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High-efficiency Gas Turbines Operating in Intermediate Duty

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1. Introduction

The efficiency and output of the combined cycle were improved in the last two decades primarily by increasing the turbine inlet temperature. A typical example is the step from the V94.2 developed in the 1980's to the V94.3A introduced 15 years later **(Fig. 1).** The best combined-cycle efficiency measured in the power station Mainz Wiesbaden in an acceptance test is 58.4 %.

(50 HZ)		V64.3A	V94.2	V94.2A	V94.3A
Speed	rpm	5400	3000	3000	3000
Introduction	Year	1997	1981	1999	1996
Exhaust Flow	kg/s	192	509	520	658
Pressure Ratio		16.2	11.1	14.0	16.9
ISO Temperature	C	1190	1060	1175	1230
Exhaust Temperature	C	580	540	570	585
NOx (gas operation)	ppm	25	25	25	25
SC Power	**MW**	**68**	**159**	**183**	**266**
GT Efficiency	**%**	**34.9**	**34.3**	**35.2**	**38.6**
CC Power	**MW**	**101**	**238**	**285**	**383**
CC Efficiency	**%**	**53.6**	**52.1**	**56.3**	**57.6**

Figure 1: Performance Data of 50 Hz Siemens Gas Turbines

This is achieved with a combined cycle using a gas turbine in simple cycle but with advanced aero, cooling and material technologies. It will be shown in the paper that this approach allows the proven design features to be retained and refined, which enable fast startup and loading of the gas turbine. These and other requirements for intermediate-duty operation are summarized in **Fig. 2.** The preferred methods for improving power and combined-cycle efficiencies are those which are not in conflict with these requirements. Of course our customers also want a low-cost power station and everybody demands the lowest possible environmental impact (NOx, CO, noise, etc). Therefore, the designer has to find the optimum solution in a range of many parameters.

The operation of a power station can be in the continuous base load regime with many hours per year and a small number of starts.

- Quick start and load increase
- Moderate stress levels induced by start stop cycles
- Start from every cool down status without blade rubs
- Use of natural gas with different compositions and various liquid fuels without hardware and software changes
- Switch from natural gas to fuel oil and back at load
- Low NOx and CO emissions between 50% and 100% load
- Quick load response for grid control

Figure 2: Requirements for Gas Turbines in Intermediate Duty

The other extreme is a peak load operation with only some hundred hours per year. The regime in between is called intermediate duty. The power station is operated between 20% and 80% of the time and has daily or weekly starts and stops. It has to change its load quite often and fast according to the need of the electric grid.

2. THERMODYNAMIC CYCLES, PARAMETERS AND LIMITATIONS

Better materials and cooling technologies led to increasing gas temperatures in the blading and today the flame temperature and the maximum gas temperature at the vane 1 inlet are identical. This process cannot continue to even higher gas temperature because it has two inherent and undesirable "side effects":

a) The higher combustion temperatures promote formation of nitrogen oxides, NOx.

b) The higher gas temperatures in the turbine blading increase cooling air consumption.

Although these two effects appear at first glance to be independent, they are actually conflicting. NOx minimization is not solely a task pertaining to the design of new burners. To make more air available for combustion, the cooling air flow requirement must be reduced as far as possible, despite the higher temperatures. This compounds the challenge imposed on the developer (**Fig. 3**).

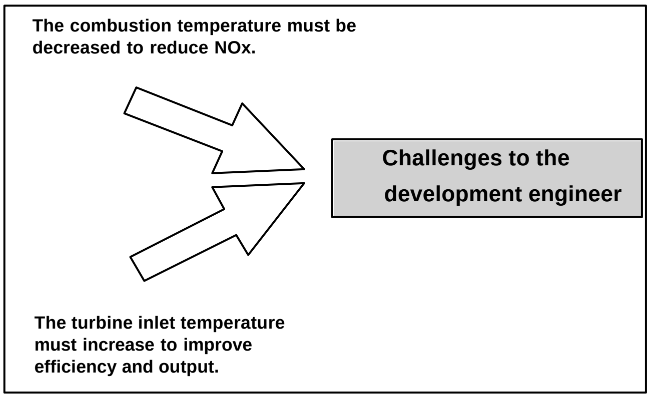

Figure 3: Contradictory Targets NOx and Rate of Efficiency

Is a cycle with 2-stage combustion a means to escape this dilemma? It may appear to be at first glance, because the temperature increase must be split between two combustion chambers with an intermediate heat input permits increasing the amount of fuel supplied without increasing the maximum temperature. One must also consider the fact that the additional expansion in the high-pressure turbine requires increasing the pressure ratio by about a factor of two. This increases the compressor outlet temperature by about 150 K. One could argue that temperatures of 550°C and pressures of 30 – 40 bar are

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definitely commonplace in other turbomachines.

Aircraft engines operate on take off with comparable high pressure-temperature conditions at the compressor outlet, but this phase only lasts for a few minutes. Much more favorable conditions prevail over the extended duration in flight and economically acceptable maintenance intervals are possible for this reason alone. The power plant gas turbine, by contrast, must operate continuously at full load.

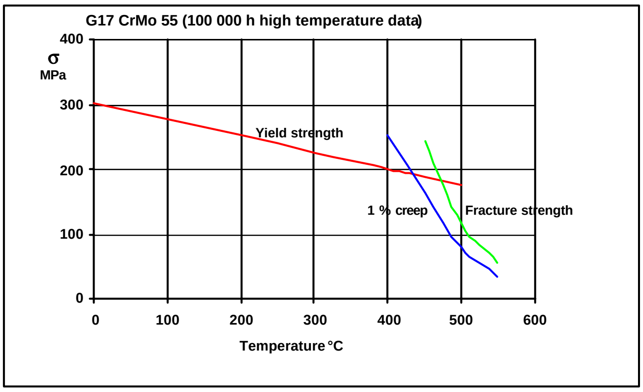


Figure 4: Strength of a Typical Casing Alloy

Can we use the **steam turbine** as a model? It achieves a long service life despite high steam pressures and temperatures. Let us consider a typical high-alloy steel such as is used in steam turbine casings and rotors **(Fig. 4)**. The 0.2% yield strength at elevated temperature which does not result in a life limitation intersects the 1% creep limit at 430°C and the fracture strength at 470°C, both of data to a gas turbine that is operated continuously in the tropics at an ambient temperature of 30°C, this yields pressure ratios of 16 to 20 in the case of today's compressor efficiencies. If the applicable limit for the respective part is exceeded, strength declines dramatically and the requisite wall thickness of casings increases correspondingly. This combination of low strength and high wall thickness results in high thermal stresses under typical startup and load change gradients. The steam turbine is only capable of handling this problem if correspondingly slow load change rates and appropriate steam temperatures are implemented. In a gas turbine the permissible numbers of load cycles before cracks form and fracture occurs decline dramatically and the typical intermediate duty with daily to weekly start-stop cycles causes profoundly-accelerated consumption of service life. The gas turbine ultimately loses its intermediate load capability.

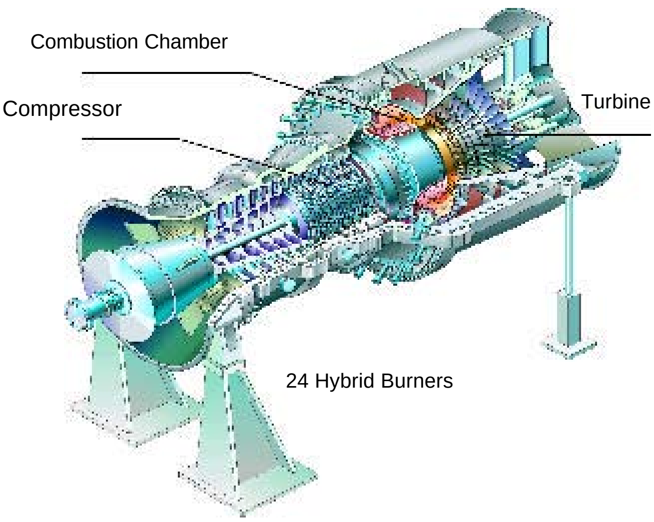


Figure 5: Siemens Vx4.3A Gas Turbine

Now let's consider this development in the context of intermediate duty plants where fast changes in output are required. Plant output targets can vary drastically over periods of minutes. As fuel is traded in a similar manner, this parameter can also change without notice. The gas turbines of the 3A Family **(Fig. 5)**, with their maximum compressor outlet temperature of approx. 440°C, still have a sufficient margin to the temperature limits in Fig. 4 and can thus withstand casing stresses caused by internal pressure as well as thermal stresses for a sufficient number of cycles. Because of this it was possible to retain the low factor of 10 for calculating the number of equivalent operating hours accrued per start.

The widespread utilization of gas turbines in power plants at locations around the world requires increased design "robustness" [2]. The conditions we are accustomed to in the industrialized countries, i.e. well-trained operating and assembly personnel and a sophisticated infrastructure, are not encountered everywhere in the world. The development engineer should strive for a robust design which also mitigates potential negative consequences on operating reliability under unfavorable operating conditions, enabling intervention in the form of service and maintenance before serious consequential damage can occur.

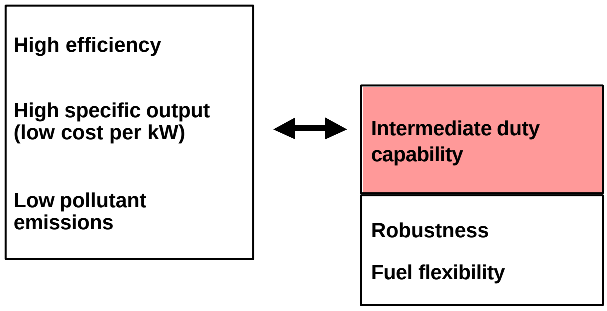


Figure 6: Requirements for Modern Combined Cycle

Robustness and fuel flexibility are two important aspects of intermediate duty capability (**Fig. 6**). Experience shows that even plants which were initially intended for base load operation must face these operating conditions after several years of service. Special design criteria and construction methods can be derived from these requirements for each gas turbine component. Several especially important considerations are covered in the following sections.

3. RADIAL CLEARANCES IN BLADING

It is well known that close radial clearances in the compressor and turbine result in high component efficiencies. However, the radial clearance cannot be allowed to "go negative" under specific operating conditions, as the resulting contact between components has a profound impact on operating characteristics.

3.1 PASSIVE CLEARANCE CONTROL IN THE COMPRESSOR

The key lies in striking a good balance between the warm-up and cool-down behavior of the casing and rotor components. This is known as a passive clearance control.

Fig. 7 shows typical compressor disks in the rear portion of a amount of material present is rapidly warmed up by the intensive flow in the blade throats and in the shaft glands.

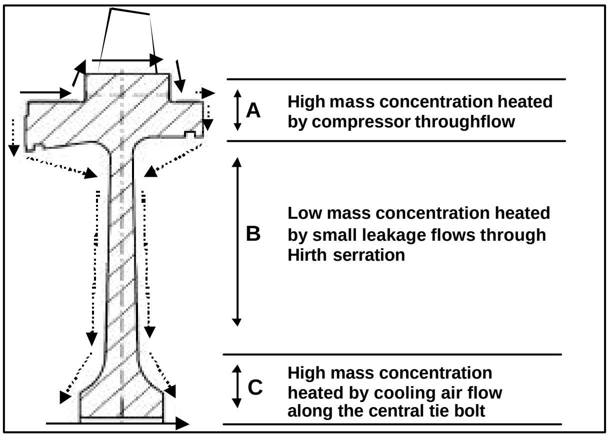


Figure 7: Heat Transfer to a Typical Compressor Disk

Hirth facial serrations at the disk-to-disk interface are deliberately designed to permit a slight leakage flow. This flow is insignificant in terms of thermodynamic impact, however it is sufficient to heat up the relatively thin disk material in region B during a cold start or to cool down this material during turning gear operation, which we perform at a relatively high shaft speed. The inner disk region C, to which cooling air that is extracted in stage 13 is fed, heats up and cools down quickly. Consequently the temperature change of the rotor is nearly identical to that of the stationary blade carriers which were deliberately designed as massive parts in terms of their thickness.

3.2 PASSIVE CLEARANCE CONTROL IN THE TURBINE

Fig. 8 shows this flow of cooling air through the rotor of a V94.3A via three separate paths to the disks of turbine stages 1, 2 and 3+4. Extraction of cooling air from stages **10** and **12** of the compressor affords significant advantages:

- The compression work and thus the power consumed by the compressor is decreased.
- The cooling air remains cooler and the turbine disks can be constructed of steel. This is much less costly than disks made of nickel-base alloy.
- The turbine disks are enveloped by a flow of cooling air and respond to temperature changes rapidly in a manner similar to that of compressor disks as described in the foregoing. This enables us to use tight radial clearances in the turbine as well.
- The centrifugal force field in the rotor functions as a perfect particle separator. No particles larger than about 1µm pass through this centrifugal separator and thus cannot reach the cooled rotor blades and clog the latter's cooling air passageways.

A combination of flow deflection baffles and extraction pipes in the diffuser perform filtration of the cooling air supplied to the turbine vanes. This system was described in detail in reference [2]. Measures presented there pertaining to the robust design of course also minimize wear phenomena that are associated with frequent

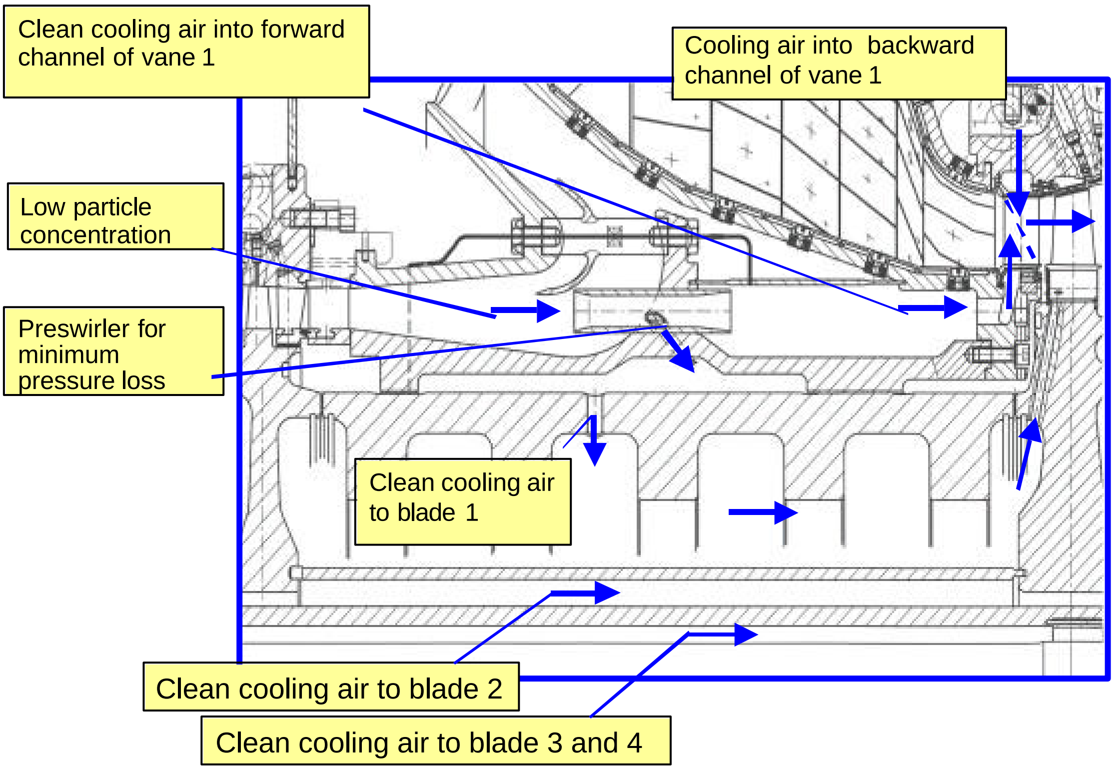


Figure 8: Cooling Air Supply Stage 1, VX4.3A(2)

If this internal flow is not provided in the case of other rotor designs, the gradients of the temperature changes of the rotor and casing differ by about one order of magnitude. This increases the radial clearances on cold startup, with the aforementioned negative thermodynamic consequences restricted to the initial hours of operation.

start-stop operations and thus make an important contribution to the intermediate load capability of SIEMENS designs

3.3 STARTUP OF THE PARTIALLY COOLED DOWN GAS TURBINE

The consequences of a hot startup after an extended operating phase and cooling for over several hours persist much longer. The parts which have previously heated up slowly cool even more slowly during turning gear operation. A solid rotor still remains hot and "fat" for many hours, while the thinner-walled casing already cools significantly, with a corresponding reduction in diameter. If the shaft speed is increased in this condition and the rotor diameter increases still further under the effects of centrifugal forces, severe rubbing contact can occur. The resultant frictional heating of the blades causes them to grow longer, further exacerbating the contact. The result is then abrasive wear of the blade tips or the abradable coatings in the casing or both, with a corresponding release of particles into the air flow. Therefore a quick thermal response of the rotor parts avoids this problem.

3.4 CIRCULAR AND CONCENTRIC CASING

In industrial turbines the casings which support the stationary blades of the compressor and turbine have a horizontal casing joint. The associated accumulation of mass at two opposed points around the circumference causes ovality on fast temperature changes. This can be counteracted in the design by making the casing flanges as small as possible and then also placing the equivalent of the additional mass still remaining on the casings at the top and bottom. Consequently, deformation is the same in both main axes, and the deviation from the ideal circular geometry becomes very slight (**Fig. 9**). The situation is similar for the extraction points for bleed air and cooling air, which also affect the temperature field in the casing.

- Rotor with internal flow changes its temperatures nearly as quickly as the casing
- Diameter and length of rotor and casing change simultaneously during heating and cooling (passive clearance control)
- Radial Hirth serrations facilitate thermal expansion of concentric rotor discs
- Symmetrical casing design avoids ovality and eccentricity

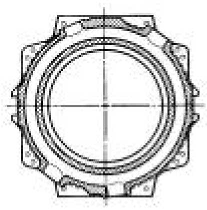


Figure 9: Hot Restart Capability and Quick Load Changes

4. COMBUSTOR

Gas turbines used in applications that demand high operational flexibility must also be able to run on various grades of natural gas. In the case of liberalized gas markets, the properties (inert gas fraction, fraction of methane and propane, heating value and Wobbe index) may change on short notice. If natural gas has a high inert gas fraction and low Wobbe index is used, more gas flows out of the premix nozzles at rated output.

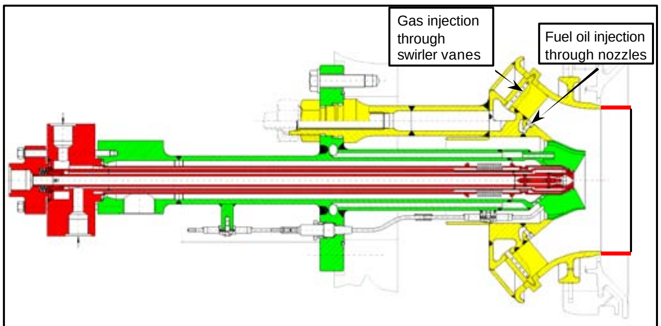


Figure 10: HR3 Burner

There are two fundamentally different directions in which the jets of fuel enter the combustion chamber and the HR3 burner (**Fig. 10**) can be used to illustrate both. The liquid fuel has a much lower volumetric flow and is injected into the premix path via 24 radially-oriented nozzles. Because the heating value of higher hydrocarbons only varies slightly, the penetration depth of the jets **(Fig. 12)** remains roughly constant at rated output. If the gas nozzles were also oriented radially, different outlet velocities would result in pronounced changes in the radial mixing profile. This would cause changes in NOx emissions and combustion stability limits. Gas nozzles would have to be changed if the Wobbe index changed substantially.

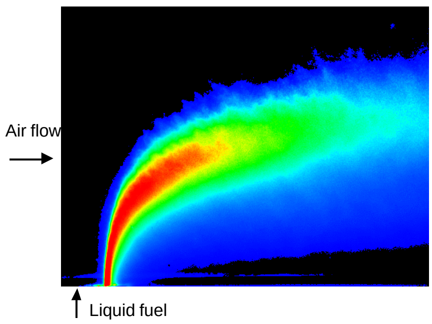
As a matter of fact, however, the 240 gas nozzle bores are oriented perpendicular to the surface of the swirler vanes and the jets thus emerge with no significant radial component. For this reason the mixing profile at the outlet of the premixing section does not change with changing gas properties, and the burner functions properly with the same hardware configuration on all common grades of natural gas (pipeline gas and LNG) without adverse impact on the combustion properties. The experience with various gas fuels confirms this fuel flexibility (**Fig. 11**).

- The heating values (LHV) range from 39,000 – 49,300 kJ/Kg
- The Wobbe indices (W_U) range from 36,000 – 44,300 kJ/(m^3(kg/m^3)$^{0.5}$)
- The widest range of an individual gas turbine is Dormagen:

  LHV 40,500 – 48,700 kJ/Kg
  W_U 36,900 – 42,000 kJ/(m^3(kg/m^3)$^{0.5}$)
- The burner hardware allows a wide range of fuel compositions.
- The control software has to be adjusted to the respective fuel compositions.

Figure 11: Experience with Different Fuel Gas Compositions

In periods of high demand for natural gas by consumers that are not flexible in terms of fuel (such as households), gas turbines equipped with combustion systems for both gaseous and liquid fuels can be changed over to a liquid fuel which, in some cases, may even be lower in price. This approach can of course also be implemented in the event of disturbances to the gas supply system. Because of the storage effect of gas lines, sufficient time is available for changeover. It should, however, be possible to perform changeover without interrupting plant operation. Activation and deactivation of the various fuel systems involve unavoidable abrupt changes in the output produced by the fuel supplied. To nevertheless remain in the permissible operating range, output is reduced somewhat. When running in the output window from 60 – 85%, Siemens gas turbines can perform fuel changeover in either direction within only a few minutes.



Objectives:

Improvement of fuel air mixing

Avoidance of fuel films on surfaces

Figure 12: Spray Pattern of Premix Nozzle HP-Injector Test Rig

While NOx emissions of 25ppm are prescribed for operation on natural gas in the dominant operating regime of gas turbines (a limit that is met by Siemens gas turbines), the requirements imposed on the combustion of fuel oil span a broad range. The fuel oil is sprayed into the premix section **(Fig. 12)** and evaporated.

The NOx emissions levels shown in **Figure 13** are achieved in dry premix mode with light distillate fuel oil or kerosene

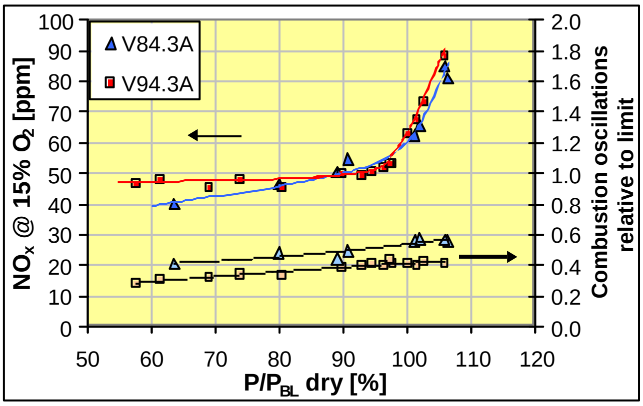


Figure 13: NOx Emissions in Dry Premix Operation Comparison V84.3A/V94.3A

Owing to the fact that the liquid fuel is rarely used in dual-fuel plants, in the majority of cases the 75ppm limit may be sufficient, which can be complied with from 60 to 100% rated load without the occurrence of combustion instabilities. Some licensing requirements involve lower limits, for example 42 ppm. Then the burners are operated with an oil-water emulsion that is formed in the fuel supply system. Emissions levels achieved in such cases are depicted in **Figure 14**.

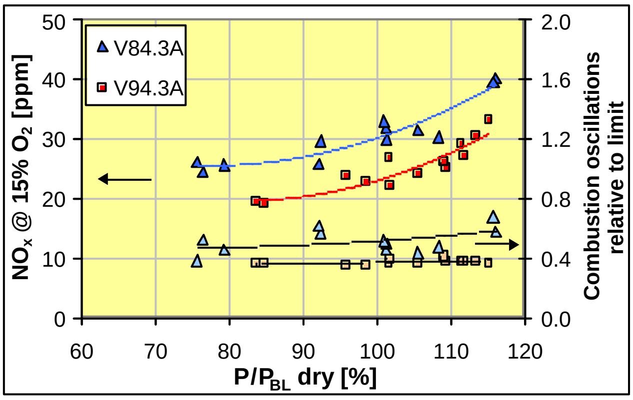


Figure 14: NOx Emissions in Emulsion Operation Comparison V84.3A/V94.3A

5. TURBINE BLADING

Depending on the length of the operating time, the course of one load cycle with startup, load application, operation and shutdown can be regarded as LCF with hold times at high temperature or as creep with cyclic interruptions.

The first stage blades are subjected to high centrifugal forces and a very hot environment. Therefore a single crystal (SC) alloy, PW 1483, is used which has a higher creep strength than conventional cast materials. **Fig. 15** shows, that the SC structure has a high LCF life. The allowable strain range including the notch effect at the film cooling holes, is higher than even that of the smooth equiaxed structure and a factor of 4 higher with comparable geometry. Therefore the blade life and the scrap rate during a refurbishment is correspondingly lower.

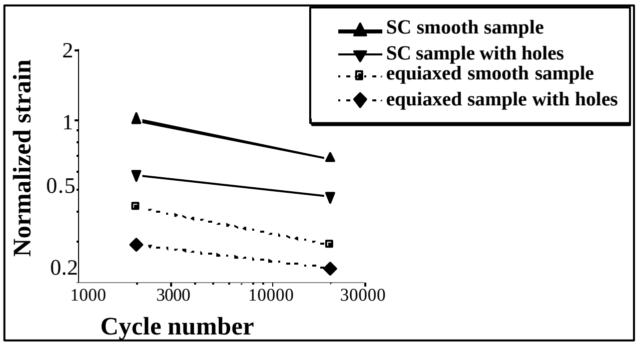


Figure 15: Comparison of a Single Crystal with a Conventional Cast Superalloy

As a matter of principle, parts which are thin and can deform under the effects of temperature differences build up less stress than structures which are rigid or have a frame-like configuration. Segments comprising 2 or more turbine vanes are included in this unfavorable group. Such effects can be clearly seen in **Fig. 16**, which shows the results of an analysis which I already obtained some time ago at an engineering meeting with Siemens Westinghouse colleagues. Like the Siemens gas turbines, the W501 gas turbines have individual vanes in the first turbine stage. A distinction is also drawn here between a normal start/stop cycle and a startup which ends with trip from full load. In both cases, the number of load cycles which individual vanes can withstand is higher than that for vane segments by more than one order of magnitude. This demonstrates how advantageous the individual vane construction is for medium-load plants. The V-series gas turbines have individual vanes in all turbine stages.

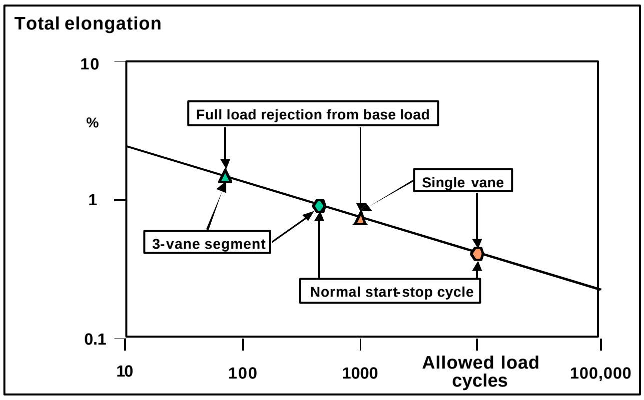


Figure 16: LCF Stress on First Stage Turbine Vanes

6. EFFECTS ON RAM RESULTS

$$\left.\begin{array}{l}\underline{R}\text{eliability}\\ \underline{A}\text{vailability}\\ \underline{M}\text{aintainability}\end{array}\right\}\quad \text{RAM}$$

Reliability includes allowance for all forced outages, while availability includes scheduled maintenance measures in addition.

These values are typically determined for the overall power plant, but also separately for its most important components. Here, we are interested in the effect of launching a new generation of gas turbines and thus in the performance data for the gas turbine, including the directly associated auxiliary systems such as the lube oil and fuel supply systems. These were not treated specifically here, but analogous design principles apply that have a profound effect on startup availability, which is especially important for medium-load plants.

The operating results of the 3A family have already been reported several times ([3] to [6]). These publications, some written by our customers and others by Siemens personnel, show that high RAM values were achieved comparatively quickly after passing the peak of the learning curve that is typical of a new gas turbine model. For example, the operator of the Tapada power plant (3x V94.3A) in Portugal was already able to state the following at the end of the year 2000: "Our projected maintenance costs fall well within world benchmarking comparative costs and also compare well with the Rye House and Killingholme plants, each equipped with V94.2 gas turbines, owned by Powergen".

Fig. 17 shows that the reliability of the new generation has already reached the values for the large fleet proven over many years. The fleet of 3A machines has grown considerably in recent years and in 2002 our RAM statistics included twice as many gas turbines as in the year 2000. Availability is essentially determined by the fraction of gas turbines that undergo a major inspection during the time frame under consideration.

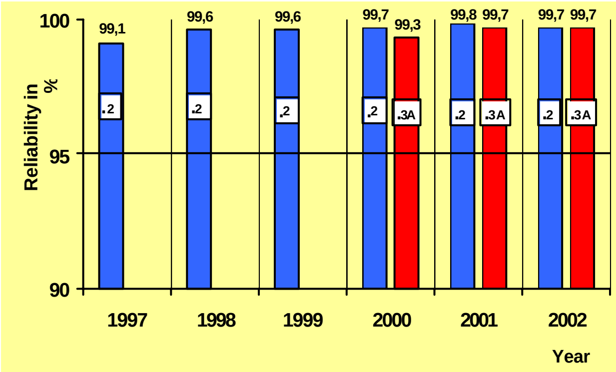


Figure 17: Reliability of the V94.2 and V94.3A Fleets

Such machines are indicated in **Fig. 18** by the symbol (1). If one assumes about 6000 equivalent operating hours during a calendar year (5000 operating hours, 100 starts) and a major inspection interval of 25,000 EOH, the major inspection frequency is 24% per year. In other words, an average of about 6 of the total of 26 gas turbines will have an extended outage during a given year per year.

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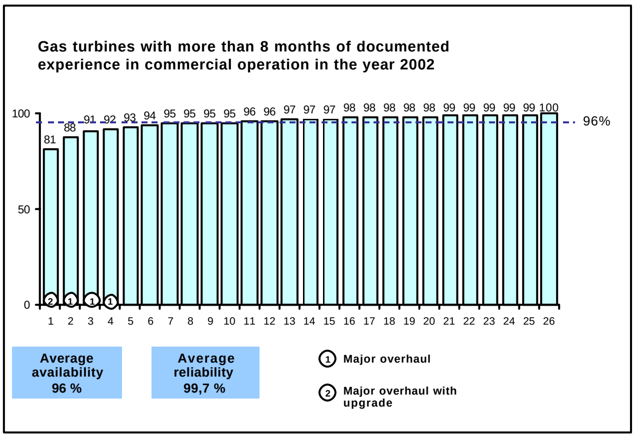

Figure 18: V94.3A Availability 2002

In the year 2002, only four plants had major inspection outages and the availability figure of 96% was better than average. The two previous years, by contrast, had a disproportionately high fraction of plants with extended maintenance outages. This indicates that an average availability of 95% can be expected. A very good correspondence can also be seen between the published figures for maintenance intervals, maintenance measures and unscheduled outages and the expected values derived by calculation. In other words, these gas turbines have clearly met the operators' expectations for availability and greatly exceeded those for reliability.

7. SUMMARY

All of the gas turbines in the V94.3A fleet are installed in combined-cycle power plants and because they are the most efficient, they are implemented preferentially by operators. On the other hand, as natural gas is a relatively expensive fuel in many countries and the base load is thus generated in nuclear, coal-fired and hydroelectric power plants, whereas the combined-cycle plants are subject to frequent startup cycles. The 90 gas turbines in the fleet were started up 23,000 times and operated for a total 1 million hours, meaning that the average plant, in operation for roughly 3 years, has been operated 250 times for a period of 2 days. This is typical for intermediate duty operation. The high reliability levels demonstrate that the 3A gas turbines can clearly meet the stiff requirements imposed by this cyclic service. The described proven design features of compressor, combustor and turbine are implemented also into the new SIEMENS family. They result in combination with modern aerodynamic, cooling and material applications in a unique combination of full operating economy, low cost and reliability.

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