

DESIGN FEATURES OF STEAM TURBINES FOR USE IN THE SUGAR INDUSTRY

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Abstract

Steam turbines employed in the sugar industry require high thermodynamic efficiency and absolute reliability. The AEG-KANIS Steam Turbine Company, situated in Nuremberg, West Germany, and for many years a supplier to the South African industry, developed such turbines based on 30 years of experience with turbines for all industrial purposes. A special feature of these typical turbines used for power generation in the sugar industry is that they have no control wheel and that they have two clearly defined load points, one at approximately 60-70 per cent load and the other at the maximum load point. Via two valves the steam is directly admitted to the reaction blading. In the maximum load point both valves are totally open. The major portion of the steam flow is, however, fed into the turbine via the second valve and in this way a small number of initial stages are bypassed, so using the high efficiency values of the reaction blading design over the total enthalpy drop in the turbine. The design which incorporates shrouded blades guarantees a paramount vibration insensitive construction together with the advantages of high thermodynamic properties.

Introduction

The AEG-KANIS Company manufactures small single-stage impulse-type turbines with outputs of a few hundred kilowatts to large steam turbines of the reaction type for industrial power plants. Such turbines can produce outputs of 60 to 100 MW. In between these two extremes are

turbines of outputs ranging from 20 to 30 MW which are also used for electric power generation and as compressor-, pump- and fan drivers. Such turbines are of a far more durable and sophisticated design, because they have to endure the rough service conditions that are very often experienced in refineries and at other petrochemical chemical plants and in sugar factories.

Approximately one year ago the company was awarded the orders for three identical turbosets with an electrical output of 8 MW each by three sugar factories in South Africa. This paper presents a lot of details of these turbines, especially their design features in respect of reliability and efficient performance.

New Turbosets for South African Sugar Mills

Three turbosets have been supplied to sugar mills in Malelane, Noodsberg and Sezela. For the sugar factory in Malelane it is the fifth turbine to go into service. All turbosets are designed to develop an electrical output of 8 MW at maximum load point.

Efficiency and Reliability

Because of the present day energy problem the efficiency of a turbine plays a decisive role when selecting a suitable thermal power generator, but when operational reliability is sacrificed for efficiency, trouble ensues. The steam turbines supplied for the above three plants constitute a well-balanced compromise, offering a high efficiency and reliability.

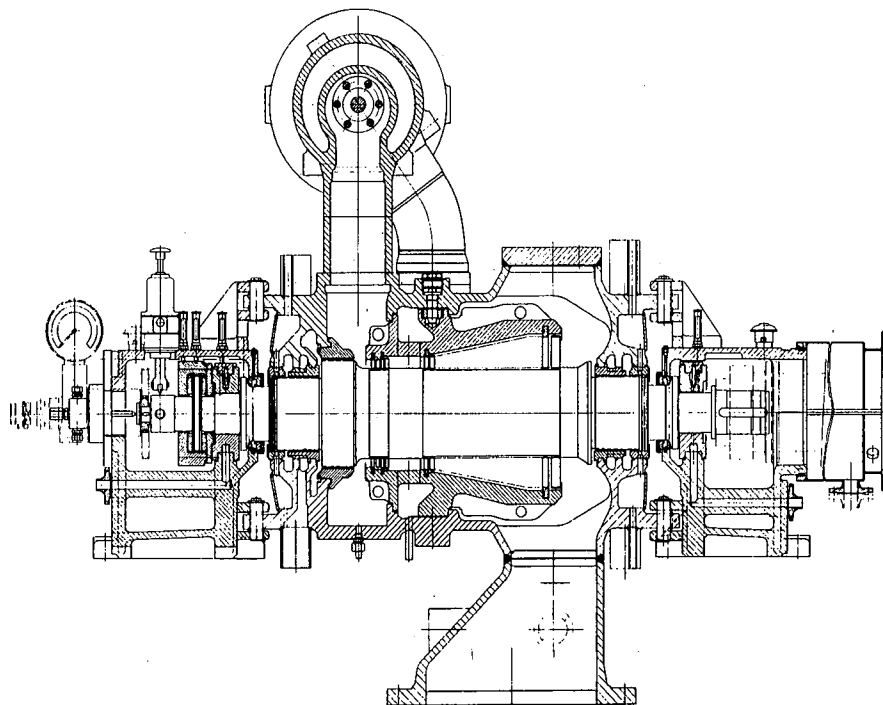


FIGURE 1 Sectional Drawing of the G25 Backpressure Turbine.

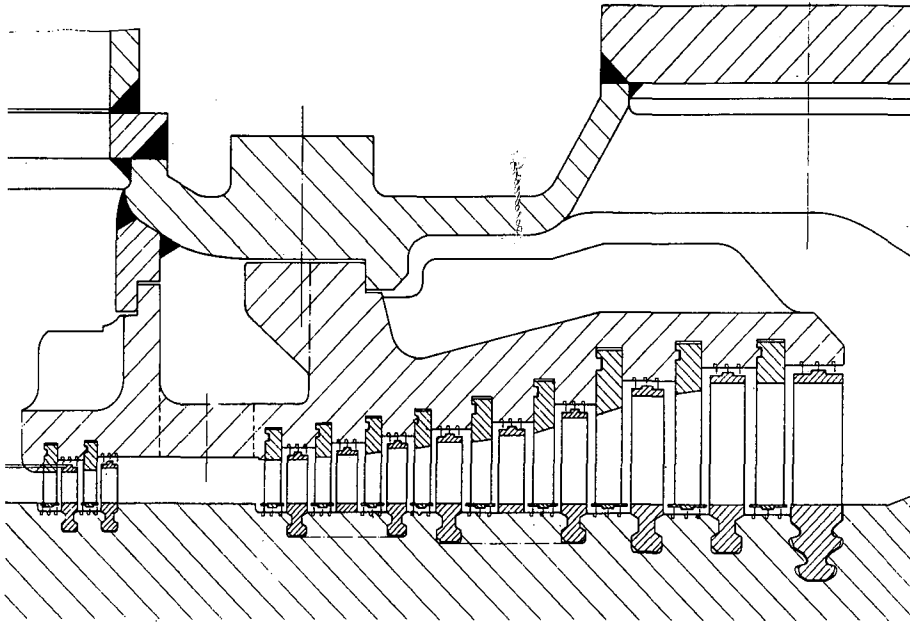


FIGURE 2 Blading Drawing of this Reaction-Type Turbine.

Design Features of these Turbines

A sectional drawing (see Figure 1) may demonstrate the principal design features. This turbine consists mainly of a horizontally split casing of cast steel. All valves, two in this special case, are firmly connected to the upper turbine section. The emergency stop valve, not shown in this drawing, is flanged on ahead of the control valves. Both bearing housings are horizontally and vertically connected to the main turbine casing by two bolts each and also by small supports in the horizontal plane. The fixed point of all these turbines is the exhaust flange, so that the total turbine can move axially if there is a change in temperature during startup or shutdown, or even in case of load variations.

The front bearing housing contains both the radial bearing and the axial thrust bearing which is to compensate for the thrust resulting from the coupling and the blading. Moreover, a trip device installed in the rotor prevents the turbine from reaching hazardous overspeeds.

The rear-end bearing housing is of similar design to the front-end bearing housing and contains only the journal bearing. As already mentioned, the front-end bearing housing contains also the thrust bearing which has to guide the rotor in axial direction. For this purpose a well-proven design incorporating a Mitchell-type thrust bearing is in use.

The journal bearings provided for this turbine model are four-lobe bearings. They have good resilience and damping behaviour as well as high loading capacity. This type of journal bearing is successfully employed in machines operating at speeds of up to 14 000 rpm. In 1967 these bearings were standardised in collaboration with the Technical University of Karlsruhe. This basic research effort proved so successful that this type of bearing is now used in all the turbines with very good results. During this research work, the bearings have been tested on a high-speed test facility in order to measure the stiffness and damping coefficients as a function of the Sommerfeld's number. This will be discussed later together with shaft vibration behaviour.

The rotor and the blades as well as the guide-blade carrier are considered the "core section" of any turbine.

It is especially these major components that are responsible for the operational reliability and efficiency of a turbine.

Turbine Rotor and Blading Design

The blading of this backpressure turbine consists of two blading sections (see Figure 2). At the front there is a stage group consisting of two rows, succeeded by a further stage group with 9 rows. It is this construction type in particular which has proved optimal for application in the sugar industry. Under full-load conditions the major part of the steam flows into the turbine through a large valve, right after the first stage section. At partial load conditions, i.e. at about 75 per cent, only the first valve is open, so that the steam flow will pass through all the eleven stages of the turbine. In this way, efficiencies are achieved, both under maximum and partial load conditions, which lie far above those values that may be attained by a turbine incorporating an impulse design feature.

Efficiency of a Reaction-Type Turbine

The design of this reaction-type turbine is also economical, because it does not require a control stage nor the pertaining nozzles. When considering further that an impulse-type blading may achieve maximum efficiencies of about 75 to 80 per cent, whereas a reaction-type blading of the type normally employed may attain efficiencies of 80 to 90 per cent, it becomes quite evident that a turbine equipped without any impulse part will obviously achieve a higher efficiency (see Figure 3). Especially in the sugar industry, where the requirement of achieving good efficiencies at extreme partial load conditions is of minor importance, it is considered that this type of turbine is an extremely reliable machine and efficient in steam consumption.

At full load condition the specific steam consumption is 7,76 kg/kWh and 7,51 kg/kWh for partial load condition at the first valve point (see Table 1). These specific steam consumption values have to be considered in direct connection with the turbine and steam design data illustrated in Table 2

Under these conditions very high efficiencies of approximately 86,5 per cent are achieved in the blading section. Nevertheless, these high efficiencies will always have to be looked at in conjunction with the operational reliability. To provide this reliability, a highly sophisticated and vibration-proof design of the blades and shaft is required.

Turbine Blading

The normal blading design consists of moving blades milled from the solid. The blade root, the blade profile and the cover plate are milled from one single piece of material (see Figure 4). When fitted into the slots, the

individual blades are tightly placed one to another. Small shims are placed underneath each root in order to provide perfect and solid positioning of the blades in the rotor slot. This results in an absolutely vibration-insensitive blading which has been operating trouble-free, even in compressor drive turbines with wide speed ranges, for about 12 years.

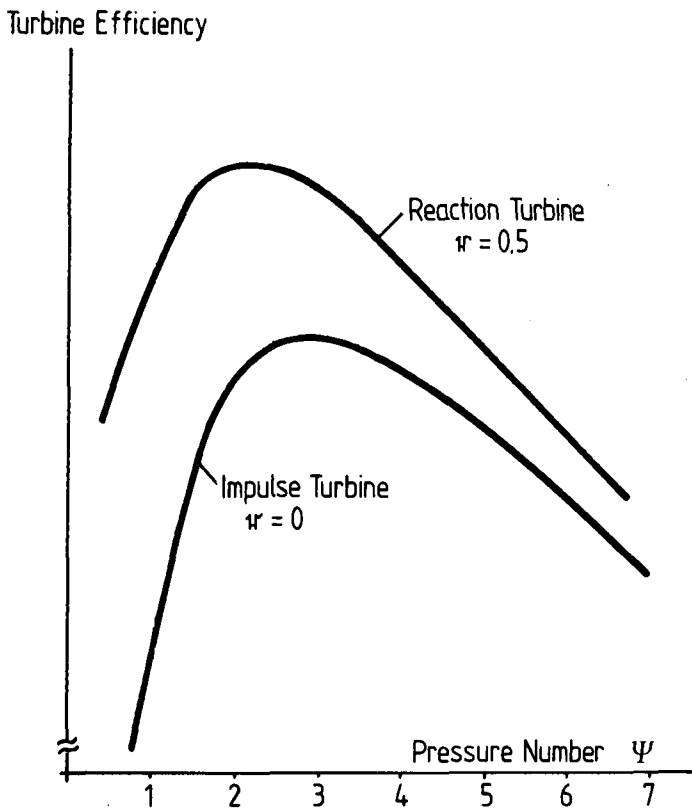


FIGURE 3 Efficiency of Impulse- and Reaction Turbines.

TABLE 1
THERMODYNAMIC PROPERTIES

	Load Point 1	Load Point 2
Output	6 MW	8 MW
Steam flow	45 t/h	62, 12 t/h
Specific steam consumption	7,51 kg/kWh	7,76 kg/kWh
Efficiency of the blading	86,5%	83,8%

TABLE 2
DESIGN DATA OF THE 8 MW TURBOSETS

Malelane, Noodsberg and Sezela	
Live-steam conditions	$P_e = 32$ bar $t_e = 400$ deg. C
Backpressure	$P_s = 2,04$ bar
Maximum steam flow	$D = 62,12$ t/h
Maximum electrical output	8 000 kW
Turbine speed	10 000 rpm

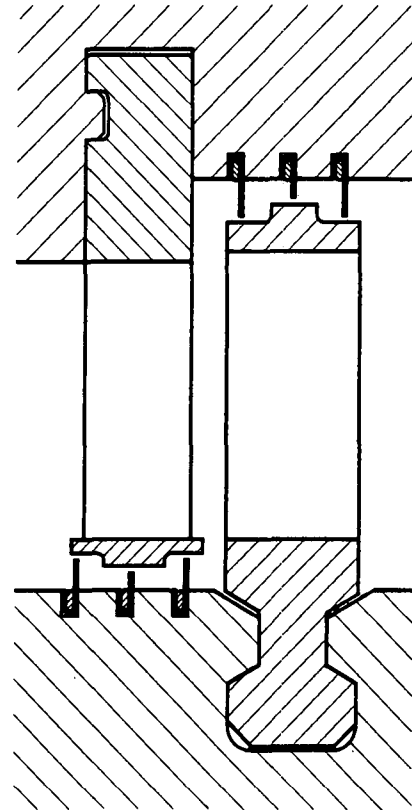


FIGURE 4 Design Principle of our Turbine Blading.



FIGURE 5 View of the Rotor- and Stator Blades.

In order to reduce the inevitable leakage flows occurring above the blade tips, sealing strips were fitted opposite both the stationary and moving blades. Together with the special shape of the shrouding, they form a labyrinth that offers sufficient sealing to the steam to make it flow through the blading (see Figure 5). The gaps between the thin sealing strips and the blade tips have to be narrow to ensure a high sealing effect; there is an actual radial gap of approximately 3 to 5 mm before rubbing between the solid material can occur. Such radial displacements are unrealistic, even at abnormal conditions in turbine operation. The use of sealing strips will increase the efficiency considerably.

The guide blades are manufactured from an extruded profile material. They are cut to the necessary length and then fitted to the guide-blade carrier with the help of spacer parts so that they are correctly spaced from one another. For the guide blades it is not necessary to use the costly type of blades that are milled from the solid. Since stationary blades are not subjected to centrifugal stresses, the shroud band may be riveted on (See Figure 4).

Experience with this Type of Blading

Blades of this construction type, i.e. moving blades milled from the solid with integral shrouds and stationary blades of drawn profile material with riveted-on shrouds have proved most successful since 1970. Up to the present day, approximately 750 000 of these blades have been used world-wide. Their success indicates that this blade design of very low bending stresses is the solution to the problem of how to obtain high efficiencies and optimal operational reliability at the same time.

Guide-Blade Carrier

The operational reliability of the turbines is also essentially attributable to the use of guide-blade carriers. They are horizontally split (see Figure 6) and contain all the stator blades with riveted-on shrouds. As illustrated in the blading drawing of the turbine (see Figure 2), the stationary blades are not inserted in the turbine casing, but fitted to a separate inner casing which is fixed by means of bolts

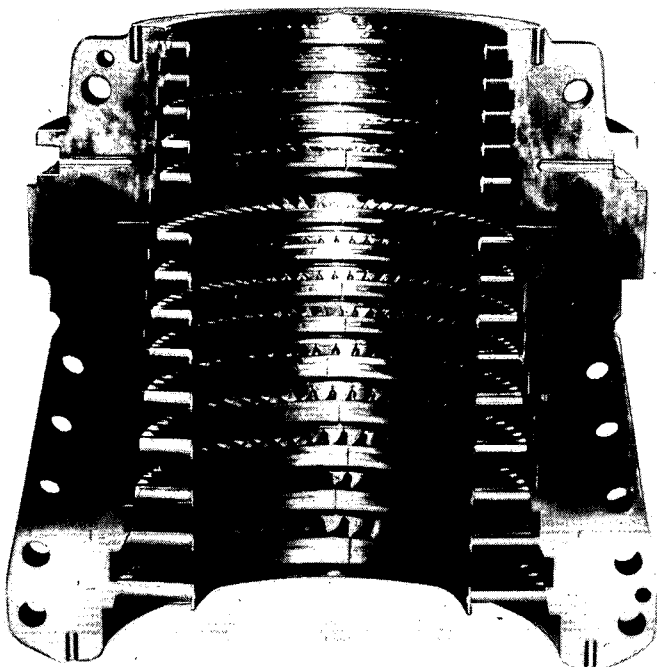


FIGURE 6 Guide Blade Carrier Half.

to the main turbine casing. The use of guide-blade carriers is one of the outstanding features of these reaction-type turbines. Much experience has been obtained in this field since guide-blade carriers were manufactured for the first time in 1950. This design idea is common to all the reaction-type turbines and has proved successful world-wide in several thousands of turbines. The international reputation of the guide-blade carrier design becomes obvious when looking at the American API standards. Here it is a must to apply this design in high-speed compressor drives for reasons of reliability.

Such guide-blade carriers offer the following advantages:

- (a) There is a speedy temperature balance between the rotor and the guide-blade carrier which is totally steam-swept.
- (b) A large temperature gradient makes this design insensitive to both temperature and load variations.
- (c) A movement in the casing partition joint does not affect the radial blade clearance.
- (d) This design offers easy servicing with simple and problem-free erection, due to reasonably sized components.
- (e) Quick replacement of a spare rotor and guide-blade carrier on site, in case of a failure, without returning the turbine to the manufacturer's workshop.
- (f) Short startup periods.

Shaft Vibration Behaviour

In addition to the choice of suitable bearings, another very important factor for the operational reliability and quiet running behaviour of a turbine is the accurate dimensioning of the shaft diameter, the bearing span and the speed. Depending on the heat gradient available, a certain number of blade stages must be assigned to the turbine type selected in order to obtain the best thermodynamic result. This will lead to a bearing span that has to be examined for its shaft-vibrational behaviour. Formerly, i.e. before 1965, only the "critical speed" on a basis of rigid bearings was determined and a shaft was considered to be acceptable if this calculated value lay approximately 20 to 30 per cent above the operating speed. In their calculations the engineers totally disregarded or overlooked the stiffness and damping effect of the journal bearings in use. In fact, depending on the type of shaft and bearings employed, actual resonant speeds are reduced to 50-80 per cent of the critical speed in rigid bearings by these special bearing characteristics. For these reasons, thousands of machines run within the resonant range all over the world. In most cases no problems have arisen, for oil film damping has helped to reduce the vibration amplitudes to such low values that they may be neglected. In present-day engineering most turbine rotors are operated above their first resonant point, if the entire system, consisting of the rotor, the bearing and the bearing housing, can be properly computed.

If balanced correctly, rotors which are operated at speeds above their first "stiff shaft" resonance show the same good, or even better, running behaviour than those whose resonant point lies above the trip speed (see Figure 7). When considering the measured stiffness and damping coefficients of the journal bearings employed for the actual turbine, it is possible, with the aid of advanced calculation methods, to predetermine most realistically the vibrational behaviour of the shaft.

In order to check this sensitivity against sudden unbalances during operation, maximum permissible balancing

quality is used as a basis before calculating the vibration amplitudes occurring in the bearings or anywhere along the shaft.

In practice, however, balancing qualities better than those taken for this calculation are obtained, so that the amplitudes in the bearings during the running test are much lower than those established in advance for the calculation.

In the event of turbines which are operated below the first resonant point, the efficiency will be reduced inevitably, because the short bearing span does not permit the installation of the number of stages required to achieve a high efficiency. As illustrated in Figure 7, the resonance behaviour is also predominantly influenced by the stiffness of the rotor itself. An impulse-type rotor has to be much more flexible than a rotor of the reaction type for efficiency reasons. This results in a quite different shaft vibrational behaviour than can be established by computer evaluation.

The steam turbines have been operated and designed according to this shaft vibration philosophy for more than 15 years. With the aid of a present-day balancing standard it is well possible to operate such turbines that run at speeds above the first "stiff-shaft" resonance.

Also, the turbine rotors of the different turbine types employed in the sugar industry are so rigid that they will not sag when passing through the first resonant point. Due to the flexible support on the bearings, the rotor will vibrate only as a rigid shaft in a vertical direction without being bent by the dynamic forces. The fact that such rotors can be balanced at low speeds is a further proof that no important deflection will occur in spite of the rotors being operated at speeds above the first "stiff-shaft" resonance. The smooth running behaviour of a shaft operating at such conditions is in any case better than that of an assumed so called "rigid" rotor. A shaft operating above a speed called "first resonance" rotates around its own centroidal axis, which, depending on the balancing quality, differs only slightly from the geometric centre line.

References for this Type of Construction

Turbines of this design, i.e. without a control stage but with a first and a second bypass, are also often used for

the topping machine with syngas compressors, and will have to withstand live steam conditions of 130 bar/540°C at steam flows of up to 300 t/h and speeds of up to 14 000 rpm (see Figure 8).

Since this principle has so far proved most successful with such heavily loaded turbines it was adopted as the standard and should stand up to the relatively less stringent performance requirements prevailing in the sugar industry.

Overall Plant Concept

The turbines supplied to the previously-mentioned sugar mills are run at approximately 10 000 rpm. The turbine speed is reduced to 1 500 rpm via a single-stage gear unit driving the alternator. The approximate dimensions of the overall unit are :

Length : 8 500 mm. Width : 3 435 mm. Height : 2 885 mm and Weight : 52 t.

These machines and larger units for the sugar industry, as delivered in Europe, are expected to achieve performances of up to 20 MW. In this case the turbine type next in size is used i.e. model size G 32. This turbine, which runs at 8 000 rpm, drives the alternator via a single-stage gear unit.

Because of the size of the unit, both the turbine and gear unit are installed on a common baseplate, whereas the alternator is placed on a separate foundation. From a certain unit size on, and to avoid an excessive concentration of weight, this solution is very practical.

The alternator is a 10,0 MVA machine with a voltage of 3,3 kV.

Safety Devices and Monitoring Equipment

To operate a turboset most reliably it is necessary to have adequate safety and monitoring equipment. Over-speed, axial displacement, shaft vibrations, and variations in oil and backpressures are the parameters most commonly used for giving alarm or even for tripping the unit. These precautions are in accordance with the international standard. In chemical plants or refineries special demands are made

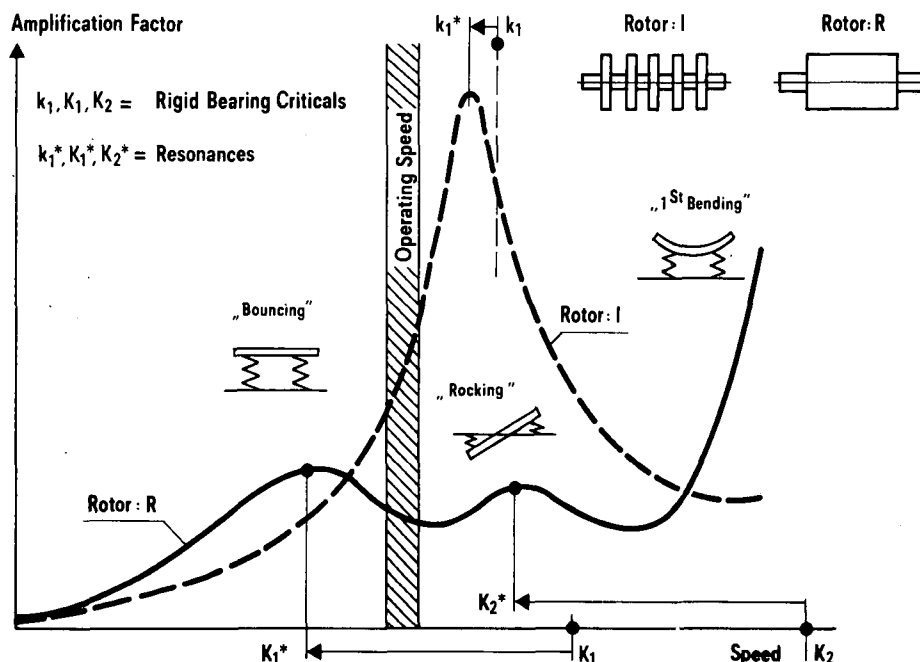


FIGURE 7 Shaft Vibrations of Impulse and Reaction-Type Turbines.

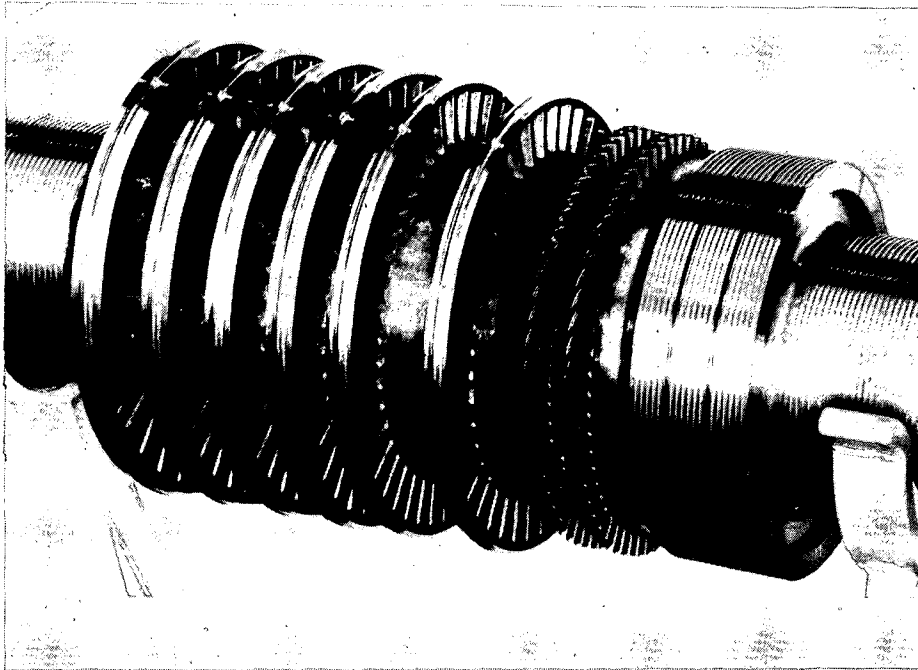


FIGURE 8 Rotor Design of a Syngas Compressor Drive Turbine.

on this type of equipment. These standards have been adopted for all the turbine products, including those employed in the sugar industry.

Conclusions

The basic concept of the turbine that has been discussed is the design principle of the machine which has been successfully employed all over the world for about 20 years.

The experience and knowledge gathered in modern turbine engineering with special regard to optimum blade design (efficiency and operational reliability) have been constantly incorporated into the various turbine models including turbines for power operation in the sugar industry.

To ensure that an alternator drive turbine is reliable in operation, the design criteria of a compressor drive turbine should be used as a basis.