



## DESIGN OF A 500 KW TURBOEXPANDER

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### ABSTRACT

The design and development of a new generation of an economical 500 kW turboexpander/generator is presented. This paper briefly discusses current pressure reduction power recovery applications and describes in detail major steps in the development of this new 500 kW Turboexpander/generator.

### INTRODUCTION

The application of a Turboexpander to gas pressure reduction has been discussed in previous papers by Cleveland (1988a)<sup>1</sup> and Swearingen (1972b)<sup>2</sup> presented at this conference. In studies which have been carried out during the past five years, the possible use of a turboexpander to recover the energy normally wasted in gas pressure reduction has been extensively evaluated. Typically, applications for turboexpanders are at city gate stations where pressures are reduced from main pipeline pressure to local distribution center pressure and at process plants where gas feed stock is used from relatively high pressures. Investigation of typical city gate station and process applications indicated that most of the units required would be in the 200-500 kW range. The vast number of turboexpander applications that could be considered as replacements for regulating valves lie within this range.

<sup>1</sup> Cleveland, A. "Turbo Expanders for Energy Recovery; Design and Installation of a 250 kW Unit," ASME Paper No. 88-GT-266, 1988

<sup>2</sup> Swearingen, J.S., "Turboexpanders and Processes That Use Them," Chemical Engineering Progress, July 1972.

### RANGE OF PRESSURE REDUCTION STATIONS

Typical pressure reduction stations based upon analysis in Alberta and for other representative locations in North America indicate that 80% of the pressure reduction applications are capable of generating significant power – in some locations, up to about 500 kW. A high percentage of these (about 65%) lies in the 200 kW range or less. The remainder can go as high as 500 kW. (Fig. 1)

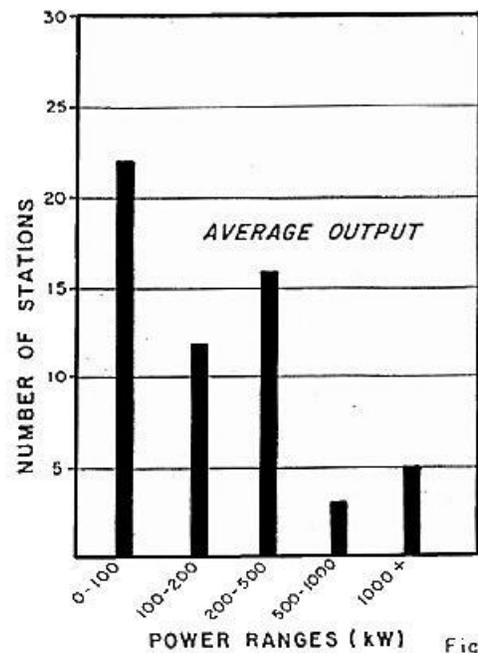


Fig. 1

With an economical, low cost turboexpander, recovery of energy in these installations becomes practical, provided electrical costs are 3 cents per kilowatt hour or greater. Other factors also have to be considered – in particular, the proximity to electrical transmission lines which utilize the power or the proximity of user plant facilities, and the availability of a preheating source, since the turboexpander produces a greater temperature drop than the corresponding regulating valve.

Given the satisfaction of these basic economic parameters and the appropriate physical locations, it is possible to install the turboexpander alongside existing pressure reduction valves in many locations and generate electricity for sale or local plant use.

The initial development of the REACT turboexpander envisaged units of up to 125 kW. The power output limitation for these units was dictated by the gearbox and not the turboexpander itself.

The REACT turboexpander is a simple, single-stage, axial-flow impulse turbine directly connected to an integral high speed gearbox which provides the necessary speed reduction to match the induction generator. The unit is self-contained, installed on its own skid and provided with all the necessary auxiliaries as a single package. (Fig. 2)

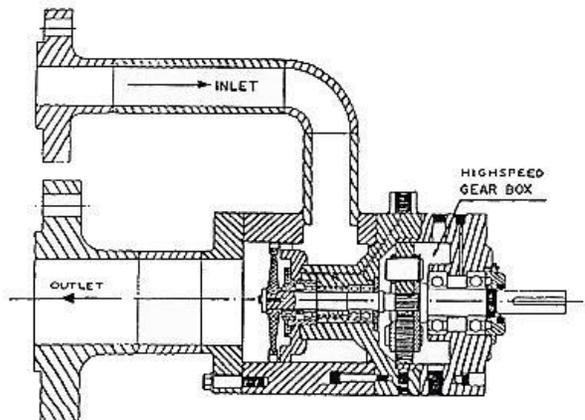


Fig. 2

This 125 kW unit is suitable for a number of sites in the small to medium range flow category, but for a significant proportion of the installations, multiple units are required if full use is to be made of the available power. It has already been pointed out that approximately 65% of the installations surveyed were capable of 200 kW or less leaving 35% which can produce 200 kW or more.

One solution was to use a double-ended unit, that is, two 125 kW turboexpanders connected to a single, double-ended induction generator. Such a unit was installed at Canadian Western Natural Gas in 1988. (Fig. 3) This allows 250 kW to be produced from a single skid. Special care was taken in the design of the connecting pipe work to ensure a balanced gas flow to each turboexpander to get equal load sharing at full power output and to control pipe movements which impose stress on the turboexpander flanges. As always with rotating machinery, it is imperative to maintain alignment of the two units with the single generator and, therefore, provisions were made in the discharge pipes to accommodate thermal growth and shrinkage.

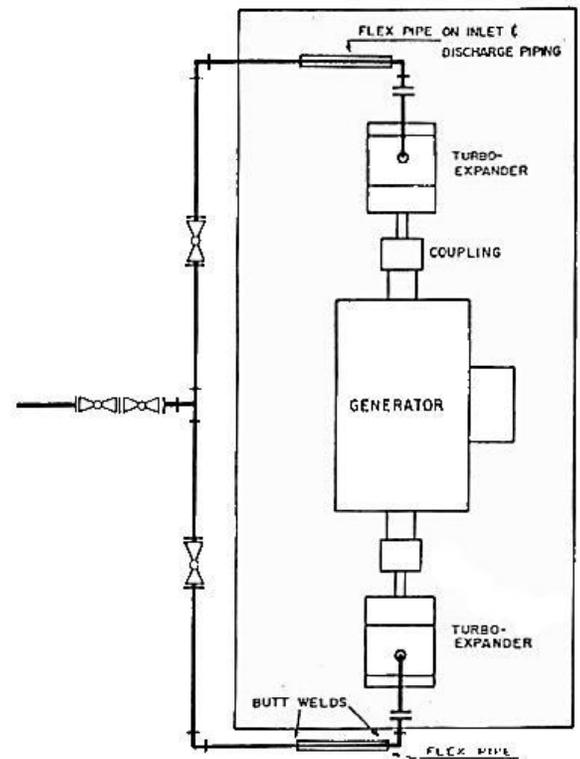


Fig. 3

The basic turboexpander wheel is capable of considerably high output than 125 kW. The power turbine itself can produce in excess of 550 kW. Therefore, it was decided to develop a 500 kW machine which would satisfy the requirements of the larger city gate stations and process plant applications. The following sections outline the considerations which led to the final design of the 500 kW unit.

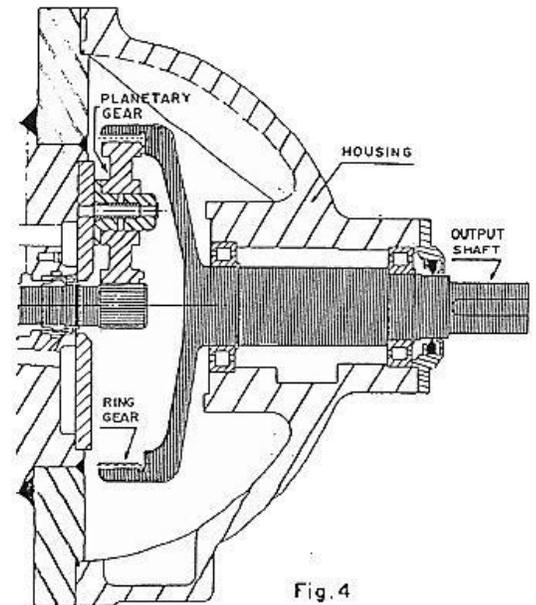
## GEARBOX

The limiting component in the existing turboexpander design was the integral epicyclic gearbox which was only rated for 125 kW. The possibilities of enlarging this gearbox to allow for increased output were first considered. It soon became obvious that the design problems inherent in simply enlarging the gears' working surfaces were significant. It was evident that a satisfactory design could not be had within the confines of the existing gearbox envelope. Therefore, it was decided to purchase a new, larger gearbox from a qualified supplier. A specification was drawn up to solicit bids for the gearbox.

The specification was based on achieving at least 500 kW continuous operation with output speed of 3000 or 3600 rpm and a gear ratio of approximately 10:1. The choice of gear ratio was based upon two considerations. First, to maximize the output from the existing turbine stage, the turbine rotor speed should be increased to between 30,000 and 36,000 rpm. This maximizes stage efficiency by nearing an optimum ratio of rotor tip speed ( $U_2$ ) to process gas isentropic spouting velocity ( $C_0$ ). This increased speed is well within the material stress limits of the turbine wheel. The second consideration was that a ratio of 10:1 is about the practical limit of a conventional single-stage, epicyclic reduction gearbox.

It was decided to continue with the concept of the epicyclic coaxial reduction gear forming an integral unit with the turbine. This concept has a number of advantages, not the least of which is no high speed coupling since the turbine wheel mounts directly on the high speed pinion (sun gear) shaft and drives directly into the planetary gears. It is compact, rigid and an easily handled design compared with a separable independent gearbox. An additional benefit is, of course, that a single lube system can be used for the gearbox and turbine. A combined oil system does, however, require a careful selection of oil type and viscosity to suit both pieces of equipment.

Proposals were obtained from a number of gear companies worldwide and finally Compact Orbital Gear Company, Ltd. (C.O.G) was selected as a supplier. The final design gear ratio which was determined after studies with C.O.G. was 9.96:1 which gives turbine speeds of approximately 30000 and 36000 rpm, with the 50 Hz and 60 Hz generators, respectively. The gearbox is contained in a hemispherical housing which attaches directly to the rear face flange of the turbine body, making a single integral unit. (Fig. 4)



In this gearbox design, the planets are rotating on fixed pins. The ring gear rotates and also forms the output shaft to the generator. Initially, a quantity of two gearboxes was made in order to carry out a back-to-back full load, full speed test for the in-shop testing in accordance with the REACT specification.

This shop test was successfully carried out in September 1988 and the first two gearboxes shipped to REACT for installation with the turbine.

## TURBINE DESIGN

While the design of the turbine wheel aerodynamically was unchanged from the earlier, small unit, the nozzle area was increased and the rotating assembly required considerable revision. To develop 500 kW power output at the design gas flow rate while maintaining reasonable stage efficiency, the nozzles were increased from partial to full admission. Figure 5 is an example of the performance of the 500 kW turboexpander/generator. The initial design utilizes subsonic, converging type nozzles. Supersonic, converging-diverging nozzles will be used when high gas flow rates result in velocity coefficients significantly over Mach 1.0.

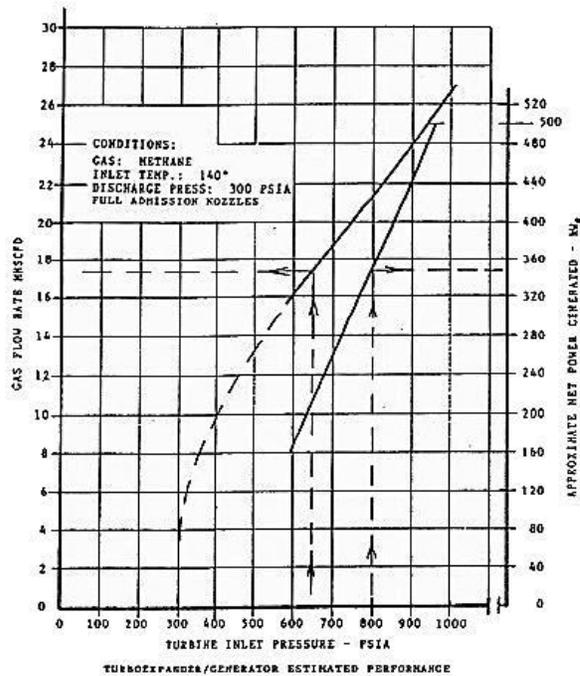


Fig. 5

To accommodate the increased torque, the turbine shaft had to be increased in diameter from .67 to .80 inches and this necessitated reconsideration of the wheel fixing to the shaft. The turbine wheel is fit on the shaft with two, stepped interference-fit cylindrical pilots. The wheel must be warmed for installation. A specially designed stretch bolt is used to provide a substantial clamping force on the wheel. Torque is transmitted from the wheel into the shaft by four modified dowel pins used as drive keys. These cylindrical pins avoid the possibility of having stress concentration as may be the case when using conventional square or rectangular keys. Along with these changes a new seal, utilizing the latest technology, is used. Increased thrust loads along with critical speed considerations required detailed investigation of alternative bearing arrangements.

### CRITICAL SPEED AND BEARINGS

A major part of the work on the design of this unit has been the evaluation of the lateral critical speed of the shaft and the selection of a suitable bearing configuration. From the outset it was decided that the criterion of the first lateral critical speed would be at least 20% above the maximum operating speed and that the current type of anti-friction bearings would be retained as far as possible.

As with all such lateral critical speed calculations, it is necessary to iterate on bearing stiffness, shaft length between bearings and overhung mass in order to arrive at a satisfactory final design. The design which evolved is a triple tandem, angular contact bearing arrangement at the turbine-end of the shaft to provide necessary capacity for axial thrust and radial stiffness. A single, large diameter, angular contact bearing is located at the gearbox-end to carry the radial loads. This arrangement results in a calculated critical speed of 43,800 rpm and adequate capacity to sustain all envisaged thrust loadings on the disk at all conditions of expected discharge pressure. The critical speed analysis had been based on a steel disk turbine wheel, although in the majority of cases aluminum will be material which is used, thus the actual critical speed will be considerably higher.

Figure 6 is an undamped lateral critical speed map for the final rotor. The single line shown on the map is the first lateral critical speed of the final rotor. The second and third lateral critical speeds were not considered as they were calculated to be considerably higher.

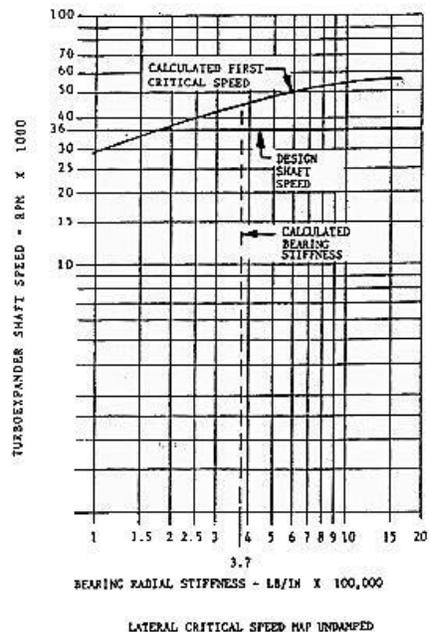


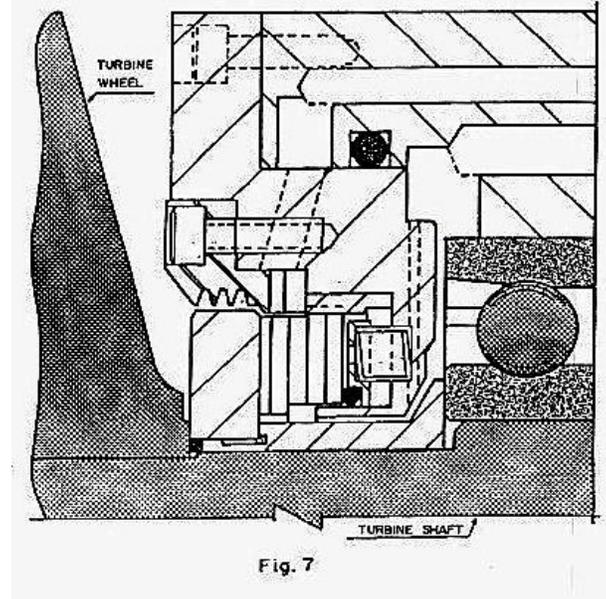
Fig. 6

## SEALS

The small 125 kW unit is provided with a mechanical face seal of carbon with a hardened mating ring. For the majority of cases this was considered adequate. Early experience with this seal indicated that under certain conditions of high discharge pressure, coupled with frequent starts and stops, excessive wear took place on the seal.

This problem was resolved by installing a modified type seal with a harder mating ring and a more direct face lubrication. With higher discharge pressure being considered for many of the 500 kW applications it was decided that rather than go to a full seal oil system, which would have been expensive, the John Crane type of dry gas seal would be used. The John Crane dry gas seal has proven itself in many high pressure gas compressor applications and although relatively expensive compared with a conventional seal, it completely eliminates the need for seal oil systems. Accordingly, John Crane was asked to design a seal for this particular machine and this seal has now been incorporated into the design. The only disadvantage of this seal lies in its slightly larger seal diameter. This larger seal diameter increased the area over which the discharge pressure acts thereby increasing the axial thrust. The axial thrust force on the rotor is, of course, a function of the differential pressure acting over the shaft seal diametral projected area. As the gear-end side of the turbine shaft is exposed to atmospheric pressure and the turbine-end of the shaft is subject to discharge gas pressure, the differential pressure is equal to the discharge gas pressure. As discussed earlier, this axial thrust force was accommodated by the proper selection of bearings.

Figure 7 is a diagram of the John Crane dry gas seal used. This seal's mating ring is positioned on the shaft by centering and clamping between the turbine wheel and the seal sleeve. The primary seal with springs and the other components is fixed in the seal housing. As an added feature, the seal has also been designed with a labyrinth located at the outside diameter of the mating ring and a purge connection in case particularly dirty gas or an extreme application is encountered.



## CASING DESIGN

In the design of the 500 kW machine the possibility of using a reverse flow arrangement was initially considered. This would have placed the cold discharge gas at the gearbox-end with the inlet directed axially into the turbine. The advantage of this reverse flow arrangement is accessibility to the nozzle block thus making rapid nozzle changes possible in the field. The disadvantages, however, are that the excessive cooling could have had an adverse affect on gearbox lubrication. Additionally, low discharge pressure applications would demand a larger exhaust area which would then require a longer shaft span between the bearings. After due considerations it was therefore decided to retain the existing flow path which has a side intake and an axial discharge. The nozzle block of the 500 kW machine is also readily change and only requires removal of the turbine wheel. The turbine casing is a massive, single-piece unit which has internal boring for lubrication passages and drains, for oil and buffer gas purge-and-vent connections for the seal. This unit was welded to a larger 24" diameter flange which forms the mounting structure for the gearbox. The casing body is braced to the flange by 4 gussets. (Fig. 8)

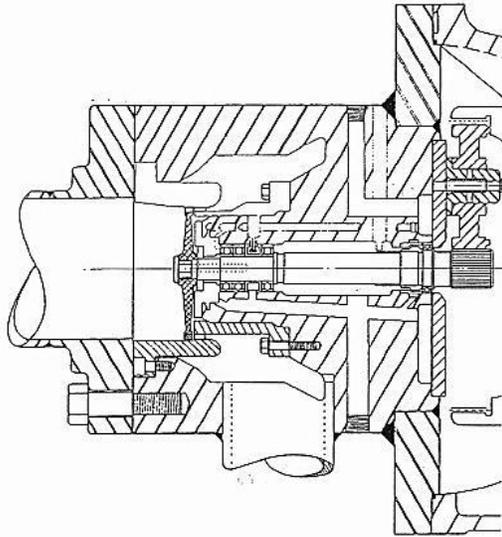


Fig. 8

Although the initial units were totally fabricated, it is planned to simplify manufacture by going to a cast assembly at an early date. The end cap which contains the discharge pipe is a standard lapped face flange to mate up with the end face of the turbine body, thus installation is greatly simplified. For access to the internals of the machine it is only necessary to provide a spool piece which can readily be removed giving access to the turbine wheel and nozzle block.

Removal of the turbine wheel is relatively simple. First, the stretch bolt is removed and then a wheel puller tool attached to the turbine wheel to withdraw the wheel from the shaft. The nozzle block can then be unbolted and removed. The turbine/gear unit is securely attached to the skid by a four point mounting with center guide arrangement. At the turbine-end the housing has two pads on the horizontal center line. This method maintains the turbine housing at its true center while allowing for thermal change of the casing. Two rigid feet are located on the bottom of the gear mounting flange. The turbine/gear unit along with the generator is bolted to a flat sole plate welded directly to the rigid steel skid frame which contains the oil reservoir. (Fig. 9)

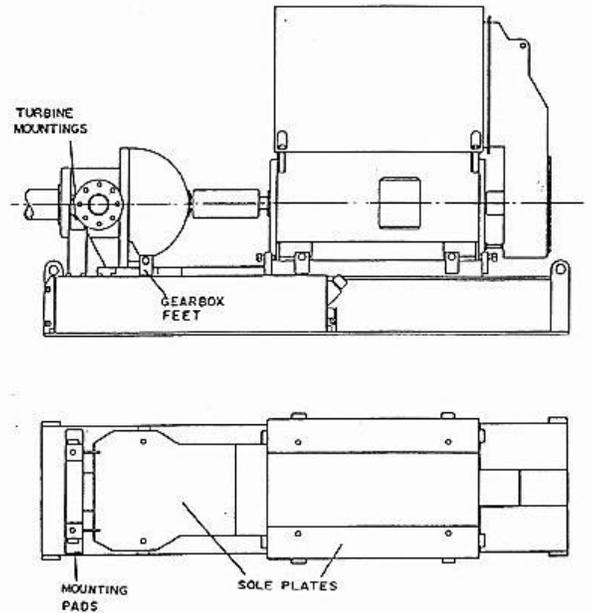


Fig. 9

The 500 kW machine can be fitted with either the current 5 1/4" turbine wheel or with the large 6" wheel when gas conditions dictate. To change wheels only requires the change of the nozzle block and the turbine shroud. All stress, loading and critical speed calculations have been carried out on the basis of a 6" diameter wheel and, therefore, the 5 1/4", which will be the standard for most applications, is operating well within its limitations.

## CONCLUSION

The development of the 500 kW turboexpander has been based upon experiences obtained with the smaller 125 kW unit. The major new component is the gearbox which, while retaining the concept of the integral epicyclic design, is completely new. The core of the unit, the turbine wheel and nozzles' aerodynamic shape remains the same as that of the smaller, 125 kW machine.

However, the large machine utilizes any increased diameter shaft, a multi-bearing arrangement for the additional thrust loads, and full admission nozzling for high gas flow. The use of dry gas seals should minimize problems with sealing up to high discharge pressures and eliminate the need for a separate seal oil system. The use of a common lubricant for gearbox and turbine greatly simplifies the lube oil system and maintenance requirements. This 500 kW REACT turboexpander couples experience with known technology and has combined them into an economically feasible package.

## ACKNOWLEDGEMENT

I would like to express my appreciation to the people at REACT ENERGY, LTD. for their help in preparing the illustrations for this paper.

In particular, I wish to thank Mr. Tony Cleveland for his considerable help.