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The 2-Stage Axial Turbine Test Facility *"LISA"*

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ABSTRACT

This paper describes the design and construction of a new two stage axial turbine test facility, christened "Lisa"[1]. The research objective of the rig is to study the impact (relevance) of unsteady flow phenomena upon the aerodynamic performance, this being achieved through the use of systematic studies of parametric changes in the stage geometry and operating point. Noteworthy in the design of the rig is the use of a twin shaft arrangement to decouple the stages. The inner shaft carries the load from the first stage whilst the outer is used with an integral torque-meter to measure the loading upon the second stage alone. This gives an accurate measurement of the loading upon the aerodynamically representative second stage, which possesses the correct stage inlet conditions in comparison to the full two stage machine which has an unrealistic axial inlet flow at the first stator. A calibrated Venturi nozzle measures the mass flow at an accuracy of below 1%, from which stage efficiencies can be derived.

The rig is arranged in a closed loop system. The turbine has a vertical arrangement and is connected through a gear box to a generator system that works as a brake to maintain the desired operating speed. The turbine exit is open to ambient pressure.The rig runs at a low pressure ratio of 1.5. The maximum Mach number at stator exit is 0.3 at an inlet pressure of 1.5 bar. The maximum mass flow is 14 kg/sec. Nominal rotor design speed is 3000 RPM. The tip to hub blade ratio is 1.29, and the nominal axial chord is 50 mm. The rig is designed to accommodate a broad range of measurement techniques, but with a strong emphasis upon unsteady flow methods, for example fast response aerodynamic pressure probes for time-resolved flow measurements.

The first section of this paper describes the overall test facility hardware. This is followed by a detailed focus on the torque measurement device including stage efficiency measurements at operating conditions in Lisa. Discussion of measurement techniques completes the paper.

NOMENCLATURE

a	Stage aerodynamic work
c	Axial chord length
c_n	Nominal axial velocity
f	Blade passing frequency
U	Circumferential velocity

greek symbols

ϕ	Throughflow coefficient	$\phi = \frac{c_n}{U}$
λ	Stage loading coefficient	$\lambda = \frac{a}{U^2}$
$\bar{\omega}$	reduced frequency	

abbreviations

CFD	Computational Fluid Dynamics
LDA	Laser Doppler Anemometry
PLC	Programmable Logic Controller
SPS	Speicherprogrammierbare Steuerung (=PLC)

1. "Lisa" follows the local Swiss tradition of giving turbomachinery test stands a feminine appelation.

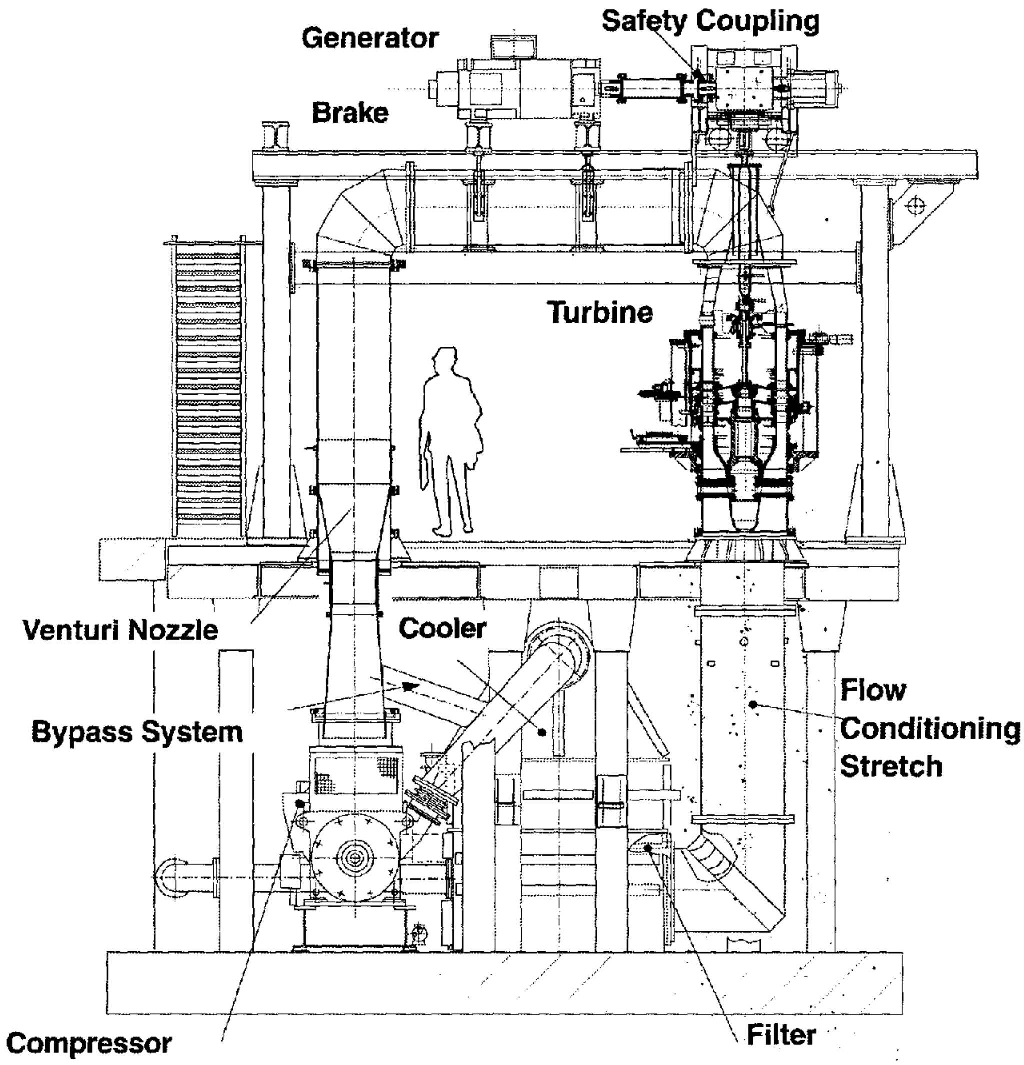

Figure 1. Test Stand Test Loop

INTRODUCTION

It is a *sine qua non* of turbine manufacturing and marketing that the customer expects the highest possible aerodynamic efficiency, and historically a significant proportion of research and development activities have been directed to this end.

The net result is that the majority of the steady flow physics dominating turbine inefficiencies are well understood and can be to a degree accounted for in design, see for example Sieverding (1984), Denton, (1993). As a consequence, highly efficient stage designs are available.

Nevertheless, the question always remains as to whether the performance and design practice can be further improved, and the issue arises whether the unsteady flow regime can be exploited to yield further design and performance improvements. There is clear evidence (e.g. Curtis et al (1996), Schulte et al (1998)) that in certain configurations such as high lift aerofoil design, this is indeed the case. Beyond this, there are a number of other unsteady flow effects which need to be systematically investigated in order to assess their relevance and impact on the design process. The ability to explore unsteady flow effects in the design of turbomachinery requires commensurate development in areas of measurement and simulation technology, and recent developments are permitting the routine study of unsteady flows. (For a few of many examples of the current explosive growth in this area see Kupferschmied et al (1999), Roduner et al (1999), Köppel et al (1999), Busby et al (1998a,1998b))

Aerodynamic design alone is not the only design driver, construction cost and complexity are also significant issues. These result in the current design trend towards more compact machines with more highly loaded blading, which suggests that unsteady flow effects, particularly blade row interactions and spacings, are becoming increasingly important issues, and should be comprehensively studied.

Despite the inherent unsteady nature of turbine flows, and the promising progress to date, it is not axiomatically obvious that *all* unsteady flow effects can be accommodated (more likely; exploited) to deliver net benefit to the customer, therefore a critical evaluation of such flows is called for (for a detailed discussion of this issue, see Wisler (1998)).

The objective of this paper is to describe the design and construction of a new two stage air turbine test rig. The contribution of this rig to design practice will be two fold, to provide data for the validation of unsteady CFD codes, but primarily to investigate experimentally the impact and relevance of unsteady flows upon performance for a range of geometric and aerodynamic conditions.

DESIGN OBJECTIVES

The design objectives framing the development of the rig can be summarized as:

- *Geometric Flexibility*: The rig requires the possibility to incorporate a wide range of designs and to change over the geometric configurations under study in a useful time frame
- *Geometric Accuracy*: Construction tolerances are tightly prescribed, typical construction and assembly tolerances are 0.1‰. This requirement arises from the need for repeatable quality in the data for CFD validation
- *Measurement Flexibility*: It is of benefit to be able to measure almost anywhere without penalties in instrumentation time, cost and particularly accuracy. Included in this is the need to measure stage efficiency on the aerodynamically representative second stage.
- *Safety:* The rig has to comfortably meet and exceed all relevant safety requirements for rotating machinery.

TEST STAND LOOP

The test stand builds upon the successful project and infrastructural foundations laid by the Zurich annular cascade (Sell et al (1996,1997)). Figure (1) shows the overall layout of the new rig and associated infrastructure. The rig is built over three floors in a vertical arrangement. The air supply and flow conditioning occupies the lower level, consisting of a 750kW (1000 HP) compressor (and a safety by-pass system, which isolates the compressor under emergency stop conditions), a cooler, an air filter and a long inlet stretch, furnished with a series of perforated plates, honeycombs and sieves. These principally act to condition the flow to produce highly uniform flow with low turbulence levels.

The second level contains the turbine, and the return flow piping to the compressor inlet. Within the return leg there is a calibrated Venturi nozzle for mass flow measurements.

At the third level, are the braking and safety systems for the developed power. This consists of a gearbox, which is used to turn the shaft and to slow the speed by two, a safety coupling, and a 12-pulse electromotor operating in a generator mode. This generator feeds the power produced by the turbine back into the electrical net, which is environmentally more responsible than dissipating the power as heat in say, a water brake. The gearbox assembly can be decoupled from both the generator and the turbine, this allows the gearbox to be rolled to the right allowing for unfettered access to the turbine assembly from above. The speed controller for the generator is optimized for the nominal turbine load points and is accurate to 0.1‰, i.e. better than 1 RPM at the main operating points.

As a consequence, the primary assembly and disassembly method is *via* an axial build of the turbine. Therefore, the majority of the turbine assembly consists of closed, uniform rings, whose construction tolerances, particularly in roundness can be guaranteed. This is an advantage over turbine rigs whose disassembly method is through an axially split casing.

PRINCIPAL CHARACTERISTICS

Table 1: Test Rig Characteristics

Maximum Shaft Power	400	[kW]
Maximum Speed	3000	[rpm]
Tip Diameter	800	[mm]
Tip-Hub Ratio	1.29	[-]
Typical Blade Axial Chord	50	[mm]
Typical Blade Count	42	[-]

Table (1) contains a summary of the characteristics of the rig. The maximum shaft power relates to the maximum power which can be braked by the generator. This is somewhat over designed to allow for unproblematic future expansion of the rig, and to give useful safety margins. The stage designs foreseen for the rig in the immediate future have nominal operating loads around half this value. The maximum speed is slow in comparison to typical aero-engine applications, but is realistic (50 Hz) for terrestrial power generation for both gas and steam turbine configurations.

Table 2: Test Rig Fluid Flow Characteristics

Maximum Shaft Power	400	[kW]
Maximum Inlet Pressure	1.5	[bar]
Typical stage throughflow [ϕ]	0.4	[-]
Typical stage loading [λ]	1.2	[-]
Exit pressure (nominal)	1	[bar]

The rig is of a useful size, such that characteristic geometric length scales are substantially larger than typical probe dimensions. This is important when considering the effect of probe blockage upon the flow physics in the rotor-stator gap.

Table (2) contains a list of the characteristic fluid flow parameters. From this table it is seen that the rig operates with quasi-atmospheric conditions at the exit, and the expansion per stage is quite low. This is quite problematic when trying to derive the stage efficiencies through thermodynamic measurements, as the effective temperature drop per stage is of the order of 10K, this leads to the requirement of precise torque and mass flow measurements which will be discussed below.

Not withstanding the relatively low speeds at inlet, the reduced frequency of a typical stage, defined as:

$$\bar{\omega} = \frac{f\,c}{c_n} \tag{1}$$

takes a value (typically) of the order of 3.0. Physically this means that the stage will possess both steady and unsteady flow effects, and will therefore have the representative flow-physics required.

RIG DETAIL.

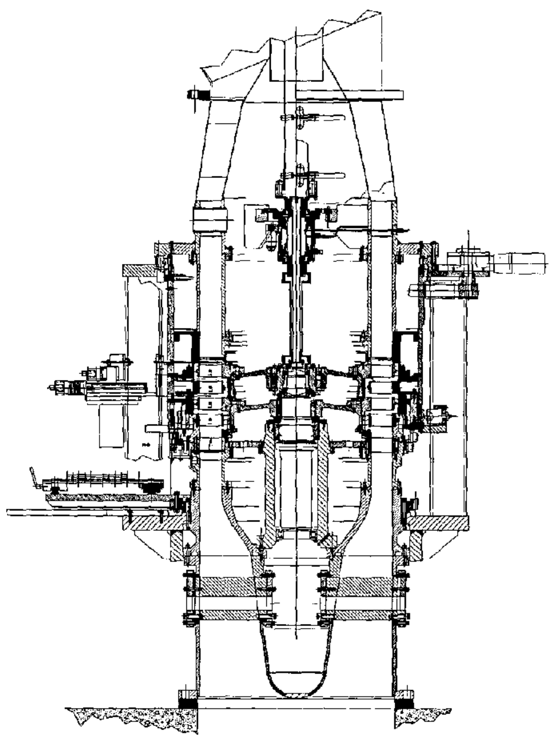

Figure 2. Test Section

Figure (2) shows the detail of the test section, with an expanded view of the measurement stretch shown in figure (3). Before the two main stages, there is a single blade row in the channel which consists of six thin profiles placed around the circumference. The purpose of this blade row is to provide structural stiffness for the main shaft bearings, with which to ensure a beneficial rotor dynamic behaviour. These support profiles have been arranged such that the wakes generated by the blading do not pass through the portion of the circumference where the probe measurements are taken. The measurement stretch has to accommodate stages representative of both steam turbine and gas turbine design philosophies. The two product areas, whilst having a degree of commonality in the aerodynamic challenges which they face, can be somewhat different in layout, for example the ratio of stator to rotor axial chord. To accommodate these changes, a number of exchangeable spacer rings permit the relative axial position of the blade rows and overall stage length to be adjusted. These also permit the study of the relevance of unsteady flow effects when the stage rows are brought closer together in the quest for more compact designs. The use of spacer rings and the accommodation of both gas and steam turbine designs is only possible because of the parallel, cylindrical hub and tip walls. For gas turbines, but particularly steam turbines, the lack of conicity in the stage annulus is an obvious design compromise over true turbine design practice, but is considered an acceptable simplification.

The core measurement concept of the rig is that the test section is encased with three separate outer casing 'skins' which can move circumferentially relative to each other. The first skin is directly connected to the first stator blade row., the second skin is connected to the second stator row and third skin comprises the outer casing of the channel. There are many reasons for the adoption of these independent skins. First of all, it permits an independent alignment of the two stator rows relative to each other ('indexing'), allowing stator-stator clocking effects to be investigated. Secondly, the third skin can be rotated circumferentially over a range of ±2.5 blade pitches. This skin is equipped with an extensive array of tapping holes, and traverse holes for probes. These holes have been selected with a large number of axial locations ahead of, in the middle of, and downstream of the measurement stage. This arrangement permits spatially dense measurements (radially, axially and tangentially) for all foreseen blade measurement options, thus achieving the design objective of measurement flexibility. The use of these skins and spacer rings posed two challenges in the design. Firstly the seals between the various rings have to be completely leakage free in order not to invalidate the measurements taken from the Venturi nozzle, but nevertheless permit the relative tangential motion. By careful experimentation with the seal design and attention to the assembly procedure. this was achieved. Secondly, the overall accumulated tolerances in the assembly from the use of so many rings had to be held within design intent. (Typically 0.05 - 0.1 mm). By attention to the manufacturing process and particularly again attention to the assembly procedures, this was also achieved. Checks of the tolerances after repeated assembly and disassembly cycles have shown that the build quality is repeatable.

There is the possibility of placing downstream of the second rotor a series of cylinders into the annulus. The objective of these clyinders is to replicate the potential effect of the leading edge of a downstream blade upon the exit flow phenomena. In addition, a rotating array of 1.5mm pins can be introduced ahead of the first stage, and these are used to introduce periodic turbulence fluctuations into the entry flow of the full two stage machine if desired.

MEASUREMENT WINDOW

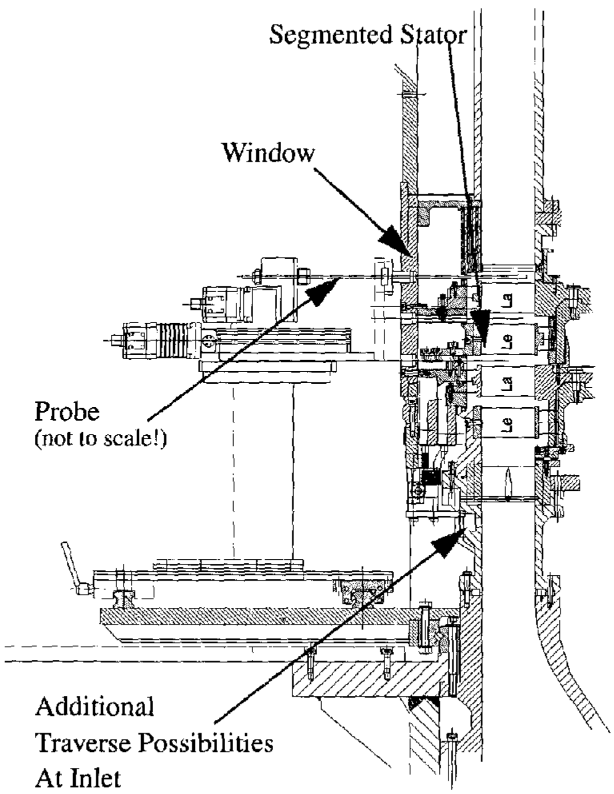

Figure 3. Expanded View Of The Test Section

As described, the test rig is disassembled axially, from above. However, not all changes strictly require an axial disassembly, and can be best affected radially. For this purpose, a so called 'measurement window' (figure (3)) has been engineered into the outer casing. This allows for the second stator ring to be split into a 300° and a 60° segment, the larger segment being fixed but the smaller capable of being mounted radially through the window. The advantage of this is that it permits for rapid changeover of instrumentation, say from a casing mounted fast response static pressure probe to a hot film gauge, and also allows for small modifications in the geometry within the stator blade row, say for example limited endwall contouring in five or six blade passages to look at the local flow physics under such changes. (But note that in these cases the global efficiency measurements will be inapplicable, and if required would have to be taken using inefficient and potentially inaccurate probe surveys). This window will also permit the introduction at a later date of optical access to the second stage, which will give options for flow visualization and LDA measurement techniques.

The window can also be used for holding the measurement probes, and it is possible to mount a probe anywhere axially within the window region. Of course, this is not the only area where probe traverses are of interest, so additional probe traversing holes are placed at the inlet and exit of the stage in order to assess inlet and exit flow conditions.

VENTURI NOZZLE

In a test rig of this nature, determination of the mass flow is a pre-requisite. For the purpose of CFD validation, the knowledge of the overall mass flow rate is a necessary condition for the correct implementation of the boundary conditions. More importantly the rig is directed towards the precise measurement of stage efficiency. However a thermodynamic determination of the efficiency for the reasons already indicated of a low temperature drop per stage is not possible, hence mass flow is a primary requirement in order to be coupled with a torque measurement.

Therefore a Venturi nozzle was built into the return leg of the pipe work. This nozzle, is designed to international (VDI) standards. Unfortunately, the inlet stretch to the venturi does not follow accepted practice in the standard, in that there is insufficient space to arrange for an undisturbed inlet flow into the nozzle. In particular the two corners preceding the venturi have the potential to produce a non uniform inlet velocity profile, and thus deviate the nozzle behaviour away from the ideal. Therefore, the venturi nozzle and the test loop including the two corners downstream of the turbine exit were calibrated as a single entity in an internationally accredited test establishment in the Netherlands. This yields the throughflow coefficient corrections as a function of the venturi Reynolds number.

TORQUE MEASUREMENTS

To complete the measurement of efficiency, torque measurements are required. However, torque measurements for performance are only valid when they are the torque developed in aerodynamically representative stages. A consequence of the defined axial inlet flow at the first stator in this (and many other rigs), is that the first stage often has an incorrect velocity triangle. Therefore measuring the total torque developed on the shaft is incorrect for the determination of the efficiency.

The usual method to deal with this problem is split the shaft in two, the first shaft connected to the unrepresentative first stage and a second to subsequent stages. Such arrangements can be seen in many test rigs (For typical examples consider the rig described in the paper of Södergard et al (1989)). Care has to be taken in such arrangements to ensure that the rotors are turning at identical speeds, something which may not always be easy in practice to achieve.

In Lisa, a dual shaft arrangement is impractical. The vertical arrangement of the rig with a long inlet stretch precludes running a shaft in the inlet stretch not only on the basis of difficult rotor dynamics but also due to lack of space in the cellar to place a secondary brake. Therefore, a different concept for the isolation of the torque upon the second rotor is needed.

The solution is shown in figure (4).

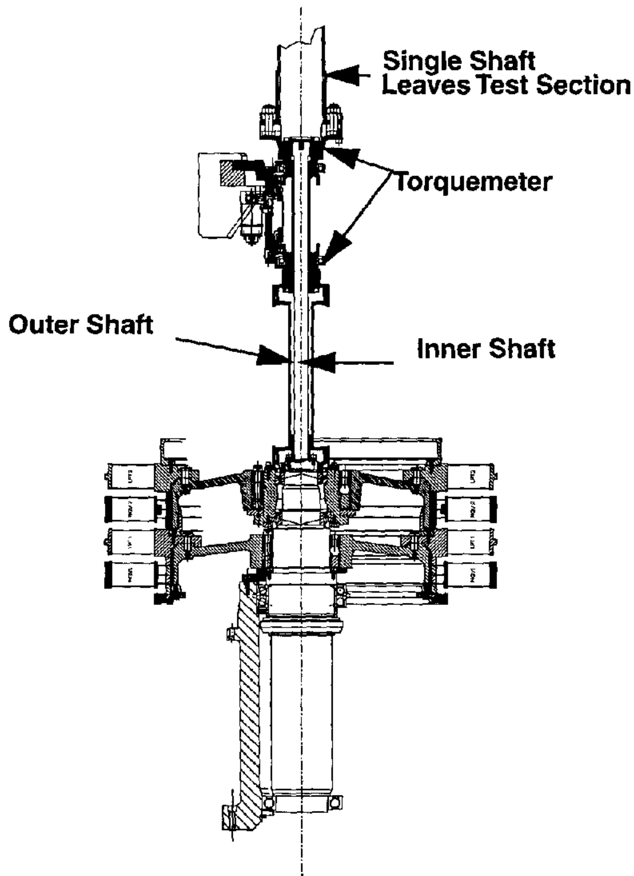

Figure 4. Split Rotor Schematic

In the measurement section the shaft has a twin shaft arrangement, a central shaft encased in a hollow outer one. The central shaft is connected to the first rotor, and the outer shaft to the second rotor. The outer shaft has low torsional stiffness, and forms the basis of a built in phase shift torque meter (van Millingen and van Millingen, (1991)), which works on the basis of measuring the phase displacement of two toothed wheels, from which the torque can be accurately derived using both static and dynamic calibrations. Behind the torque meter the two shafts come together and connect through to the braking system.

The advantages of the system are the compact nature of the torque measurements and the requirement for a single braking system. A minor problem in the arrangement is that it adds to the complexity of the rotor dynamics, but the main problem is the need for a shaft with low torsional stiffness for the torquemeter. A consequence is that the two rotors twist their respective shafts differently under load. This means in practical terms that the two rotors have slightly different leading edge alignments (or 'clock') relative to each other at different turbine operating points. This problem is addressed by two actions. Firstly the inner shaft has an equivalent torsionally stiffness, thus the two rotors clock to the same order of magnitude. Secondly both rotors provide a once per blade and once per revolution trigger signal, thus the magnitude of the phase difference between the two rotor positions is always known and can be taken into consideration in the data assessment. This information can also be used because the connecting shaft for the inner rotor allows for the possibility of clocking the two rotors in 0.1° steps, thus if necessary any clocking of the rotors due to operating point can be taken out of the system by pre-clocking the rotors in the opposite sense. Obviously this capability can be used for pre-meditated rotor-rotor clocking studies.

BLADE DESIGN AND MANUFACTURE

Considerable effort was expended in the identification of the manufacturing process for the rotor and stator bladings. The design criteria are partially contradictory, on one hand the costs have to be driven quite low to ensure that testing of multiple geometries is economically feasible, but on the other hand manufacturing processes have to be quite precise in order to achieve fine tolerances in the final blading in order to allow the differentiation of small aerodynamic effects. This section briefly outlines the choice, and the reasons therefore, in the method of the blade manufacture.

Rotor

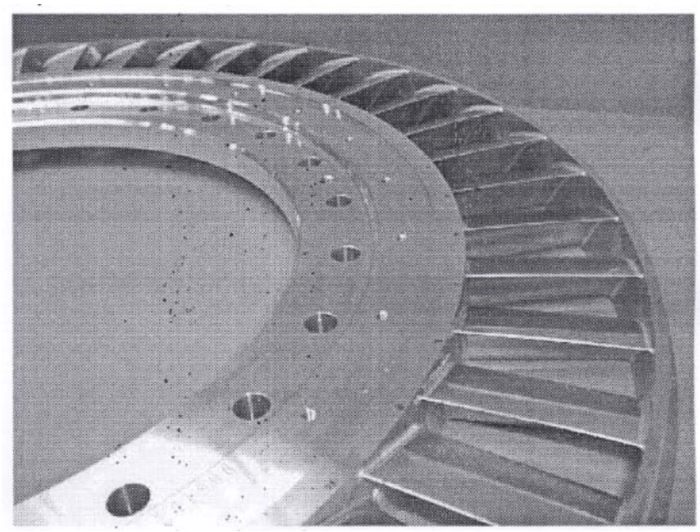

Figure 5. Rotor Blade Ring Assembly

The rotor assembly is designed as a two piece construction. Attached to the shaft is a blade carrier (see figure (2)) upon which separate blade rings can be screwed. This arrangement requires that only the blading itself, and not the entire rotor, needs to be exchanged, facilitating quick changeovers.

The rotor ring, figure (5), is constructed out of a single piece of aluminium. In a two step process first the coarse shape of the ring is milled, and then the final blade profiles are formed using electro-spark erosion. The advantages of the process are that firstly the single blade rotor piece affords higher mechanical safety than would be offered by connecting together individual blades, and secondly using electro-spark erosion, the repeatability on the profile shape is very high. All the individual blades in the ring are deviate away from the design intent at the the throat by no more than ±0.05 mm. This strong circumferential repeatability in the rotor geometries gives confidence in the geometric quality of the rotor aerodynamics and performance measurements. A minor disadvantage with the manufacturing process is that it is difficult (but not impossible) to instrument directly onto the rotor blade surfaces, and to a degree important phenomena, such as a cooling flow effects, are difficult to introduce into the rig Stator Blades

For the stator blades, figure (6), where mechanical integrity under rotational load is not an issue, individual blades are used. These blade are cast polyurethane blades. A characteristic of polyurethane is that the material shrinks unpredictably after the blade has been cast, making blade accuracy problematic. This is compensated for by using a two stage casting process, whereby a central core is cast first, and then a thin 2mm skin is cast around the core to reach the final blade design. The unpredictable shrinkage on the central core thus becomes irrelevant, and the shrinkage on the skin is both uniform (thanks to the uniform skin depth), and due to the thin depth of the second cast, an order of magnitude less than design tolerances.

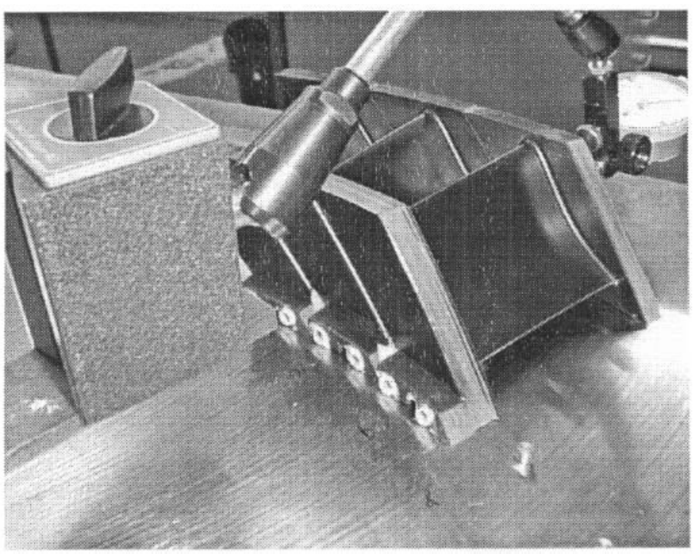

Figure 6. Stator Blade Ring In Final Assembly

Not only does this result in a blade with the same dimensional accuracy as the rotors, access to an easily machinable pre-cast core permits a simplified introduction of instrumentation directly into the blade itself, as runs for pneumatic and electrical lines can be easily set into the blade. The use of plastic cast blades is also economically advantageous, as the cost per blade is less than the cost of milling the individual blades from aluminium, by a factor of 3, despite the fact that two relatively expensive moulds are required for each blade design.

Figure (6) shows the stator blades in the final phase of mounting into an integral ring form. Extreme care is taken, using dedicated jigs and precise measurements to ensure that the blades are accurately positioned with respect to each other and to the centre line of the rig..

SAFETY SYSTEMS

As with all rotating systems, safety is the paramount consideration in operation. Multiple redundancies in safety monitoring and reaction, are, of course, mandatory in a machine of this nature.

Amongst many risks, a concern in design was the overspeed of the turbine, a result for example of the failure of the braking system. This scenario is considered likely to happen, as the feed of the electrical power into the net is dependent upon the presence of the main electrical net, and the historical statistics from the power provider point to two power losses *per annum*. To prevent catastrophic failures under these conditions, a compressor by-pass is installed which connects the turbine inlet to outlet (thus providing a less resitive path for the high pressure air at the turbine inlet). The by-pass consists of two fast acting pneumatic valves, one to isolate the main loop from the compressor inlet and one to open the link from inlet to outlet. Additionally an electrical brake capable of holding the load temporarily acts on the generator shaft. Despite this, the amount of unexpanded air in the loop is not insignificant and a rotor overspeed is expected. Simulations of the rotor behaviour in the case of a power loss point to at the most a 10% overspeed, after which the rotor windmills safely down to rest. This maximum overspeed is significantly within all the design margins for bearings, balancing, material strengths etc.

DATA ACQUISITION AND CONTROL

As is the norm in such test stands, all data acquisition and control functions are controlled by a single computer integrating together the separate elements of the system (operating point, measuring point, measurement configuration etc.). This permits an on-line monitoring of the test stand operating point and ready construction of results files appropriate for further on-line post processing or direct entry into relational databases.

Notable in the new rig is a dedicated communication between the main data acquisition computer and the generator via a profibus-DP line. Profibus-DP is a distributed bus communications system working on a master-multiple slave principle, whereby the slaves can be monitoring devices or separate distributed control systems such as PLC (SPS[2]) devices. This comminication between the data acquisition and the system-bus is helpful, as it allows monitoring of the generator conditions, (permitting a degree of forewarning of potential problems), and also gives a further check on the shaft rotational speed, providing for a further level of redundancy in

2. SPS is the (generally better known) German abbreviation for PLC

the measurements. For safety reasons, the data acquisition reads data only from the Profibus and is disabled from writing data to the Profibus. This means that there is no dependency of the generator and compressor upon the data acquisition systems, and normal functioning cannot be compromised through a data acquisition system failure or accidental misuse.

MEASUREMENT METHODOLOGY

As previously stressed, the principle use of the test rig will be to gather basic performance data, and back this up with spatially and temporally dense data sets with which to appraise the importance of any unsteady flow effects present. The measurement technology chosen for the latter task is an in-house developed fast response pressure probe system (Köppel, Kupferschmied and Roduner et al (1999a,b,c)). This permits the accurate determination of the time resolved flow using small diameter (0.84 mm) needle probes. Such probes have the advantage of being robust, with high measurement bandwidths. Being pressure based sensors, they can access directly the static pressure fields which could be of importance when assessing the impact of fluctuating potential fields, a notable advantage over competitive measurement technologies such as hot wire or LDA which can only measure directly the velocity fields. An obvious disadvantage of using probe based technology is the inherent interference effects which result from their presence in a flow field, however the probe diameters in use within the test rig are at least an order of magnitude smaller than any relevant geometrical size (for example rotor-stator spacing), and thus considered appropriate for use.The fast response probe traverses will be supplemented with pneumatic probe measurements, both using small bore needle probes acting in a virtual four hole probe mode, and cobra-form small head diameter (0.9 mm) five hole probes. Wall pressure measurements, using both pneumatic and fast response sensors will be taken. Hot film gauges will be used to resolve wall shear stresses where required.

CONCLUSIONS

This paper has presented the design and construction of a low speed two stage air turbine (figure (7)) designed to investigate the relevance and impact of unsteady flow effects upon aerodynamic performance.Several innovative design aspects have been highlighted; in particular:

- The split shaft and integral torque meter arrangement for isolated torque measurements on the aerodynamically representative second stage.
- Precise mass flow measurements via a calibrated Venturi, including most of the exit stretch.
- Single piece rotor assembly fabricated through electro-spark erosion, yielding high accuracy coupled with mechanical integrity.
- Stator blades produced through a two stage polyurethane casting process.

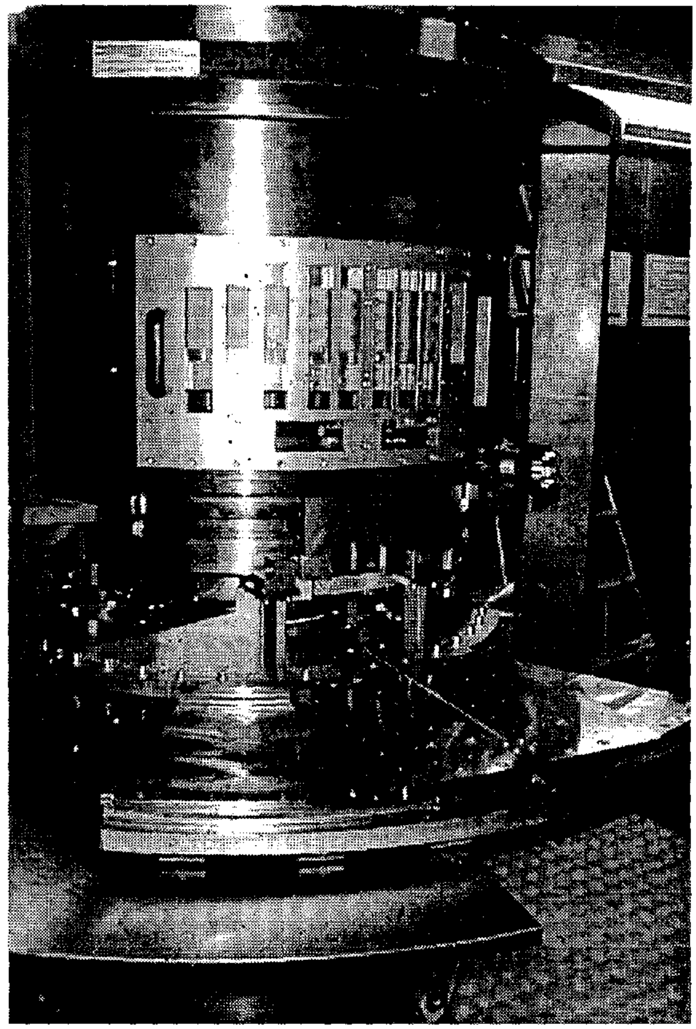

Figure 7. Completed test Rig

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