Design of High Speed Drive System for Rocket Subsystem Tests

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Abstract

A high-speed drive system driven by a radial turbine was designed for testing high speed seals and bearings for rocket engine turbopump. The drive system uses angular contact ceramic ball bearing pairs with high speed capability for supporting the rotor and is lubricated using oil mist lubrication method. The radial turbine design was carried out by 1-D mean line method, to match the output of the available air supply from a screw compressor. The preliminary design of the radial turbine was analysed using CFD with CFX software. Structural analysis was performed using ANSYS to ensure that stresses are benign at the operating conditions. Rotordynamic analysis of the rotating system was carried out with in-house FORTRAN code and Campbell diagrams obtained showed that sub-critical operation was achieved. The hardware was realised at in house fabrication facility and balancing of the rotating system was carried out to achieve minimum unbalance levels. The facility realization required for the drive involved integrating procured pneumatically operated valves and sensors with necessary instrumentation to the fabricated feed lines and test bed. The testing of the high-speed drive was performed by remote operation and the test parameters were logged using a data acquisition system. The testing carried out achieved the targeted speed of greater than 60,000 rpm. Test Data showed minimal temperature rise at the bearing location and large margins for the valve openings, suggesting that even higher speeds are feasible. Post-test observation revealed good health of the hardware.

Keywords: *High speed drive, radial turbine, testing*

1. Introduction

A high-speed drive system capable of achieving speed greater than 60,000 rpm for carrying out high speed tests on bearings and seals was developed in LPSC. Here system engineering concept takes significance, due to the multi-disciplinary branches of engineering analysis has to be applied for the design acceptance. Preliminary design using analytical methods followed by modelling, CFD, structural and validation testing was rendered towards the product development.

2. Literature Review

The preliminary design of radial inflow turbines is discussed in [1]. A non-dimensional design is developed based on the initial specification of the turbine pressure ratio and the stage efficiency. The inlet velocity triangle is defined by radial blades with an incidence flow selected for minimum inlet Mach number. The relative exit Mach number was used for the design of the exhaust guide vanes. The relative velocity ratio is then specified to define the outlet velocity triangle. The rotor non-dimensional geometry is now worked out based on the assumed efficiency of the stator. The design is optimized by adding various loss models like curvature, tip leakage etc.

The design procedure in [2] followed a solution of equations for speed parameter, flow angles, blade width ratio, mass flow parameter using a constrained optimization algorithm. A computer program was written to optimize the axial length of the rotor by minimizing the

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loss of stagnation pressure. This also gave the hub and shroud contours of the rotor. The nozzle-less casing was designed using the equations of mass flow rate, energy, and angular momentum, where isentropic free vortex flow was assumed.

Detailed measurements of the flow in a procured radial turbine were performed in [3], where details measurements were made at volute inlet, volute exit, rotor exit and diffuser exit. 2 to 4% of the inlet stagnation pressure was lost from volute inlet to impeller inlet, and the swirl angle was a function of the velocity ratio. The diffuser effectiveness was very low at 0.2 to 0.4. Rotor efficiency peak was observed at an incidence of -40°, and at a reaction of 0.6-0.65 with U/C_0 of 0.7.

The design and testing of a very high-speed radial turbine for a rocket turbopump is described in [4]. The design features a vaneless volute, an unshrouded rotor, and an exhaust with guide vanes to reduce swirl losses. The turbine has only 7 primary parts, with cast housings. Testing with GH2 at 35% rated power level met the targeted efficiency.

Radial turbine tip clearance flow was studied in [5]. Flow visualization was performed with ammonia injection, and static pressure measurements were performed on the blade surface at the gap region. Additionally, static pressure was measured in the rotor casing. The flow in the gap was also measured using hot wire traverse. The tip gap mass flow was much lower for the radial turbine case compared to the axial turbine for the same gap, due to the “scraping” effect of the flow.

An experimental comparative study of oil mist lubricated all steel and hybrid ball bearing performance was made in [6]. The measured friction torque was lower for hybrid ball bearings for the high-speed range. The centrifugal loads on ceramic balls were lower compared to steel balls due to lower density. Consequently, the contact loads and frictional forces were lower in hybrid ball bearings.

The performance of hybrid ball bearings for turbocharger applications at speeds up to 120,000 rpm was evaluated in [7]. Comparisons with all steel and floating bush bearings were also made. Oil jet lubrication was used. The frictional loss in hybrid bearings was 30% lower than floating bush bearings. Hybrid ball bearings did not seize even at 10% of the oil feed amount of where the steel ball bearings seized and under 5% of oil feed amount where bush bearings seized. Hence anti-seizure performance was superior to them.

The advantages of hybrid ball bearings over steel ball bearings were discussed in [8]. Hybrid ball bearings are electrically non-conducting, highly wear resistant, and have a high bearing stiffness and long service life. Hybrid bearings have lower coefficient of thermal expansion which results in more stability over temperature change and wear rate of hybrid bearings under contaminated lubrication is comparably very low than steel bearings which makes it suitable for high speed rotating machines because of low friction.

The performance of hybrid and steel ball bearings with oil-mist lubrication for sensitivity with different oil viscosity, oil flow, mist supply pipe length etc was studied in [9]. It was found that hybrid bearings had lower temperature rise compared to steel bearings for all operating conditions. The optimum length of pipe of 1.5m and oil viscosity of 100 cSt gave the lowest temperature rise for both ceramic and steel bearings.

3. Design, Analysis and Manufacturing Process

3.1. Configuration

A radial turbine rotor powered by compressed air is supported on hybrid ball bearings in a housing cooled with an oil mist spray. Figure 1 shows the schematic of the high-speed drive. It consists of a radial turbine rotor mounted on a shaft. The shaft is in turn supported on two pairs of angular contact hybrid ball bearings. During operation, these bearings are cooled using an oil mist which is sprayed on the bearings through the mist injector (Fig. 1). A mist generator uses compressed air to generate mist using oil stored in its reservoir. The mist is admitted to the high-speed drive through plastic tubes. The drive system owes its high-speed capability to these angular contact ball bearings.

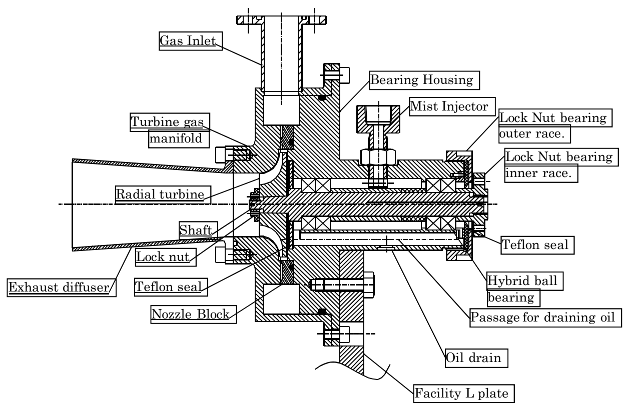

Figure 1: Schematic of high speed drive

Figure 2 shows the schematic of the test facility established for running the high-speed drive. It consists of a diesel engine powered screw compressor which can deliver compressed air at 200 g/s at 11 bar. Compressed air from the screw compressor is admitted to the turbine inlet manifold through a 1” SS pipe. Flow is controlled by a control valve in the line. Emergency shut off of the high-speed drive can be done by an electro-pneumatic valve. Flow rate through the turbine is monitored using a flow meter in the line.

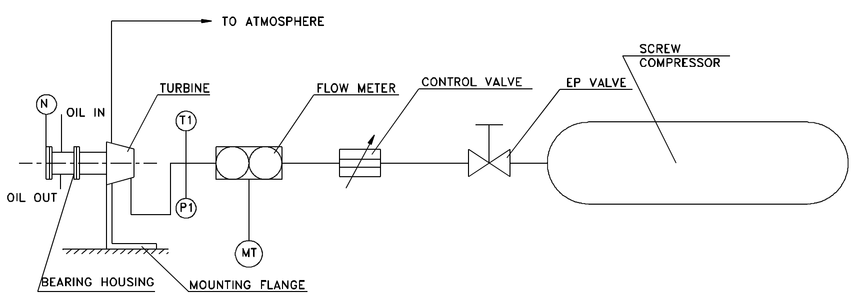

Figure 2: Schematic of the test facility for carrying out high speed test

3.2. Design Details

3.2.1. Design of radial turbine

The design inputs for the rotor are given in Table 1.

Table 1: Design inputs for the rotor

Parameter	Value
Working fluid	Air
Inlet total gas pressure (P_0), bar	10
Inlet total gas temperature (T_0), K	335
Overall turbine pressure ratio (PR)	10
Speed of rotation of the rotor (N), rpm	100,000
Mass flow rate (m), g/s	200

CFD analysis of the design was carried out. Flow pattern across the turbine was smooth, with no abrupt changes in fluid variables (Fig. 3).

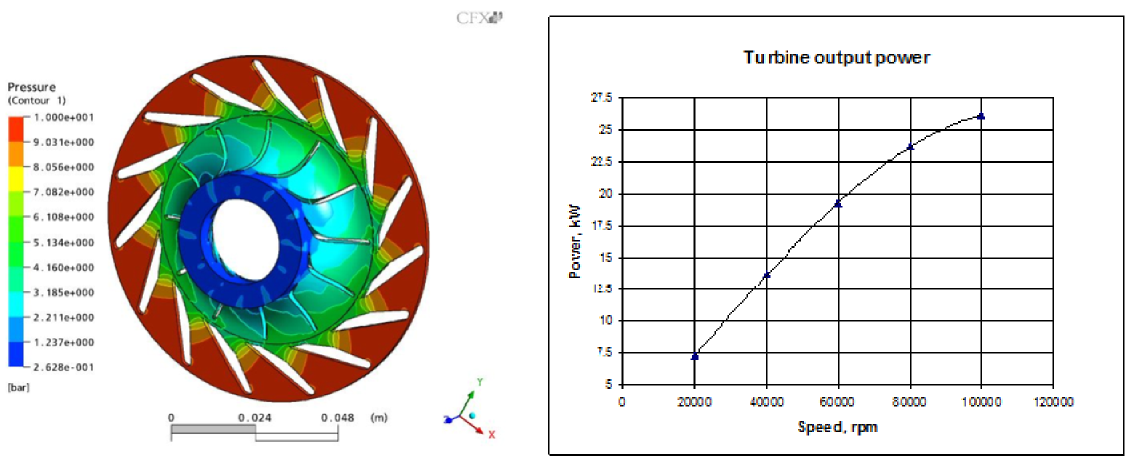

Figure 3: CFD analysis of the radial turbine

Configuration of turbine rotor arrived after detailed design is given in Fig. 4.

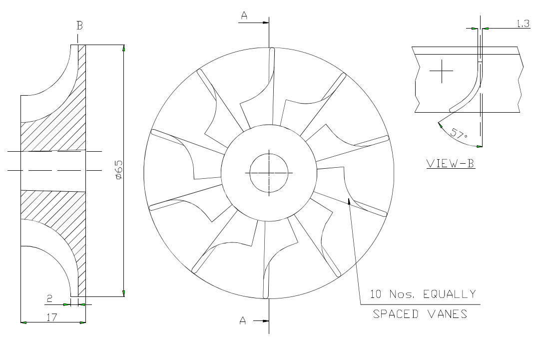

Figure 4: Configuration of turbine rotor

Structural analysis of turbine rotor was carried out for estimating the safe operating speed. Turbine rotor was made of D16T aluminum alloy. Mechanical properties of the turbine rotor are given in Table 2.

Table 2: Mechanical properties of materials at RT

Sl No	Part Name	Material	Young's modulus	Poisson's ratio	YS	UTS	%el
1	Rotor	D16T	70,000	0.33	265	470	10

Results of the FEM analysis carried out using ANSYS are given in Table 3. Assuming a factor of safety of 4 on uniform elongation of 5%, the failure speed of the rotor was estimated as 1,17,500 rpm.

Table 3: Summary of stress analysis

Parameter	Value
Axial deformation (mm)	-0.27
Radial deformation (mm)	0.02
Axial stress (N/mm^2)	-33
Radial stress (N/mm^2)	65
Hoop stress (N/mm^2)	177
von Mises stress (N/mm^2)	170

3.2.2. DESIGN OF SHAFT/BEARING SYSTEM

A detailed rotor dynamic analysis of the rotor is also carried out to estimate the critical speeds of the rotating system. The FEM analysis of the rotating system is carried out by considering each element as a Timoshenko beam element (Fig. 5). Turbine rotor, sleeves on shaft OD and lock nut are considered point masses. The shaft material is SS 202, turbines rotor is D16T aluminum alloy, lock nut for bearing inner race for locked bearing is Ti6Al4V (Titanium alloy) and other rotating elements are made of SS321 stainless steel. Rotordynamic analysis was carried out using in-house developed rotordynamics code in FORTRAN.

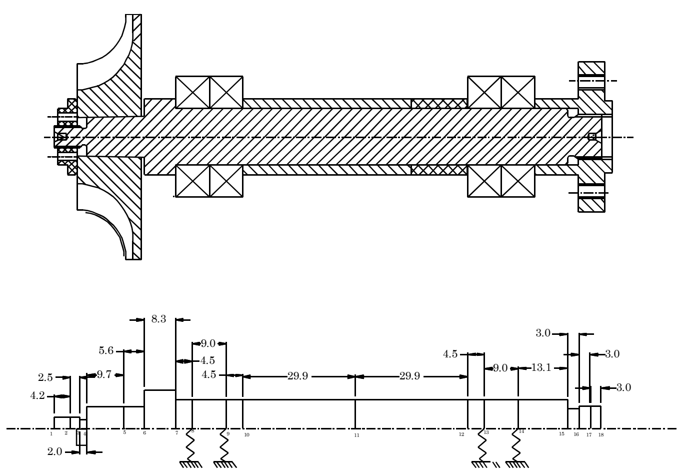

Figure 5: Finite element model for carrying out rotor dynamic analysis

The Campbell diagram (Fig. 6) shows that the first critical speed for the rotating system is 1,40,000 rpm whereas the second critical speed is 1,58,000 rpm.

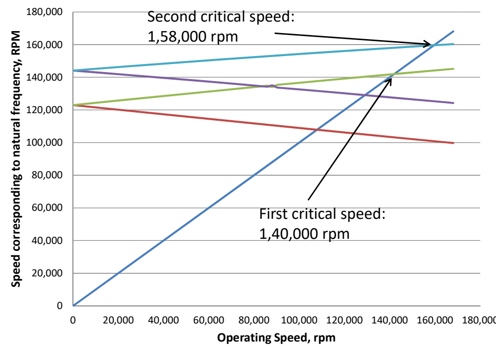

Figure 6: Campbell diagram for rotordynamic analysis

3.3. *Manufacturing of parts*

Test rig elements including rotor, shaft, nozzle block, casing and bearing housing were realized in-house (Fig. 7). Other items like valves and bearings were procured (Fig. 8).

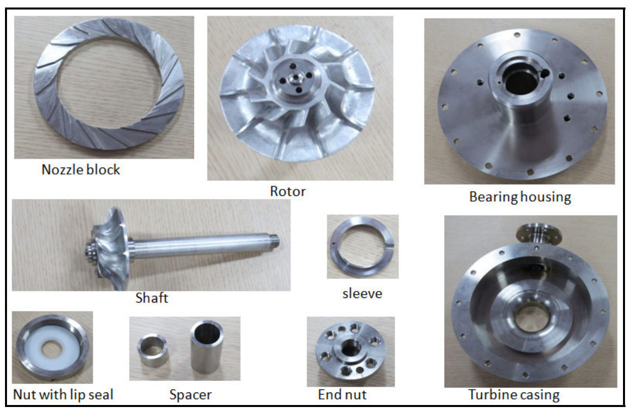

Figure 7: Test rig elements realized

Figure 8: Procured items for high speed drive

3.4 TESTING

3.4.1. BALANCING OF THE ROTATING ASSEMBLY

Balancing is a crucial activity for system operating at very high speeds. Two plane balancing was carried out at 2800 rpm, to achieve a turbine side unbalance of 0.25 g.mm and 0.192 g.mm on the nut side. Here, balancing is done at low speeds as the turbine operates below its first critical speed.

3.4.2. TESTING

The photograph of the facility established for the test is shown in Fig. 9. The facility consists of the following elements: 1) A diesel engine driven air screw compressor (flow rate of 200 g/s at 11 bar). 2) An EP valve for stopping of the turbine. 3) A control valve to control the flow of air to the turbine. 4) A flow meter to measure 5) Temperature and pressure sensors at turbine inlet and outlet 6) Mist lubrication system to supply oil mist for cooling of hybrid ball bearings. 7) Thermocouple for measurement of bearing temperature (locked bearing). 8) Accelerometer to monitor vibration 9) Optical sensor for speed measurement 10) GN2 cylinders and associated lines for actuation of pneumatic valves.

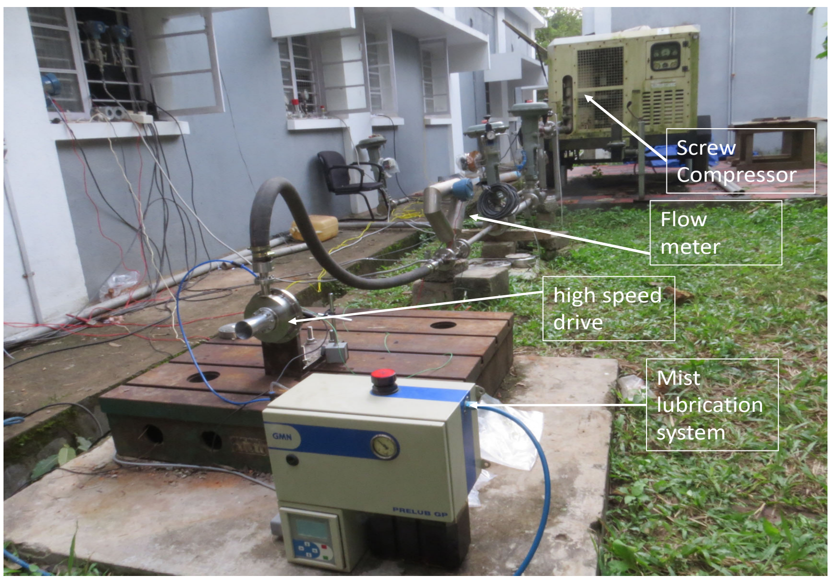

Figure 9: Facility established for testing of high speed drive

Detailed measurement plan for the test is shown in Table 4. All measurements were acquired remotely using a data acquisition system.

Table 4: Measurement plan

Parameter	Legend	Range	Nominal value	Accuracy	Sensor Type
Turbine Inlet Pressure	PTI	0-15 bar	10 bar	±1 % FS	Absolute Pressure Transmitter
Turbine Inlet Temperature	TTI	300-360 K	335 K	± 0.5 K	TP 198-05 –Probe type RTD
Surface temperature of	TBI	300-360 K	335 K	± 2.5 K	K type

Table 4 *contd…*

locked bearing					thermocouple
Parameter	**Legend**	**Range**	**Nominal value**	**Accuracy**	**Sensor Type**
Turbine speed	STI	0-125,000 rpm	100,000 rpm	0.5 % rdg	Optical speed
Turbine Inlet flow	FTI	0-250 g/s	200 g/s	0.5 % rdg	Mass flow meter
Vibration measurements	VBX, VBY	± 100 g	20 grms	± 5 % rdg	CE 252

4. Results and Discussions

4.1. Test procedure

The high-speed drive was assembled as per the schematic shown in Fig. 1. The test article was mounted on the test bed. The facility elements were connected as per schematic shown in Fig. 2. Instrumentation of the facility was performed and data acquisition was set up. Facility valves, namely the EP value and control valves were kept in the closed position. The mist lubrication system was turned on. The diesel air screw compressor was used to supply air at the required pressure. The EP valve was opened, following which the control valve was opened gradually in small steps and the speed was monitored. The control valve was opened slowly to reach the target speed of 60000 rpm and the speed was maintained for 10s. To stop the test, the EP valve was closed to stop air supply to the turbine. The mist lubrication system was then switched off, and finally, the screw compressor was switched off.

4.2 Test results

The turbine speed reached close to 60,000 rpm at 106s. The speed was maintained up to 119s for a duration of 13s. The speed was achieved for no load conditions at turbine inlet pressure of 1.37 bar and flow rate of 18.5 g/s. The rise in bearing temperature was gradual and well within the safe limits. A total temperature rise of 7K was observed. The operating power level was estimated based on the pressure ratio and mass flow rate. The efficiency of the turbine could not be estimated as no torque measurement or power absorbing devices were available for this speed. Vibration levels were found to be benign throughout the test.

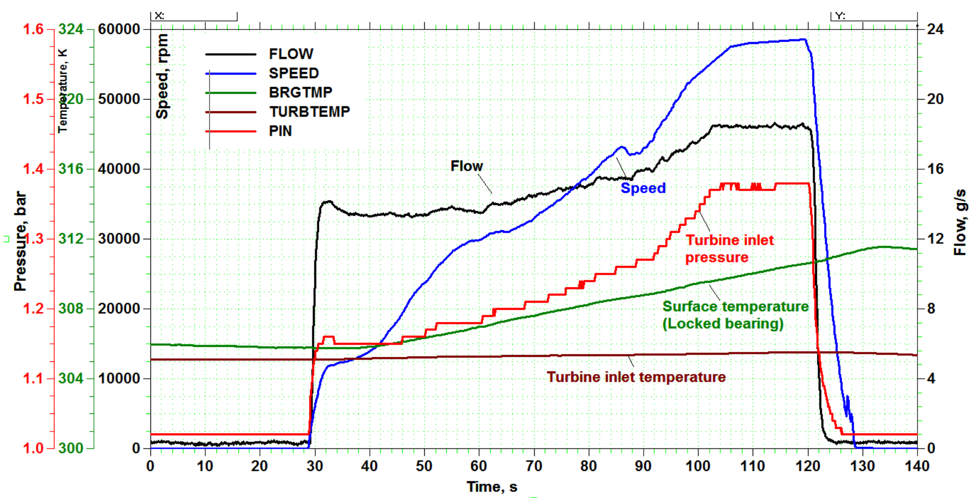

Figure 10: Test results

5. Conclusions

Design, modeling, analysis and realization of the radial turbine including bearing housing, shaft and lubrication system were completed. Facility realization and instrumentation was also carried out. Finally testing of the high-speed drive system was carried out at 60,000 rpm. This drive system can be used for high speed testing of seals and bearings, towards future programmes.

This project gave hands on experience in developing a new drive system with high speed capability and its associated test facility. The design of the radial turbine was carried out for the first time in LPSC. And for the first time the concept of mist lubrication was implemented for a drive system.

Acknowledgements

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