible isentropic efficiency. Unlike aircraft applications, where the turbines have to operate under widely varying conditions, a cryogenic turboexpander operates with fixed inlet and exit conditions throughout its life. In small and medium sized cryogenic plants, the throughput and head combinations lead to small values of specific speed , where the radial inflow configuration provides the highest efficiency .

During operations there are many losses which decreases the efficiency .The three major losses [43]are as follows:

1. Rotor passage loss.
2. Rotor incidence loss.
3. Rotor clearance loss.

3.2.Nozzle

The required inlet velocity and swirl can only be obtained by providing a set of static nozzles around the turbine wheel. The flow is subsonic, the absolute Mach number being around 0.95. Filippi [58] has derived the effect of nozzle geometry on stage efficiency by a comparative discussion of three nozzle styles: fixed nozzles, adjustable nozzles with a centre pivot and adjustable nozzles with a trailing edge pivot. At design point operation, fixed nozzles yield the best overall efficiency. Nozzles should be located at the optimal radial location from the wheel to minimize vaneless space loss and the effect of nozzle wakes on impeller performance. Fixed nozzle shapes can be optimized by rounding the noses of nozzle vanes and are directionally oriented for minimal incidence angle loss.

The throat of the nozzle has an important influence on turbine performance and must be sized to pass the required mass flow rate at design conditions. The continuity equations determines the exit flow angle and exit velocity from nozzle. The throat velocity should be similar to the stator exit velocity and this determines the throat area by continuity [59]. Turbine nozzles designed for subsonic and slightly supersonic flow are drilled and reamed for straight holes inclined at proper nozzle outlet angle [60]. In small turbines, there is little space for drilling holes; therefore two dimensional passages of appropriate geometry are milled on a nozzle ring. The nozzle inlet is rounded off to reduce frictional losses.
Thomas used the inlet nozzle of adjustable type. In this design the nozzle area is adjusted by widening the flow passages. The efficiency of a well designed nozzle ring should be about 95% while the overall efficiency of the turbine may be about 80% .

Fig 1[43]: Nozzle diffuser

3.2.1Different Nozzle Configurations[42]

Several configurations of variable area nozzles have been studied by Kato et. al. [61]. They classify the nozzles into three types. In a type A nozzle, the discharge angle is set by rotating the vane about a pivot. The type B nozzle is of partial admission type, where the region of admission is controlled, while in Type C the flow cross section is manipulated by changing the nozzle height by movement of the lower plate. Type B nozzles are used in turbochargers and in large cryogenic plants. Leakage of process gas around anchor pins of the actuating mechanism is substantial. Hence Type C nozzles are preferred in helium applications . Luybli and Phillipi have shown that for fixed nozzle designs, very high nozzle efficiency can be attained at the design point; but the efficiency drops sharply under off-design conditions. The nozzle ring with trailing edge pivot provides the flattest curve for the efficiency with changing mass flow rate. The Rotoflow Corporation uses variable nozzle design for their helium turbines. The plate covering the nozzles uses special springs to provide clamping force and prevent “blow-by” over a wide range of operating conditions [62]. The helium turbine developed by CCI Cryogenics also uses an adjustable height type nozzle.

The space between the nozzle and the rotor, known as the vaneless space, has an important bearing on turbine design. Watanabe et. al. empirically determined that the maximum efficiency occurs at a value of the interspace parameter k given by the relation :

$$k = \Delta r / b_n \cos\alpha_1 = 2$$

where Δr is the radial clearance between the nozzle exit and the rotor tip. Whitfield and Baines have concluded from others’ observations that the design of vaneless space is a compromise between fluid friction and nozzle-rotor interaction. They have recommended the assumption of free vortex flow in the design of the vaneless space.

There is always a tendency of foreign particles to accumulate in the space between the nozzle and the wheel which may cause surface damage by erosion. In severe cases, the trailing edges of the nozzles have been completely worn away. The use of stainless steel nozzles reduces the rate of deterioration but the only satisfactory cure is the prevention of particle entry by filtration .

3.3 Diffuser

The diffuser acting as a compresso r, converts most of the kinetic energy in the gas leaving the rotor to potential energy in the form of pressure rise. The design of the exhaust diffuser is a difficult task, because the velocity field at the inlet of the diffuser (discharge from the wheel) is hardly known at the beginning.The expansion ratio in the rotor is thereby increased with a corresponding gain in efficiency.

The efficiency of a diffuser may be defined as the fraction of the inlet kinetic energy that gets converted to gain in static pressure. The Reynolds number based on the inlet diameter normally remains around 10^5. The efficiency of a conical diffuser with regular inlet conditions is about 90% and is obtained for a semi cone angle of around 5° to 6°. According to Shepherd, the optimum semi cone angle lies in the range of 3°-5° [63]. A higher cone angle leads to a shorter diffuser and hence lower frictional loss, but enhances the chance of flow separation. Whitefield and Baines and Balje have given design charts showing the pressure recovery factor against geometrical parameters of the diffuser.

Ino et. al. [64] have given the following recommendation for an effective design of the diffuser: Half cone angle should be 5° - 6° and Aspect ratio : 1.4 – 3.3.

The inner radius is chosen to be 5% greater than the impeller tip radius and the exit radius of the diffuser is chosen to be about 40% greater than the impeller tip radius this proportion is roughly been the representative of what is acceptable in a small aero turbine application.

3.4 Shaft

The major inertia force is due to the force acting on the turbine shaft due to the revolution of its mass center and around its geometrical center constitutes. A restoring force equivalent to a spring force for small displacements, and viscous forces between the gas and the shaft surface, act as spring and damper to the rotating system. The film stiffness depends on the relative position of the shaft with respect to the bearing and is symmetrical with the center-to-center vector.

In order to eliminate the need for a heavily loaded thrust bearing,Winterbone has suggested that the diameter of the shaft be made the same as the diameter of the turbine wheel. Shaft speed is limited by the first critical speed in bending . This limitation for a given diameter determines the shaft length, and the overhang distance into the cold end, which strongly affects the conductive heat leak penalty to the cold end. In practice, particularly in small and medium size turbines, the bending critical speeds are for above the operating speeds. On the other hand, rigid body vibrations lead to resonance at lower speeds, the frequencies being determined by bearing stiffness and rotor inertia.

The important criteria in choosing the material for shaft are:

I.The critical frequency should be greater than the operating frequency so as to avoid damage.

II.The stress calculated over the surface should be less than the yield stress of the material chosen.

The material of the shaft is 410 stainless steel or K-monel. stainless steel 410 which was chosen because of its desirable combination of low thermal conductivity and high tensile strength [66]. Prevention of contact damage between the journal and bearing at start up is very important while designing the shaft. . The 18/8 stainless steel is also frequently used as a shaft material since its low thermal conductivity is advantageous in limiting heat flow into the cold region of the machine. To improve its bearing properties,it is necessary to treat the

3.5 Brake Compressor[43]

The power developed in the expanders may be absorbed by a geared generator, oil pump, viscous oil brake or blower wheel . Where relatively large amounts of power are involved, the generator provides the most effective means of recovery. Induction motors running at slightly above their synchronous speed have been successfully used for this service. This does not permit speed variation which may be desirable during plant start up or part load operation.

A popular loading device at lower power levels is the centrifugal compressor . The centrifugal compressor is ideally suited for the loading of small turbines because of its simplicity and ease of control. It has the additional advantage that it can operate at high speeds. An electrical brake can be used for small turbines whose work output exceeds the capacity of a centrifugal gas compressor.. The compression ratio ranges between 1.2 and 2.5 depending upon the speed.

3.6 Bearings[43]

3.6.1 Aerostatic thrust bearings

The gas bearing is mostly suited for supporting the rotors of these machines. Kun, Amman and Scofield [69] have both described the development of a cryogenic expansion turbine supported on gas bearings at the Linde division of the Union Carbide Corporation, USA during the mid and late 1960's. They used aerostatic bearings to support the shaft.

L'Air Liquide of France began its developmental efforts on cryogenic turboexpander from the late 1960's . The high speed rotors were supported by Gas lubricated journal and thrust bearings. This bearing system assured an unlimited life to the rotating system, due to total elimination of contact between the parts in relative motion.

In recent times, Thomas [70] reported the development of a helium turbine with flow rate of 190 g/s, working within the pressure limits of 15 and 4.5 bar. Both the journal as well as the thrust bearings used process gas for external pressurisation. The journal bearings with L/D ratio of 1.5 were designed for a shaft of diameter 25.4 mm. The bearing clearance was kept within 20 and 25 μm. The bearing stiffness was measured to be 1.75 N/μm.

3.6.2 Tilting pad journal bearings

In this design, a converging film forms between the pads and the shaft and generates the required pressure for supporting the radial load. A fraction of the bearing gas from each converging film is fed to the back of the pad, thus forming a film between the pad and the housing..

The pad floats on this film of gas. This tilting pad bearing is characterised by the absence of pivots in any form.

Gas lubricated tilting pad journal bearings were also used to support the rotor of a large helium turboexpander developed by the Japan Atomic Energy Research Institute and the Kobe Steel Limited [72]..

Contact damage between the journal and the bearing should be prevented at start up which is very important. No flaws should be found in the contact surface of the bearing and the shaft. Ino et. al. found no flaws on the heat treated surface of the journal combined with the ceramic tilting pads after 200 start/stop cycles and then it was confirmed that the combination of shaft and ceramic bearing provides significant improvement in their contact damage and thus design is safe.

The materials of Bearings are of nickel silver, which are used largely for ease of accurate machining and compatibility with respect to its coefficient of contraction in cooling. The material is selected for its anti friction properties which reduces scoring during initial testing .

3.7 Seals

In a small turboexpander, Proper sealing of process gas, is a very important factor in improving machine performance. For lightweight, high speed turbomachinery, requirements are somewhat different from heavy stationary steam turbines [73]. The most common sealing systems are labyrinth type, floating carbon rings, and dynamic dry face seals. Due to extreme cold temperature, commercial dry face seal materials are not suitable for helium and hydrogen expanders and a special design is needed.

Effective shaft sealing is extremely important in turboexpanders since the power expended on the refrigerant generally makes it quite valuable. Simple labyrinths can be used with relatively good results where the differential pressure across the seal is low. More elaborate seals are required where relatively high differential pressures must be handled. In larger machines, static type oil seals have been used for these applications in which the oil pressure is controlled by and balanced against the refrigerant pressure .

4. DESIGN PROCEDURE

The development of an expansion turbine system consist of the design of wheel, nozzles, diffuser and the brake compressor,then the prediction of performance under varying operating conditions. It has been well presented by many authors like Kun ,Hasselgruber, Katsanis, Ghosh .

In this chapter we will see the various calculations involved in the design of:

i. Turbine Wheel: calculation of rotational speed and gross dimension like major diameter,hub diameter,tip diameter etc.
ii. Design of diffuser.
iii. Design of nozzle.
iv. Design of blade profile.
v. Design of brake compressor and shaft.
vi. Off design performance prediction.

4.1NOTATION

m Mass flow rate in (Kg/s)

η Efficiency

V Volume flow rate (m^3/s)

H Available head (J/Kg)

P Pressure (Pa)

T Temperature (K)

h Enthalpy (J/Kg)

s Entropy (J/(Kg K))

ρ Density (Kg/ m^3)

ω Rotational speed (rad/s)

R Gas constant (J/(Kg K))

C_p Gas property (J/(Kg K))

γ Gamma

ξ Loss coefficient

f Function e.g. h=f (P, T) implies h is a known function of P and T

Subscripts

m Meridional component

θ Circumferential component.

r Radial component.

in Inlet

1 Nozzle exit

2 Impeller inlet

3 Impeller exit

ex Diffuser exit

s Isentropic process

Angles

α Absolute flow angle (angle between C and U)

β Relative flow angle (angle between W and U)

δ Blade lean angle (angle between W_m and axis of rotation)

θ Angular coordinate of cylindrical coordinate system

Dimensions

A Area

D Diameter

r Radius

b Blade height

t Blade thickness

Velocities

C Absolute velocity

W Relative velocity

U Circumferential velocity

4.2 Turbine Wheel

Input Parameters:

a. Inlet Pressure: Pin
b. Inlet Temperature:Pout
c. Exit Pressure: Pout.
d. Expected efficiency: n. ,and mass rate flow Mr and exit velocity Cex.

From inlet pressure and temperature, other state properties at the inlet can be evaluated.

$[h_{in}, s_{in}, \rho_{in}] = f(P_{in}, T_{in})$ {1}

For an isentropic process, the exit entropy is equal to the inlet entropy

$s_{exs} = s_{in}$ {2}

Exit state properties for an isentropic process are calculated from exit pressure and exit entropy.

$[T_{exs}, h_{exs}, \rho_{exs}] = f(P_{exs}, s_{exs})$ {3}

Expected efficiency is used to evaluate actual exit enthalpy from isentropic exit enthalpy and inlet enthalpy.

$h_{ex} = h_{in} + \eta \times (h_{exs} - h_{in})$ {4}

In order to determine the gross dimensions of the turbine wheel we need to take the help of Ns-Ds curve as obtained by Balje.

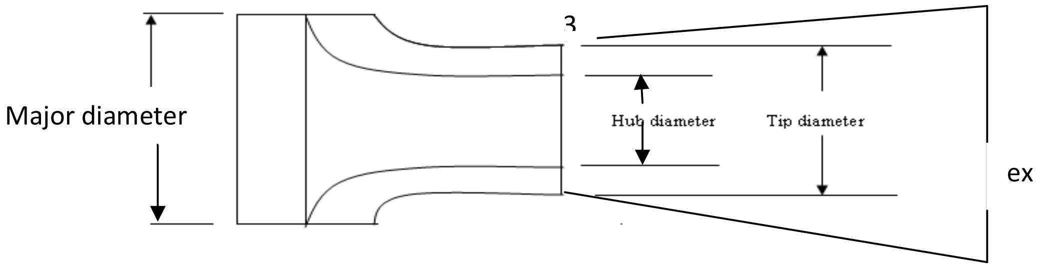
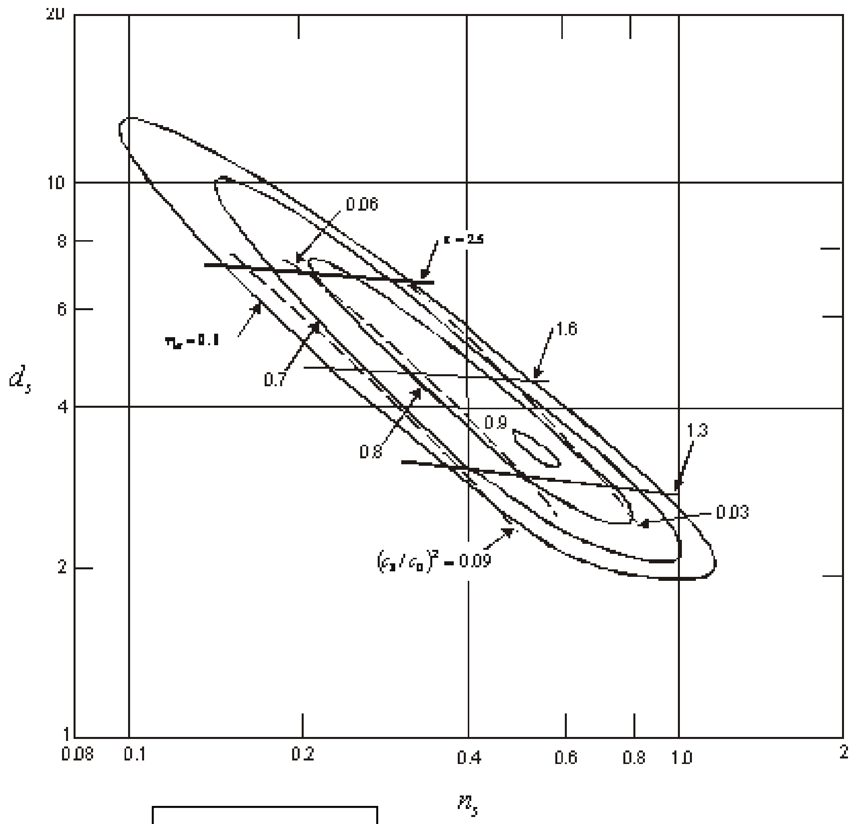


Figure d[a].



Ns-Ds Curve

The Balje's n_s-d_s diagram, shows lines of optimum geometry along with contours of constant efficiency. The following observations may be noted:

1. The validity of maximum efficiency occurs only when the Reynolds number exceeds a certain value and the Laval number is less than a certain quantity.
2. Efficiency penalties are considered for stress-limited wheels, wet turboexpanders and sub-optimum installations.
3. The diagram are based on certain values of clearance ratio, trailing edge ratio and surface roughness ratio.

4. A major advantage of Balje's representation is that the efficiency is shown as a function of parameters which helps in calculating the rotor frequency and its diameter.

Corresponding to expected efficiency the value of Ns and Ds is determined from the above graph.

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:

$$\text{Specific speed} \quad n_s = \frac{\omega \times \sqrt{Q_3}}{\left(\Delta h_{in-3s}\right)^{3/4}} \qquad 1.[43]$$

$$\text{Specific diameter} \quad d_s = \frac{D_2 \times \left(\Delta h_{in-3s}\right)^{1/4}}{\sqrt{Q_3}} \qquad 2.[43]$$

Procedure:

Take k1,k2 which accounts for the difference between state 3,ex caused by pressure recovery and consequent rise in temperature and density in the diffuser.

The specific volume (v) can be found from the chart at initial conditions. So the volumetric flow rate Qex is determined by the formula Qex=Mtr*v.

Q1=K1*Qex;

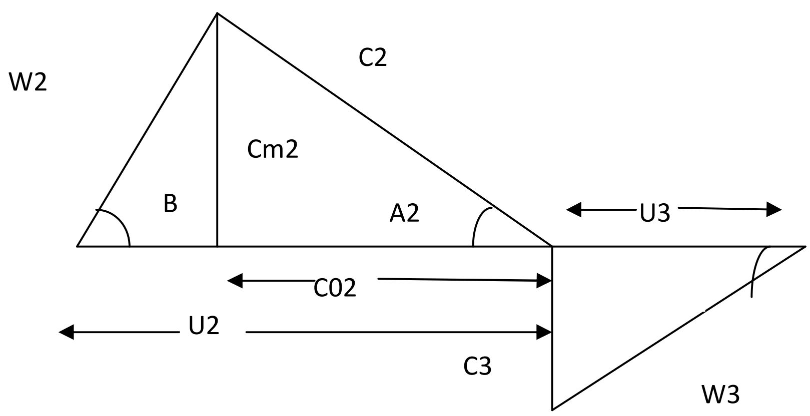
H01, Hex is found out from chart with gIven initial and final conditions.

The isentropic enthalpic drop from inlet to turbine exit $= \Delta h_{in-3s} = k_2\left(h_{0in} - h_{exs}\right)$

Dt= Inner diameter of turine wheel is calculated by the formula 1.[43] and the angular velocity "w"is calculated by 2[43].

The blade velocity at inlet of turbine U2= w*Dt/2;

<u>Velocity Triangle at inlet</u> <u>Velocity triangle at outlet</u>



Note:

A2=α2,B2=β2,A3=α3

The ratio of eye tip diameter to inlet diameter should be limited to a minimum of .7[43] to avoid excessive shroud curvature.

Dtip= .6*Dt;……………………………………3[43]

The exit hub diameter to tip diameter ratio should be maintained above a value of 0.4 to avoid excessive hub blade blockage and energy loss.

Dhub=.425*Dtip;……………………...............4[43]

Mean outlet diameter=(Dhub+Dtip)/2;

The Number of blades and the blade thickness is chosen to be 10,.6mm respectively.[43]

From geometrical considerations:

$$A_3 = \frac{\pi}{4}\left(D_{tip}^{\;2} - D_{hub}^{\;2}\right) - \frac{Z_{tr} t_{tr}\left(D_{tip} - D_{hub}\right)}{2\sin\beta_{mean}}$$

where

Z_{tr} = number of blades,

t_{tr} = thickness of the blades, and

β = exit blade angle

Now by writing equation (3.12) in the form Q_3 results:

$$Q_3 = A_3 C_3 = C_3\left[\frac{\pi}{4}\left(D_{tip}^{\;2} - D_{hub}^{\;2}\right) - \frac{Z_{tr} t_{tr}\left(D_{tip} - D_{hub}\right)}{2\sin\beta_{mean}}\right]$$

$$Q_3 = C_3\,\frac{\pi}{4}\left(D_{tip}^{\;2} - D_{hub}^{\;2}\right) - \frac{Z_{tr} t_{tr}\left(D_{tip} - D_{hub}\right)}{2} \times W_3$$

Parameter	Recommended range	Source
α_1	68°-80°	[78]
β_{3m}	50°-70°	[79]
$D_{3h}/D_{3\,tip}$	< 0.4	[78]
$D_{3\,tip}/D_2$	< 0.7	[78]
$D_{3\,m}/D_2$	0.53-0.6	[79]
b_2/D_2	0.05-0.15	[78]
U_2/C_0	0.55-0.8	[78]

Parameter	Recommended range	Source
W_3/W_2	2-2.5	
U_3/U_2	0.15-0.5	[79]
ξ_R	0.4-0.8	[78]
ξ_N	0.06-0.24	[78]
D_{3m}/D_2	0.6 – 0.67	[78]
b_2/D_2	0.09 – 0.07	[80]
Cross sectional area ratio	0.01	[80]

The Velocity triangle at turbine inlet

Assuming the incidence angle α2=26;

C2= 1000(h02-h03)+ u3*c3*cos(α3).

C02= C2 cos(α2).

Cm2= C2 sin(α2);

Tan β2= Cm2/(U2-C02).

Velocity triangle at turbine outlet

Assuming the incidence angle, α2=26

Absolute velocity at the turbine outlet,

C3=Cm3/sin(α3)

C03=C3 cos(α3)

Thermodynamic state at wheel discharge (state 3)

At the exit of the diffuser,Qex,Aex

The exit velocity is defined as :

$C_{ex} = Qex/Aex.$

This velocity is below 20 m/s as suggested by Balje [8].

Exit stagnation enthalpy:

$$h_{0ex} = h_{ex} + \frac{C_{ex}^{\ 2}}{2}$$

Exit stagnation pressure:

$$p_{0ex} = p + \frac{1}{2}\rho_{ex} C^2{}_{ex} \approx p$$ (because velocity C_{ex} is small)

From the stagnation enthalpy, h_{03}, and stagnation pressure p_{0ex}, the entropy s_3 is estimated .

And static enthalpy:

$$h_3 = h_{03} - \frac{C_3^{\ 2}}{2}$$

Tip circumferential velocity

$$U_{3tip} = \frac{\omega D_{tip}}{2}$$

Relative velocity at eye tip

$$W_{3tip} = \sqrt{U_{3tip}^{\ 2} + C_3^{\ 2}}$$

Highest Mach No

$$\frac{W_{3tip}}{C_{s3}}$$

$$\beta_{3tip} = \tan^{-1}\frac{C_3}{W_{3tip}}$$

4.3 Design of Diffuser

Kinetic energy at rotor outlet should be recovered using a diffuser.Generally a diffusing angle of 5-5 degrees is used which minimizes the loss in pressure recovery .The aspect ratio is 1.4-3.3.The diameter of the diffuser at inlet is equal to the diameter of the turbine wheel at inlet with recommended clearance of 2%of the exit radius.Diffuser outlet diameter is equal to the outlet piping diameter which gives the length of the diverging section.

The different geometric parameters to be computed are:

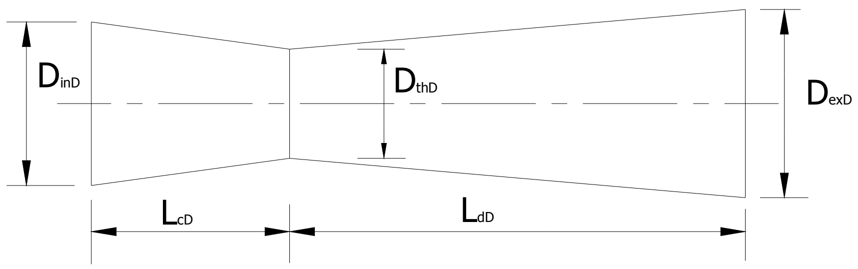
i. Diffuser exit diameter.
ii. Diffuser inlet diameter.
iii. Diffuser profile for 1^{st} and 2^{nd} section.
iv. Total diffuser length.

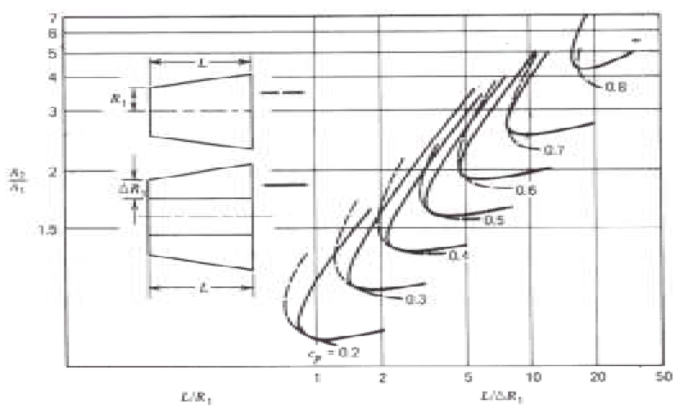
The Datas entered by the user are:

Mass rate flow:Mtr.

Discharge velocity: Cex As calculated.

Discharge Density:ρex.





Performance diagram for diffusers(reproduced from balje[8])

In order to assess the validity of the above dimensions of the diffuser, the Fig. 3.5 is reproduced from Ref [8]. From the figure, in the divergent section, the length to throat radius ratio of 8.31 and exit area to throat area ratio 2.98 give a stable operation of recovery factor of 0.7. This confirms the design of the diffuser.

1. $D_{ex} = \sqrt{\dfrac{4\times V_{ex}}{\pi \times C_{ex}}}$; diffuser exit diameter……………………………….d[1]
2. Inlet Diamter Din =Dt+ 2*radial clearance………………d[2]

   Area at inlet of diffuser =π/4 * Din*Din.
3. Diameter at throat =Dtd=Dtip+2*radial clearance………d[3]
4. Let taper angle=5degree= ¥

   Length of the diverging section of diffuser:

   Ld= (Dex-Dtd)/ (2*Tan ¥)………………………………d[4]

4.4 Nozzle Design[42]

In order to avoid incidenc loss ,the flow form the nozzle should come on to the wheels at correct angle α2.Hence, a well-designed nozzle is very necessary for an efficient turbine.

One of the important forcing mechanisms in evaluating the fatigue conditions at the wheel is the nozzle excitation frequency. As the wheel blades pass under the jets emanating from the stationary inlet blades, there will be periodic excitation proportional to the inlet nozzles and the speed of the wheel. To reduce the effect due to this periodic excitation a thumb rule is that the number of nozzles should not be integral multiple of the number of turbine blades.

Following [] the nozzle cascade height is taken as:

$$b_1 = 0.9\times b_2$$

This is in order to leave some margin for expansion in annular space above the wheel and for axial misalignment. Providing 4% vane less space the throat diameter (D_t):

$$D_t = 1.08 \times D$$

The above values (0.9 and 1.08) are commonly used but the designer is free to change them. Let C_{mt} be the meridional component at the throat of the nozzle. From the continuity equation:

$$C_{mt} = C_2 \times \frac{\rho_2}{\rho_t} \times \frac{D}{D_t} \times \frac{b_2}{b_t}$$

From the conservation of angular momentum, the tangential component of velocity at throat:

$$C_{\theta t} = r_2 \times \omega \times \frac{D}{D_t}$$

From conservation of energy:

$$h_t = h_2 + \frac{C_2^2}{2} - \frac{C_{\theta t}^2}{2} - \frac{C_{mt}^2}{2}$$

$s_t = s_2$

$[T_t, h_t, \rho_t] = f\,(P_t, s_2)$

The absolute velocity at the nozzle throat:

$$C_t = \sqrt{C_{mt}^2 + C_{\theta t}^2}$$

Using continuity throat width (W_t) and angle (α_t) are calculated as:

$$W_t = \frac{m}{b_t \times Z_n \times \rho_t \times C_t}$$

$$\alpha_t = \tan^{-1}\left(\frac{C_{mt}}{C_{\theta t}}\right)$$

From the conservation of angular momentum we get the radius of the

$$r_t = \frac{r_2^2 \times \omega}{C_t \times \cos(\alpha_t)}$$

From the cosine rule of triangles the radius of cascade discharge is given by:

$$r_1 = \sqrt{r_t^2 + \frac{W_t^2}{4} + \frac{W_t \times r_2^2 \times \omega}{C_t}}$$

Blade loading δ_u is defined as:

$$\delta_u = \cot(\alpha_t) - \cot(\alpha_0)$$

Let the mean velocity angle be β_∞, then:

$$\cot(\beta_\infty) = \frac{\cot(\alpha_t) + \cot(\alpha_0)}{2}$$

Then the stagger angle β_s:

$$\beta_s = \beta_\infty - 4^o$$

Chord length of the nozzle vane

$$c = \frac{4 \times \pi \times \delta_u \times r_1}{\Psi_z \times Z_n \times \left\{1 + \left(\cot(\beta_\infty) + \frac{\delta_u}{2}\right)^2\right\} \times \sin(\beta_s)}$$

4.4 Design of shaft[43]

It is believed that the strength of materials improves at low temperature and thus stress consideration are taken as unimportant. In reality, cryogenic turbines, because of the moderate to high-pressure ratio and low flow rates operate at high rotational speeds, leading to significant centrifugal stresses in the shaft. The shaft transmits the torque produced by the turbine to the brake compressor.. Also the turboexpander is vertically oriented and bending load is neglected due to the absence of any radial load. Important considerations in the design of the shaft are:

- number and size of components linked with the shaft,
- tangential speed on bearing surfaces,
  stress at the root of the collar

critical speed in shaft bending mode,

The major dimensions of the shaft include:

- diameter of the shaft,
- diameter of the collar and
- length of the shaft

Ino et. al [82] have chosen a shaft diameter of 16 mm for their helium turbine rotating at 2,30,000 r/min, while Yang et al have chosen 18 mm for their air turbine rotating at 180,000 r/min. A shaft of diameter 16 mm and length 88.1 mm with a thrust collar of diameter 30 mm has been selected in the present case.

V_{Surf}=w*d/2 m/s. ....s[1]

and that on the tip of the collar is 2*Vsurf. ....s[2]

A preliminary calculation considering the collar as a solid disk gives [83]

$\sigma = 1/3*\rho_{ss}*V_{surf}^2$ Ma s[3]

This value is more than recommended design stress of 230 MPa for stainless steel SS 304 , justifying the need for other material. Hence K-Monel-500 for the shaft material is chosen having design stress of 790 Mpa. By using K-Monel-500 as a shaft material the possibility of yielding of the shaft is very less.

Shaft speed is generally limited by the first critical speed in bending. This limitation for a given diameter determines the shaft length. The overhang distance into the cold end, strongly affects the conductive heat leak penalty to the cold end .

The first bending critical speed for a uniform shaft is given by the formula

$$f = 0.9\left(d/l^2\right)\sqrt{\frac{E}{\rho}} \quad \text{Hz} \qquad \text{s[4]}$$

where d is the diameter of the shaft, l is the length, E is the Young's modulus and ρ is the density of the material. Considering the shaft to be a K-Monel-500 cylinder of diameter 16.0 mm and length 88.1 mm, the bending critical speed is

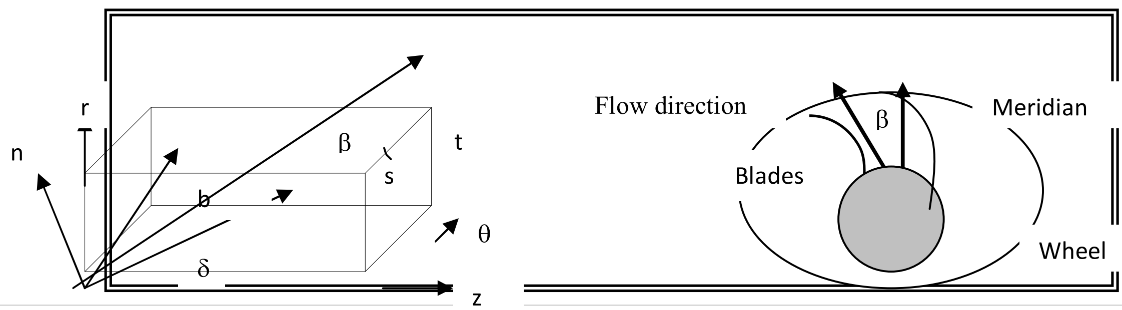
$$f = 0.9\left(\frac{0.016}{0.0881^2}\right)\sqrt{\frac{18\times10^{10}}{8440}} = 8544 \text{ Hz} = 5{,}12{,}640 \text{ r/min}$$

This is well above the operating speed of 2,18,775 r/min.

The gas lubricated bearings of a cryogenic turbine need to be maintained at room temperature to get the necessary viscosity. This requires a strong temperature gradient over the shaft overhang between the lower journal bearing and the turbine wheel. The rate of heat flow can be reduced by (a) using material of lower thermal conductivity (b) reducing the shaft diameter below the lower journal bearing and (c) by using a hollow shaft in that section.

4.5 Blade design

In the design of a turbo expander, the vital part is the turbine wheel, because losses mainly occur in the flow passage due to improper blade design. Therefore, design of the blade should be such that it will produce the flow angles and velocities required by the velocity diagrams. An analytical procedure has been outlined by Hasselgruber . This procedure requires the major dimensions of the wheel and the relative flow angle at the wheel inlet and exit. Then the blade profile is calculated.



In the figure:

t	Direction and arc length of relative streamline
s	Direction and arc length of meridian streamline
n	$\perp$ to t and s
b	$\perp$ to t and n

The profile is determined based on the theory of frictionless flow in a rotating wheel and the basic boundary conditions. The equations of motion in a moving coordinate frame have been derived equating pressure forces and the inertial forces acting on the fluid element.

The pressure forces:

$$\left\{-\frac{\partial P}{\partial t}, -\frac{\partial P}{\partial b}, -\frac{\partial P}{\partial n}\right\}$$

The inertial forces consist of

1. Forces due to relative motion with respect to the impeller.
2. Centrifugal force.
3. Coriolis force.

The governing equations come out to be:

$$-\frac{\partial P}{\partial t} + \rho \times \left(-W \times \frac{\partial W}{\partial t} + \frac{U^2}{r} \times \sin(\delta) \times \sin(\beta) \right) = 0$$

$$-\frac{\partial P}{\partial b} + \rho \times \left(-\frac{W^2}{r} - \frac{U^2}{r} \times \sin(\delta) \times \cos(\beta) + 2 \times \omega \times W \times \sin(\delta) \right) = 0$$

$$-\frac{\partial P}{\partial n} + \rho \times \left(\frac{C_u^2}{r} \times \cos(\delta) - \frac{C_m^2}{R_m} \right) = 0$$

Three characteristic functions used by Hasselgruber for calculation of the profile:

1. The first function depicts the variation of relative acceleration of the fluid from turbine wheel inlet to the wheel exit

$$f_1\left(\frac{s}{s_2}\right) = \sqrt{(\mathrm{cosec}(\beta_{3m}))^2 + \left\{(\mathrm{cosec}(\beta_2))^2 - (\mathrm{cosec}(\beta_{3m}))^2\right\} \times A}$$

where,

$$A = \left[\frac{\frac{s}{s_2} \times (k_h + 1) \times \mathrm{cosec}(\beta_2) + (\mathrm{cosec}(\beta_{3m}) - \mathrm{cosec}(\beta_2)) \times \left\{1 - \left(1 - \frac{s}{s_2}\right)^{k_h+1}\right\}}{k_h \times \mathrm{cosec}(\beta_2) + \mathrm{cosec}(\beta_{3m})} \right]^{k_e}$$

2. This function gives the relative flow angle along the flow path.

$$f_2\left(\frac{s}{s_2}\right)=\frac{1}{\operatorname{cosec}(\beta_2)+\{\operatorname{cosec}(\beta_{3m})-\operatorname{cosec}(\beta_2)\}\times\left(1-\frac{s}{s_2}\right)^{k_h}}$$

3. This function is a combination of the first two.

$$f_3\left(\frac{s}{s_2}\right)=f_1\left(\frac{s}{s_2}\right)\times\sqrt{1-f_2^2\left(\frac{s}{s_2}\right)}$$

4.6 Design of brake compressor

The shaft power generated by the turbine must be transferred to a braking device mounted on the shaft.For relatively large amount of power ,an electrical generator is mostly used as a raking device.A brake compressor is the most common device for small turbo-expander.

Design Input Parameters

a. Process Gas.
b. Power to be dissipated.(P)
c. Angular speed.(w)
d. Inlet total pressure.(p_{01})
e. Inlet total temperature(T01)
f. Expected efficiency($N_{b)}$Assume isentropic enthalpin drop as Δh_{0s}

Specific speed $n_s = \dfrac{\omega\sqrt{Q_4}}{\Delta h_s^{3/4}}$ b[i]

Specific diameter

$$d_s = \frac{D_5\ \Delta h_s^{1/4}}{\sqrt{Q_4}}$$ b[ii]

Balje has pointed out that mixed geometry is necessary to obtain the highest efficiency at these Values N_s,$D_{S.}$

From above equations , Q_4 and D_5 is determined.

Where D_5 is the diameter of the impeller at the exit.

From input parameters, P01,Mn,R,T01

Density ρ_4=.94*ρ_{04}..B[1]

Where ρ_{04}= (P01*Mn)/(R*T01)…………………B[2]

Mass rate flow M_b= ρ_4 *Q_4. B[3]

Peripheral Speed at exit=U_5= w*D5/2 B[4]

Assuming zero swirl at inlet , Power Input P;

$P = \phi\sigma_{sf}\ \dot{m}_b\ U_5^2$ B[5]

where, ϕ = power input factor = 1.02

σ_{sf} = slip factor $= \dfrac{C_{\theta 5}}{U_5} = 0.78$

$C_{\theta 5}$ = Tangential component of the absolute velocity at exit

U_5 = peripheral speed at exit = $\omega D_5/2$

Assuming exit to inlet diameter ratio as 2.25 and blade height to diameter ratio at inlet as .2.

Inlet Diameter D4=D5/2.25 B[6]

Inlet blade height B4= .2*D5 B[7]

Blade thickness of .75mm and number of blade Z_B=12 recommended.

Inlet velocities

Assuming number of blades, Z_b = 12 and a uniform thickness t_b = 0.075 mm, the radial absolute velocity C_{r4} (which is also equal to the absolute velocity C_4 in the absence of inlet swirl) is given as:

$$C_{r4} = C_4 = Q_4/((\pi D_4 - Z_b t_b)\times b_4)$$ B[8]

The peripheral velocity at inlet is computed to be:

$$U_4 = D_4\omega/2$$ B[9]

The inlet blade angle β_4 and the inlet relative velocity W_4 are computed from the inlet velocity triangle

$$\beta_4 = \tan^{-1}\frac{C_{r4}}{U_4} \quad ,$$

$$W_4 = \sqrt{U_4^2 + C_4^2} \qquad \text{B[10]}$$

The relative Mach number at inlet

$$M_{W4} = W_4 / \sqrt{\gamma R T_4}$$

This value indicates that the flow is subsonic in nature.

Exit velocities

The absolute exit velocity:

$$C_5 = \sqrt{2(h_{05s} - h_{5s})} \qquad \text{B[11]}$$

Using the value of 0.82 for the slip factor, the tangential velocity:

$$C_{\theta 5} = 0.82\, U_5 \qquad \text{B[12]}$$

$$C_{r5} = \sqrt{C_5^2 - C_{\theta 5}^2} \qquad \text{B[13]}$$

The exit blade angle:

$$\beta_5 = \tan^{-1}\left(\frac{C_{r5}}{U_5 - C_{\theta 5}}\right) \qquad \text{B[14]}$$

and the absolute exit angle :

$$\alpha_5 = \tan^{-1}\left(\frac{C_{r5}}{C_{\theta 5}}\right) \qquad \text{B[15]}$$

The relative velocity at exit:

$$W_5 = C_{r5} \operatorname{cosec} \beta_5$$

Exit temperature $$T_5 = T_{04} + \frac{\Delta h_{adst}}{C_p}$$

and exit pressure: $$p_5 = p_{04} \left(\frac{T_5}{T_{04}} \right)^{\frac{\gamma}{\gamma - 1}}$$

Density at exit: $$\rho_5 = \frac{p_5}{RT_5}$$

The required blade height at exit:

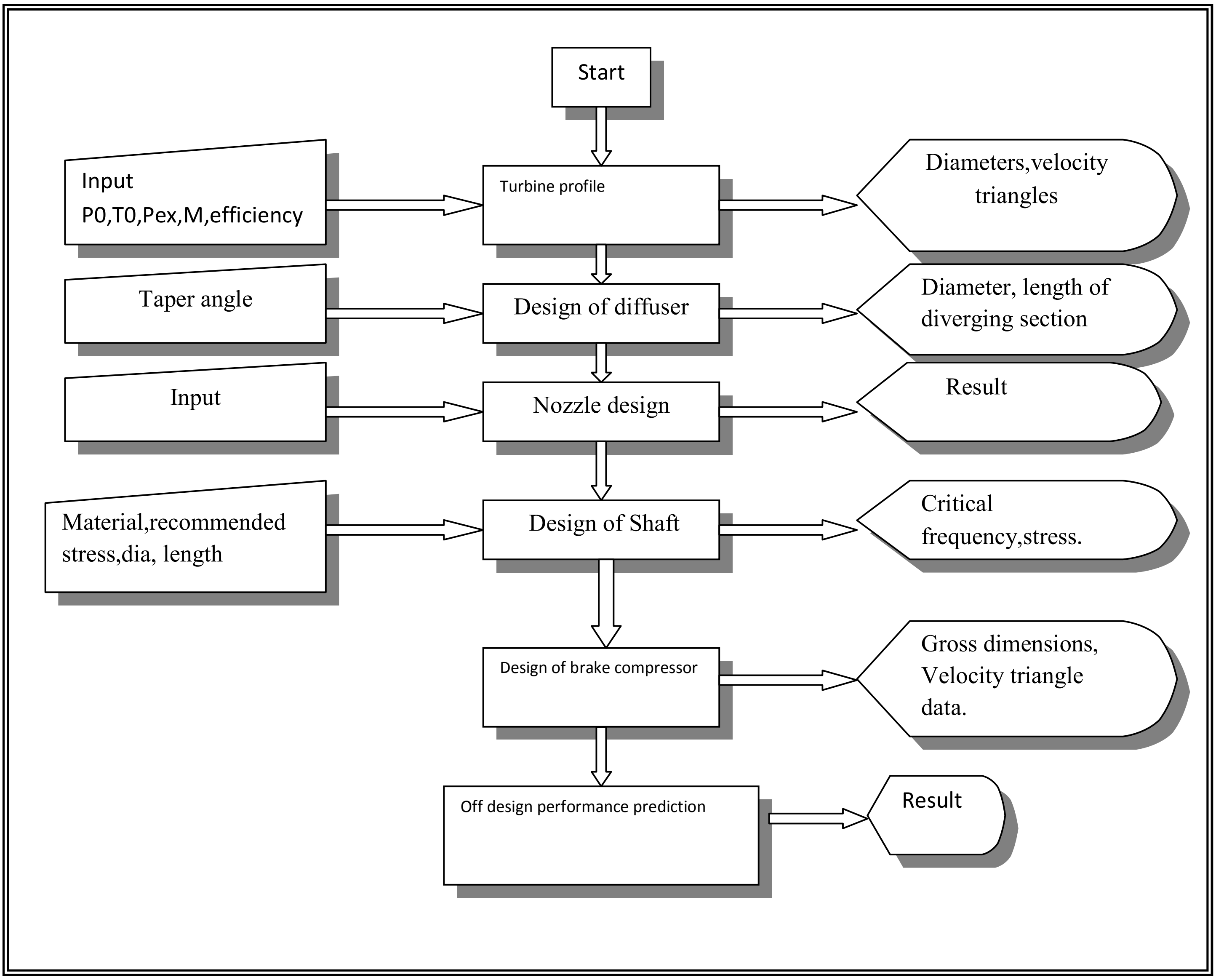
$$b_5 = \frac{\dot{m}_b}{(\pi D_5 - Z_b t_b)\rho_5 C_{r5}}$$

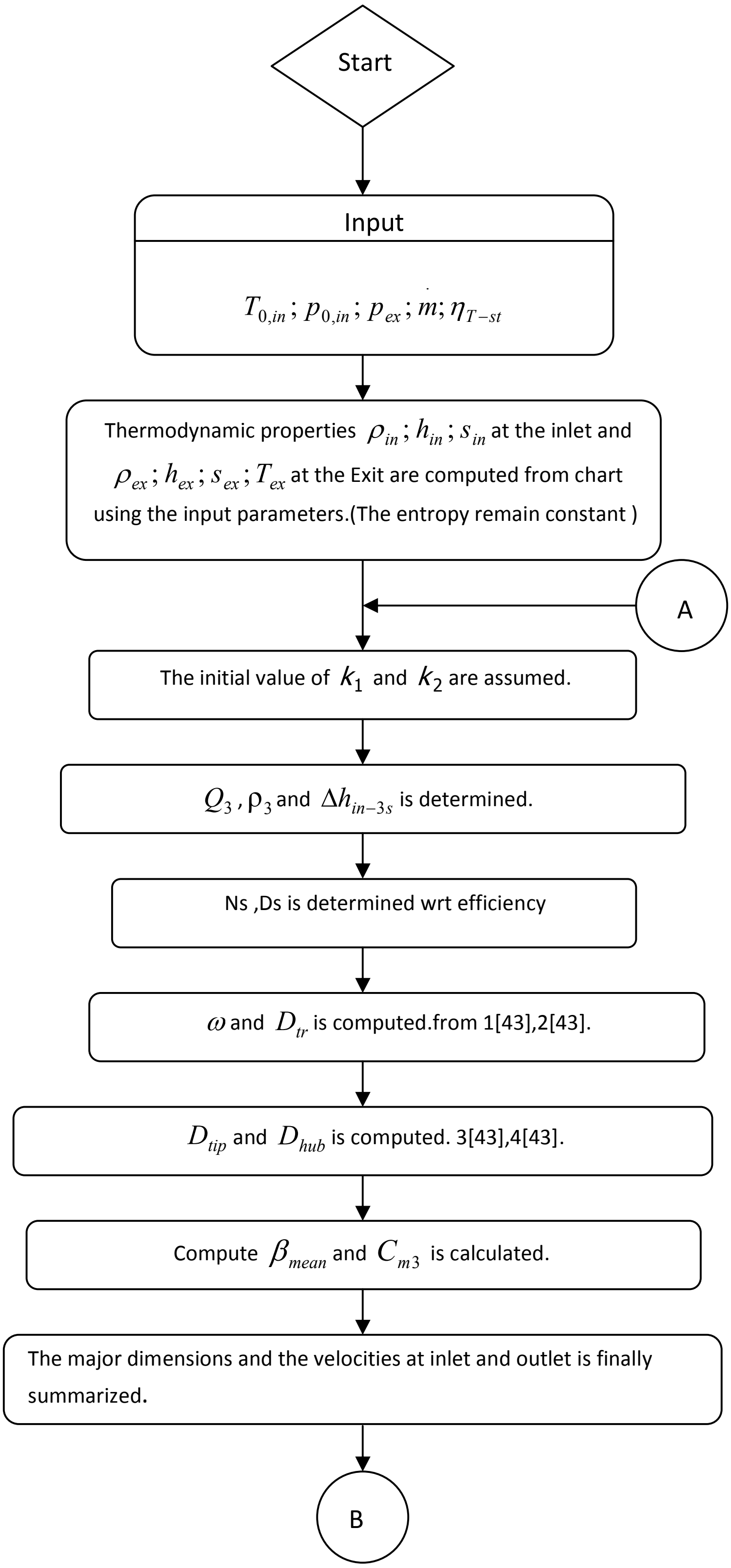
4.7 Off Design Predictions

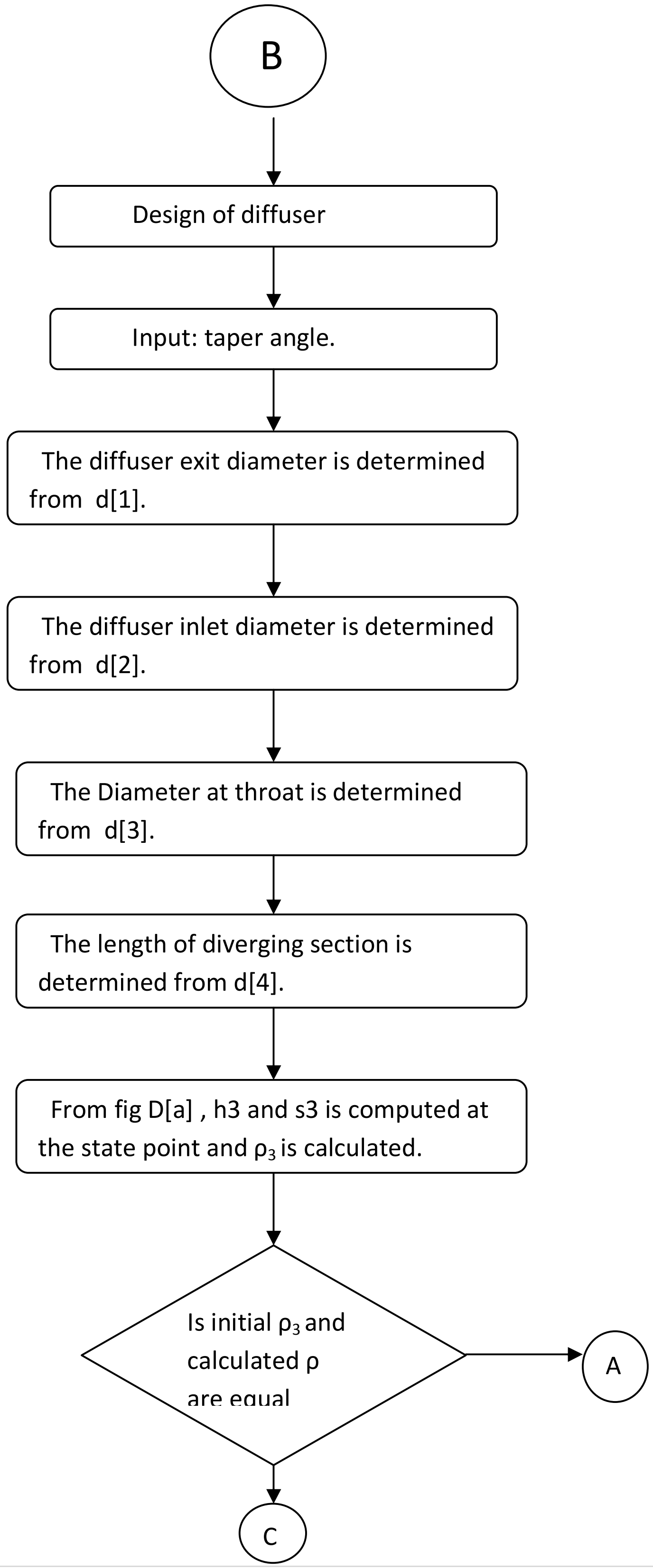
It is also important for the designer to predict the complete performance map of a machine so that alternative designs can be compared, assessed and implemented.A turbo expander is one of the key components of cryogenic process plants, which run under varying operating conditions. This necessitates the study of the performance of the turbine at conditions different from the designed ones such as to examine the start transients.

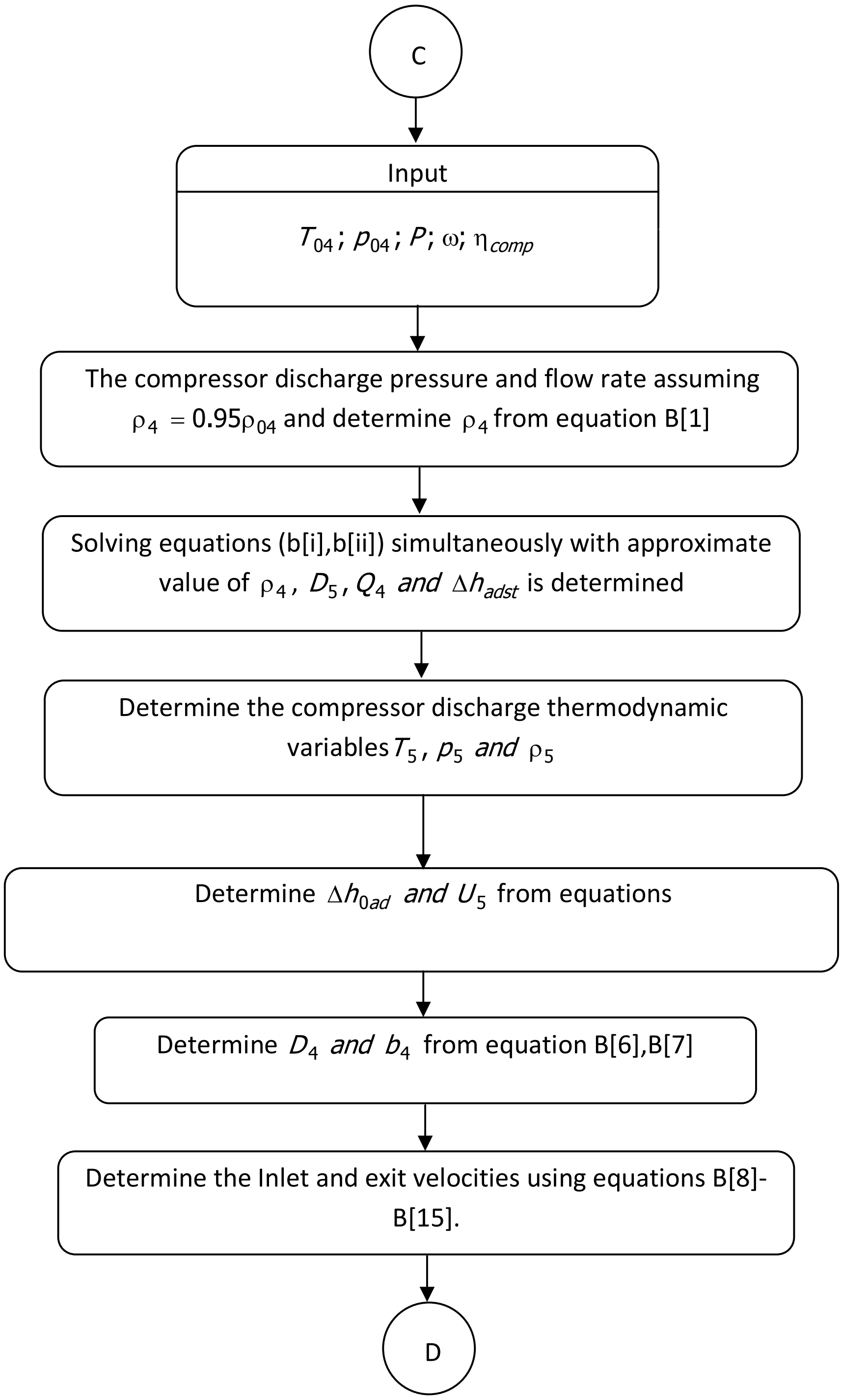
5.ALGORITHM AND FLOW PROCESS

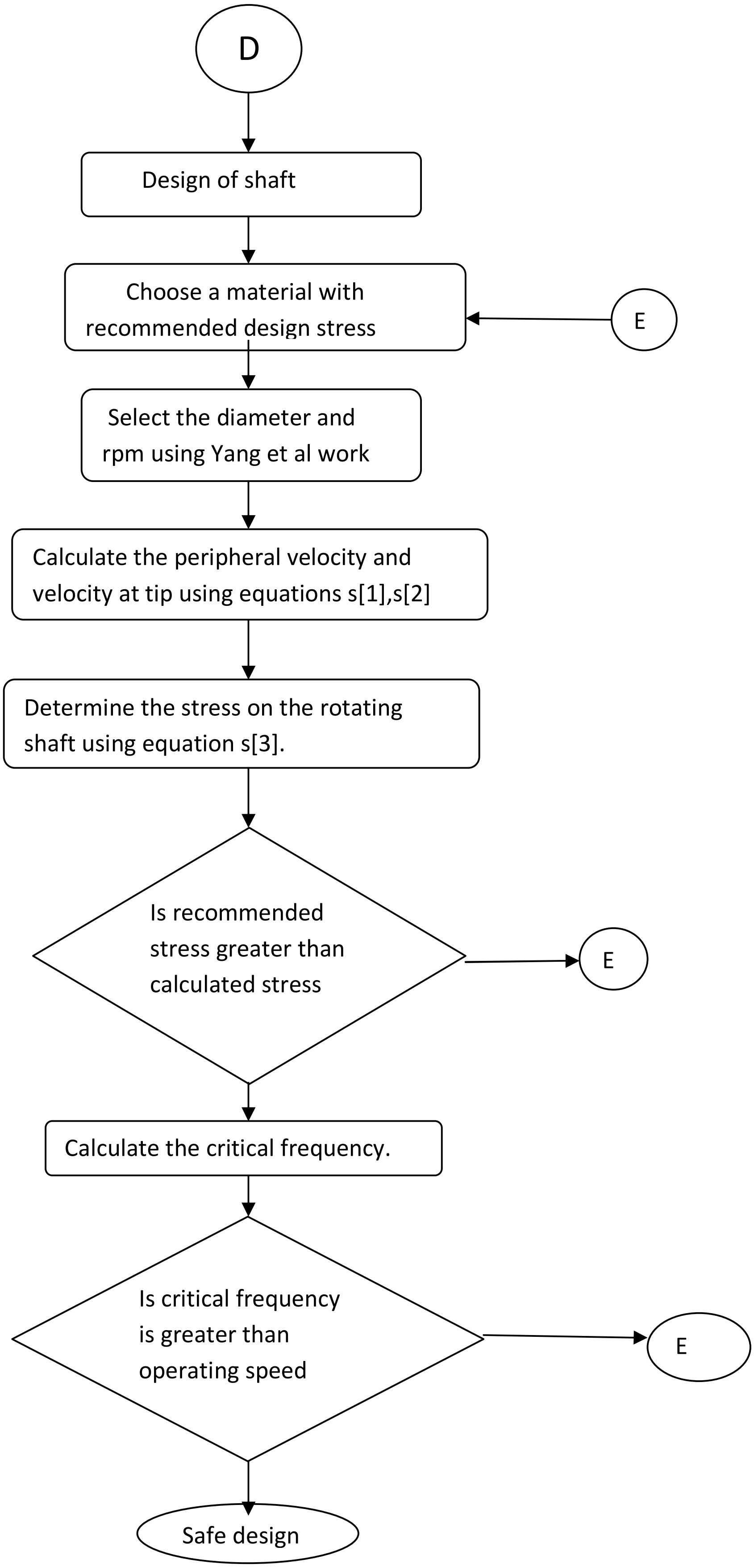
A systematic procedure is followed to calculate the various parameters associated with the design of Turbo exapander. The help is taken from previous discussed design procedures.Help has been taken from various works of Shri S.K.Ghosh and Shri Partho Sarathi to compile them in a systematic manner.




Start
Input
$T_{0,in}$; $p_{0,in}$; p_{ex}; $\dot{m}$; η_{T-st}
Thermodynamic properties ρ_{in}; h_{in}; s_{in} at the inlet and ρ_{ex}; h_{ex}; s_{ex}; T_{ex} at the Exit are computed from chart using the input parameters.(The entropy remain constant)
A
The initial value of k_1 and k_2 are assumed.
Q_3, ρ_3 and Δh_{in-3s} is determined.
Ns ,Ds is determined wrt efficiency
ω and D_{tr} is computed.from 1[43],2[43].
D_{tip} and D_{hub} is computed. 3[43],4[43].
Compute β_{mean} and C_{m3} is calculated.
The major dimensions and the velocities at inlet and outlet is finally summarized.
B


B
Design of diffuser
Input: taper angle.
The diffuser exit diameter is determined from d[1].
The diffuser inlet diameter is determined from d[2].
The Diameter at throat is determined from d[3].
The length of diverging section is determined from d[4].
From fig D[a] , h3 and s3 is computed at the state point and ρ_3 is calculated.
Is initial ρ_3 and calculated ρ are equal
A
C


C
Input
T_{04}; p_{04}; P; ω; η_{comp}
The compressor discharge pressure and flow rate assuming $\rho_4 = 0.95\rho_{04}$ and determine ρ_4 from equation B[1]
Solving equations (b[i],b[ii]) simultaneously with approximate value of ρ_4, D_5, Q_4 and Δh_{adst} is determined
Determine the compressor discharge thermodynamic variables T_5, p_5 and ρ_5
Determine Δh_{0ad} and U_5 from equations
Determine D_4 and b_4 from equation B[6],B[7]
Determine the Inlet and exit velocities using equations B[8]-B[15].
D


D
Design of shaft
Choose a material with recommended design stress
E
Select the diameter and rpm using Yang et al work
Calculate the peripheral velocity and velocity at tip using equations s[1],s[2]
Determine the stress on the rotating shaft using equation s[3].
Is recommended stress greater than calculated stress
E
Calculate the critical frequency.
Is critical frequency is greater than operating speed
E
Safe design

VI.STRUCTURE OF THE SOFTWARE

The software has been designed using Turbo C++.This is a object oriented programming language which is preferred approach for most software projects.It offers a new and powerful way to cope with complexity.Instead of viewing a program as a series of steps to be carried out ,it views it as a group of objects that have certain properties and can take certain actions.

Major Elements Of C++

A. Objects.
B. Classes.
C. Inherintence
D. Reusability.
E. Polymorphism
F. Overloading.

The design codes have been written to develop each components of a turbo expander.Help has been taken from various recent thesis to perform the systematic design procedure for the development of Turbo-Expander.The output can be potrayed in a visual screen with the help of visual C++.The designer is asked to enter the parameters and the output is displayed at every design of the component. This software is easily upgradable and can be used for newer versions.

6.1 CODES FOR THE DESIGN OF TURBO EXPANDER

```
#include<iostream.h>

#include<conio.h>

#include<math.h>

#include<string.h>

void main()

{  clrscr();

 cout<<" design of turbo expander"<<endl;

cout<<"entr working fluid"<<endl;

char a[10];

cin>>a;

cout<<"entr constants k1, k2"<<endl;

double k1,k2;

cin>>k1>>k2;

cout<<"entr  turbine inlet temperature,inlet pressure,discharge pressure,efficiency"<<endl;

double t1,p1,p2,e;

cin>>t1>>p1>>p2>>e;

cout<<" find the corresponding ns and ds wrt efficiency"<<endl;

double ns, ds;

cin>>ns>>ds;

cout<<" from mollier chart entr the inlet enthalpy and exit enthalpyin KJ/Kg"<<endl;

double h01,h02;

cin>>h01>>h02;

cout<< "entr  exit volume rate as m3/sec" <<endl;
```

```
double q02;
cin>>q02;
double h,q ;
q= k1*q02;
h=k2*(h01-h02)*1000;
double dt;//inner diameter of turbine wheel
dt= ds*pow(q,.5)/ pow(h,.25);
//determination of angular velocity
double w= ns* pow(h,.75)/pow(q,.5);
double u2= w*dt/2; // blade velocity.
double dtip=0.6*dt;
double dhub= .425*dtip;
double dmean=(dhub+dtip)/2;
cout<<"-----------------design results--------------"<<endl<<endl;
cout<<"enthalpy drop--"<<h<<endl<<endl;
cout<<"inner diameter--"<<dt<<"m"<<endl<<endl;
cout<<"angular speed---"<<w<<"rpm"<<endl<<endl;
cout<<"blade velocity at inlet of turbine----"<<(int(u2*100))/100<<"m/sec"<<endl<<endl;
cout<<"tip dia taking shi as .6---"<<dtip<<"m"<<endl<<endl;
cout<<"hub dia meter taking lamda as 0.425--- "<<dhub<<"m"<<endl<<endl;
cout<<"mean diameter---- "<<dmean<<endl<<endl;
// mean exit angle blade =b;
double b=45.922*3.14/180;
double c3;
 double a3=95*3.14/180;   //assumed.
```

```
//c3= q/(sin(a3)*(3.14*(pow(dtip,2)-pow(dhub,2))/4- 10*.006*(dtip-dhub)/(2* sin(b))));

c3=110.32;

cout<<" exhaust velocity"<<c3<<endl<<endl;

double u3= w*dmean/2;

cout<<" mean blade velocity"<<u3<<"m/sec"<<endl<<endl;

// the velocity triangle at turbine inlet.

cout<<" discharge velocity from diffuser as cex as m/sec"<<endl;

double cex;

cin>>cex;

double h0ex;//exit stagnation enthalpy

h0ex= h02+ pow(cex,2)/2000;

double c2;

double a2=26*3.14/180;

c2= (1000*(118.92-h0ex)+ u3*c3*cos(a3))/(u2*cos(a2));

double c02,cm2;

c02= c2*cos(a2);

cm2= c2*sin(a2);

double w2=pow( cm2*cm2+ pow(( u2-c02),2),.5);

double tanb2= cm2/(u2-c02);

cout<<endl<<endl;

cout<<" data for velocity triangle at inlet of turbine"<<endl;

cout<<"c2--"<<c2<<"m/sec"<<endl<<endl;

cout<<"c02--"<<c02<<"m/sec"<<endl<<endl;

cout<<"cm2--"<<cm2<<"m/sec"<<endl<<endl;

cout<<" w2----"<<w2<<"m/sec"<<endl<<endl;
```

```
cout<<" tan of blade angle"<<tanb2<<endl<<endl;


// velocity triangle at turbine outlet.
double cm3= c3*sin(a3);
double c03=c3*cos(a3);
double w3= pow(( u3*u3 + c3*c3 - 2*u3*c3*cos(a3)),.5);
cout<<" data for velocity traiangle at outlet "<<endl;
cout<<" cm3----"<<(int(100*cm3))/100<<"m/sec"<<endl<<endl;
cout<<" c03----"<<(int(100*c03))/100<<"m/sec"<<endl<<endl;
cout<<" w3-----"<<(int(100*w3))/100<<"m/sec"<<endl<<endl;
// thermodynamic state at wheel discharge.
cout<<" entr mass flow through turbo expander as kg/sec"<<endl;
double m;
cin>>m;
double  h03=h0ex;
double h3= h03 - pow(c3,2)/2000;
double d3= m/q;
cout<<" stagnation enthalpy   "<<h03<<endl<<endl;
cout<<" static entalhpy   "<<h3<<endl<<endl;
cout<<" density  "<<d3<<endl<<endl;
cout<<" find properties using above from charts"<<endl<<endl;
 //design of diffuser
 cout<<endl;
 cout<<" design of diffuser"<<endl<<endl;
cout<<" taper angle"<<endl;
```

```
double t;
cin>>t;
double dex,din,dtd,ld;
double aex;
aex=q02/cex;
dex= pow((4*aex/3.14),.5);
din=dt+2*.001;
double ain;
ain=3.14*din*din/4;
dtd=dtip+ 2*.001;
ld= (dex-dtd)/(2*tan(t*3.14/180));
cout<<"---------design result--------"<<endl;
cout<<" inlet diameter--"<<din<<"m"<<endl<<endl;
cout<<" area at inlet of diffuser--"<<ain<<"m2"<<endl<<endl;
cout<<" diameter at throat ---"<<dtd<<"m"<<endl<<endl;
cout<<" exit diameter"<<dex<<"m"<<endl<<endl;
cout<<" exit area"<<aex<<"m2"<<endl<<endl;
cout<<" length of diverging section---"<<ld<<"m"<<endl<<endl;
//design of shaft.
cout<<endl;
cout<<"design of shaft"<<endl;
start:
cout<<" diamter of the shaft and operating speed"<<endl;
double d,w1;
cin>>d>>w1;
```

```
double vs,vt;
vs= w1*d/2;
vt=vs*2;
cout<<" choose the material and mention the recomended design stressin MPa"<<endl;
char n[10];
double s,den;
cin>>n>>s;
cout<<" entr the density of chosen material"<<endl;
cin>>den;
double str;
str= den*vt*vt/(3*1000000);
cout<<str<<endl;
if ( str>s) { cout<< "change the material"<<endl;
              goto start;
              }
              else
  cout<<"entr length of shaft,youngs modulus"<<endl;
  double ln,el;
  double f;
  cin>>ln>>el;
  f= .9*( d/pow(ln,2))*pow((el/s),.5);
  if (f<w1) { cout<<" design not safe entr values again"<<endl;
                    goto start;
                    }
cout<<"---------design result--------"<<endl;
```

```
cout<<" surface velocity"<<vs<<endl<<endl;

cout<<" velocity at tip of the collar"<<vt<<endl<<endl;

cout<<" calculated stress"<<str<<endl<<endl;

cout<<" critical frequency"<<f<<endl<<endl;

cout<<"design  is safe with "<<n<<" material having design stress as"<<s<<" and having diameter and length as  "<<d<<" m   "<< ln<<" m respectively"<<endl;

//design of brake compressor.

cout<<endl;

cout<<" design of brake compressor"<<endl<<endl;

cout<<" input parameters"<<endl;

cout<<" process gas"<<endl;

char pr[10];

cin>>pr;

cout<<" power required"<<endl;

double pw,ip,it,w4,n2;

cin>>pw;

start1:

cout<<" entr inlet total pressure in N/m2,inlet  Temperature in kelvin ,angilar speed,expected efficiency"<<endl;

cin>>ip>>it>>w4>>n2;

double d8;

cout<<" molecular weight of the gas"<<endl;

double mw;

cin>>mw;

 d8= .94*ip*mw/(8314*it);

 cout<<" asumption ofdrop in enthalpy"<<endl;
```

```
double h8;
cin>>h8;
cout<<" corresponding to expected efficiency find Ns  and Ds"<<endl;
double ns1,ds1;
cin>>ns1>>ds;
double q4,d4,d5,mb,b4;
q4=pow((ns1*pow(h8,.75)/w4),2);
d5=ds*pow(q4,.5)/pow(h8,.25);
mb=d8*q4;
double pw1;
cout<<" entr power factor, slip factor"<<endl;
double z,y,pv;
cin>>z>>y;
pv= w4*d5/2;
pw1= z*y*mb*pow(pv,2);
if(pw1<pw) { goto start1;
           }
d4=d5/2.25;
b4=.2*d5;
double zb,tb;
cout<<" entr number of  blades and blade thickness in m"<<endl;
cin>>zb>>tb;
//inlet velocities.
double cm4,u5,w6;
cm4=q4/((3.14*d4-zb*tb)*b4);
```

```
 u5=d4*w4/2;
 w6=pow( ((u5*u5)+(cm4*cm4)),.5);
cout<<” ------design result------“<<endl;
 cout<<" diameter of impeller at exit "<<d5<<"m"<<endl<<endl;
 cout<<" mass  flow"<<mb<<"kg"<<endl<<endl;
 cout<<" power input"<<pw1<<" watt"<<endl<<endl;
 cout<<" inlet diamter"<<d4<<" m"<<endl<<endl;
 cout<<" blade height"<<b4<<" m"<<endl<<endl;
 cout<<" radial velocity"<<cm4<<" m/sec"<<endl<<endl;
 cout<<"  peripheral velocity"<<u5<<"  m/sec"<<endl<<endl;
 cout<<" inlet relative velocity"<<w6<<" m/sec"<<endl<<endl;
 cout<<" number of blades"<<zb<<endl<<endl;
 cout<<" blade thickness"<<tb<<" m"<<endl<<endl;
 getche();
  }
```

Results:

Design of Turbo-Expander

Enter working fluid:

Nitrogen.

Enter Constants k1,k2:

1.11

1.03

Enter turbine inlet temperature,inlet pressure, discharge pressure,efficiency.

120.45

7.95

1.3

50

Find the corresponding Ns and Ds wrt efficiency:

.547

3.47

From chart enter the inlet enthalpy and exit enthalpy as kJ/kg:

131.35

81.615

Enter exit volume rate as m^3/sec:

.0148

-------Design Results-------------

Enthalpy Drop-------------------51227.05 J/kg

Inner diameter------------------0.02956 m

Angular speed--------------------14531.80 rad/sec

Blade velocity at inlet of turbine--------214 m/sec.

Tip diameter -----------------------0.0177 m

Hub diameter---------------------0.00753 m

Mean diameter------------------ 0.01263 m

Exhaust velocity-----------------110.32 m/sec

Mean blade velocity------------91.827 m/sec.

Enter discharge velocity as m/sec:

17

Data for velocity triangle at inlet of turbine.

C_2-------187.93 m/sec

C_{02}------ 168.92 m/sec

C_{m2}-------82.34 m/sec

W_2------ 94.25 m/sec

Tan of blade angle----1.795

Data for velocity triangle at outlet.

C_{m3}------- 109 m/sec

C_{03}-------9 m/sec

W_3-------149 m/sec

Enter mass flow through turbo expander as kg:

.0765

Stagnation enthalpy-----------------81.7595 KJ/kg

Static enthalpy------------------------75.67 KJ/Kg

Density at exit ------------------------4.65 kg/m^3

Design of diffuser

Enter taper angle in degree:

5

-------------design result------------------

Inlet diameter -----------------0.03156 m

Area at inlet of diffuser------0.00078 m^2

Diameter at throat   ---------0.019738 m

Exit diameter-------------------0.0333 m

Exit area-------------------------0.000871 m^2

Length of diverging section-------0.07756 m

Design of shaft

Enter diameter of shaft and operating speed:

.02

14345

Choose the material and mention the recommended design stress as MPa:

k-Monel

790

Enter the density of chosen material as Kg/m^3:

10700

Enter the length of shaft ,young modulus

0.1

2*10^11

-------------design result-------------

Surface velocity------------143.45 m/sec

Velocity at tip of collar-----286.9 m/sec

Calculated stress---------------293.57 N/m^2

Critical frequency--------------28640 rad/sec

Design is safe.

Design of brake compressor.

Process gas:

nitrogen

Power required in Watt:

2000

Enter inlet total pressure in N/m^2, inlet temperature in Kelvin, angular speed,expected frequency:

112000

300

14345

60

Enter molecular weight of process gas:

28

Assumption of drop in enthalpy in J/kg:

13463.8

Corresponding to expected efficiency find Ns,Ds

1.95

2.9

Enter power factor,slip factor.

1.02

0.78

Enter number of blades and blade thickness

10

0.00075

----------design result-------------

Diameter of impeller at exit-------------0.04574 m

Mass flow----------------------------------0.034 Kg/sec

Power input------------------------------2921.86 watt

Inlet diameter --------------------------0.02033 m

Blade height----------------------------0.009148 m

Radial velocity------------------------ 56.013 m/sec

Peripheral velocity-------------------145.815 m/sec

Inlet relative velocity----------------156.203 m/sec

Conclusion

The Software has been built for the designing the components of a turbo expander : turbine wheel, nozzle, diffuser,brake compressor,shaft design, off design prediction.Care has been taken to produce the data in a systematic manner. Still the work is left to give it a visual form to insert graphs . This may be performed using the visual C++ codes.

Future Scopes

.Future work can be done to validate the design with respect to manufacturing point of view.The nozzle design can be perforemed using the complied algorithm and the design procedures mentioned.Further work can be made to design the bearings and seal of the product.

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