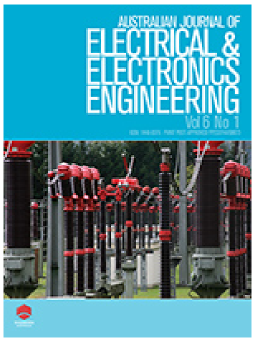

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Hydro generator high voltage stator windings: Part 3 – stator winding slot support systems *

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***SUMMARY**: Appreciable radial electromagnetic forces are exerted on the slot sections embedded in the stator slots (Calvert, 1931). On rapidly loaded hydro generators used for peaking duty, there may be significant differences in axial and radial thermal expansion between quickly heated stator windings and the stator core (which takes a longer time to heat up), imposing significant compressive stresses on winding insulations and possibly causing insulation creep, with resulting reduction in size (Grabner & Kofler, 2001). If a stator winding is allowed to become loose within the core slots, the radial electromagnetic forces will cause winding slot section vibration. The slot radial electromagnetic forces are acting at twice the electrical frequency of the machine, so that loose winding slot sections may experience up to 8.64 million cycles of force in any day of full operation. If even the smallest winding slot section looseness is present, the slot electromagnetic forces will cause radial movement/vibration of winding at double frequency, causing rapid abrasive wear between winding insulation and stator core laminations. If corrective measures are not implemented, rapid insulation failure will result. The slot support systems consisting of slot wedging system, radial slot strips and slot section side packing are designed to prevent in-slot winding vibrations, and to ensure positive slot section grounding for prevention of in-slot partial discharges.*

1 SLOT BAR RADIAL ELECTROMAGNETIC FORCES

The current flow in the conductors of coils/bars, combined with the cross slot leakage flux, produce appreciable radial forces on the slot sections embedded in the stator slots (Calvert, 1931; Grabner & Kofler, 2001).

There are also small circumferential (tangential) forces on the slot sections amounting to only 10% of the radial forces, which are caused by the interaction of the rotor excitation field and the current in the stator coils/bars (Stone et al, 2004). The radial force on the bottom bar is always toward the bottom of the slot. This can be easily explained from first principles by considering figure 2.

The current flow in the stator winding is the consequence of an electromagnetic force produced by the principal magnetic flux. This current produces a leakage flux with a circuit across the slot, down the tooth and back across the slot. The leakage flux direction relative to the direction of the stator current is determined by the Ampere's right hand rule. Since leakage flux is a consequence of the current flow in the stator winding, its relationship to the stator current will remain constant and will follow the reversals of the current for each half of a cycle. Given that cross slot leakage flux increases toward the top of the slot (refer to figure 1), and applying the basic electromagnetic principles for the direction of force exerted on the current carrying conductor in a magnetic field (Fleming' left hand rule), it is obvious that the force on the bottom bar caused by slot leakage flux will always be towards the bottom of the slot irrespective of the main current reversals each half cycle, and its frequency of oscillations will be twice the power frequency (ie. 100 Hz in Australia).

The direction of force on the top bar will be toward the bottom of the slot for the slots containing the coils/bars of the same phase, and can be upward or downward for the slots containing slot sections of different phases. This will depend on the direction of instantaneous currents in the top and bottom slot sections fitted in the same slot. These slot forces are sinusoidal in nature as they are resulting from sinusoidal current flow in the generator coils. When the conductor bars are not wedged and side packed with sufficient tightness in the stator slots,

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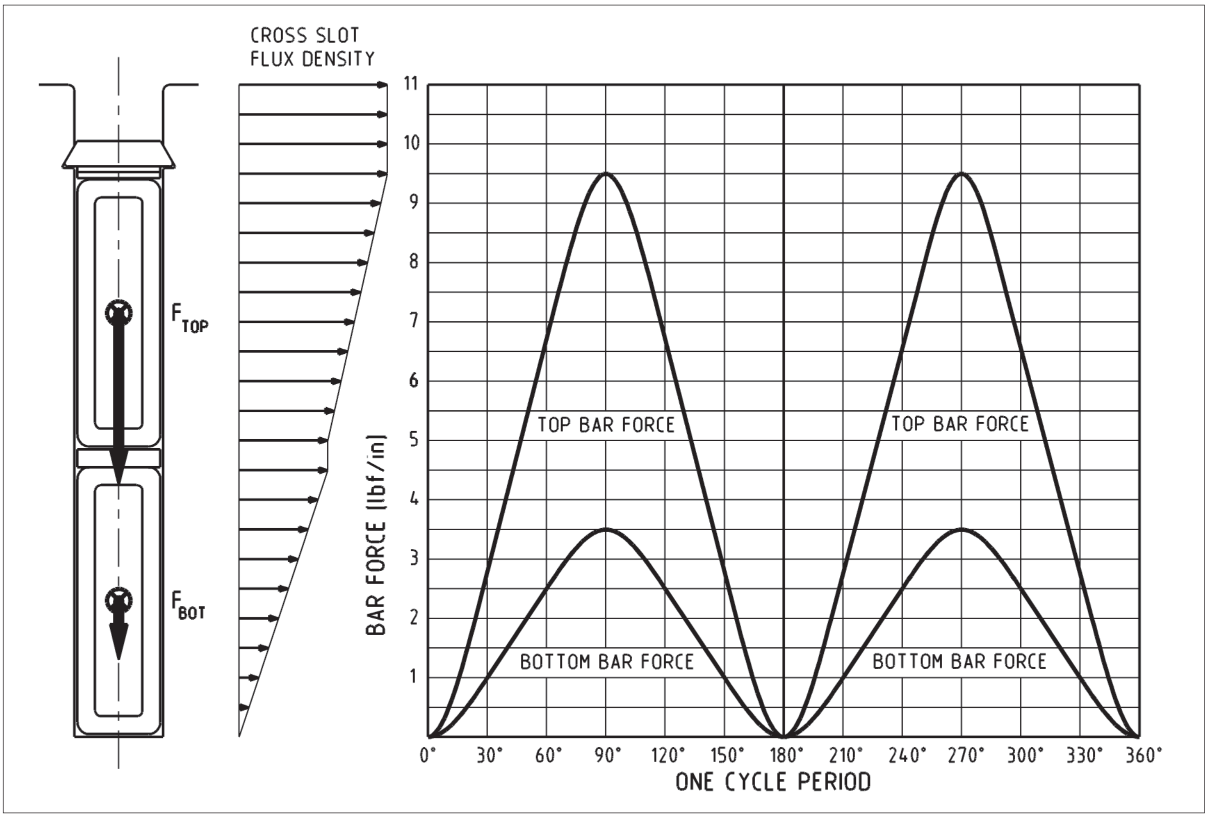


Figure 1: Cross slot flux distribution and resulting radial forces on stator winding bars.

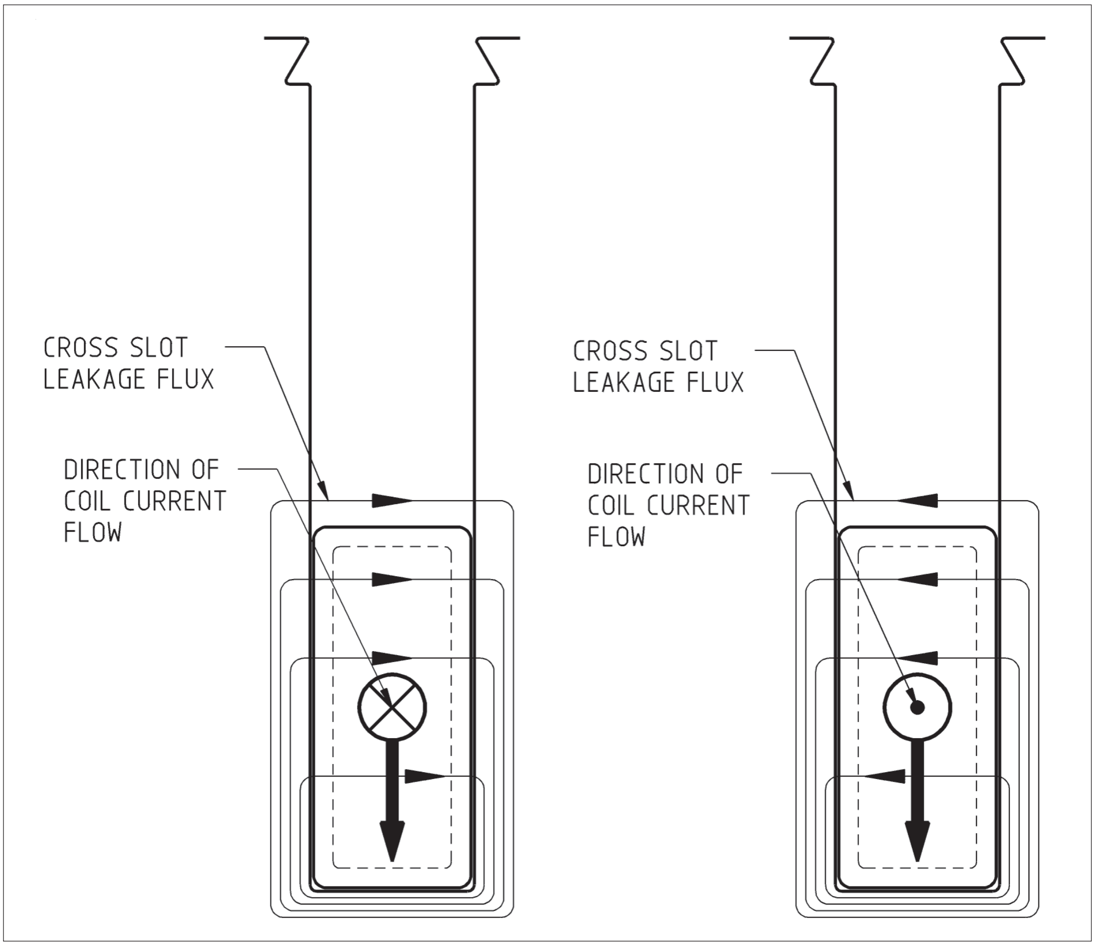
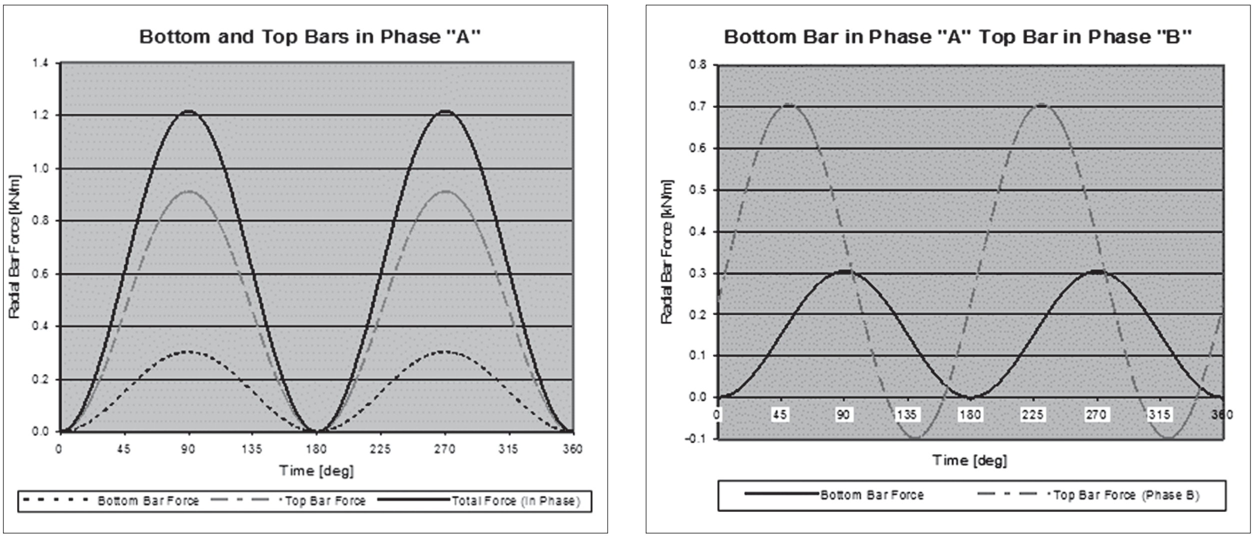


Figure 2: Radial forces on the bottom bar are always toward the bottom of the slot.



(a) (b)

Figure 3: Radial bar force plots for 175 MVA, 11 kV, 10-pole hydro generator – (a) positive forces are toward bottom of the slot, and (b) negative forces are toward top of the slot.

the electromagnetic forces between conductors and the stator magnetic field result in the slot vibration with gradual mechanical erosion and consequent destruction of insulation by repeated mechanical impacts (Mitsui et al, 1981). The resulting bar vibrations are twice that of the machine operational frequency (double frequency vibrations). This is one of the major problems with hydro generators, particularly with modern windings manufactured with thermosetting materials and hard slot cells (resin rich epoxy mica, or vacuum pressure impregnated (VPI) using either epoxy or polyester synthetic resins), which due to their hardness are difficult to tightly side pack into the equally hard stator core slots.

Although the predominant bar forces are radial (up and down in the slot), the unsupported slot section will vibrate in both a radial and a tangential direction, causing insulation erosion on all four sides of the slot section.

From figure 1, it is evident that the top bar force is about three times bigger than the bottom bar force for the slots containing the top and bottom bar in the same phase. This is due to the higher cross-slot leakage flux density toward the top of the slot (nearest to the bore).

The downward slot bar forces impose compressive stress on the ground and strand insulations, while the upward slot bar forces result in compressive load on spring loaded under wedge packing, and shear loading on the slot wedge itself (Bernard & Ferguson, 1995); see figure 3.

The slot forces are normally calculated during the winding design process, for both normal operation and fault conditions, to ensure that the spring loaded slot wedging system exerts adequate pressure on the coils in the slots, and that loading on the slot wedge will not exceed its shear loading capability.

For comparative purposes it is customary to express the sum of bottom and top bar downward forces in the same phase per unit of the stator core length. In the metric system this is expressed as kN per meter of core length (kN/m), and in the imperial system the equivalent unit is pounds of force per inch of core length (lb/in). For standard air-cooled hydro generator windings typical values for bar forces range between 1.0-2.5 kN/m (5.7-14.3 lb/in), and for water-cooled stator windings with higher current densities, the forces may be up to 4.4 kN/m (25 lb/in). These forces are not great, and the slot wedge system with ripple springs will exert the forces many times greater than the sum of the bottom and top bar downward forces.

The following equations from Calvert (1931), Bernard & Ferguson (1995) and Evans (1981) can be used by interested readers to calculate bar forces for their generators. The total bar force is expressed as the sum of bottom and top bar downward forces in the same phase per unit of the stator core length:

$$F_{tot(d)(m)} = 25.088 \times 10^{-7} \times \frac{\left(\sqrt{2} \times I_{ph} \times \frac{T_c}{n_p}\right)^2}{w_s} \quad \left[\frac{\text{kN}}{\text{m}}\right] \tag{1}$$

$$F_{tot(d)(i)} = F_{tot(d)(m)} \times 5.701 \quad \left[\frac{\text{lb}}{\text{in}}\right] \tag{2}$$

where $F_{tot(d)(m)}$ = total downward slot bar forces for the slots containing the bars of the same phase (metric units) [kN/m]; I_{ph} = rated stator current per phase [A]; T_c = number of turns per coil; n_p = number of parallel paths in each phase winding; w_s = width of stator slot [mm]; and $F_{tot(d)(i)}$ = total downward slot bar forces for the slots containing the bars of the same phase (imperial units) [lb/in].

The experience gathered over the last 30 years seems to indicate that in order to achieve a wedge tightness life of 30 years, for normal full load operation the ratio of the radial wedge force per unit of the stator core length should be 8 to 10 times greater than the calculated sum of bottom and top bar downward forces per unit of the stator core length (Bernard & Ferguson, 1995).

2 THE THERMAL SLOT EFFECTS

Due to different thermal expansion coefficients for copper and steel, and quicker heating of rapidly loaded hydro generator windings when compared to the surrounding core iron, axial and radial differential expansion of stator winding must be considered with the slot wedge design (Bernard & Ferguson, 1995). The thermally induced compressive forces on the slot insulation, which may potentially result in "insulation creep", must also be considered (Dalal, 1981).

2.1 Axial slot section expansion

The fact that hydro generators can be brought to full load in a matter of a few minutes from start-up, makes them eminently suitable for peak demand duty. This mode of operation, however, is the most onerous when considering thermally induced differential expansion stresses on the stator winding. With full load applied immediately following the start up, the stator winding copper will heat up rapidly and much quicker than the stator core iron, which needs a few hours to heat up and stabilises at operational temperatures. In addition, the problem is aggravated by the fact that copper's coefficient of thermal expansion is much higher than that of steel (17.6×10^{-6} for copper and 12.0×10^{-6} for steel). Quite often, the number of starts for peaking duty of a machine is considered one of the most important factors when judging a machine windings' life expectation (Bernard & Ferguson, 1995).

Taking, for example, a hydro generator with stator core length of 2500 mm, and assuming winding temperature increase from 20-120 °C in the first half of an hour, while stator core temperature increases from 20-60 °C in the same period of time, the copper total increase in length will be 4.40 mm (2.20 mm per stator core side). The corresponding stator core increase in length will be 1.20 mm (0.60 mm per stator core side). The differential expansion between stator winding and core iron at each end of stator core is therefore 1.60 mm. In order to accommodate this relative movement between the winding and stator core, slip planes are designed between the stator winding and slot wedge system to avoid ratcheting of slot wedges out of the slot wedge grooves.

2.2 Radial slot section expansion and insulation creep

From the above it follows that a similar differential expansion between copper and core iron will occur in a radial direction (depth-wise in the slot). Assuming copper total depth for two bars of 150 mm, and slot depth under wedge of 170 mm, and for the same temperature changes as in the previous axial example, the copper thermal expansion will be 0.264 mm, the stator slot core iron thermal expansion will be 0.0816 mm, giving a differential expansion of slot copper greater by 0.1824 mm when compared to that of slot steel.

If the winding slot sections were rigidly constrained in the slot by the slot wedging system, before the temperature was increased, the differential thermal expansion must be accommodated by a compressive strain in copper and slot insulation.

The compressive stress in copper is given by:

$$\sigma_{CU} = \frac{\Delta L}{L} \times E \quad [\mathrm{Pa}] \tag{3}$$

where σ_{CU} = compressive stress in copper [Pa]; ΔL = increase in length at new temperature [mm]; L = the original length [mm]; and E = Young's modulus of copper = 117×10^9 [Pa].

Using equation (3) and applying above values for the copper thermal expansion, the compressive stress in copper (which will be transmitted through slot section high voltage insulation) is 205.92 MPa. This is a very high stress, far exceeding insulation's compressive strength, particularly at elevated temperatures where epoxy resins experience considerable softening at levels close to the glass transition temperatures. Under such conditions, the insulation would be crushed at the local high points between insulation and steel laminations, resulting in insulation extrusion and creep. The final outcome would be looseness of the slot contents following the generator's cooling to the room temperature.

Although the above example is slightly pessimistic, it clearly illustrates the requirement for some flexibility and "follow up" in slot wedge system design.

Long-term insulation creep at elevated temperatures may occur when much lower continuous pressures are exerted on slot insulation. The slot wedge system follow up mechanism (radial ripple springs) must therefore be designed not to exert pressures much higher than about 1 MPa onto the slot insulations.

3 SLOT WEDGING SYSTEMS

The essential purpose of the slot wedging system is to minimise the tendency of movement and vibration of the winding slot sections in radial direction. The slot wedges are inserted axially into specially profiled slot wedge grooves, which are located at the inside diameter of stator core, facing the air gap.

Slot wedge design received particular attention with the introduction of thermosetting insulation systems (resin rich and VPI) in the early 1960s, where the problems were experienced almost universally with "hard slot sections" becoming loose after short periods of time in operation, when employing the traditional slot wedging systems without the "follow up" characteristic (Evans, 1981; Lyles, 1985a; 1985b).

A properly designed slot wedging system needs to accommodate both insulation physical shrinkage, and rapid thermal expansion, without becoming loose. The "follow up mechanism", to account for both reduced and increased sizes of slot bars, is an essential characteristic of correctly designed and applied slot wedging systems for stator windings with hard slot sections.

With the contemporary slot wedging systems, the follow up mechanism is provided by the use of top radial (under wedge) ripple springs, which accommodate all thermally induced physical changes in the slot bar insulations, whilst maintaining adequate and precisely controllable pressure on the slot bars to prevent vibration and movement due to slot bar electromagnetic forces. In addition, the ripple spring pressure is adjusted to a level that is not too high to cause bar insulation creep at operational temperatures.

As outlined above, correctly designed and applied hydro generator winding slot wedges are of critical importance to prevent slot bar vibration and premature degradation by destruction of winding corona protection layers leading to partial discharge attack, and physical destruction of slot side insulation by abrasion with slot core iron.

A carefully balanced slot wedge design process should produce a slot wedge system with a life expectancy of 30 to 35 years.

3.1 Slot wedge materials

Modern practice almost invariably employs G-10 and G-11 epoxy glass laminates as defined by NEMA LG1 Standard (NEMA, 1989). The laminates are constructed from multi layers of glass cloths made of woven glass fabric impregnated with a specially formulated epoxy resin system, which provides superior thermal resistance and endurance for the given temperature class. In general, G-10 materials can be safely employed with the machines operating at Class B temperature rises (130 °C), and G-11 materials are suitable for the machines operating up to Class F (155 °C) temperature rises. The materials offer excellent mechanical and dielectric properties.

The most important typical mechanical properties for G-11 materials relevant to slot wedge design are:

- Flexural strength at 155 °C = 270 MPa
- Compressive strength = 550 MPa
- Shearing strength perpendicular to glass cloth layers = 120 MPa
- Modulus of elasticity in flexure at 155 °C = 13,000 MPa.

In addition, the materials have excellent dielectric properties, with very high surface and volume resistivity, and typical dielectric breakdown strengths of 18 kV/mm. Physical thermal stability is also very good, as is material homogeneity and very low water absorption characteristics.

Epoxy glass laminates are abrasive and concerns are sometimes raised about the possibility of stator core damage in slot wedge groove caused by epoxy glass slot wedges, due to stator core vibrations. In some instances where hydro generator designed core stiffness may be of concern, epoxy glass wedges covered with abrasion resistant Kevlar may be specified.

3.2 Ripple springs materials and characteristics

Top ripple springs are manufactured from several layers of specially constructed glass fibre roving fabric bonded with a completely cross-linked and cured polyimide resin with high thermal stability. The glass fabric roving content is about 15 times higher in the warp (stress) direction, with total glass content of a corrugated sheet being about 70%.

The top ripple springs are manufactured in 0.8 and 0.9 mm thickness. The full wave cycle is 30 mm, and spring deflection (compression allowance) from a fully relaxed position to a fully compressed state is 1.8 mm (refer figure 4).

Ripple spring characteristic curves (spring deflection versus spring pressure) are provided by ripple spring manufacturers to facilitate slot wedge design (refer figure 5).

From figure 5, ripple spring material thermal stability is obvious, as there is very little difference between room temperature and 120 °C characteristic.

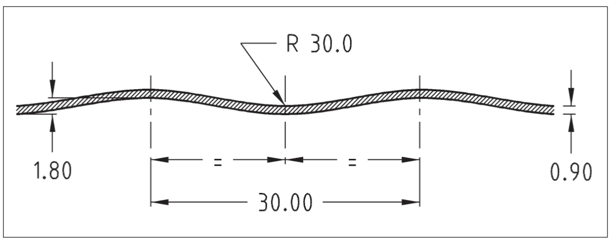


Figure 4: Ripple spring physical dimensions (thickness, wave cycle and deflection allowance).

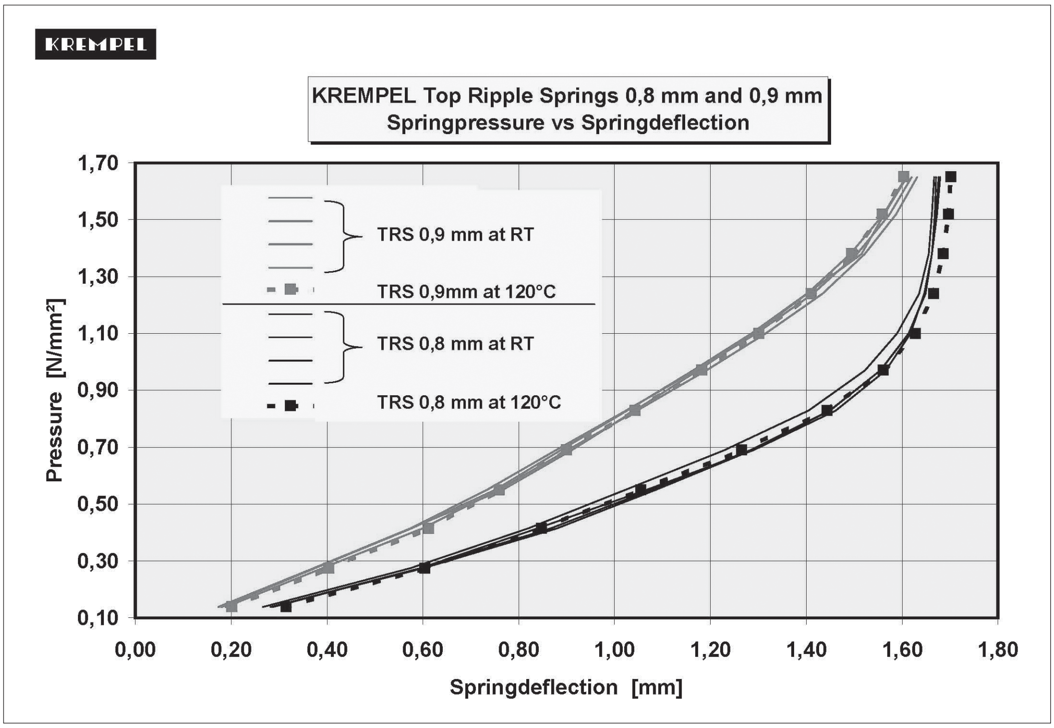


Figure 5: Manufacturer's curves for 0.8 and 0.9 ripple spring deflection versus pressure at room temperature and at 120 °C.

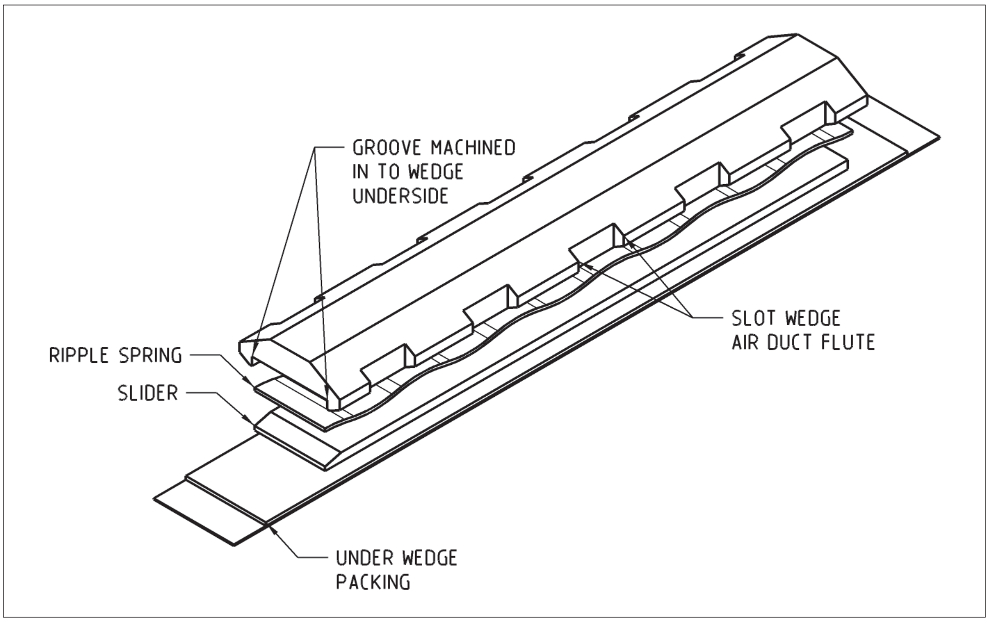


Figure 6: Slot wedge system incorporating grooved wedge, ripple springs and parallel slider for ease of fitting.

3.3 Slot wedging systems

In his practice, the author has implemented and refined the slot wedge system consisting of a specially profiled slot wedge, which accommodates both the ripple spring and sliding wedge tightener in the groove milled into its underside face (refer figures 6 and 7).

The advantages of this system are:

- Provision of adequate radial pressure and follow up pressure to restrain the winding slot section movement for envisaged life of winding.
- Accurate gauging of installed ripple spring pressure.
- Provision for periodic checking of ripple spring deflection.
- There is no driving of the wedge along the wedge groove with minimised potential for shorting of core laminations due to abrasion.
- Improved speed of installation to ensure meeting of requested outage schedules.
- Lowered risk of RSI injuries due to lesser forces needed to drive the parallel slider, when compared to driving the wedge into the wedge groove.

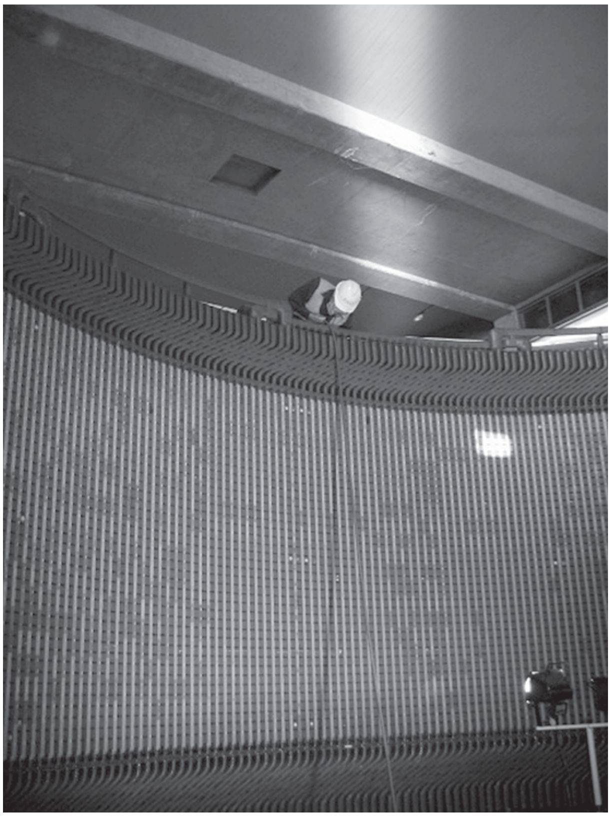


Figure 7: Completed slot rewedging process for 350 MVA hydro generator stator.

Nominal slot wedge design is carried for ripple spring compression of 80%. The ripple springs should be fitted with compression tolerance of no less than 75%, and no more than 90%. Ripple spring compressed to 90% will still allow 0.18 mm for slot content thermally induced radial expansion, which is sufficient for most cases. For the purpose of measuring ripple spring compression during installation, the author has implemented the use of tapered slot wedge fitting gauges.

Two to four slot wedges per slot should be provided with five holes drilled through the slot wedge body, spaced 7.5 mm apart. This allows for periodic checking of the ripple spring compression. Out of five measurements, the difference between the highest and lowest measurement represents the amount of the room required for full ripple spring compression. From here, the remaining percentage of ripple spring compression and slot system forces can be recalculated. Experience shows that up to a 25% reduction in slot wedge system forces may be experienced during the winding operational life. This is due to thermal degradation caused by the slot content shrinkage, as well as some ripple spring force relaxation after many years of exposure to high temperatures.

As far as practicable, the "slip plane" should be provided between the slot wedge system and top winding bar. This is usually accomplished by continuous (as long as possible) lengths of G-11, under the wedge packers.

Slot end wedges (at both ends) should be cemented into the stator core slot wedge groove only, by using special slot wedge locking epoxy. This is to prevent end wedges working loose due to the ratcheting effect caused by repeated slot content (winding) thermal expansion.

4 RADIAL SLOT STRIPS AND SLOT SIDE PACKING

For voltages above 4 kV, all slot radial and side packing must be conductive to enhance functionality of the corona protection system, and ensure positive grounding of coil/bar slot section conductive layers.

The *bottom slot strip* is fitted to even out the somewhat irregular surface at the bottom of the slot caused by lamination stacking, thus preventing random high pressure points on the bottom slot section, which may lead to insulation creep, wear and reduction in thickness. The thickness of the bottom slot strip is kept as small as possible (usually between 0.3 and 0.6 mm) to allow as much room as possible in the slot for coil slot sections (refer to figure 8).

The purpose of the *middle slot strip* is separation of the bottom and top slot sections to prevent partial discharges between the two where they exit the stator core. These strips are usually ground on thickness to very fine tolerances to ensure good electrical and mechanical contact surface between top and bottom slot sections. Middle slot strip thickness is between 3.0 to 4.0 mm for 11 kV machines, 4.0 to 4.5 mm for 13.8 kV machines, and 5.0 to 6.0 mm for 16 kV machines.

The *under wedge slot strips* are used to ensure good load distribution between the slot wedge system and winding slot sections within the slot. Principal slot strip 1.0 mm thick is always used on top of radially inner slot sections, followed up by thinner strips to achieve the required compression of radial ripple springs. For conventional slot wedge systems with radial ripple springs, the required allowance for under wedge packing is between 2.5 and 3.0 mm, and for grooved slot wedge systems with parallel slider the allowance can be reduced to as little as 1.5 mm (refer to figure 8).

The *conductive slot side packing* can be in the form of conductive flat solid side pack, conductive side ripple spring, interference fit conductive silicone rubber moulded directly to one side of the slot section during the coil manufacturing process, or conductive rubber pasted onto slot sections, enclosed into a conductive wrapper and immediately fitted before it sets.

The *conductive flat solid side pack* is more labour intensive to fit since various thicknesses must be combined to achieve the required tightness (usually "no go" with 0.05 mm feeler gauge). It has the best possible thermal conductivity properties, since there are no air voids, but can come loose after some years of service due to slot section content shrinkage

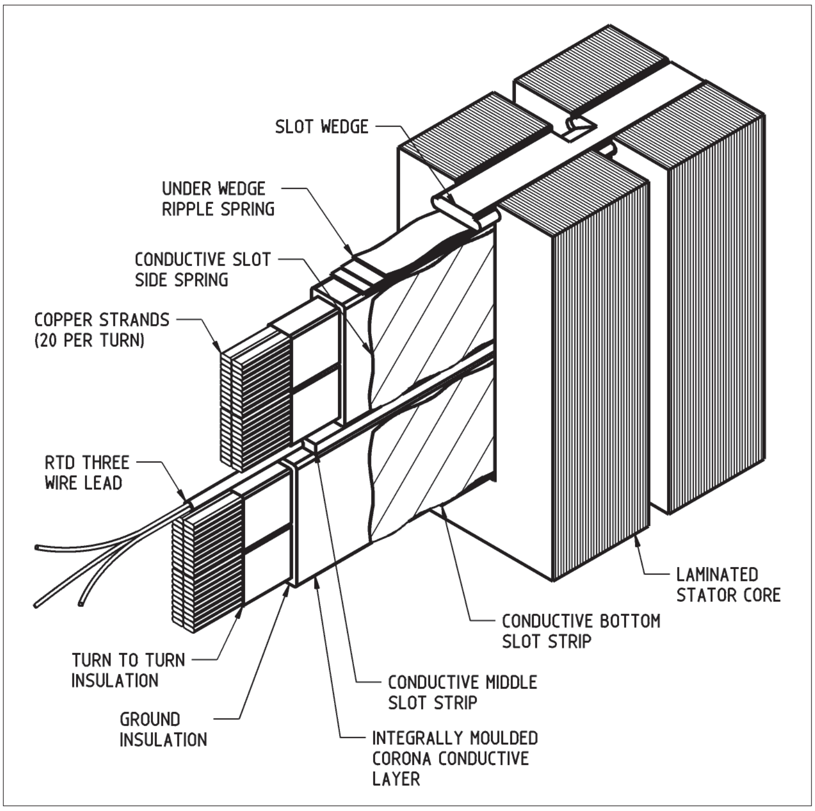


Figure 8: Typical modern hydro generator slot support system designed to eliminate slot section vibration, including spring loaded under-wedge packing and conductive slot side springs.

caused by thermal degradation. The required slot side space allowance is between 0.6 and 0.8 mm.

The *conductive side ripple spring* is easier to fit, and will provide better durability in service (up to 30 years). The known disadvantages are a result of the slightly larger slot side allowance, and the presence of air pockets between material ripples, which are known to impede heat dissipation. This has to be taken into account for a machine where thermal dissipation is a critical design parameter. The required side space allowance for 0.8 mm conductive side ripple spring is 1.2 mm (refer to figure 8).

The *conductive silicone rubber systems* provide excellent grounding of the slot sections and due to the interference fit have some side-wise follow up characteristics.

Common to both multi-turn and bar windings employing thermosetting "hard" slot sections and some form of side packing are difficulties in removing coil sides or bars when it becomes necessary to replace failed coils or bars. Since a lot of effort is used during the winding process to ensure that the slot sections are firmly side packed in the slots, it follows that precisely that slot packing presents an impediment for their removal for repairs. In the author's experience it is rarely possible to properly remove side packed coils or bars without damage. They are invariably discarded and replaced with spare coils and bars. Failed top bars or coil sides are easier to replace, whereas replacement of the bottom coil side or a bar requires a number of top bars or coil sides to be removed.

Generally, bar windings are easier to repair since they are designed to be fitted and joined as half bars. Multi-turn coils, on the other hand, require a lot more skill, given that a number of bottom coil sides cannot be moved. The solution is to cut and remove top halves of those coils, and splice in replacement coil sides by connecting strand by strand and insulating with fully cured mica tapes and air drying epoxy resins. A high degree of skills are required to succeed with such repairs. In the author's experience, to ensure successful repairs, it is best to practice on a few spare coils outside of the machine before attempting the actual repairs.

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Michael was born in Croatia, where he completed his electrical apprenticeship in 1968. Since emigrating to Australia and for the past 32 years he has worked with TGE Energy Services in Perth, Western Australia. TGE Energy Services (formerly F. R. Tulk and Co) is a joint venture between Transfield Services Australia and GE Energy Services (Australia). In the early 1980s, Michael was instrumental in the establishment and development of a high voltage coil and bar manufacturing facility, which now has clients in 22 countries around the world. He is currently engineering manager for all three TGE Energy Services facilities (Perth and Bunbury in Western Australia, and Sydney in New South Wales). Michael's current interests are focused on design of high voltage windings for large electrical machines, applied research on high voltage insulations for rotating electrical machines, and applied engineering for upgrades and uprates of hydro generators.

Michael is a fellow of Engineers Australia and registered chartered professional engineer in Australia.