GER-3705

GE Power Generation

GE Steam Turbine Design Philosophy and Technology Programs

R.S. Couchman
K.E. Robbins
P. Schofield
GE Company
Schenectady, NY

GE STEAM TURBINE DESIGN PHILOSOPHY AND TECHNOLOGY PROGRAMS

R.S. Couchman, K.E. Robbins and P. Schofield
GE Company
Schenectady, NY

INTRODUCTION

The need to provide State-Of-The-Art products for a diverse and ever-changing market is the challenge facing steam turbine-generator manufacturers in the '90s. While GE is uncertain what the industry will need as far as size, steam conditions, and technology mix, we have learned the value of strict adherence to a design philosophy based on long-term reliability and efficiency measurements. Technology, on the other hand, needs to be dynamic and responsive to support the needs of the power generation industry.

A look back reminds us that change is nothing new.

The market between 1960 and 1990 was everything but constant. The decade of the '60s was characterized by growth in unit size in both the traditional fossil-fuel market and the newly developing nuclear market (Figures 1 and 2). This growth in size was driven by utilities' strategy of reducing generation costs by taking advantage of economy-of-scale to satisfy a constantly increasing load demand which required power generation capability to double every 10 years.

During the 1970s nuclear units continued to grow somewhat in size (Figure 2), but the maximum-size fossil unit did not exceed the largest unit installed in the '60s. Rather than being a decade of continually increasing unit size, the '70s could be characterized as one devoted to reliability and availability improvement as it became evident that the large plants installed in the '60s did not meet expectations.

During the '70s load growth continued, but overall only at half the rate of the '60s. However,

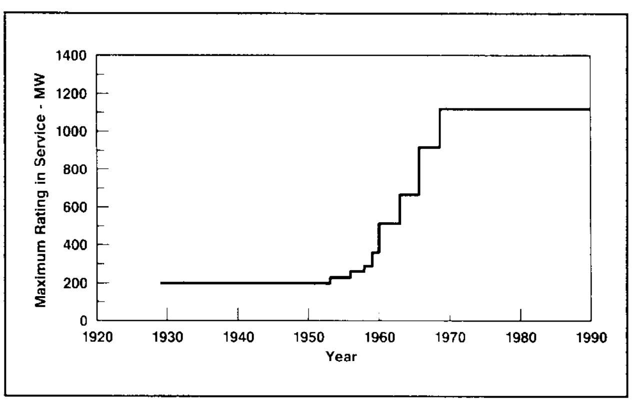


Figure 1. Growth in MW rating of GE fossil units vs. service year

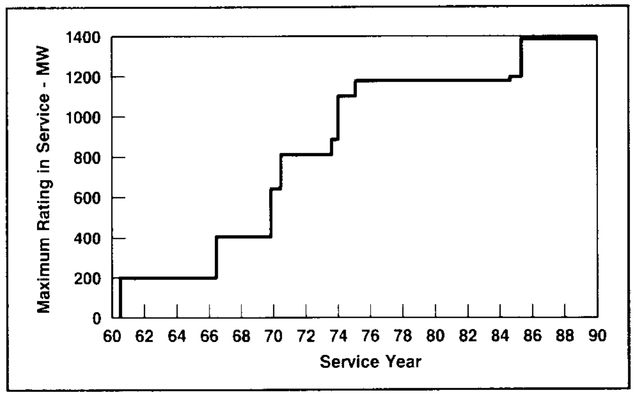


Figure 2. Growth in MW rating of GE nuclear units vs. service year

new generation equipment planned and ordered during the early '70s created a total market for steam turbines which nearly doubled that of the '60s as shown in Figure 3. The drive for improved reliability and availability resulted in the industry backing away from both the 1050°F steam temperatures which were common in the late '50s and early '60s and the 3500 psig supercritical throttle pressure which was utilized in the late '60s and early '70s. By 1975 steam conditions of 2400 psig/1000°F/1000°F for utility turbines had become the standard although the performance was poorer than the steam conditions which had been used previously.

The '80s were a period of uncertainty. During the late '70s and early '80s load growth virtually disappeared as the industry coped with the energy crisis; air, water and environmental concerns; nuclear problems; conservation efforts; and a difficult economic environment. Slow load growth compounded with excess generation capacity installed in the late '70s created a situation where the new equipment needs of the power generation industry were only 30% of that of the '70s (Figure 3), thus manufacturers restructured their businesses accordingly. This was also a period when there was much industry activity and discussion regarding what the power generation equipment of the future would look like. Substantial work was done for advanced steam cycles of 4500 psig, 1050°F and 1100°F, single and double reheat, based on pulverized coal boilers. There was also an accelerating interest in combined cycles as an operating experience base was rapidly being built on units installed in the '70s and '80s and as the promise of advanced combined cycles with very attractive performance levels became a reality. Much attention was also given to advanced combined-cycle features such as coal gasification and fluidized bed boilers.

GE's steam turbine technology programs have always strongly supported industrial and utility steam turbine customer needs. Reliability and performance improvement programs have continued throughout the three decades discussed above as well as several earlier ones. During the '60s emphasis was given to increasing unit size. During both the '60s and '70s heavy emphasis was also given to developing cost effective compact designs, primarily through the development of longer last-stage buckets. The longer buckets eliminated the need for cross compound designs for the largest ratings and permitted tandem compound units to be built in more compact arrangements with fewer low pressure sections. During the '70s many programs were aimed at the special areas unique to nuclear applications. The '80s received heavy emphasis on developing hardware suitable for retrofitting, particularly in the long-bucket area where substantial performance improvements were possible through the application of modern technology.

As we enter the '90s we again see a decade of change although it is anything but clear what the exact requirements of the steam turbine market will be. A somewhat increased level of shipments compared with the '80s is anticipated as shown in Figures 3 and 4.

During the '90s GE expects that about 20% of the MWs shipped will be for nuclear fueled units, although this represents less than 5% of the number of units shipped as shown in Figure 4. These units will be for international customers as it is not anticipated that there will be any new nuclear unit shipments required domestically prior to the year 2000, even though there are some indications that

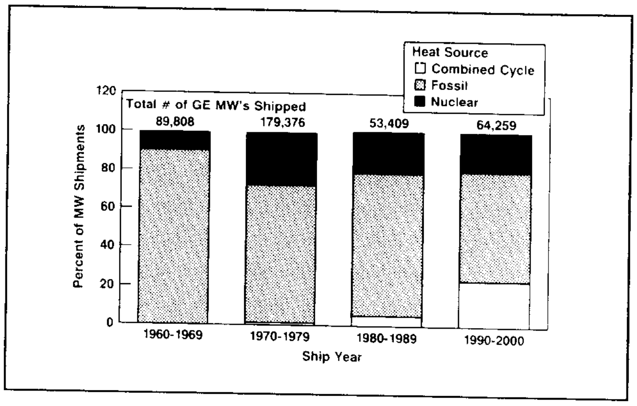


Figure 3. Steam turbine shipments – GE Power Generation MW count

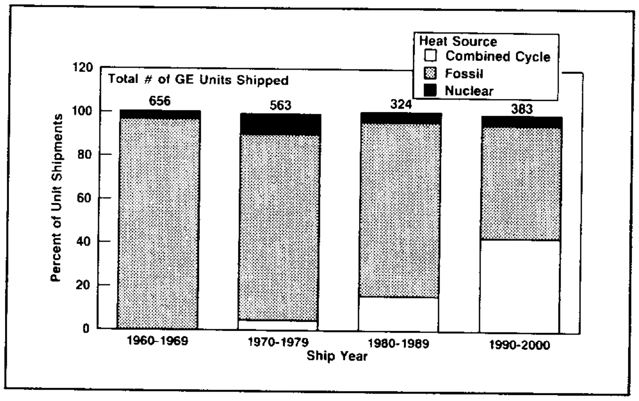


Figure 4. Steam turbine shipments – GE Power Generation unit count

there may be a revival of the market toward the end of the century.

A substantial increase in combined-cycle units is anticipated for the '90s. For the 10-year period 1980-1989 about 5% of the MW shipped, corresponding to 18% of the number of units, were for combined-cycle applications. During the period 1990-2000 it is anticipated that combined cycles will constitute an increasingly larger share and will account for 25% of the MW shipped and 45% of the number of units as shown in Figures 3 and 4. The remaining 55% of the MW and 50% of the units will be made up of industrial and utility units using traditional fossil fuels.

Not only was there a change in the robustness of the market in the '60s, '70s and '80s, but there were significant shifts, ebbs and flows, in the technology mix, the market segmentation and unit size. This market required larger, more efficient units (fossil and nuclear) for the utilities; combined-cycle units for the utilities and industrials; fast-track packaged units for the industrial and emerging NUG market; and parts and product improvements for upgrades and repowering. Design philosophy and technology played a very important role in serving this diversity.

As we move through the '90s one thing seems clear — there will be even greater diversity required in steam turbine designs as the need to serve traditional markets continues and as the need to respond to new applications, particularly in the combined-cycle area, develop.

This paper discusses the basic steam turbine design philosophy used by GE and summarizes some of the key technology programs which will support steam turbine designs for the '90s.

OVERALL DESIGN APPROACH

The design of reliable, efficient steam turbines requires the application of many diverse areas of technology. There are many competing design and material requirements that must be thoroughly evaluated, so that optimum trade-offs can be achieved. As new design requirements are imposed, and if in-service problems occur, the turbine manufacturer must possess the technical competence to reconcile them quickly and effectively. This means a competent, experienced technical staff is required, with adequate supporting laboratory facilities to permit keeping current with rapidly changing technology and to push back technological frontiers or barriers, when the need arises. The overriding objective in all steam turbine design activities is to produce turbine designs which minimize the life-cycle cost of ownership.

Life-Cycle Cost Objectives

The total cost of ownership of a steam turbine-generator can be considered to be made up of two components. The first of these is the first cost, or purchase price, for acquiring and installing the equipment. The second component is the subsequent operating costs for fuel, operation and maintenance, and unplanned outages. To understand the total cost of ownership of equipment like a turbine-generator, it is necessary to identify the life-cycle process related to the design, manufacture, installation, operation, and maintenance.

From customer requirements, the detailed design is completed. The unit is then built by the manufacturer for the agreed price. Installation of the unit may be by the supplier, customer, or third party. This is also part of the first cost. Subsequent costs include the value of fuel, replacement parts and equipment, as well as maintenance.

Of primary significance is that factors such as reliability, efficiency, availability, maintainability and operability, which are functions of the design and construction of the equipment, will affect the operating and maintenance costs of the user. Through use of the life-cycle cost concept, the influence of design alternatives on these indicators are evaluated and decisions reached, always with the objective of minimizing the overall life-cycle cost.

Component Design Considerations

GE's mechanical design philosophy is quite straightforward. It consists of comparing the type of duty to which turbine components are exposed in service with their ability to withstand these various types of duty. Usually this reduces to comparing in-service stresses (or strains) with the corresponding material strengths (or strain capabilities).

For some time a statistical or probabilistic approach for establishing design bases has been used, rather than simply comparing a single value of stress to a single value of strength and considering the difference between the two as a margin of safety. Long ago it was recognized that this latter approach is quite naive. Regardless of the degree of sophistication employed in calculating or measuring stresses (or strains), there remains a considerable amount of uncertainty about their actual magnitude in service under different operating conditions. Similarly, one cannot assume a single value of strength (or strain capability). Heat-to-heat variations and even variations within a single large component, such as a rotor forging or turbine shell, introduce unavoidable uncertainties in material capability. Thus, it has become necessary to treat the problem statistically, as illustrated in Figure 5. The "permissible" probability of failure, or failure rate, depends on many factors, including the consequences of failure. All material strengths are evaluated statistically. An example is shown in Figure 6, which illustrates the rupture strength distribution of a rotor material.

The most up-to-date available technology is employed in evaluating in-service duty and corre-

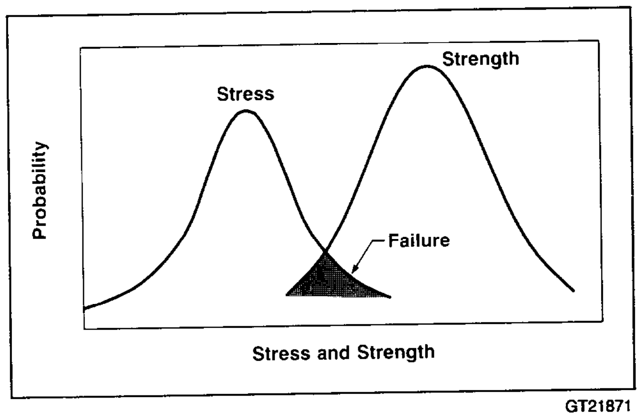


Figure 5. Probabilistic comparison between stress and strength

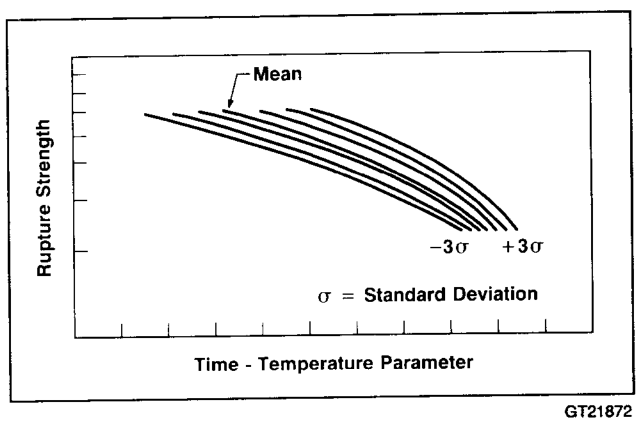


Figure 6. Illustrative statistical distribution of rotor material rupture strength

sponding material capabilities. For example, finite element methods are used for calculating stresses, vibratory frequencies, mode shapes, and temperatures. Fracture mechanics technology is used to evaluate the ability of turbine components to tolerate flaws of various types.

Stresses in a component can be generally categorized as (1) steady-state operating stress (2) dynamic stress and (3) thermal stress. The relative importance of each of these varies from component to component, and even from location to location within a component. The approach that is used in all cases is a comparison between stress and strength.

As an example,Table 1 illustrates the major types of stress to which a turbine rotor is subjected, together with the corresponding required material properties.

The rotor is subjected to steady-state centrifugal stresses within the body of the rotor, its wheels, and bucket dovetails. The corresponding material requirements are tensile strength and tensile ductility in the portion of the rotor that operates at temperatures below the creep-rupture temperature range (generally below about 700-800°F) and rupture strength and rupture ductility at temperatures within the creep-rupture range.

Because the temperature of the steam in contact with the surface of the rotor changes significantly during startups, load changes, and shutdowns, transient temperature gradients are introduced which, in turn, produce thermal stresses and strains. These require an adequate level of low-cycle fatigue strength.

Shaft-bending stresses caused by the gravity sag of the rotor require an acceptable level of high-cycle fatigue strength and similarly, the transient torsional stresses imposed due to electrical transients require an acceptable level of torsional fatigue strength.

In designing any turbine component all of the relevant factors have to be examined in great detail. Stress is determined by the geometric design of the component together with its operating condition and strength is determined by the material selected. Also, over the life of the component the material strength can be affected by corrosion, resulting in stress corrosion cracking or corrosion fatigue. A successful design is one which considers all these factors and results in good service experience.

TABLE 1
ILLUSTRATIVE SUMMARY FOR A STEAM TURBINE ROTOR

Stress	Property Requirement
Centrifugal	Tensile strength and ductility at lower temperatures Rupture strength and ductility at higher temperatures within the creep-rupture temperature range
Thermal	Low-cycle thermal fatigue strength
Shaft Bending	Tensile High-Cycle fatigue strength
Torsional	Torsional fatigue strength

Material Considerations

There are three basic properties that any material must possess at a satisfactory level in order to perform satisfactorily in any application. These are:

- Strength
- Ductility
- Toughness

The material must obviously possess sufficient strength to withstand the gross stresses imposed on it. There are various types of stresses that must be compared to the corresponding strengths.

Because all practical material applications contain discontinuities caused by such things as changes in section, holes, grooves, or fillets, the material must have sufficient ductility to tolerate the concentrated strains that result at these discontinuities, without cracking. In other words, the material must possess sufficient strain capability to tolerate the local strains imposed on it. These local strains will be in general much larger than gross or nominal strains.

Because no material is absolutely perfect, it must possess sufficient toughness to tolerate a reasonable level of imperfections without fracturing. Forged materials, regardless of the degree of care exercised in their production, contain material "defects" or inhomogeneities such as chemical segregation, small undetectable forging tears, etc. Castings also contain unavoidable defects such as shrink, porosity, and sand. Some of these defects are detectable by non-destructive testing and may be removed or repaired, but the material must have sufficient toughness to tolerate, as an absolute minimum, defects of sizes and types that may occur and may not be detectable. The material must also be tough enough to tolerate a reasonable amount of damage that may occur in service. Small nicks or cracks due to foreign material left in the steam path should not precipitate a gross failure. The material should be able to tolerate small fatigue or stress-corrosion cracks that may be produced in service without failing, so that there will be a reasonable chance of detecting and correcting them during planned inspections.

In addition to the three basic material requirements there are many other supplementary characteristics or properties required of a material. The following is only a partial list:

- Machinability
- Corrosion resistance
- Metallurgical stability
- Cost
- Fretting resistance
- Hardness
- Erosion resistance (moisture and foreign particle)
- Weldability
- Damping
- Density

The importance of these supplementary material properties relative to each other and to the three basic material properties depends on the specific application. The successful application of a material depends on a knowledge of the duty to which it will be exposed. A material must be evaluated in terms of the service requirements.

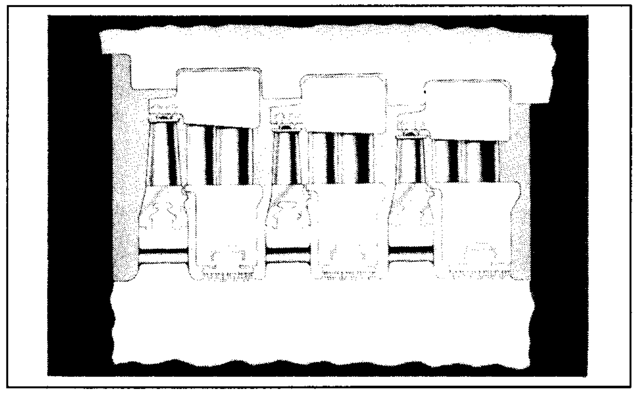


Figure 7. Typical impulse stages, wheel and diaphragm construction

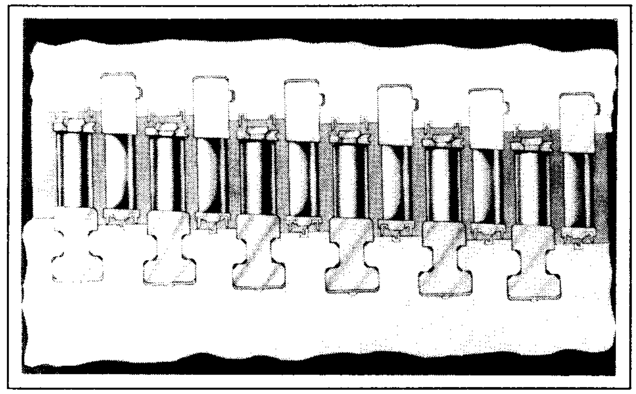


Figure 8. Typical reaction stages, drum rotor construction

Thermodynamic Design

All stages of GE steam turbines are designed to have most of the pressure drop occur in the nozzles and relatively little pressure drop in the buckets (or moving blades). This design is often referred to as "Impulse" in contrast to a "Reaction" design, which has approximately equal pressure drop in the nozzles and buckets.

Because there is little pressure drop across the buckets, it is possible to mount them on the periphery of a wheel without generating significant axial thrust. The nozzles are then mounted in a diaphragm with a packing located at the inner diameter, which controls leakage along the shaft between adjacent wheels. This "Wheel and Diaphragm" method of construction used by GE is shown in Figure 7.

A "Reaction" design has a significant pressure drop across the buckets, and there would be excessive thrust if the buckets were mounted on wheels. A "Drum" rotor, as shown in Figure 8, is, therefore, universally used by manufacturers adopting the reaction design concept. Even with a drum rotor, a reaction design is still liable to have high thrust so a "balance" piston is normally built into the rotor, unless the turbine section is double flowed.

Peak efficiency is obtained in an impulse stage with more steam energy per stage than in a reaction design, assuming the same diameter. It is normal, therefore, for an impulse turbine section to require either fewer stages on the same diameter or the same number of stages on a smaller diameter.

With less pressure drop across the bucket, the leakage across the bucket tip of an impulse design is obviously less than for a reaction design. In addition, spring-backed spill strip segments, which are designed to move away from the rotor to minimize the effects of a rub and can be very easily replaced if they become worn, are mounted in the diaphragm.

There is more pressure drop across the diaphragm of an impulse design than for a reaction design; however, the leakage diameter is considerably less. In addition, there is sufficient room in the inner web of the diaphragm to mount spring-backed packings with a generous amount of radial movement and a large number of labyrinth packing teeth. In total, the leakage at the root of an impulse stage is less than that of a reaction stage. The loss in efficiency corresponding to this leakage, furthermore, is much less with an impulse design because the leakage flow passes through a balance hole in the wheel and does not re-enter the steam path. In a reaction design, the leakage flow is forced to re-enter the steam path between the nozzles and buckets, which results in a significant, additional loss.

Leakage flows in a turbine stage can easily be calculated and the results show that, for HP stages typical of today's large steam turbines, tip leakage flows are 2 to 4 times greater for a reaction design than for an impulse design, both of which are designed for comparable duty. Similarly, shaft packing flows are 1.2 to 2.4 times greater for the reaction design without allowing for the re-entry loss discussed in the preceding paragraph. The absolute value of leakage losses, of course, becomes smaller as the volume flow of the stages increase for both reaction and impulse designs. On a relative basis, however, the leakage losses on a reaction stage will always be greater than those on an impulse stage designed for comparable application.

Overall Turbine Configuration Design

The term "overall configuration" means the general turbine arrangement in terms of its major features, such as the number of turbine casings, how the turbine sections (HP, IP and LP) are arranged within the casings, the arrangement of valves and the main steam piping between the valves and the turbine shells, the type of inlet admission/automatic extraction provisions the unit may have, and generally the size of the last-stage bucket and number of LP flows. The overall configuration for a specific unit is generally composed of standard features or arrangements that may be common to many different overall configurations. Discussed below are a few of the principal features utilized by GE in establishing overall configurations.

HP/IP Stages

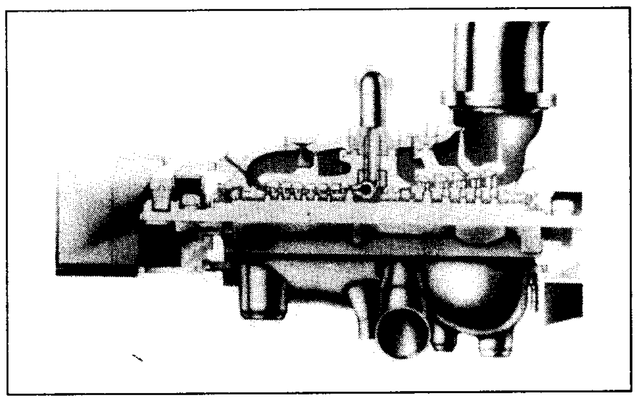
The impulse design concept advanced by GE results in relatively few high-pressure (HP) and intermediate-pressure (IP) stages, leading to very compact configurations. This allows units up to about 600 MW to be designed with all the HP and IP stages contained in a single casing in an "opposed-flow" arrangement.

In the "opposed-flow" design, a single-flow high-pressure section is combined with a single-flow reheat section within the same bearing span, as shown in Figure 9. On larger single-reheat units, a single-flow high-pressure section and a double-flow reheat section are separated, each having its own bearing span.

The first unit with the GE opposed-flow innovation went into service in 1950. Figure 10 shows GE operating experience with this design from 1950 through 1990.

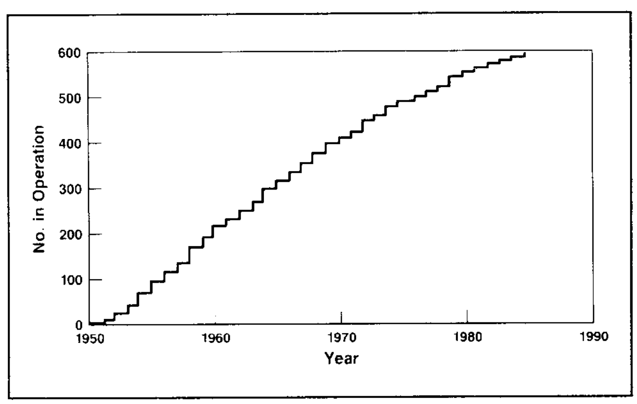
In the opposed-flow arrangement, high-pressure steam enters the center of the section and flows in one direction while steam reheated to about the same temperature also enters near the center and flows in the opposite direction. This arrangement combines the highest temperature steam to a single central location and results in a gradual controlled temperature gradient from the center towards the ends. This results in relatively cool steam adjacent to the end packing and bearings.

The opposed-flow design is more compact than a design with separate high-pressure and reheat sections, and tests have shown that this leads to a lower rate of temperature decay after shutdown and, hence, allows more rapid start up because of the smaller mismatch between steam temperature and metal temperature.



GT22032

Figure 9. Arrangement of opposed-flow, high-pressure and intermediate-pressure sections



GT21874

Figure 10. GE opposed-flow experience

Partial-Arc Control-Stage Design

In a turbine designed for partial-arc admission, the first-stage nozzles are divided into separate nozzle arcs (typically four to eight), and each arc is connected independently to its own control valve. For units operating with constant initial pressure, load is reduced by closing these valves in sequence. Typically the largest utility unit in any configuration would close one valve, with the remaining valves closing together to give a two-admission unit. Intermediate-size units would have two valves closing in sequence, with the remaining valves closing together to give a three-admission unit.

A single-admission (or full-throttling) machine controls load by throttling on all of the admission valves equally. The pressure ahead of the nozzles and after the first stage for a constant initial pressure is shown schematically on Figure 11 for both a single-admission and a typical four-admission machine. As load is decreased on the single-admission unit, an increasing amount of throttling takes place in the control valves. In the partial-admission unit on the other hand, considerably less pressure is lost due to throttling so more pressure is available to produce power in the first stage, with a corresponding improvement in overall heat rate.

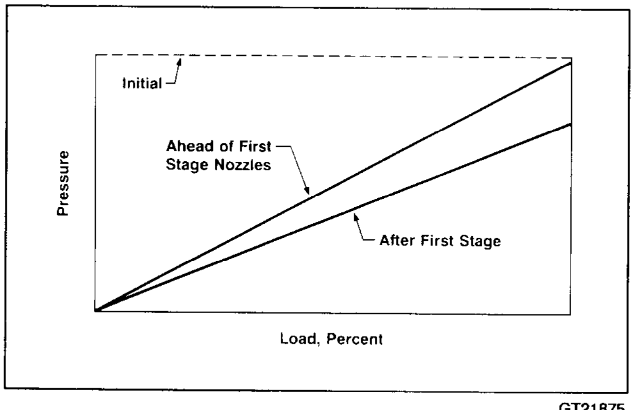


Figure 11a. Variation of pressure at a single-admission first stage

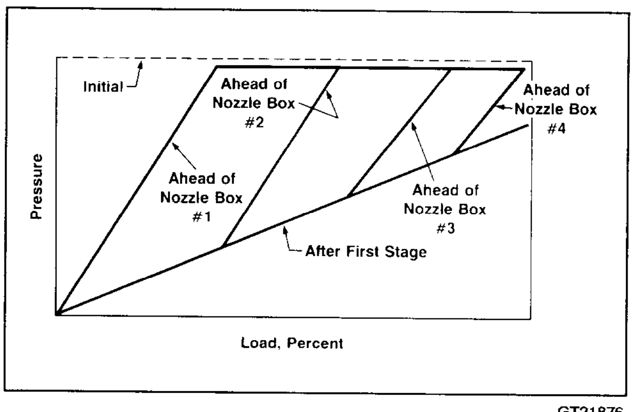


Figure 11b. Variation of pressure at a partial-admission first stage

The nozzle boxes for large modern GE fossil units are made from two 180° segments, each containing two separate inlet chambers. A nozzle box for a double-flow first stage consisting of two 180° segments is shown in Figure 12. Each 180° segment is made from a single 12% chromium forging. A ring of bridges is welded to the nozzle-box openings followed by a ring of nozzles that is welded to the bridge ring. This method of construction is very rigid, yet it allows free expansion and has proven to be extremely reliable in service. Other admission arrangements are discussed later in this paper.

It is possible to change unit output by varying the throttle pressure with some resulting benefits in load changing ability. With variable-pressure operation, the benefit of the partial-admission feature is sometimes questioned. However, if load is reduced by varying pressure in the valves wide-open condition, load increase can only be achieved by increasing boiler pressure, which is a relatively slow process, and the unit cannot participate in system frequency control. This problem is avoided by operating the turbine with the control valves somewhat less than fully open as throttle pressure is varied to control load. This provides some "throttle reserve" to permit more rapid response to changes in system frequency. With a partial-admission unit, it is attractive to close one or two valves and then vary pressure with one or two valves closed as shown on Figure 13. This method of providing throttle reserve also gives significantly improved performance.

It is GE practice to offer designs with partial-arc admission on all units except those of extremely large rating (900 MW and over) designed for base-load operation.

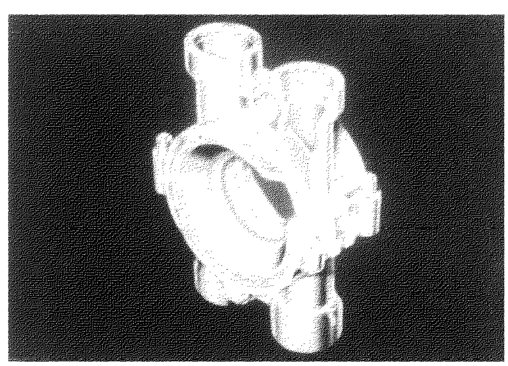
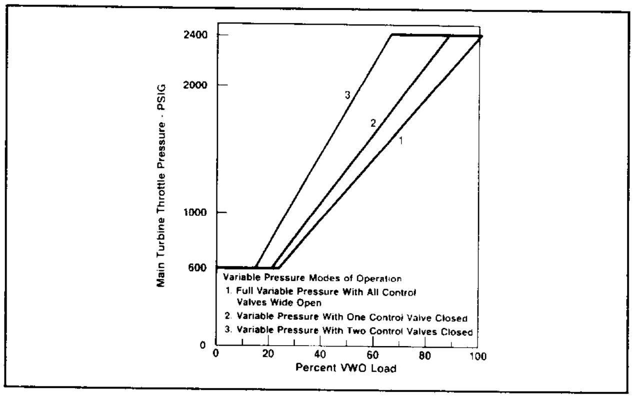


Figure 12. GE 180° nozzle box



GT21877
Figure 13. Variable pressure modes of operation

In addition to providing partial-arc control stages at the steam-admission point to the turbine for improved part-load performance, they are also provided at the first stage following an automatic steam-extraction point to improve the overall unit efficiency. See Figure 14.

Valve Arrangements

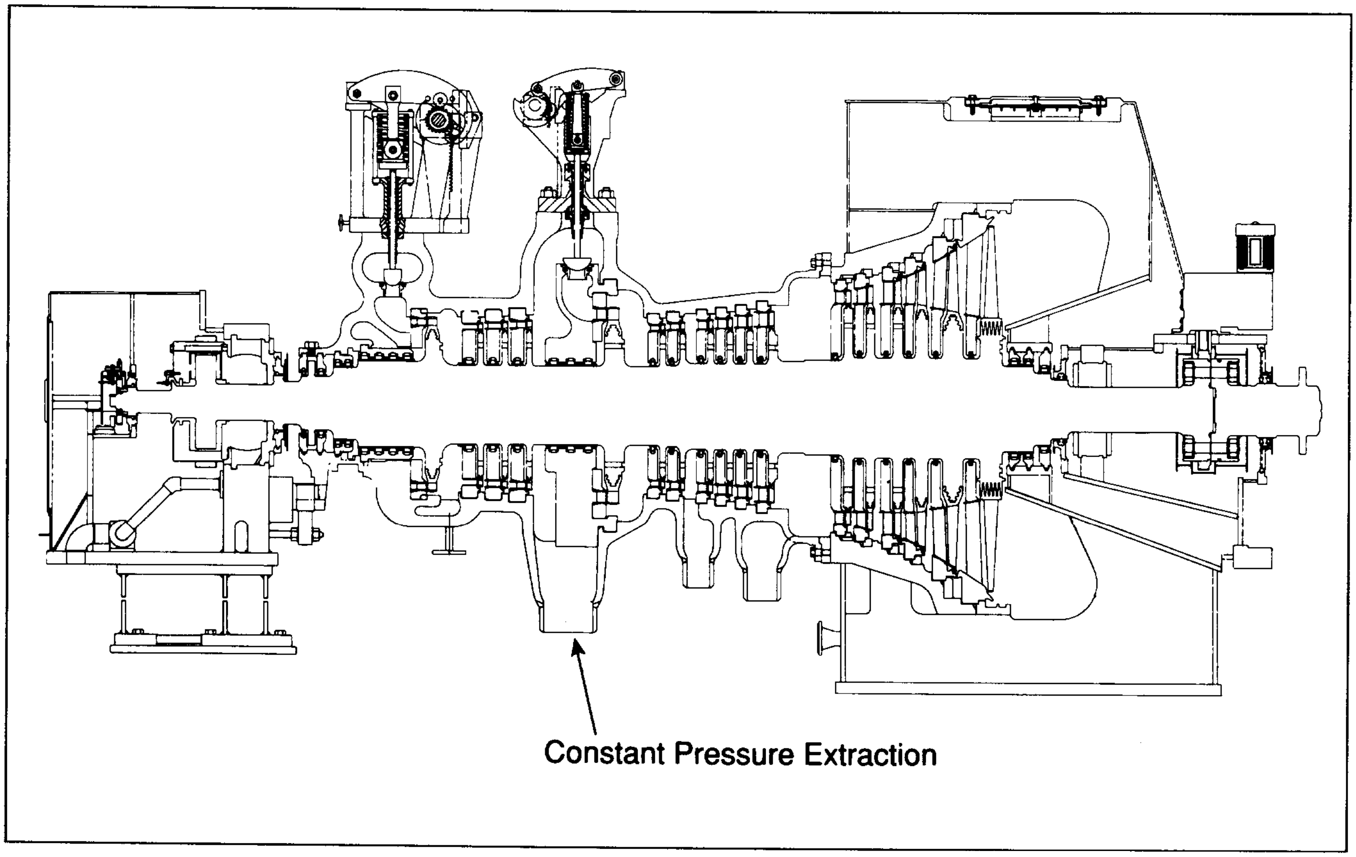
GE's Product line of valves and valve arrangements is guided by two criteria, the first being to provide as few valve casings as possible while providing high efficiency partial-arc control, and the second being that all valves should be designed with vertical stems and horizontal steam joints to permit easy disassembly using the station cranes. In addition, for lever operated valves, the valve steam parts can be completely disassembled without disturbing the servomotors. Such ease of disassembly improves maintainability and avoids the reliability risks inherent in disconnecting the electrical and high-pressure hydraulic connections.

On smaller units, both industrial and utility, the control-valve chest is designed integral with the shell. In addition, control valves have been eliminated from the lower half-shell wherever possible to eliminate the need to work overhead during maintenance and assembly. See Figure 15.

On larger units the control valves are either in a separate chest or in separate individual casings, which, in either case, are welded to the stop valves to make a stop-valve/control-valve assembly. This assembly is usually located below the floor, and is supported on solid-rod hangers. It is free to move to contribute to the flexibility needed between the superheater outlet and the turbine.

Typical stop-valve/control-valve assemblies are shown in Figures 16 and 17.

All modern reheat inlet valves are of the combined type, with both the reheat stop-valve and intercept-valve functions in the same casing (Figure 18). There are normally two combined-



GT22033
Figure 14. Steam turbine with partial-arc admission following controlled extraction

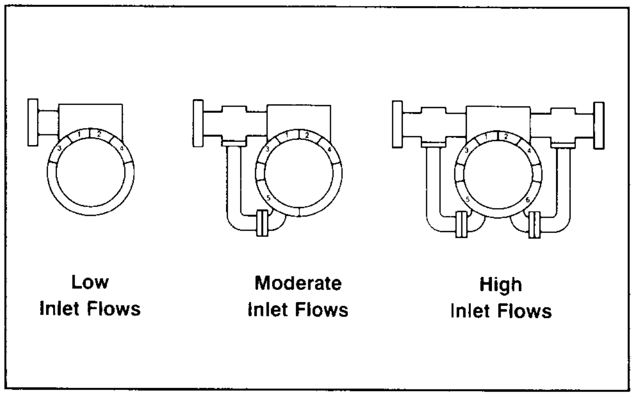


Figure 15. Industrial units – new inlet module concept

reheat valves which are close coupled to the turbine as shown in Figure 19.

GE overall configuration designs result in machines that are easy to install and maintain. Because of the features described above the main-turbine casings are small and compact. The valve and piping designs result in arrangements that are uncluttered with most of the valves under the turbine deck, making it very easy to do maintenance on both the valves and turbine main casings. All turbine casings are split at the horizontal joint making assembly and disassembly convenient, and do not require any unique disassembly stands or provisions. All valves are cast and designed with horizontal joints for easy maintenance.

Operational Characteristics

Changes in the operating conditions of a steam turbine, whether at start-up from cold stand-still or simply due to a change in process steam extraction or generator output, produce changes in the steam flow, pressure and temperature throughout the machine. These changes may result in changes to the steady-state stress in many components, but more importantly produce transient thermal effects which must be accounted for in the design of the turbine.

There are three primary areas in which temperature response of the turbine components affect turbine operation. They are:

1. Differential Expansion
2. Rotor Vibration
3. Thermal Stress

GE steam turbines have many features, some obvious and some not so obvious, which have been developed and perfected over the years addressing each of these areas.

Of the three above effects, differential expansion is perhaps the one most easily addressed.

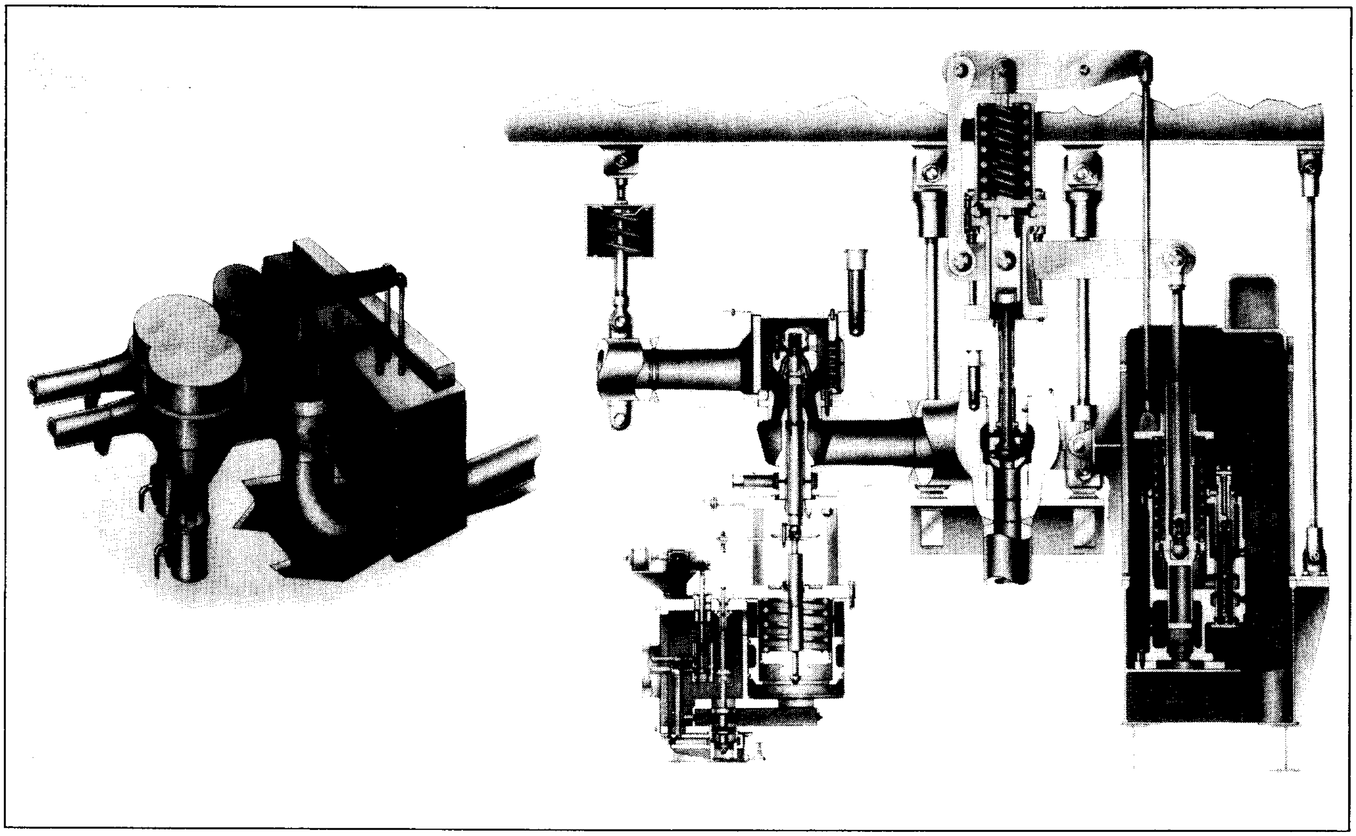


Figure 16. Typical stop-control valve assembly with off-shell valve chest

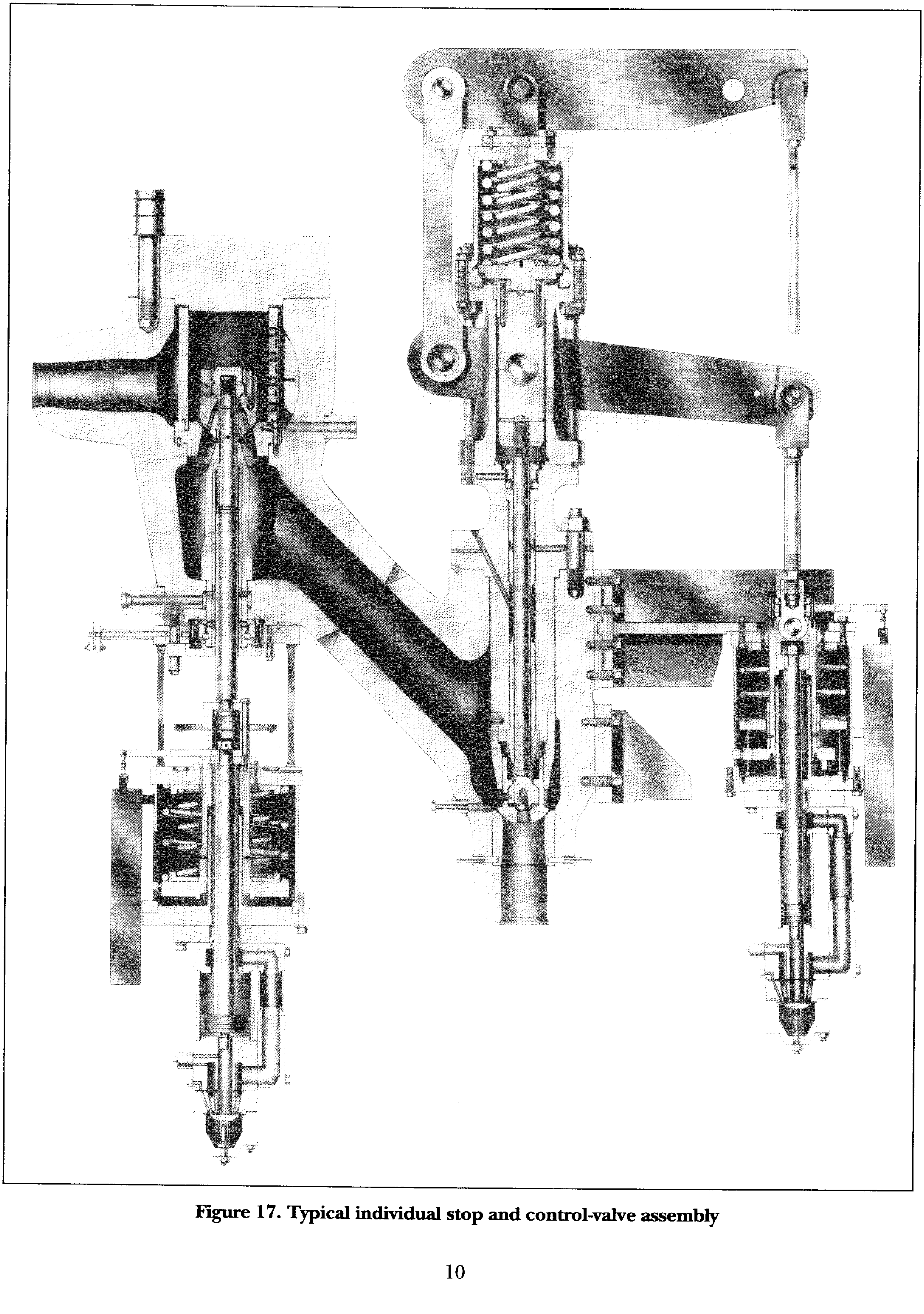


Figure 17. Typical individual stop and control-valve assembly

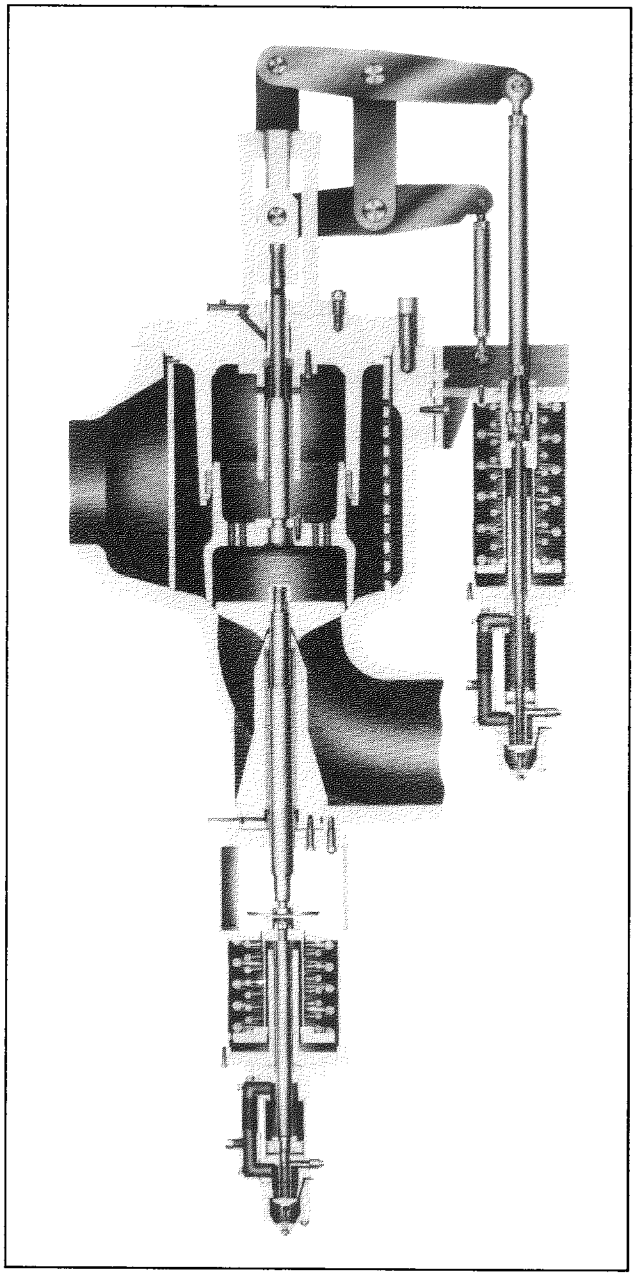


Figure 18. Combined reheat valve

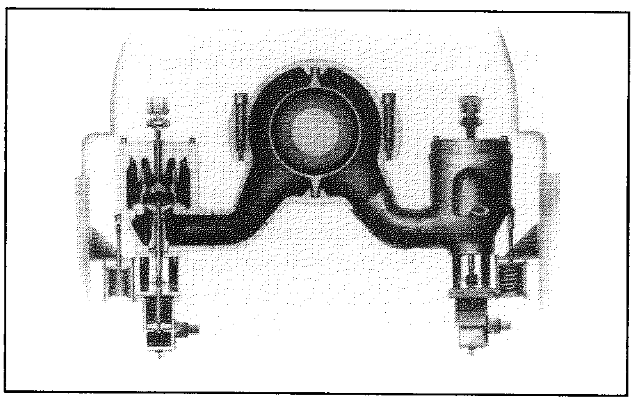


Figure 19. Reheat valves and turbine inlet arrangement

During the design of a steam turbine, limiting differential expansion is calculated relative to the thrust bearing position and clearance is then provided to assure that rotating and stationary components do not come into contact.

Rotor vibration can result during changes in the operating conditions of a steam turbine primarily from:
- internal rubbing caused by distortion of the stationary parts
- thermal bowing of the rotor
- rotor instability due to changes in the forces acting on the rotor
- alignment changes.

Thermal bowing is primarily a materials issue which has been effectively dealt with and is generally not a problem. Distortion of stationary parts, force acting on the rotor and alignment changes are primarily influenced by design features and great effort is put into these areas to assure they do not generally limit manueverability.

The third item above, thermal stress, is by far the most important and also the most complicated to deal with. There are many design features which influence thermal stress; however, for any design, no matter how good, there is always some thermal ramp which has the potential to inflict low-cycle fatigue damage on turbine components.

The bottom line is, then, that you must have excellent turbine design characteristics and reasonable care must be exercised in changing the loading conditions on a steam turbine if trouble-free long life is desired. Briefly discussed below are some of the principal design features used to enhance the maneuverability of GE steam turbines.

Impulse Wheel and Diaphragm Construction

The impulse wheel and diaphragm construction used on GE steam turbines is by far the most important single design concept to minimize thermal stress and therefore low-cycle fatigue damage. Specific characteristics resulting from this design philosophy are:
- Rotor body diameters are small, thereby significantly reducing thermal stress.
- Bucket dovetails are located on the wheel peripheries away from the rotor surface. This results in separating dovetail stresses from rotor thermal stresses.
- There are fewer stages and smaller stage diameters, permitting more compact designs with generally smaller components, in particular shells.

- Wheel shapes on periphery of the rotor act as fins and improve heat transfer to the interior of the rotor, thereby reducing thermal stress.
- The use of centerline-supported diaphragms allows movement of the diaphragms with respect to the shell without rubbing. If the stationary blading is held in a blade carrier, as is true in reaction designs, thermal distortion of the blade carrier results in internal rubbing.
- Ample room between wheels permits the use of large-wheel fillets to minimize thermal stress.

Thermally Flexible Inlets

GE uses various kinds of construction at the high-temperature inlets to its turbines depending principally on inlet pressure, temperature and flow, and whether the admission is to a full-arc or partial-arc stage. Inlets to a full-arc stage are subject to much less severe duty than those to a partial-arc stage since the steam temperature is uniform circumferentially. Typical types of construction used for inlets to a partial-arc stage are shown in Figures 20 (a), 20 (b) and 20 (c). The objective in all of these arrangements is to provide a structure that is strong enough to withstand the steady-state and dynamic stresses produced by the steam, while being flexible enough to assure low thermal stresses. All three arrangements shown are based on the concept of having a smaller pressure vessel within a larger one. The smaller pressure vessel contains the main-steam inlet pressure and temperature, while the larger vessel contains steam at a lower pressure and temperature. This concept allows the inner pressure vessel to be designed with thin walls entirely free of the large horizontal joint flanges of the outer vessel. This provides flexibility and the freedom of the inner chest to expand without inducing large thermal stresses.

Centerline Support

The temperature of various turbine components can change considerably as the turbine is started and loaded causing radial "differential expansion". If no special provisions are made, the rotor will move relative to the stationary components so that it is positioned eccentrically. This eccentricity would represent a significant loss in efficiency since extremely large sealing tooth clearance would have to be established and because of misalignment between nozzles and buckets. Additional risk of rubbing could also contribute to decreased reliability as well as a further loss in efficiency.

As a result, General Electric utilizes various concepts to ensure that all stationary and rotating

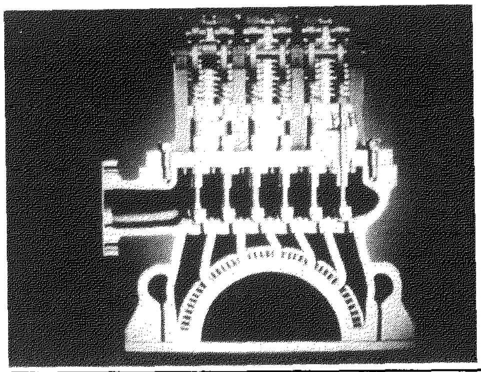


Figure 20a. Free-expanding chest for industrial and small utility units

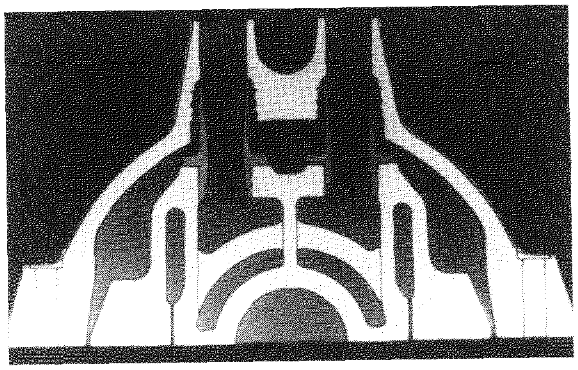


Figure 20b. Free-expanding nozzle chest for medium size utility units

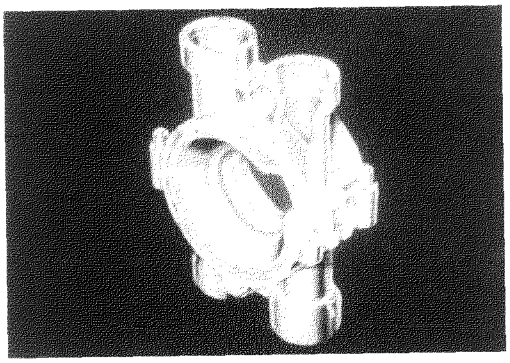
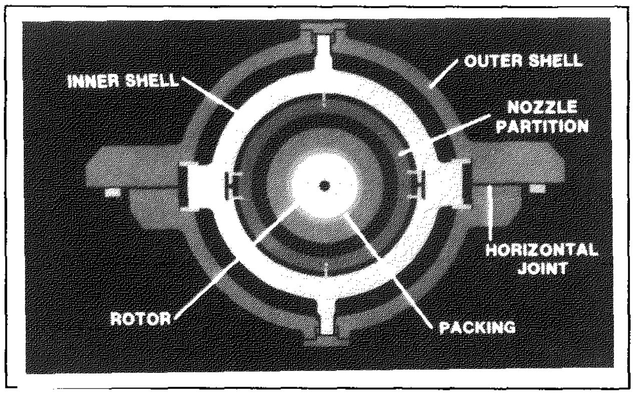


Figure 20c. GE 180° nozzle box for large utility units

components remain concentric as temperatures vary. Typical support details for diaphragms and inner shells are shown in Figure 21.

The weight of high-pressure and intermediate-pressure shells is carried on arms that extend from the flange at the horizontal joint. For high-temperature shells the shell arm is carried from the upper half-shell for true "centerline" support. For intermediate-temperature shells the shell arm is carried from the lower half-shell, providing approximate centerline support. This



GT22030

Figure 21. Method of support of stationary components for precision alignment

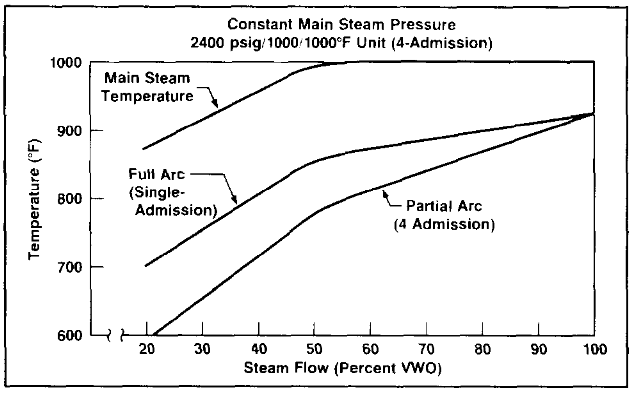
simplifies maintenance for cases where true centerline support is not required, since temporary blockage for the lower half-shell is not required when the upper half-shell is unbolted and lifted. Very-low-temperature components, for example exhaust hoods, need not be centerline supported because their low temperatures minimize the potential for differential expansion.

Admission-Mode Selection

The ability to control first-stage shell temperature is strongly influenced by the type of admission control employed. This effect is shown in Figure 22. For example, dropping load from 100% to 20% in the full-arc (FA) mode causes the first-stage shell temperature to drop by about 225°F, whereas if the same load change is accomplished using partial-arc (PA) admission, the drop in first-stage shell temperature is increased to about 325°F.

AMS is a control feature which allows the selection of either FA or PA operation of the turbine control valves throughout the load range. The transfer then proceeds automatically at a rate preselected by the operator.

In the FA mode, all control valves open together and the turbine can be started and loaded in



GT21879

Figure 22. First-stage shell steam temperature characteristics

this mode. The turbine can be brought to the valves-wide-open condition while still in FA mode. In the PA mode, the control valves open sequentially. PA operation is inhibited during start-up until the unit is synchronized to ensure uniform heating of the stationary parts. Once the unit is synchronized, the transfer from FA to PA and back to FA can be performed at any load. This function can be used to help control thermal stress that may develop in the HP rotor due to the effect of major load changes, or due to rapid main-steam temperature changes.

By expanding the range of control over the first-stage shell steam temperature during shutdown, as well as during start-up, it becomes easier to achieve the proper steam-to-turbine metal temperature matching over the full range of start up conditions encountered. In addition, by using the FA mode during the shutdown portion of a cycle, it is possible to reduce the amount of cooling the rotor will experience, which in turn will help to shorten the startup time.

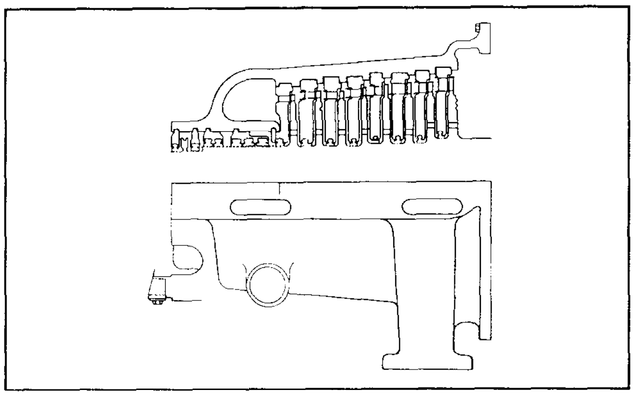
Equally important is the fact that with this dual capability the inherent high efficiency of the HP turbine design is not compromised. During part-load operation, PA mode can be selected for high efficiency.

Full Arc Heads for Combined-Cycle Applications

In combined cycles, the steam turbine normally operates under variable-pressure control. The steam turbine is therefore designed for full-arc admission with off-shell control valves. The full-arc admission allows completely uniform heating of the admission parts, thus facilitating rapid loading. Such an inlet arrangement is shown in Figure 23.

Thermal Stress Control

GE steam turbines are designed with many features aimed at minimizing thermal stresses



GT22029

Figure 23. Full-arc admission arrangement for combined-cycle steam turbines

resulting from a given temperature change. The thermal stresses in any particular component, however, are directly influenced by the steam temperature to which they are exposed regardless of design. Thus an important way of minimizing damage caused by thermal stress is to control the temperature exposure. This is commonly referred to as "thermal-stress control". The GE SPEEDTRONIC™ Mark V control system has been developed to include microcomputer algorithms to continuously calculate stresses in the high temperature rotors based on speed and metal temperature measurements in the turbine-inlet region(s). The control system automatically provides the functions required so that a startup, shutdown, or major load change is monitored and controlled by the thermal stresses, as well as other pertinent operational criteria. Automatic stress-control logic includes strategy that uses AMS and/or load holds to control starting and loading in a manner designed to minimize start-up times. Thermal stress control has significant benefits for applications involving cyclic duty, combined-cycle units and all large utility applications. Current thermal-stress control strategies are based on extensive experience dating from 1968, when the first computer control approach was utilized on a GE steam turbine.

Maintainability

The optimum time to consider turbine maintenance is in the design stage of the steam turbine. Many maintenance features are designed into GE steam turbines, some of which have been described earlier. Recent technological advances have permitted additional improvements in the maintainability of GE steam turbines to be made. Some of these improvements are described below.

Minimizing Solid-Particle Erosion (SPE) Damage

SPE damage in the control stage and the first reheat stage frequently causes severe performance deterioration, increases the risk of component failure and extends the length of maintenance outages. Through a four-pronged development effort that included inspection of eroded steam-path components, the analysis of particle trajectories, the development of erosion-resistant coatings and the implementation of design changes, GE has succeeded in developing turbine-stage components that are significantly more erosion-resistant than previous designs.

Figure 24 shows a modified control-stage partition that has been developed based on trajectory analysis confirmed by field experience. With the modified design the majority of the particles impact the nozzle partitions at a lower velocity well before the trailing edge. Fewer particles impact the trailing-edge region, and those that do, impact it at a reduced angle. The nozzle partition is further protected by an erosion-resistant diffusion coating of iron boride. The modified profile and the diffusion coating will extend the life of a control-stage nozzle partition by about three times.

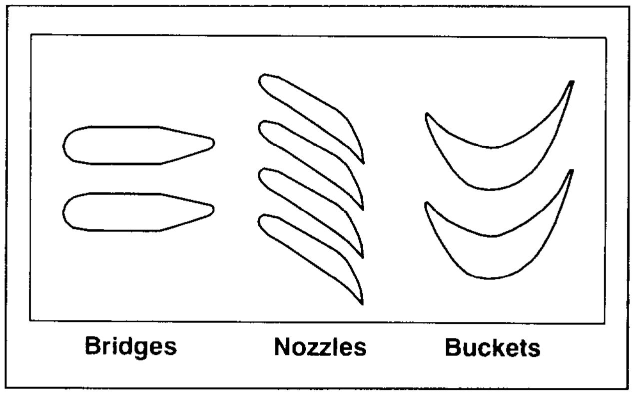


Figure 24. Modified first-stage partition for SPE

Trajectory analysis also demonstrated that first-reheat-stage nozzle-partition erosion results from rebounding of particles from the leading edge of the bucket toward the convex side of the nozzle partitions, where they impact the surface at a high velocity and shallow angle, a very erosive condition.

Theoretical calculations indicated that increasing the spacing between the nozzles and the buckets would greatly reduce the damage to the partition, and that additional erosion protection would be provided by applying plasma-spray chromium-carbide coating. The benefits of these design changes were later confirmed by field evaluation and are now part of GE standard design practices.

Erosion-resistant plasma-spray-chromium carbide coating can also be applied to buckets that are susceptible to SPE damage.

Full-Flow Oil Filters

Clean lubricating oil is essential to prevent bearing, journal and thrust-runner damage that can cause forced outages, or if not forced outage, extended maintenance outages to repair the damage. Dirt can enter the lube-oil system in several ways. It can be carried by air sucked into the drain side of the system, or it can enter from a dirty oil storage tank and inter-connecting piping. No matter where it enters, a full-flow oil-filtration system can combat such dirty-oil-related damage.

A full-flow oil-filtration system is especially beneficial after major maintenance outages have occurred. During maintenance operations, some amount of contaminant particles inadvertently get into the lube-oil system while the system is left open. The full-flow oil-filtration system safeguards the turbine since it removes the contaminants from the circulating oil as the lube-oil system is restarted. Oil flushes that are often only partially successful, as well as very time consuming, become unnecessary.

Borescope Access Ports

Occasionally it is desirable to visually inspect steam path components (turbine buckets and nozzle partitions) for damage or deposits. These conditions can significantly affect availability if they are not identified until the unit is opened for a regular scheduled outage. This is particularly true if new parts are needed but have not been ordered. One feature available to help in planning for maintenance outages is the borescope-access port. These ports permit the insertion of optical devices (i.e., borescopes) for the visual inspection of buckets and nozzle partitions without having to disassemble the major turbine components.

TECHNOLOGY PROGRAMS

Technology is the key to improving turbine designs of the future. GE has always strongly supported the power industry although the emphasis has changed over the years. This is not only true for the for the steam turbine business, but for other organizations within GE as well, bringing to bear the full strength of GE. Components like Aircraft Engine, Nuclear Energy and Corporate Research and Development have all played a role, although in many instances this "behind-the-scene" type of activity may not be apparent to those outside of GE.

Now, perhaps more than at any time in the past, other organizations within GE are contributing to steam turbine technology. Advanced fluid dynamics computation programs developed for aircraft gas turbines are equally applicable to steam turbines; high temperature materials developed for gas turbines may be used in steam turbines; advanced computer algorithms developed by the GE Corporate Research and Development component can be used to design steam turbines.

Described below are a few of the key technology programs in the steam turbine area currently receiving attention. It should be realized that many of these are supported by the entire GE Company, not just the Steam Turbine organization.

Aerodynamic Developments

During the 1980s GE extensively utilized the resources of organizations throughout the GE company to further steam turbine technology. The expertise of the Aircraft Engine Department, the Gas Turbine Department and the Corporate Research and Development Center all supported the Steam Turbine Department in it's tradition of continuous steam turbine development. Some of the important aerodynamic developments that have taken place recently include:

Controlled-Vortex Stages

Traditionally, steam turbine stages have been designed free-vortex, that is designed to have radial equilibrium within the turbine stage. The authors' company has for many years designed non-free-vortex stages in LP turbine sections to account for the significant divergence of the stages. Applying these techniques to design a turbine stage to bias the flow toward the more efficient mid-section of the stage results in improved performance. These results occur because of the reduced turning angle at the root due to higher root reaction, and to reduced tip leakage because of the lower tip reaction. In 1984 an intermediate pressure (IP) section of a 650-MW unit was designed with controlled-vortex staging. Figure 25 shows the streamlines associated with the controlled-vortex design of that IP section.

A single-stage development test conducted in GE's Aerodynamic Development Laboratory of a controlled-vortex stage designed with the same methods as was used for the IP section mentioned above demonstrated an encouraging 1.5% efficiency improvement compared to an equivalent free-vortex designed stage. GE now applies controlled-vortex staging on the IP sections of all large utility units.

Contoured Sidewalls

Analytical investigations based on three-dimensional formulation of the inviscid Euler and Navier-Stokes equations, described in References 11 & 12, have been successfully used to predict

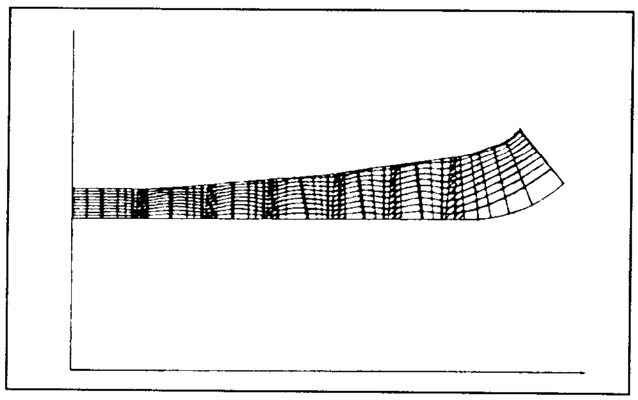


Figure 25. Streamline plot for controlled-vortex stage design

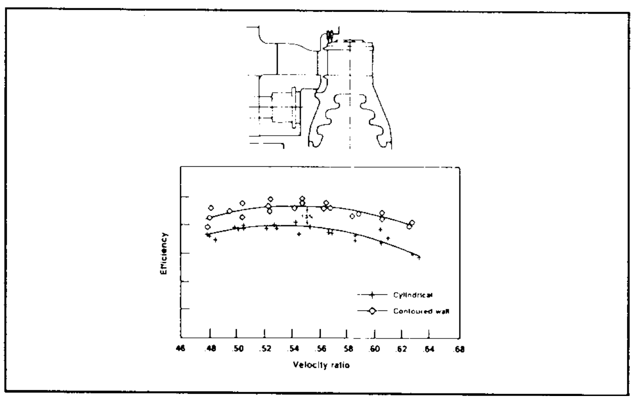
flow fields in steam turbine blade rows. Using these computational methods it has been demonstrated that the correct contouring of the outer sidewall of a nozzle passage can result in improved performance by reducing secondary losses. This is accomplished by reducing the cross-channel pressure gradients which both weaken the strength of the secondary flows and suppress the secondary vortex on the inner sidewall.

The first running-stage test of a steam turbine using contoured sidewalls was performed in 1983. This test was performed on a small low-energy two-stage commercial drive turbine, which included a converging-endwall contour on the first stage and a diverging-converging contour on the second stage. The contours were designed using guidelines presented in Reference 13. An overall efficiency gain of 0.7% was realized.

In 1985, after additional wind tunnel cascade tests were conducted to verify an optimal outer sidewall contour shape developed analytically, a running stage was tested in the authors' company's development facilities. Figure 26 shows the shape of the control-stage outer sidewall contour and the results of tests comparing a cylindrical nozzle with the contoured nozzle. Contoured-sidewall nozzles are being applied to HP stages of mechanical-drive turbines on a regular basis.

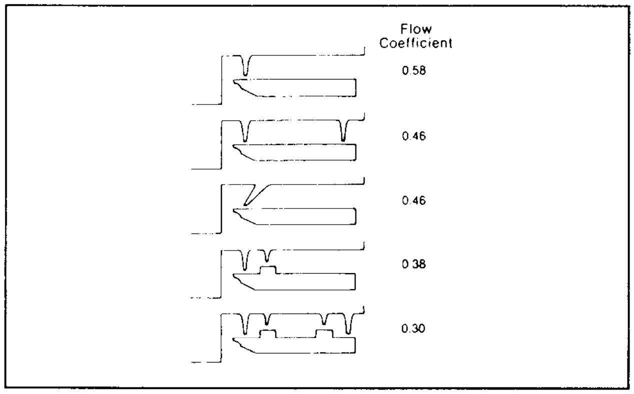
Improved Bucket Tip Leakage Control

Traditional bucket tip-leakage controls have either a single radial tip spill strip or two spill strips, on either side of the bucket cover tenon. To investigate improved tip-leakage controls, the authors' company conducted a series of tests with different sealing configurations. The results of these tests are shown on Figure 27. They show a significant benefit when a stepped or high-low spill strip is used to minimize bucket tip leakage. Stepped-tooth radial-tip spill strips are now used on the HP stages of all large utility units that have adequate axial space.



GT22027

Figure 26. Contoured-wall stage design



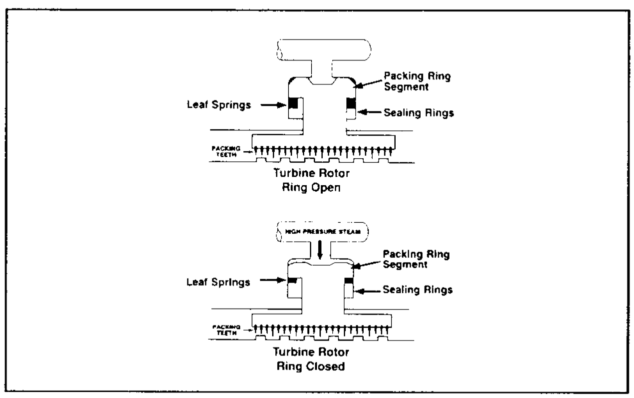
GT22028

Figure 27. Results of tip-sealing tests

Positive-Pressure Variable-Clearance Packing

Labyrinth seal packings close to the mid-span of a high-temperature steam turbine rotor are susceptible to rubbing. Operation below the first critical, acceleration through criticals and boiler temperature variations all occur at start up, making the packing most vulnerable during this period. Excess clearance caused by rubbing during the start up of the unit results in increased fuel costs and a reduction in unit capacity. In addition, vibration problems associated with packing rubs can prevent the turbine from accelerating through it's critical speeds, prolonging the start up of the unit. Positive-pressure variable-clearance packing provides a large clearance during start up and reduced clearance after the unit has synchronized. This arrangement minimizes rubs associated with turbine startups while providing optimum sealing when the unit is loaded. A cross section of a positive-pressure packing ring is shown in Figure 28.

Positive-pressure variable-clearance packing utilizes a combination of the pressure drop across the packing and an additional pressure force, when required, to close the packing rings after synchronization. Also, an external control of the packing rings in a mid-span packing of an



GT22026

Figure 28. Positive-pressure packing cross section

opposed-flow unit can be provided. This allows the unit to be pre-warmed by pressurization on turning gear after a prolonged outage without the unit rolling off turning gear. A more detailed description of the positive-pressure variable-clearance packing is given in Reference 14.

Long-Bucket Advancements

With the slow growth rate in the 1980s many utilities were upgrading and extending the life of existing turbines. In this period, the long-bucket development programs were therefore focused on the modernization of older designs to improve efficiency and reliability. In the 1990s, turbine size is not expected to increase over the largest of those already in operation and some indications show that new unit sizes may be primarily in two ranges, 300-500 MW and 800-1000 MW. Development programs currently include new longer buckets to permit more cost-effective configurations, as well as modernization of older designs.

Long-Bucket Modernization Programs

The long-bucket modernization programs have included the 3600-rpm 20-inch, 23-inch and 26-inch last-stage buckets, the 1800-rpm 43-inch last-stage bucket and the 3000-rpm 26-inch and 33.5-inch designs, References 15, 16. The 60- and 50-Hz last-stage buckets were redesigned to include the latest aerodynamic and structural-design concepts to improve efficiency and reliability. The redesigned buckets can be retrofit into most existing units.

The predominant features were based on the modern 30-inch and 33.5-inch last-stage buckets and include:

- Transonic convergent-divergent supersonic flow passage design with high solidity, Figure 29.
- Modern bucket-vane aerodynamic design optimized for centrifugal untwist.
- Modern radial distribution of mass flow to optimize efficiency, Figure 30.
- New nozzle-vane aerodynamic and steam-path design.
- New tip-leakage control designs which allow moisture removal, Figures 31 and 32.
- Vane designs and mechanical connections to prevent vibration at off-design operating conditions.
- Continuous-coupling designs for structural damping and vibration control, Figure 33.
- Self-shielded or flame-hardened erosion protection.

All of these bucket redesigns will have been tested and shipped by the end of 1992 and others are already accumulating service experience. Over 150 rows of the 23-inch and 26-inch 60-Hz and 33.5-inch 50-Hz redesigns have been shipped with over 180 row-years of experience.

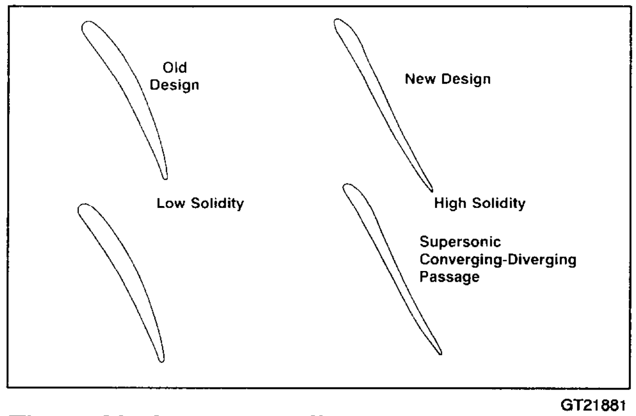


Figure 29. Convergent-divergent supersonic tip

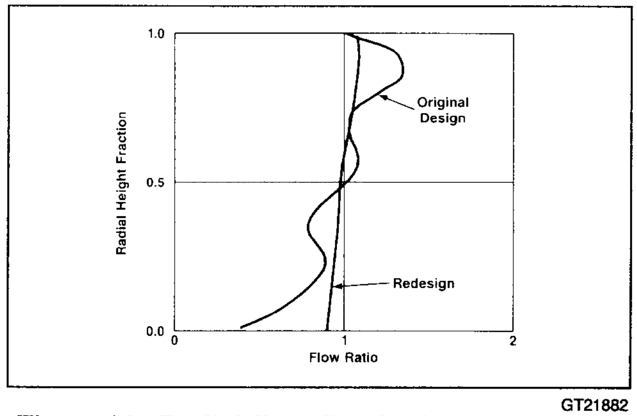


Figure 30. Radial-flow distribution comparison

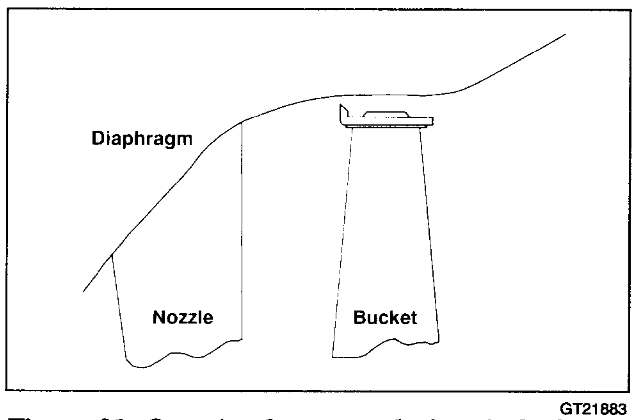


Figure 31. Over/under cover design tip leakage control of 20- and 23-inch redesigns

New Longer-Bucket Design Programs

New longer-bucket development programs have been initiated to support more cost-effective configurations with the required annulus area on large new units. The 50-Hz long-bucket programs have included new 41.9-inch (3000 rpm) and 52-

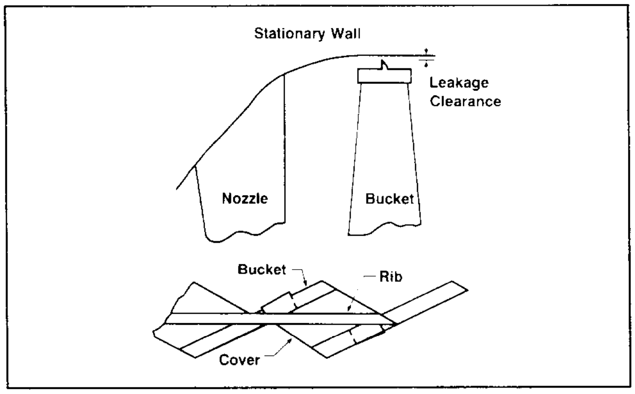


Figure 32. Side entry cover design tip leakage control of 26- and 33.5-inch redesigns

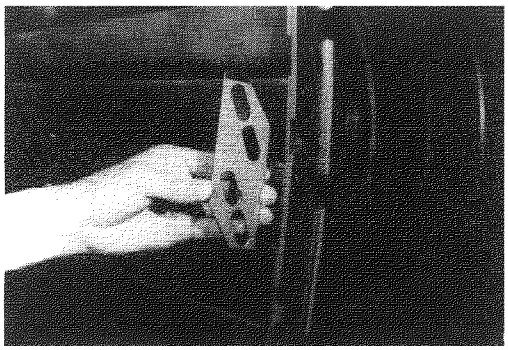


Figure 33a. Over/under design of 20- and 23-inch redesigns

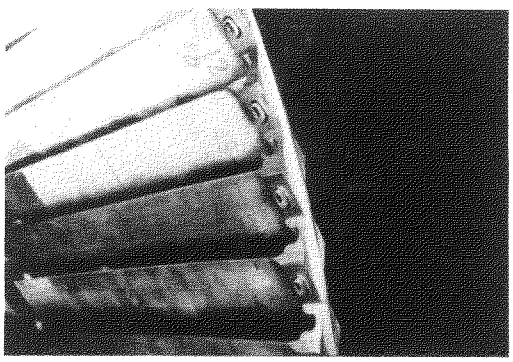


Figure 33b. Side entry cover design of 26- and 33.5-inch redesigns

inch (1500 rpm) designs. For the 60-Hz market, the new long-bucket development programs include the 3600-rpm 40-inch titanium last-stage bucket, begun in the late 1970s, Reference 17. Each of these designs are complete and have been offered for new large units.

The design features of the new 40-inch titanium and 41.9-inch were based on the modern 30-inch and 33.5-inch last-stage buckets and include the features mentioned above.

The new 52-inch 50-Hz design was based on the 43-inch 60-Hz redesign and utilizes similar design features.

Structured Industrial and Combined-Cycle Turbines

A reduction in construction time and cost is always an important goal for industrial and combined-cycle turbines. Supporting these objectives is particularly challenging to the turbine manufacturer, since turbines for these applications are already shipping in cycles as short as 12 months. Fortuitously, these same families of turbines are the ones that can most readily be predesigned and packaged. Recent work has been completed in the areas of modularized design and packaged base-mounted turbines.

Modularized Turbine Design

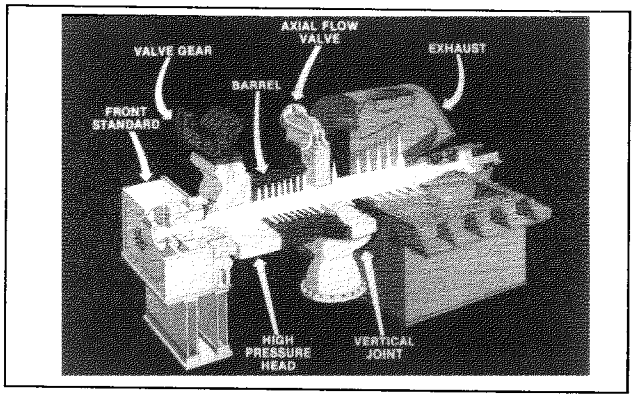
The use of predesigned components in the manufacture of steam turbines is generally advantageous to all functions. In engineering, for example, the design of a standard module can be done on a longer schedule permitting the detailed optimization of features, resolution of issues, and the completion of any required laboratory tests prior to committing the design for production. In manufacturing, standard components result in making parts and assemblies which everyone is familiar with and has experience with. There are similar advantages in all other functions, including installation and operation. These advantages result in a significant reduction in life-cycle costs.

Modularized components have been developed for all straight-condensing and non-condensing units, single-and double-automatic-extraction condensing units, and single-and double-automatic non-condensing units. In addition, small reheat units have been modularized. Condensing units include single- and double-flow low-pressure configurations.

The build-up of a typical modularized single-automatic-extraction single-flow condensing unit is shown in Figure 34.

Base-Mounted Turbines

With the standard modules described above it is entirely feasible to ship some of the smaller-size units completely assembled and mounted on a base or skid. With the exception of the low-pressure turbine modules, all the other modules have been designed to be within normal shipping limits for rail transportation. GE single-flow low-pressure turbines with 20-inch or shorter last-stage buckets with a down exhaust hood, and 30-inch or shorter last-stage buckets with an axial-



GT22025

Figure 34. Modularized single-automatic-extraction condensing steam turbine

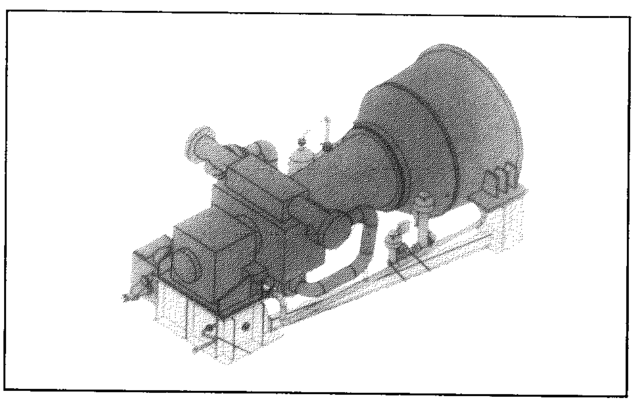
flow exhaust hood are all within rail shipping limits. A high percentage of industrial and combined-cycle turbines are, therefore, within shipping limits and thus can be base mounted.

A typical base-mounted unit is shown in Figure 35. The entire steam turbine is fully assembled. In addition, a large amount of the secondary piping and electrical wiring is completed before shipment from the factory. The turbine is completely aligned in the factory on the base and this alignment is documented and duplicated at installation in the field. The gland-seal system and steam-seal regulator valves are mounted on the base and piped to the turbine. The connections to the base are minimized with the goal of reducing the installation cycle to an absolute minimum. The bearings and lube-oil piping are completely flushed at the factory, thus drastically reducing the time needed for field flushing.

New Control Systems

SPEEDTRONIC™ Mark V System

Regardless of the application or size of a steam turbine, a complete family of digital control systems has been developed to meet the plant's control needs - from electronic governors offering simple speed- and load-control to triply-redundant systems which offer increased reliability for critical applications compared with any previous system design offered. The cornerstone of GE steam turbine controls is the SPEEDTRONIC Mark V digital control system, now being offered as standard equipment on all new steam turbine units.



GT22024

Figure 35. Base-mounted turbine design with pre-piped auxiliaries

Available as the SPEEDTRONIC Mark V TMR (Triple Modular Redundant) and as Mark V Simplex, a non-redundant system, the new SPEEDTRONIC Mark V offers a new standard for performance and reliability in steam turbine controls.

GE's SPEEDTRONIC Mark V control system combines advanced digital microprocessors with proven control software which results in several operating benefits to GE steam turbine owners. These include:

- Improved Operation. A wider range of control, faster response, greater speed-setting accuracy and reduced electronics drift add up to a more responsive, more reliable turbine control.
- Greater Efficiency. Automated sequencing improves the efficiency of start up, shutdown and synchronization. It also reduces wear and tear on the rotor and other vital parts during mode changes.
- Self-diagnostics. Mark V's self-diagnostics capability warns operators of problems before they affect availability. If required, repairs to the TMR version can be made on line.
- Compactness and Reduced, Simplified Maintenance. The system's streamlined design uses fewer electronic devices and permits easy removal/replacement. Cabinet layouts for a typical Simplex system and a typical TMR system are shown in Figure 36, demonstrating the modularized concept and the ease of installation and maintenance provided by the cabinet design. Figure 37 shows an assembled TMR cabinet.
- Flexibility, Adaptability. More control logic in software, less in the hardware, so functions can be easily added or modified and tested at the turbine site, with no need to modify hard wire-circuitry.

Retrofit Considerations

Previous generation control systems used mechanical-hydraulic or, later, analog electronics technology. While these systems provided many years of dependable service, they often cannot meet today's need for greater operating flexibility and increased performance.

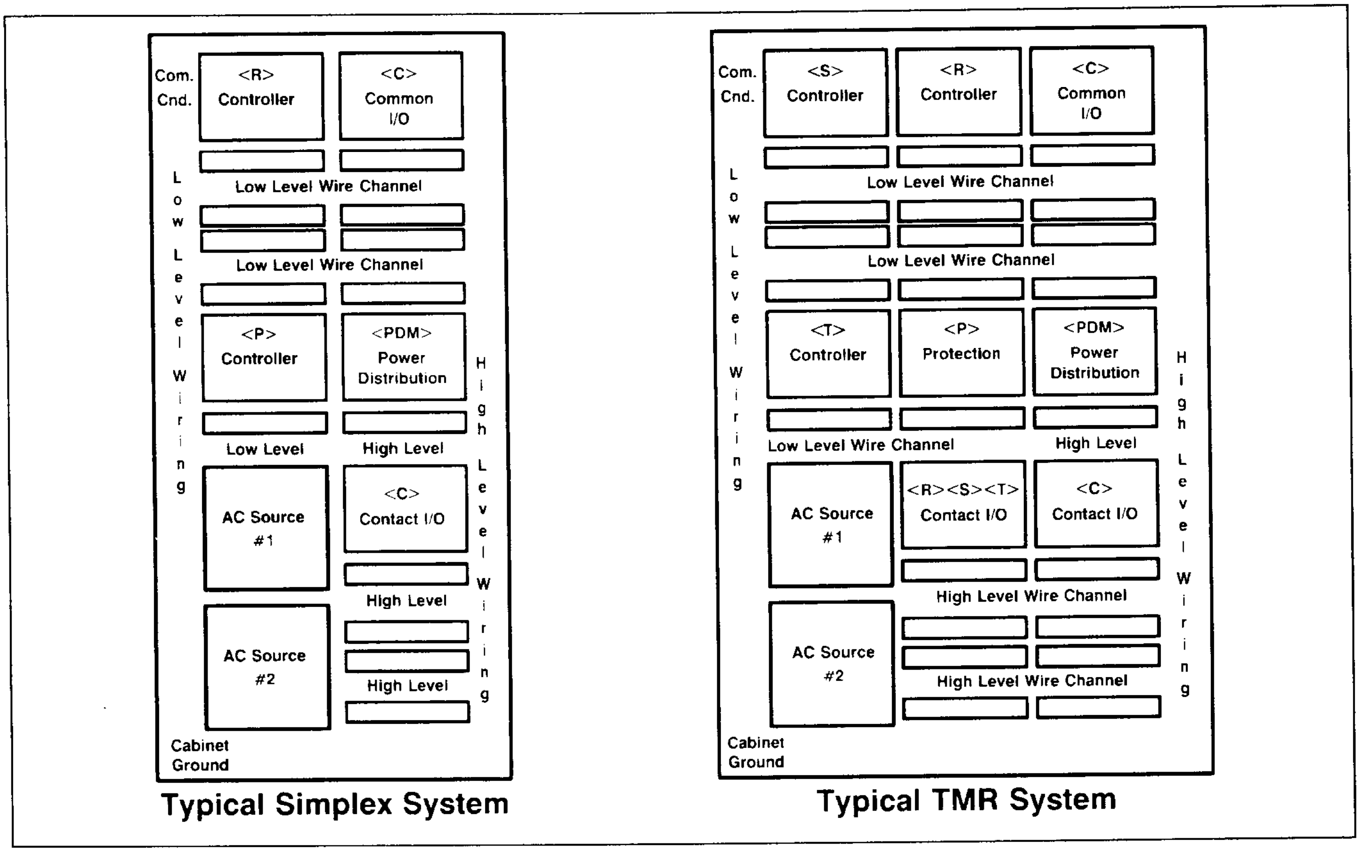


Figure 36. Control cabinet layout showing compactness and standardization

GE's SPEEDTRONIC Mark V Control System can be retrofitted to existing turbines to take advantage of recent developments in digital electronic controls technology. The same advanced microprocessor based control systems that are standard on new GE turbines are utilized for retrofit applications for both industrial and utility units.

Single-Shaft Stag™ Designs for the '90s

Combined-cycle power plants have been built with the gas turbine, steam turbine, and generator connected end-to-end to form a machine having a single shaft. To date, these plants have utilized a non-reheat steam cycle and a single-casing steam turbine connected to the collector end of the generator through a flexible shaft coupling as shown schematically in Figure 38. Both the gas and steam turbines are of essentially conventional design, requiring that the generator be in the middle of the machine because neither turbine is capable of transmitting the torque produced by the other through its rotor to the generator. The steam turbine must be physically moved out of the way to remove the

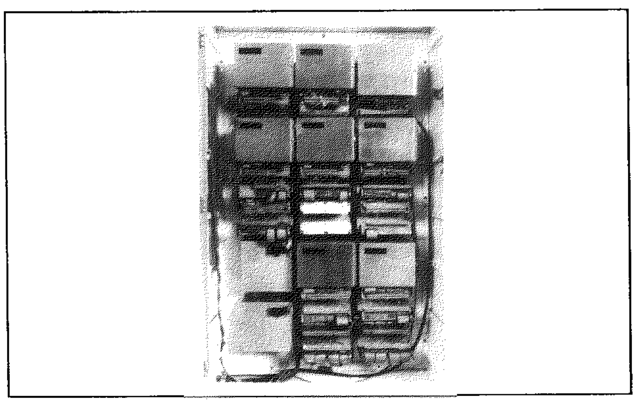
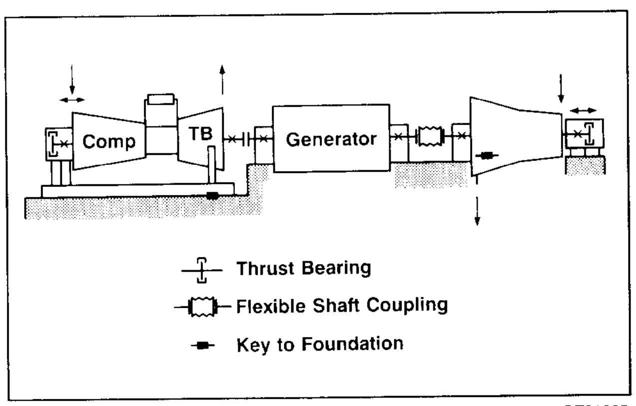


Figure 37. Control cabinet for typical TMR SPEEDTRONIC Mark V Control System



GT21885

Figure 38. Configuration of previous single-shaft units

field from the generator. This is practicable because of the small size of the single-casing turbine and the small number of piping connections, both due to the fact that the cycle is non-reheat. The steam and gas turbines have their own thrust bearings to maintain the proper axial position of rotors relative to casings and stationary nozzles. A flexible coupling is required somewhere in the shaft between the steam turbine and the gas turbine to permit the steam and gas turbine rotors to expand and contract axially independently of each other.

The arrangement discussed above works well for smaller ratings; however, it has limitations when applied to larger, more complex units.

As a result, GE has developed a product line of single-shaft gas turbine/steam turbine arrangements based on the MS7001FA 60-Hz and MS9001FA 50-Hz advanced gas turbines. A product line of arrangements for each speed is required to accommodate site conditions, primarily condenser pressure, although the gas turbine is of standard design in all cases.

Two basic configurations are required to accommodate various steam turbine low-pressure section designs. These are shown schematically in Figures 39 (a) and 39 (b).

Gas Turbine

The GE MS7001FA gas turbine is an advanced design, heavy-duty machine for electric power generation. In the initial planning of its development, it was anticipated that combined-cycle installations would represent an increasingly significant portion of new generating capacity, and that the design must be thermodynamically optimized for high combined-cycle thermal efficiency and must be mechanically suitable for both multi-shaft and single-shaft applications. At a firing temperature of 2350°F (1288°C), the pressure ratio of 14.7 is optimum for maximum specific power, defined as the ratio of power output to compressor inlet mass

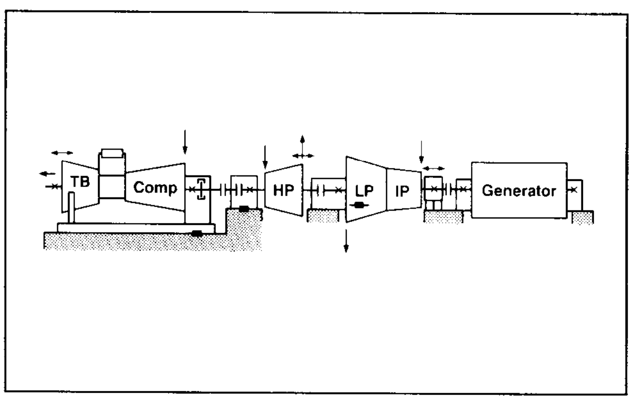


Figure 39a. Fully integrated gas turbine–single flow reheat steam turbine design

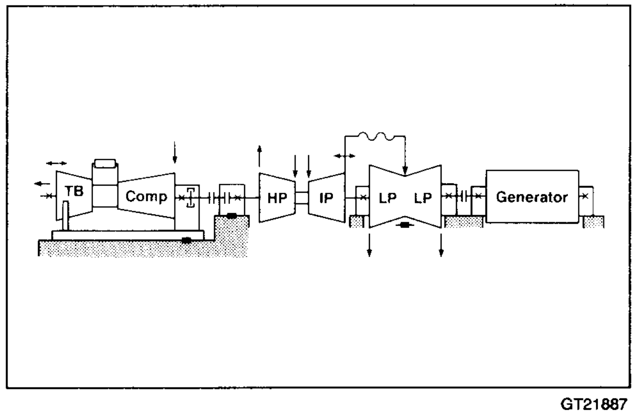


Figure 39b. Fully integrated gas turbine–double flow reheat steam turbine design

flow. Maximizing specific power for a given firing temperature results in a design having the highest combined-cycle thermal efficiency and lowest installed cost for simple-cycle operation.

At the gas turbine exhaust end is an axial exhaust diffuser. The thrust bearing is located on the opposite end at the inlet to the compressor, where the machine is anchored axially to the foundation, providing good clearance control in the compressor. This arrangement, plus the fact that the thrust bearing is conservatively sized for thrust loading in either direction, permits the identical machine to be either directly coupled to the generator for simple-cycle or multi-shaft combined-cycle applications, or to the steam turbine rotor for single-shaft combined cycle.

The gas turbine nominal rating is 159 MWe. The 50-Hz model, the MS9001FA is geometrically similar to the MS7001FA and has a nominal rating of 229 MWe.

Cycle and Performance

Economic and equipment-design studies led to the selection of a reheat, three-pressure steam cycle with nominal throttle conditions of 1450 psig (9996 kPa), 1000°F (538°C), with reheat to 1000°F (538°C); intermediate-turbine admission at the reheat pressure of 300 psig (2172 kPa); and low-pressure admission at 60 psig (517 kPa). The most efficient recovery of gas turbine exhaust heat in a combined cycle is achieved with full feedwater heating by exhaust heat rather than by extraction steam. The absence of extraction-steam feedwater heating removes one variable that is commonly the subject of economic optimization in conventional Rankine-cycle plants, and makes equipment standardization easier.

The MS7001FA gas turbine exhaust temperature is approximately 1100°F (593°C) at base-load conditions, nearly 100°F (56°C) higher than that of gas turbines previously applied with non-

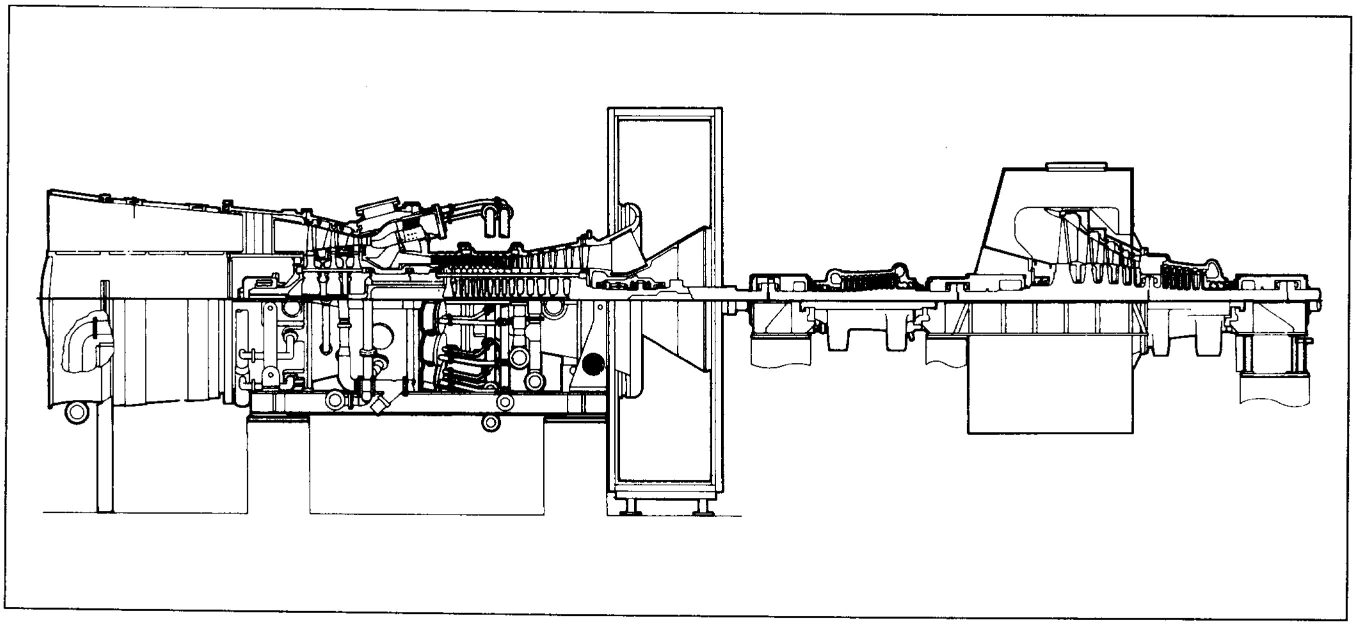


Figure 40. Cross section of GE single-shaft 107FA gas turbine with single flow LP steam turbine

reheat steam cycles. This higher exhaust-gas temperature increases the thermodynamic gain which can be achieved with a reheat steam cycle. Selection of three pressure levels with a reheat cycle avoids the compromise of two-pressure-level systems, in which the lower admission pressure is less than the thermodynamically optimum reheat pressure and higher than that required for efficient recovery and utilization of low-level exhaust gas energy. The three-pressure reheat cycle has a thermal-efficiency advantage of about one percent over two-pressure reheat and two percent over two-pressure non-reheat. Reheat has the additional important advantage for a single-shaft unit of reducing the moisture content of the steam in the low-pressure turbine stages, permitting use of longer last-stage buckets.

Cycle and performance parameters are summarized in Table 2.

Table 2
CYCLE PERFORMANCE PARAMETERS – NATURAL GAS FUEL

Rating	
Net Plant Power	234 MWe
Net Plant Heat Rate	6450 Btu/kWh (6805 kJ/kWh)
Thermal Efficiency (LHV)	53.0%
Steam-Cycle Conditions	
Throttle	
Pressure	1450 psig (10100 kPa)
Temperature	1000°F (538°C)
Reheat	
Pressure	300 psig (2172 kPa)
Temperature	1000°F (538°C)
LP Admission	
Pressure	60 psig (517 kPa)
Temperature	475°F (246°C)

Note:
Site conditions = 59°F (15°C), 14.7 psia (101.4 kPa), 60% RH

Steam Turbine

For both the single-flow and double-flow configurations (Figures 39 (a) and 39 (b)) the steam turbine consists of two casings. The single-flow configuration consists of a single-flow high-pressure (HP) section and a single-flow combined intermediate/low-pressure (IP/LP) section. The double-flow configuration consists of an opposed-flow HP/IP section and a double-flow LP section. A cross section of the single-flow configuration is shown in Figure 40. For a single-shaft machine, design features that minimize length are important. Reheat permits use of a single-flow exhaust with the longest available last-stage buckets without excessive erosion due to moisture. Impulse-stage design with wheel and diaphragm construction is particularly well suited because fewer stages and shorter axial length are required for the steam path compared to reaction stage designs.

Single-shell construction is used in both the HP and IP sections. Throttle steam pressure varies with load and the steam turbine operates with the inlet steam valve normally fixed in the full-open position. The inlet section is designed for full-arc admission without a control stage. The simple shell geometry in this region minimizes the thermal stress imposed by cyclic operation.

The inlet stop and control valves are located off the shell below the operating floor. The absence of shell-mounted valves, and the fact that all steam inlet piping connections are made to the lower half, eliminates the need for bolted piping connections and facilitates removal of the shell upper half for disassembly of the machine.

With the low dynamic thrust of impulse-stage design and the opposed-flow arrangement of the sections, the unbalanced thrust of the steam turbine is quite low. Packing diameters have been established such that the steam turbine thrust is less than that of the gas turbine and in the opposite direction, so that the thrust-bearing loading of the combined machine is less than that of the MS7001FA gas turbine in simple-cycle applications.

As with any condensing steam turbine, the annulus area of the exhaust must be matched with the condenser pressure. Three single-flow low-pressure section designs with 26-, 30-, and 33.5-inch last-stage buckets and two double-flow low-pressure section designs with 26- and 30-inch last-stage buckets are available for 60-Hz applications. For 50-Hz applications two single-flow low-pressure section designs with 33.5- and 42-inch last-stage buckets and three double-flow low-pressure sections designs with 26-, 33.5-, and 42-inch last-stage buckets are available. These variations permit optimizing steam turbine performance over a wide range of condenser pressure applications.

Eliminating the flexible coupling and positioning the steam turbine between the gas turbine and the generator created several special challenges in the design of the steam turbine.

Figure 39 (a) shows the single-flow arrangement consisting of a high-pressure (HP) section and combined intermediate-pressure (IP) and low-pressure (LP) section. The need for a crossover pipe is thereby eliminated.

Figure 39 (b) illustrates the design for a single-shaft reheat STAG unit with a double-flow LP turbine. Here the two casings are an opposed-flow HP-IP and the double-flow LP. Again, both casings are anchored to the foundation. Provision is made for relatively large movement between the two casings and between the stationary parts and rotor in the LP.

STEAM TURBINE OPERATING EXPERIENCE

The acid test of any turbine design is (1) how well it operates over its lifetime and (2) how efficient it is.

In regard to the first item, two key measurements of how well a unit operates are its reliability (measured by forced outage rate) and availability. These are quantities that can be defined and measured. It is highly desirable to have forced outage and availability rates based on owner furnished information to assure unbiased reporting of information. Currently the world's most useful source of such information is the North American Electric Reliability Council (NERC) Generating Availability Data System (GADS).

The NERC GADS data base has information on more than 4,000 electric generating units representing over 91% of the installed capacity in North America. Investor-owned, municipal, cooperative, state, provincial and federal utilities participate periodically reporting operating information on their units using standard reporting practices and computation methods. The GADS "Generating Availability Report" is the primary publication containing reliability and availability data. However, special requests for information will be honored. Information received from GADS is very insightful for manufacturers to identify areas where product improvements should be pursued, and also for owners or potential owners to make comparisons between competing manufacturers. Current GADS information covers units through the year 1989.

Figures 41 and 42 show the results of the analysis of the the reliability and availability data, respectively, for the five-year period 1985-89. This information is broken down by MW size ranges, and shows the percent forced outage and availability advantages GE has over all other

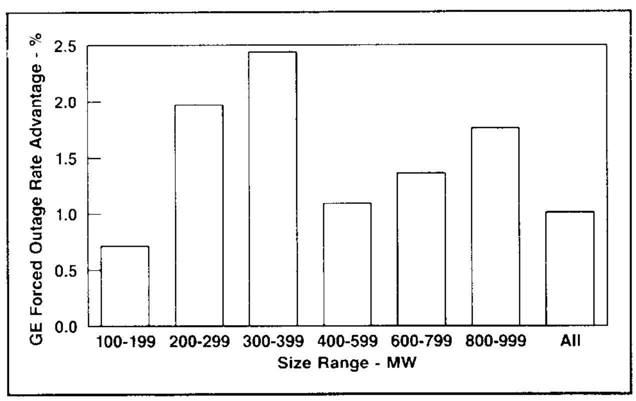


Figure 41. T-G forced-outage rate – GE advantage over other manufacturers

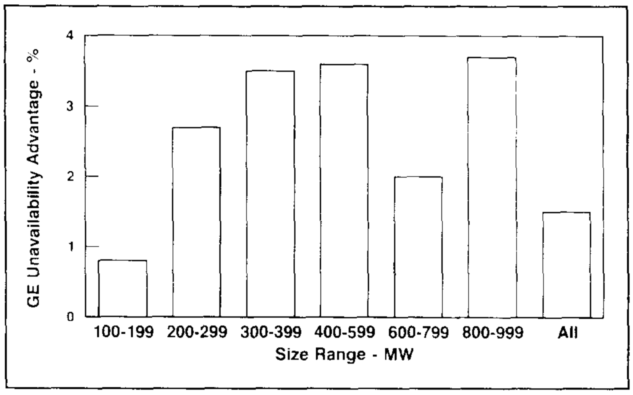


Figure 42. T-G Unavailability – GE advantage over other manufacturers

manufacturers. This data clearly shows an overall advantage of 1.0 percentage point in forced outage rate (FOR) and 1.5 percentage points in availability in favor of GE units. In some size categories the FOR and availability advantages become much larger. For example, in the size range 300-399 MW the FOR advantage is 2.4 percentage points while the availability advantage increases to 3.5 percentage points.

The NERC-GADS data, together with internal GE data, have enabled engineers to direct their attention to problems, allowing them to be resolved on newer units and in many cases have also enabled substantial improvements on older units. Figure 43 shows the difference in FOR by component for older designs (those completed prior to 1975) and for newer designs (those completed after 1975). Over the last 15 years GE has focused technology toward improvements on many components including buckets, piping, alignment, valves and shells. It is clear from Figure 43 that this effort has resulted in improved reliability even though more recent improvements such as full-flow filters and digital controls are probably not fully reflected in the data. It is also clear that the efforts of other manufacturers have not resulted in component designs which are as reliable as GE's as shown in Figure 44. Although they have been endeavoring to improve their component reliability, they have had inconsistent results as demonstrated in Figure 45.

Tracking of turbine efficiency over the lifetime of a unit is much more difficult. Accurate periodic efficiency data would require frequent expensive tests to be conducted and is not warranted except in unusual situations Consequently the best measure of efficiency is high accuracy ASME heat-rate acceptance tests conducted by the owner in cooperation with the manufacturer to determine if guaranteed contractual commitments have been met. Many owners track efficiency on an individual-unit basis, but this is difficult to draw overall conclusions from since daily performance is heavily influenced by such things as load on a unit and ambient temperature.

In the area of efficiency, GE has had a longstanding program of supporting accurate ASME acceptance tests in cooperation with owners. Figure 46 shows acceptance test results covering the period 1980 to 1990. The chart shows that the

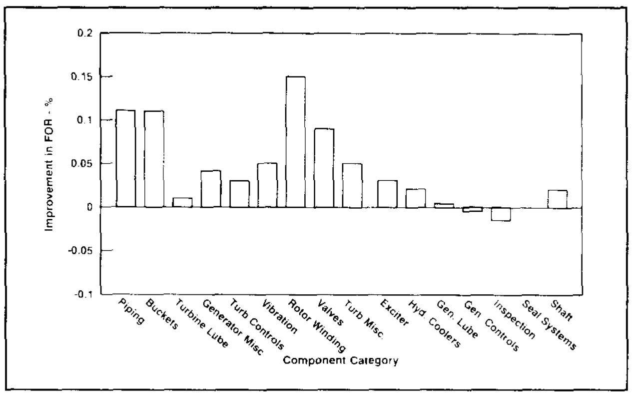


Figure 43. GE component reliability improvements since 1975

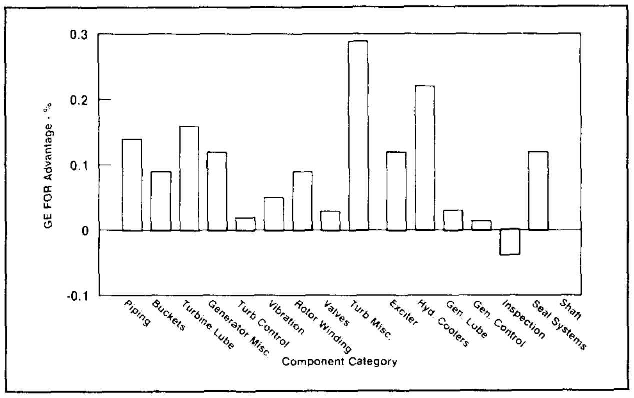


Figure 44. GE component reliability advantage over other manufacturers since 1975

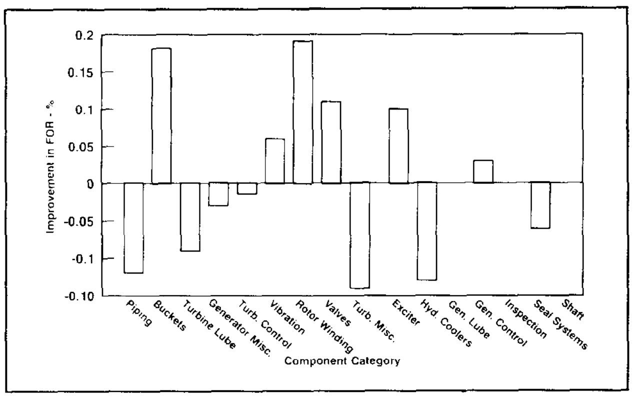


Figure 45. Other manufacturers component reliability improvement since 1975

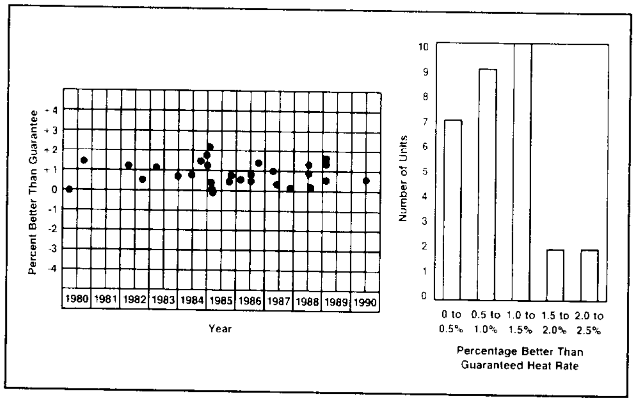


Figure 46. Results of acceptance tests on GE steam turbine-generators, 100 MW and larger, for period 1980 to 1990

vast majority of GE turbine-generators test significantly better than guarantee. What the curve does not show, however, is the improvement in guarantee level which has occurred over the same time period reflecting the results of aerodynamic development programs which have been heavily emphasized by GE over several decades.

GE strongly encourages precision ASME acceptance tests. These tests not only confirm contractual commitments but also provide GE engineers with valuable design information for making accurate performance predictions and for confirming aerodynamic improvement perfected in the laboratory.

SUMMARY

The design of steam turbines requires many technical disciplines and many tradeoffs between design requirements that affect cost, reliability, availability, efficiency, maintainability and operability in very complex ways. Through a thorough understanding of life-cycle cost concepts, a vast array of design features that have been developed and perfected over many years, and a strong commitment to invest in new technology, GE has maintained its leadership position as a world-class supplier of steam turbines. It is our goal to produce machines which are superior to the competition in all aspects so that GE will be the natural choice of customers. Although we may not know exactly what turbine designs will be needed by customers over the next ten years, we believe GE has the right technology, the right philosophy and the resources to respond to all possibilities.

REFERENCES

1. Timo, D.P., "Design Philosophy and Thermal Stress Considerations of Large Fossil Steam Turbines", 1982 Large Steam Turbine Seminar, GE.
2. Booth, J.A., and Holman, R.J., "Design Features of General Electric Large Steam Turbines for Fossil Applications", 1982 Large Steam Turbine Seminar, GE.
3. Carson, R.L., "Design and Experience with Steam Valves for Large Steam Turbines", 1982 Large Steam Turbine Seminar, GE.
4. Booth, J.A., Couchman, R.S., and Sturges, D.C., "Features and Experience with Large Steam Turbine-Generators Designed for Two-Shift Operation", April 1979, American Power Conference, GE Publication GER-3088.
5. Wheeler, E.B., "Cycling Operations of Modern Fossil Utility Steam Turbine-Generators", 1985 Large Steam Turbine Seminar, GE.
6. Livingston, R.G., "Computer Control of Turbine-Generator Startup Based on Rotor Stresses", September 1973, ASME/IEEE Joint Power Conference, GE Publication GER-2948.
7. Schofield, P., and Sumner, W.J., "Improving the Thermal Performance of Older Steam Turbines", 1990 ASME Joint Power Generation Conference, Vol. 10, pp. 89-97.
8. Sumner, W.J., Vogan, J.H., and Lindinger, R.J., "Reducing Solid Particle Erosion Damage in Large Steam Turbines", 1985 American Power Conference, Chicago.
9. Schofield, P., and Johnson, T., "Experience with Coatings and Turbine Modifications to Minimize Solid Particle Erosion Damage on Muskingum River Unit #5", 1988 EPRI Solid Particle Erosion Conference, New Orleans.
10. Bievenue, R.T., "Large Steam Turbine Maintainability - A Means to Improve Unit Availability", 1982 Large Steam Turbine Seminar, GE.
11. Holmes, D.G., and Tong, S.S., "A Three-Dimensional Euler Solver for Turbomachinery Blade Rows", ASME Journal of Engineering for Gas Turbines and Power, Vol. 107, April 1985, pp. 258-264.
12. Hah, C., "A Navier-Stokes Analysis of Three-Dimensional Turbulent Flows Inside Turbine Blade Rows at Design and Off-Design Conditions", ASME Journal of Engineering for Gas Turbines and Power, Vol. 106, April 1984.
13. Dejch, M.E., et al., "Method of Raising the Efficiency of Turbine Stage with Short Blades", Teploenergetika, February 1960, pp. 18-24. Inf. Service, G.E. Trans. 4563.
14. Morrison, B.L., Booth, J.A., and Schofield, P., "Positive Pressure Variable Clearance

Packing", 1989 EPRI Heat Rate Improvement Conference, Knoxville, Tenn.
15. O'Connor, M.F., Robbins, K.E., and Williams, J.C., "Redesigned 26-Inch Last Stage for Improved Turbine Reliability and Efficiency", 1984 GE Publication GER-3399.
16. O'Connor, M.F., Williams, J.C., Dinh, C.D., Ruggles, S.G., and Kellyhouse, M.W., "An Update on Steam Turbine Last-Stage Redesigns for Improved Efficiency and Availability", 1988 GE Publication GER-3577.
17. Morson, A.M., Williams, J.C., Pedersen, J.R., and Ruggles, S.G., "Continuously Coupled 40-Inch Titanium Last Stage Bucket Development", 1988 GE Publication GER-3590.
18. Moore, J.H., "An Integrated Steam/ Gas Turbine-Generator for Combined-Cycle Applications", ASME Paper 89-JPGC/GT-3, October 1989, ASME/IEEE Joint Power Generation Conference.

For further information, contact your GE Field Sales Representative or write to GE Power Generation Marketing

GE Industrial & Power Systems

General Electric Company
Building 2, Room 115B
One River Road
Schenectady, NY 12345

8/91 (3M)