Recent Developments in Large Generator Repair Solutions and Monitoring Systems

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Abstract

The upcoming utility deregulation in the United States has increased the incentive for utilities to look for possible cost savings in the operation of their generating facilities. This in turn has resulted in lower budgets for maintenance activities. Plants are being forced to run for longer periods between outages and also to shorten outage times. Also major upgrades are only going to be possible if it can be shown that the payback period is short.

For this reason the author’s company has focused on generator repair solutions which address these goals. One of these repair solutions is the generator rotor “Tooth Top” Modification - without Rewind. Another solution is the “Stator End Winding Support System” . These solutions have been designed to be cost effective and to address the specific problems which have been encountered. Major capital spending (such as machine rewinds) are avoided by these solutions.

In addition because of the increasing age of utility plants and the increased time span between outages it is very important to obtain the maximum amount of useful information as possible and to present this to the operations personnel in a readily usable form to as far as possible avoid costly equipment failures. Therefore the author’s company has developed specialized monitoring tools to enable power equipment owners and operators to achieve these goals.

This paper describes both the repair options mentioned above and also a series of generator monitoring modules specifically designed to maintain the overall health of this important equipment.

1. Tooth Top Modification

When tooth top cracking was first discovered in large generator rotors in the early 1980's, it was thought that the only possible solution to repair this was to strip the rotor completely and to machine off the rotor tooth tops. Thus a full rewind of the rotor was necessitated. This is expensive and also requires down-time to perform.

The author's company has been able to establish that the repair of this phenomenon can in fact be performed without removal of the rotor windings.

The causes of tooth top cracking are described in detail in references [1], [2].

A brief summary of the cause of this cracking mechanism is as follows:

At standstill, the retaining ring presses radially downwards on the rotor teeth, thus producing high compressive stresses in the radii of the tooth shoulders. At rated speed, the shrink seat is relieved, and hence the compressive stress is correspondingly reduced. In addition, a centrifugal force from the winding occurs. This produces a tensile stress in the radii of the tooth shoulders, which is superimposed on the previously mentioned compressive stress.

During every start/stop cycle high stress changes occur. Depending on the magnitude of these stress changes and on the number of start/stop cycles, cracking will occur at some point in time due to low cycle fatigue (LCF).

The solution described in the above mentioned papers can be performed on site or in shop without a rewind and requires new longer retaining rings. This solution is termed the "long ring modification without rewind" (Fig. 1).

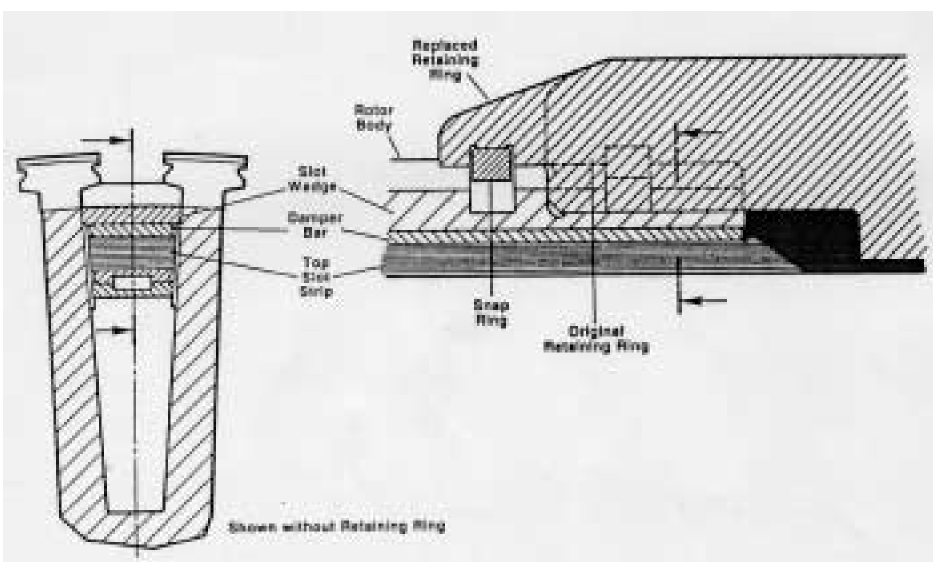


Fig. 1: Modified shrink-fit area for the long ring modification.

Another solution to the “tooth top” cracking problem is however possible should the customer already have installed the “short” 18Mn18Cr retaining rings or if, for some reason it becomes necessary to reuse the existing “short” 18Mn5Cr retaining rings (for example if replacement 18Mn18Cr forgings are unavailable). This solution is referred to as "tooth top shoulder machining".

Both of these procedures do not require a rotor rewind and have been patented by the author’s company [2].

Because the solution with longer replacement retaining rings is described in detail in reference [1] some more details on the "shoulder machining" procedure will be given in this paper.

This solution can be applied if the cracks in the rotor tooth have not progressed beyond a certain critical depth.

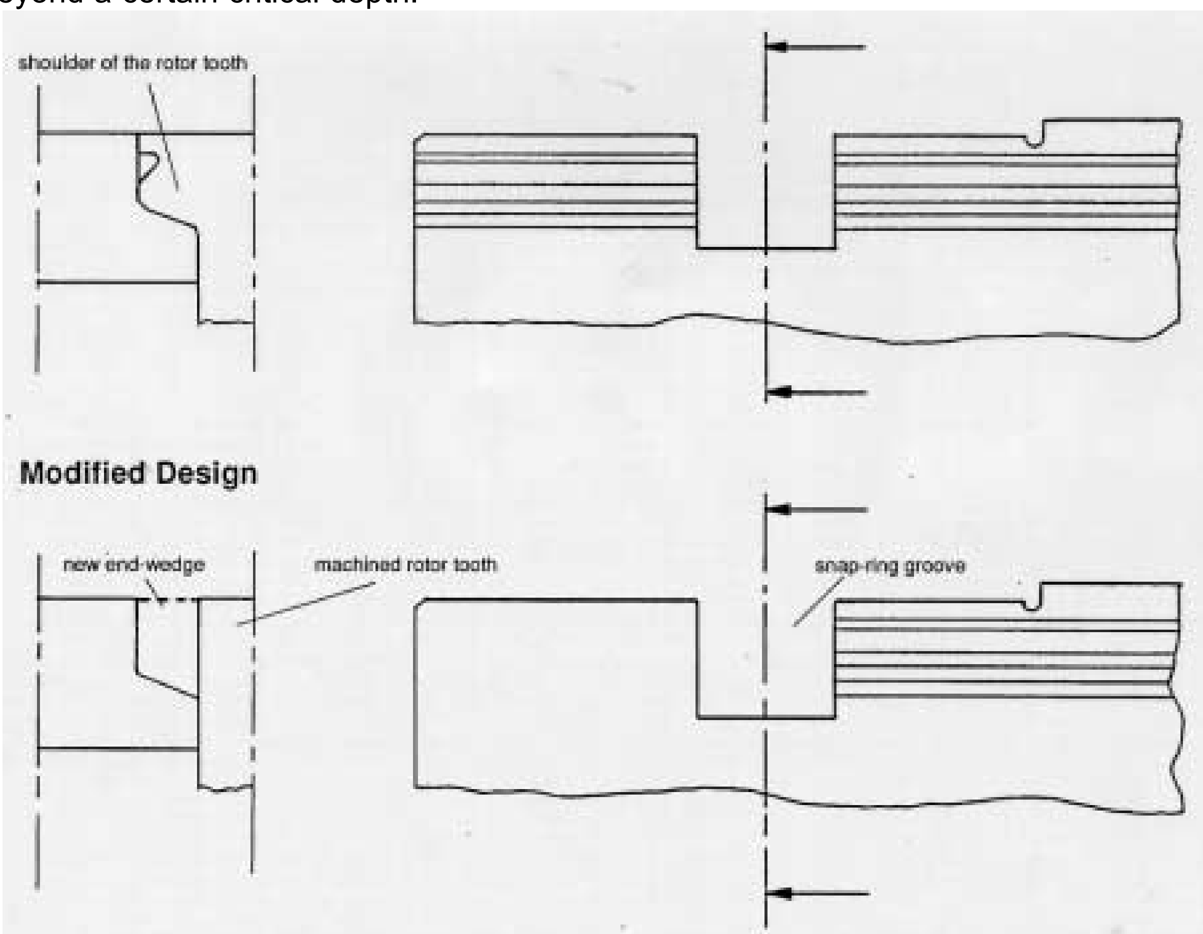


Fig. 2: Comparison between the original design and the proposed modification. In the proposed modification the shoulders of the rotor teeth between the snap-ring groove and the end of the rotor body are machined off. The original end-wedges may be replaced by new end-wedges adapted to the new slot geometry.

In order to remove the cracks and at the same time to relieve the rotor teeth in the region between the end of the rotor body and the snap-ring groove, the tooth shoulders in this region are machined off (see Figure 2). Thus the excessive stresses due to the notches are eliminated.

On the other hand, the end-wedges in this region bear directly on the retaining ring, thus slightly increasing the load on the ring. For this reason, the stresses to be expected in the retaining ring following the proposed modifications were also determined in order to ensure that they are acceptable.

Any minor cracks which still penetrate the remaining tooth material can be ground out locally.

The end-wedges can be replaced after machining of the rotor body by new end-wedges adapted to the new slot geometry.

The tooth top modifications described in the paper have been evaluated in detail with Finite-Element-Programs. Retaining ring stresses and shrink-fit forces at standstill and at rated speed were determined utilizing an axisymmetrical model. Friction forces caused by differential expansion of the windings and the retaining ring and also the rotor body during different operation modes were also considered in the calculation model. In all of these calculations nonlinear stress-strain curves were used.

By machining off the shoulders of the rotor teeth at the retaining ring shrink-fit area, the transition radii are eliminated. The stresses in the remaining tooth section are considerably lower, so that the danger of crack formation has been eliminated.

Table 1 gives a comparison of various tooth top repairs which are available. For example on a 40 inch rotor the tooth top cracks are initiated after 240 start stop cycles. After approximately 850 cycles the cracks propagate completely through the tooth tops which means there are broken off tooth segments under the retaining ring. Table 1 shows also that only the long ring modification and the shoulder machining solve the problem.

Table 1: Comparison of Various Tooth Top Repair Procedures 60 Hz-Generators

Rotors	37"	40"	43"	60"	67"
Number of cycles to crack initiation	500	240	350	450	950
Additional cycles by remachining the fillet radius	500	400	230	400	450
Additional cycles by shoulder machining	>10000				
Long ring modification	>10000				

The so called short ring modification where the fillet radius of the tooth tops are remachined does not solve the problem and is not recommended. For example with this solution the cracks will initiate again after 400 additional start stop cycles on a 40 inch rotor. This becomes very important for power plants which may be repowered for an additional 20 or more years, especially if the operation mode is changed.

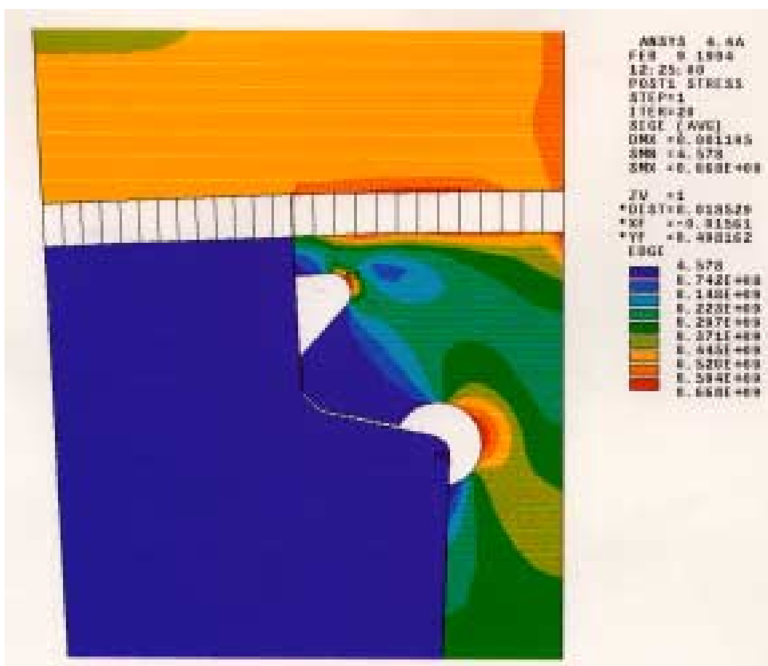


Fig. 3: Equivalent elastic-plastic Mises Stresses (SIGE in N/m^2) on a 40" rotor on the remachined rotor tooth at standstill. Radius of the machined fillet is 0.1 inch. Highest stress is 96.9ksi.

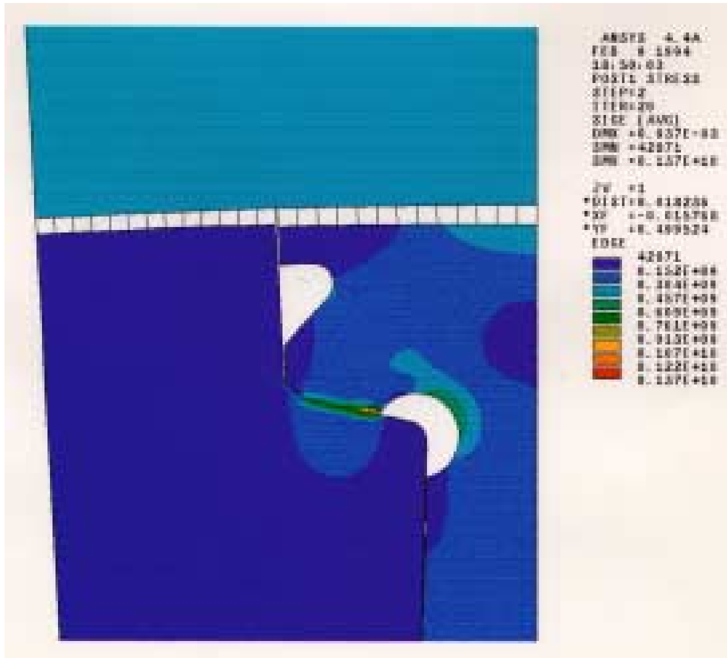


Fig. 4: Equivalent elastic-plastic Mises Stresses (SIGE in N/m^2) on a 40" rotor on the remachined rotor tooth at rated speed. Radius of the machined fillet is 0.1 inch. Highest stress is 198.7ksi.

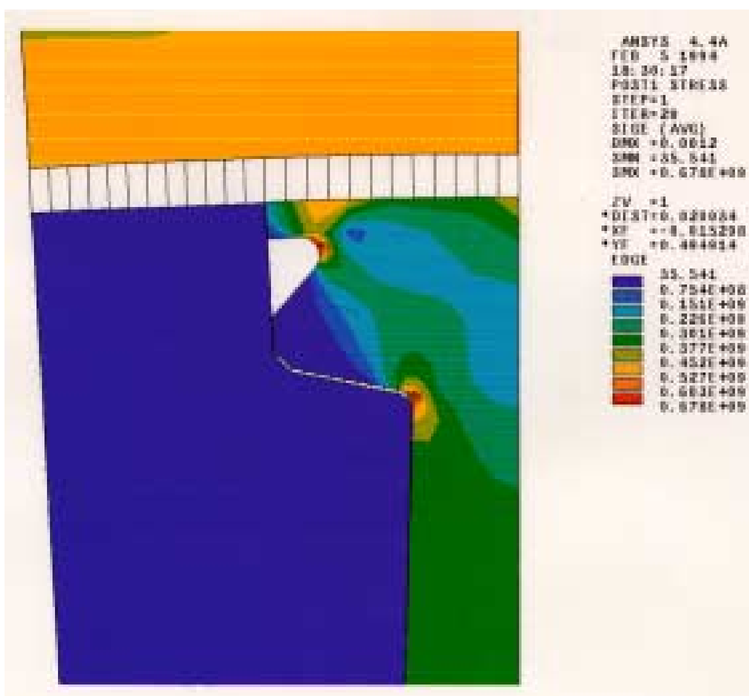


Fig. 5: Equivalent elastic-plastic Mises Stresses (SIGE in N/m^2) on a 40" rotor at standstill. Highest stress is 98.3ksi.

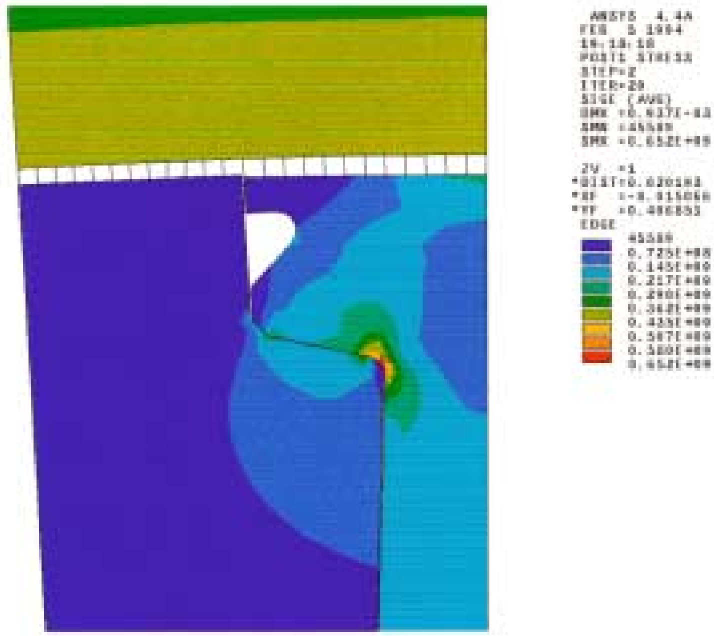


Fig. 6: Equivalent elastic-plastic Mises Stresses (SIGE in N/m^2) on a 40" rotor at rated speed. Highest stress is 94.6ksi.

For the remachining of the fillet radius normally a cutter with a 0.1 inch radius is used. Figure 3 shows the stress distribution on the machined rotor tooth at standstill and Figure 4 at rated speed. Both calculations show that with the remachined fillet radius the stresses are somewhat less compared to the original design (see Figure 5 and Figure 6) but still too high to prevent crack initiation from re-occurring. The crack initiation time with the fillet radius is not increased by a significant margin.

It should also be considered if such a short ring modification is applied and new retaining rings are installed this may mean that another new set of retaining rings may be required during the life of the plant.

The numbers of cycles to crack initiation shown in table 1 are calculated values but have also been demonstrated from actual experience on all sizes of rotors.

The issue of the bending forces on the glass-epoxy-mica slot liner has also been given attention recently although no known failures of this component have occurred in service. In the case of the "long ring" modification performed without a rewind the end wedge in the rotor is no longer supported by the rotor teeth. A change in the retaining ring shrink fit (in the case of rotors which are fitted with replacement retaining rings) can address this point. Thus on a 40 inch diameter rotor increasing the shrink from it's original value of 0.0022" per inch of rotor diameter to a value of 0.003" per inch of rotor diameter means that it is possible to maintain (or even to reduce) the slot liner deflection at it's original value. Several rotors are in service in this country with this increased value of shrink and have seen many thousands of hours of operation.

Table 2 shows a comparison of the slot liner calculated maximum deflections which occur with the various diameters of rotor with the original value of shrink .

Table 2:

Maximum Slot Liner Deflection of Various Rotors

Rotor OD	37"	40"	43"	60"	67"
Shrinkage at standstill per inch of diameter	0.0021"	0.0022"	0.0026"	0.0026"	0.0026"
Max slot liner deflection at rated speed	0.0472"	0.0516"	0.0591"	0.0709"	0.0866"
Max slot liner deflection at 20% overspeed	0.0551'	0.0661"	0.0669"	0.0748"	0.0906"
Shrinkage at rated speed	no	no	no	minor	no

A significant observation from this table is that the largest values of slot liner deflection in fact occur on those 1800 rpm rotors with 60 inch and 67 inch body diameters. For example the slot liner deflection in a 67 inch diameter rotor when the machine is at rated speed and utilizing the original value of shrink of 0.0026" per inch of rotor diameter is in fact 68% greater than on the 40 inch diameter rotor. There are machines with rotor diameters of 67 inches have had the "long ring" modification applied without a rewind. Also these machines have the original glass epoxy mica slot cell and this has apparently proved to not be a problem. See also reference [4].

It should also to be pointed out that machines utilizing values of 0.0026" per inch of diameter (or less) for the retaining ring shrink will (in the case of 3,600 rpm machines) be in a condition of having zero shrink fit at the rated speed. For example a 40" diameter rotor with this value of shrink will have no shrink at 2,950 rpm. Thus the possibility of fretting of the retaining ring is increased when using this value of shrink and is reduced by using a higher value as is recommended by the authors.

Another phenomenon which has been found to occur on this series of rotors is that of cracking of the rotor shaft under the blower hub fit area. This is also not a new problem and is described in detail in reference [5].

The reasons for the occurrence of these cracks are:

1. The increased pressure at the ends of the shrink-fit between the blower hub and the shaft.

2. The stress concentration effect due to the sharp radii of the axial slots at the shaft surface.

These two effects lead to high stresses. These high stresses, together with the presence of the alternating bending stresses, have led to cracks being found in several rotors.

Therefore design modifications to the rotor are required to reduce the high stresses at the end of the shrink-fit and in the axial slots on rotors with this geometry [5].

2. Stator End-Winding Modification

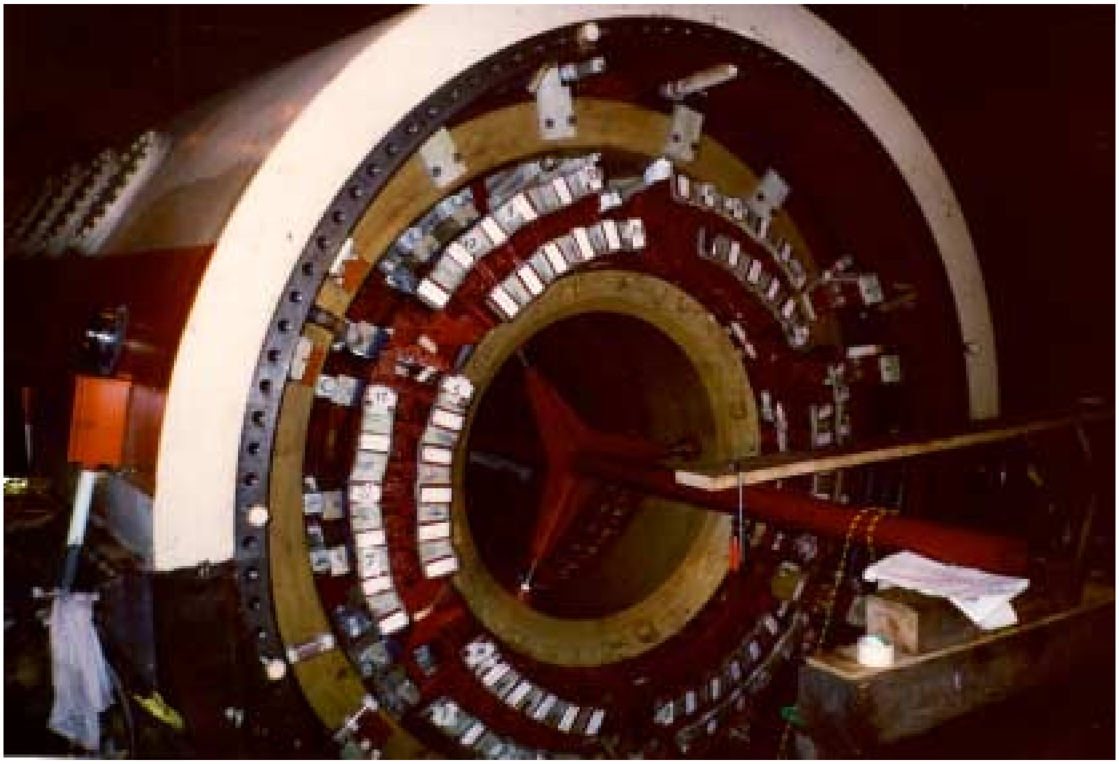
Many generators in operation have high stator end-winding vibration, which can lead in time to loosening of the complete end-winding structure. This in turn causes insulation abrasion and damage.

Generators with loose end-windings have been temporarily repaired and finally rewound by the OEM. Such rewinds are expensive and require long outage time. The generators affected are in the range 350 MVA to 1000 MVA.

The author's company has developed a series of repair alternatives which do not require a rewind of the stator and which can be done on site during a regular overhaul.

Different options of the Stator End Winding Support System (as offered by the author's company) are available depending on the different sizes of generator and also on specific customer needs.

The stator end-winding vibration problem is caused by low stiffness of the end-winding support system and high forces resulting from the closeness of the natural frequency to the operating frequency (120 Hertz in 2 pole 60 cycle units).



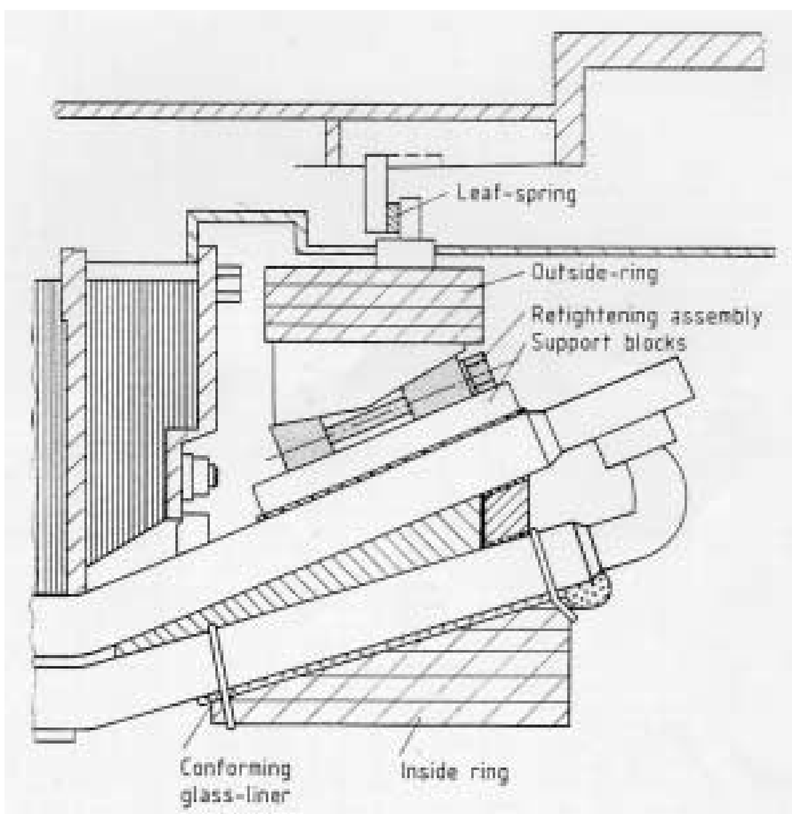


Fig. 7: Modification with inner- and outer ring and parallel ring modification.

A detailed description of a patented solution already successfully applied to several generators is given in [1] and [3]. This papers describe the design philosophy and the installed components in detail.

The installation of the outside ring and the retightening assembly described in [3] requires the complete modification of the parallel rings (see Figure 7).

Many vibration problems can be solved with a very similar solution described in [3] but without the necessity of including the outside ring and the retightening assembly (wedge bolt arrangement). In this case the end-winding can be retightened with the inner ring. Thus a simplified version of the original system can be applied which allows for a shorter installation time and is also more cost effective. The authors company has developed a “building block” approach to the options available with the end winding support system described in this paper.

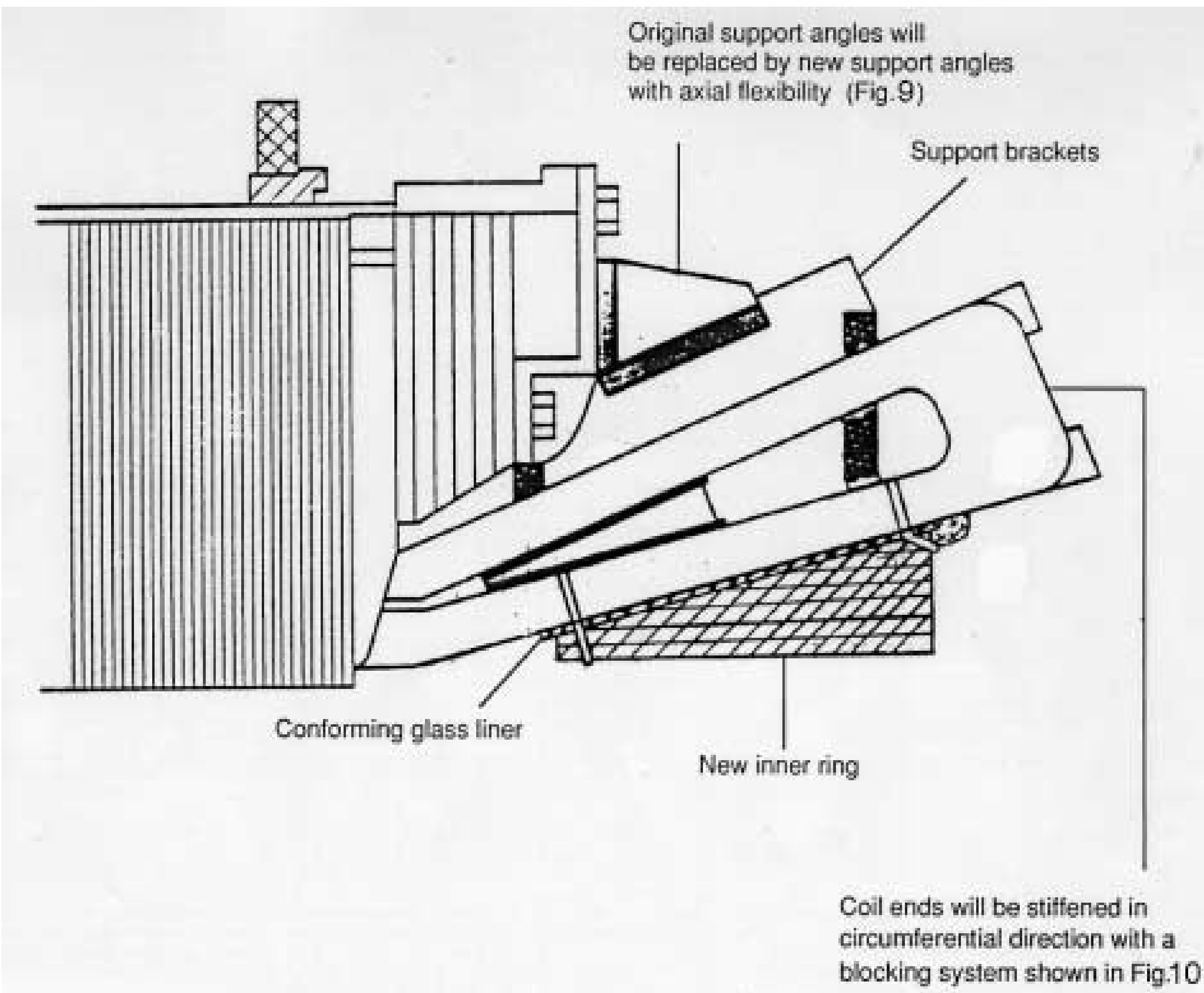


Fig. 8: General end-winding arrangement (with the inner ring only)

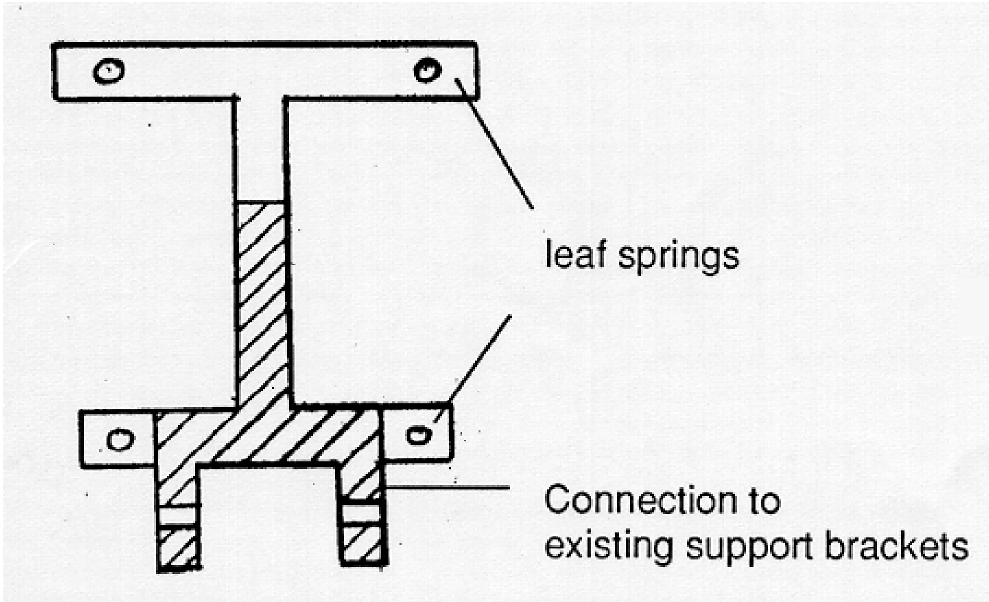


Fig. 9: New support angles with axial flexibility.

In order to provide high radial stiffness and axial flexibility for the thermal expansion of the windings, the existing support angles are replaced by new axial flexible support angles. In addition a stiff inside wound glass epoxy ring in combination with a conforming glass liner filled with epoxy resin is installed. This consolidates the end-windings to eliminate movement between critical components. This is shown in the Figures 8 and 9. The two inner rings on the turbine end and exciter end may be different because of different cone angles of the end-winding baskets. Both inner rings are shaped in such a way that the hydrogen gas flow is not affected.

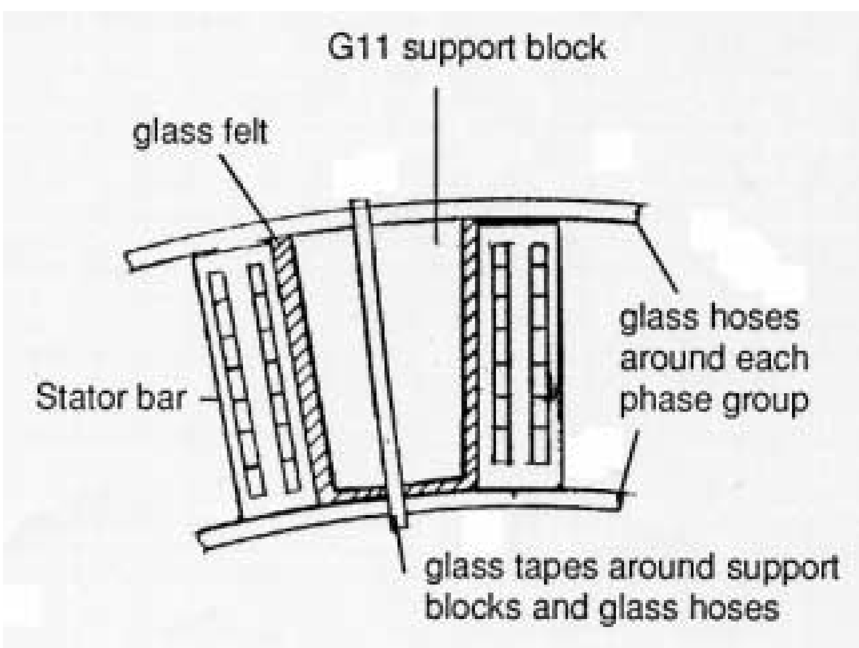


Fig. 10: New circumferential blocking system.

The stiffness in the circumferential direction is increased by a blocking system shown in Figure 10. This blocking system is installed between all the stator bars of each phase group on the outboard end of the end-windings. The G11 support blocks are adapted to the actual geometry in a manner shown in Figure 10. An important point with this alternative solution is that it easily facilitates the inspection and retightening of the core bolt nuts.

3. Generator Monitoring

The economic performance of turbo generators and hydro generators can be considerably enhanced by on-line monitoring techniques.

Vibration signals as well as process data such as temperature, hydrogen pressure, hydrogen purity, flow rate, partial discharge, shaft voltage and field winding shorted-turns can be monitored and used for condition assessment of the entire generator.

The author's company has developed several monitoring modules in order to meet the different customer requirements. All of these modules can be customized to specific customer needs.

The following modules are available:

1. Vibration Monitoring Module
2. Temperature Monitoring Module
3. Field Winding Shorted Turn Module
4. Core Monitor Module
5. Shaft Voltage Module
6. Partial Discharge Module
7. Cyclic Operation Module
8. Power Chart Module
9. Efficiency Module
10. Stator Cooling Water Unit Module
11. Seal Oil Unit Module
12. Hydrogen Unit Module
13. Abnormal Event and History Module

Because vibration, especially stator end-winding vibration is such an important index of the overall soundness of rotating electrical machines it may be used for the early detection of abnormalities and trouble. Therefore this module is described here in more detail.

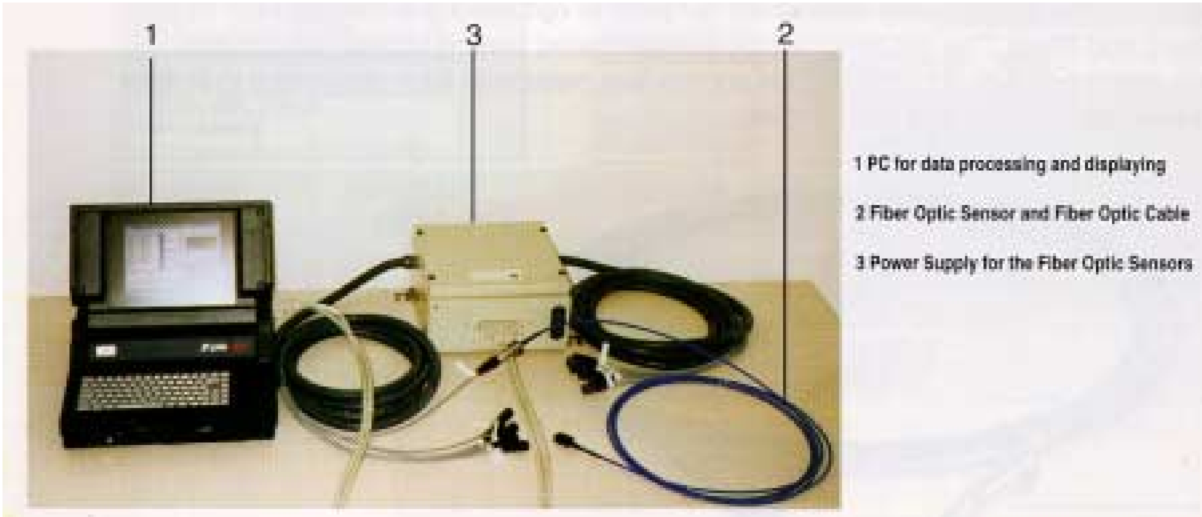


Fig. 11: Fiber optic system with 12 fiber optic measuring channels.

A modern, reliable and efficient system of measuring, displaying, evaluating and storing of large amount of measurement data has been developed to achieve the goal of end winding vibration monitoring (see Figure 11).

The system allows long term evaluation and statistical evaluation of the end-winding vibration behavior.

The operator gets a clear overview and display of process data, machine condition as well as operation recommendations. The operator is not required to interpret long and difficult survey readings.

A Fiber Optic Vibration System is used to measure the vibration of high voltage generator stator end-windings where conventional hardwired transducers cannot be safely mounted. Because they do have an impact on end-winding vibration additional process data mentioned above can be integrated into the system. The extent of such additional data depends on the individual generator design.

The generator end-windings of the stator experience forced mechanical vibrations during operation. The frequency of this vibration is twice the electrical synchronous frequency of the generator.

Therefore stator end-winding vibration occurs at 120 Hz for 60 Hz systems and 100 Hz for 50 Hz systems.

High vibration can lead to loosening of the entire end-winding support system, deterioration of supports, insulation wear, rupture of coil conductors or fatigue cracking of conductors which would require extensive out-of-service repairs.

The Fiber Optic Vibration System provides a data base which is helpful in anticipating generator end-winding vibration problems and predicting future maintenance needs, extending inspection and minimizing down time for maintenance.

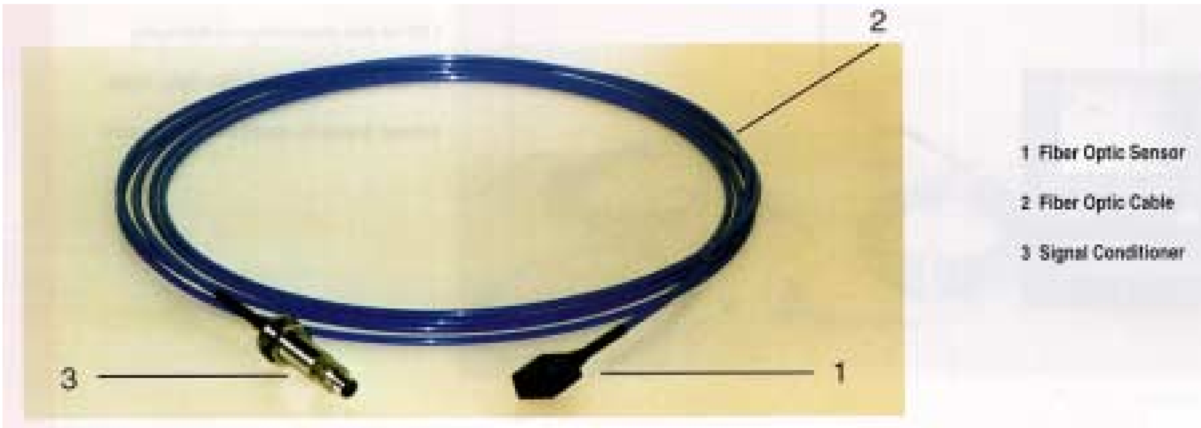


Fig. 12: Fiber optic sensor with fiber optic cable and signal conditioner.

The vibration sensor is shown in Fig. 12. The device is sensitive to vibration motion in one direction. Small size and the electrical isolation of the optical circuit allow the sensor to be mounted directly to the stator coil ends. The sensor head is located at the end of a three single strand multimode optical fiber glass cable. One fiber carries the light generated by the conditioning electronics for illumination. The sensor head returns two optical signals of variable intensity through the remaining fibers.

When the fiber optic accelerometer is subjected to vibration, the force proportional to the absolute acceleration encountered causes angular deflection of an elastic cantilever. The incoming beam is reflected at an angle proportional to the magnitude of acceleration. The conditioning electronics consists of optoelectronic detectors, amplifiers and filters.

The resulting output is a calibrated analog acceleration signal which is interfaced with a computer. An appropriate passband filter attenuates the noise and resonance frequency of the cantilever.

A software system for Windows installed on a PC is used for data processing and display (see Figure 11). It provides outputs proportional to vibration acceleration and displacement. A digital signal processing performs a Fast Fourier Transform (FFT) and the data is presented in the form of a amplitude-frequency spectrum to the operator.

The system can be used as an independent stand-alone system or interfaced with a master computer.

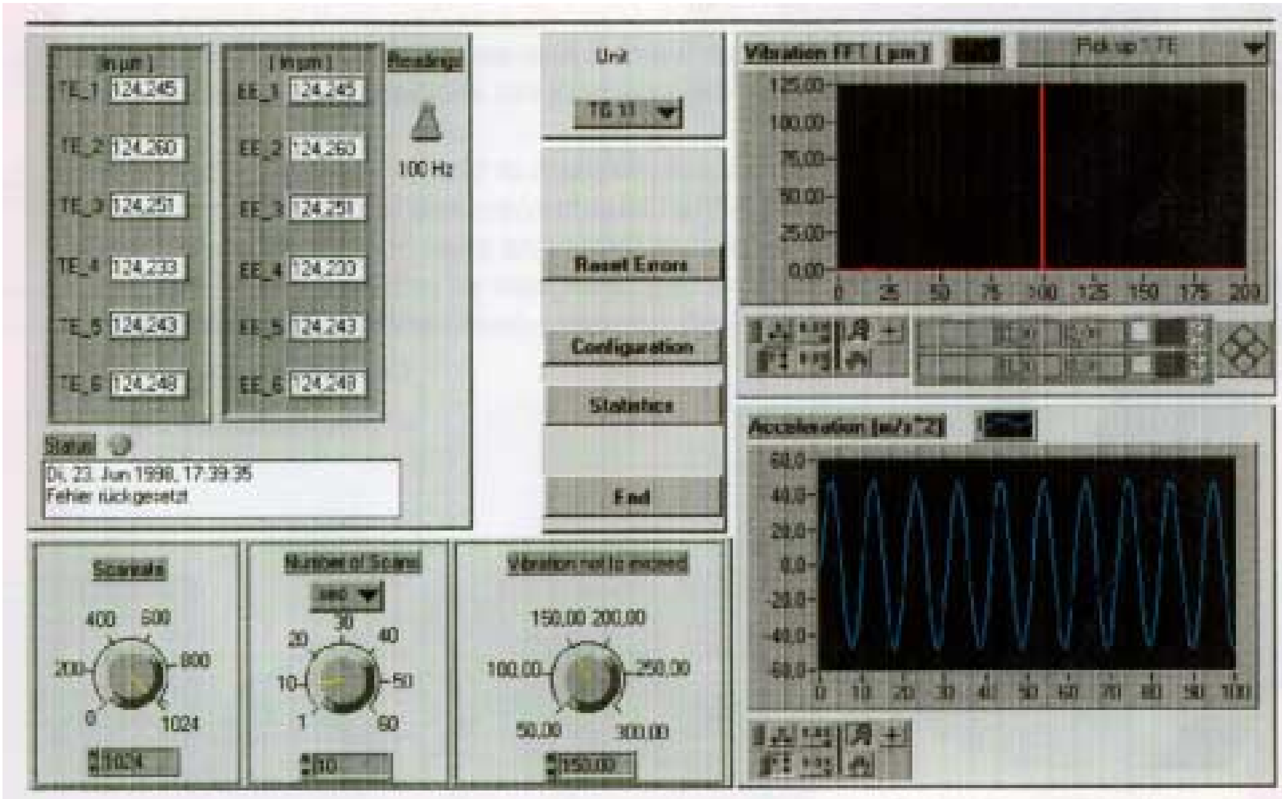
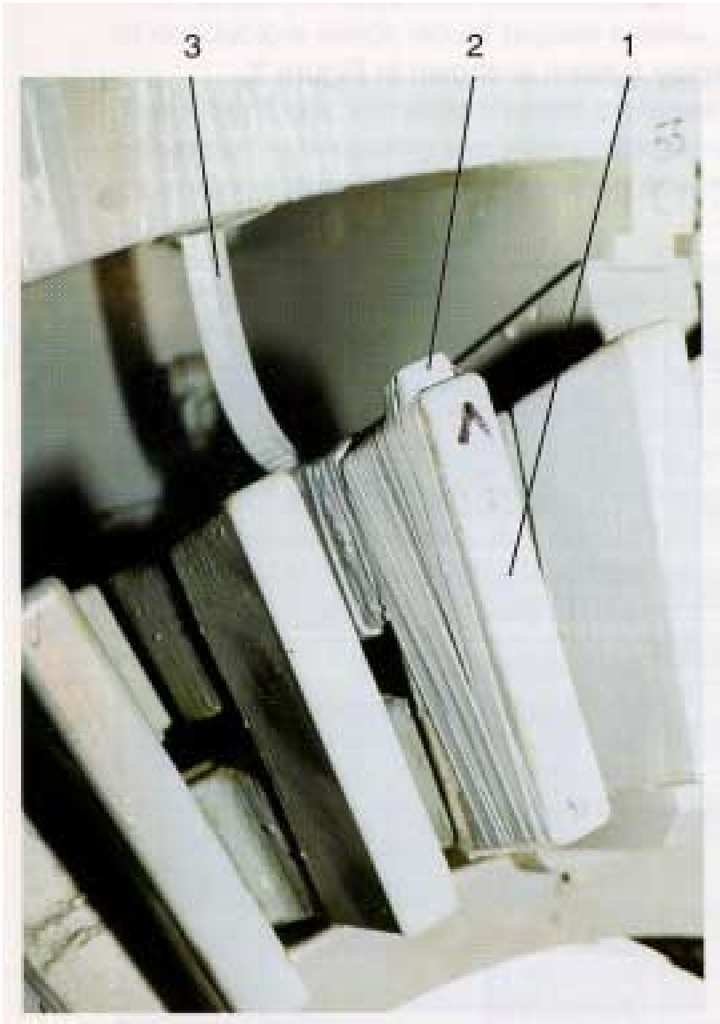


Fig. 13: Front panel of the generator end-winding monitoring system.

The front panel of the data processing and display system is shown in Figure 13. The Fiber Optic system can be installed in generators during an outage and be upgraded to a complete Generator Monitoring System.

Figure 14 shows a Fiber Optic Sensor installed on a high voltage stator end-winding.

Complete systems with 12 Fiber Optic channels, 6 on the turbine end and 6 on the exciter end have been successfully installed in nuclear power plants for measuring, displaying, long term storage and evaluation of the vibration behavior of generator stator end-windings.



1) High Voltage Stator End-Winding

2) Installed Fiber Optic Sensor

3) Routing of the Fiber Optic Cable

Fig. 14: Installed fiber optic sensor.

As mentioned above the monitoring systems which have been developed by the author's company have the ability to be customized to customer needs. Therefore specific parameters which are known to exist on the various manufacturer's equipment can be focused on.

For example direct capacitive measurement of stator bar vibration **in the stator slot** is possible which is a known problem on one specific manufacturer's large water cooled machines where abrasion of the winding by the side ripple spring has been a serious problem.

Another example is the hydrogen purity module which can assist utility customers in preventing energy losses which are known to be several megawatts in some cases which we have encountered.

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