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A MODEL OF TORSIONAL BEHAVIOR IN THERMO-MECHANICAL BENDING

By

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ABSTRACT

An effort was made to model the behavior of high voltage transmission cables to the torsional loads which occur in the in situ condition of the cable buried in a pipe. Tests were run on a torsion machine in the laboratory which was able to accommodate a reversal of load. The behavior of the cable was studied as it went through cycles of being twisted in opposite directions alternately. Strain in the tape was measured, as were torque and angle. Mathematical models were derived to determine the torsional stiffness of the cable and the distribution of radial stress in the cable insulation. An attempt was made to compare these models to the test results. The torsional stiffness model compared favorably; however, the results for stress could not be analyzed due to the fact that the material constants which were needed could not be obtained. When these constants become available, the mathematical model will undergo further analyzation to determine its feasibility.

INTRODUCTION

We have run a variety of tests on high-voltage buried power transmission cables, in an effort to determine the effects of the thermo-mechanical bending (TMB) phenomenon on these cables. During a normal twenty-four hour load cycling, the cables are subject to thermal expansion and contraction. The cables are buried in pipes, with three cells placed in each pipe. When the cable expands, it flexes or "snakes" due to the fact that the joint casing will not accommodate the excess length of the cable. This motion within the pipe causes the insulating tapes to move relative to one another, forming "soft spots" in the insulation where the thickness may be reduced by as much as 50%. This creates an area of low dielectric strength and results in electrical failure of the cable.

The movement of the cables within the pipe may be modeled in the laboratory by a number of different tests. Among them are the axial compression test, the torsion test, the fatigue test, and the flexure test. This report will concentrate on the cable's response to torsion, which is basic to describing the motion within the pipe.

The cable contains a copper wire core, with wires of about 0.1" diameter being divided into four "pie-shaped" sections. Each section contains 61 individual wires. Two of the sections are insulated with two or three layers of paper tape. These two sections are placed diagonally opposed to each other and separated by the two bare segments. The four sections are then spiralled into a cylindrical

section, and bound by high strength bronze or stainless steel tape.

The insulation of the cable consists of many oiled paper tapes approximately 1" in width and 0.006" in thickness. The taping is done by precision machinery, except at cable joints, where it is done by hand. The dry tape is applied to the core of the cable, the whole thing is vacuum dried, and then the tapes are impregnated with oil. Finally, the insulated core is wound with moisture barrier tapes and metal shielding tapes.

Knowing the construction of the cable, it is easy to understand the problems associated with TMB. The lengthening of the cable produces an axial compressive load on the cable. Work done in the laboratory shows that most of the axial load is taken by the conductor. Due to the configuration of the conductor, which is wound into sort of a helix, the wires of the conductor exert a circumferential force on the insulating tapes surrounding them. Thus, the behavior of the cable in compression is highly dependent on the properties of the insulation.

Similarly, for a cable which is being twisted slightly, the insulating tapes may play a significant role in the behavior of the cable. The cable has different torsional properties in different directions, therefore it is important to be able to reverse the direction of twist in the torsion test. Since the conductor is twisted into a helix initially, twisting in one direction tends to tighten the helix, creating contact forces between the wires. Twisting in the other direction loosens the helix, exerting circumferential forces on the insulation, as in the compression test.

In the first case, where the helix is tightened, we can use the work of George A. Costello in twisted wire cables to model the behavior of the conductor. The insulation may be modeled as a thick-walled tube. Stiffnesses for each of these parts may be found separately, and added together, as there is little interaction between them in this case. However, since the stiffness of the insulation is sufficiently less than that of the copper core, it does not play a significant role in this case.

In the second case, that of loosening the helix, the analysis becomes more involved. The contact forces which in the previous case acted on the wires themselves, now act on the insulation surrounding the wire core. To aid in analysis, we can use the work of Lekhnitskii for an anisotropic thick-walled cylinder under internal pressure.

The test results for torsional analysis of the cable and the actual analysis of results are discussed in the sections following.

PIPE TYPE CABLES

-Single Conductor Detail - Pipe Type Cable

Static Rating 3 PHASE 345 KV
Approximately 550 MVA. (920 AMPERES)

TEST PROCEDURE

Testing of the cable's torsional properties was done on the torsion machine in the laboratory. Many modifications to the machine needed to be made over the period in which the testing was done. Originally, the machine was designed to twist a specimen in one direction only. The grips were such that, as the load was increased, the grip would tighten on the specimen. If the material was twisted in the opposite direction, the grips would loosen up, and the specimen would not be securely held. Due to the fact that the cable has very different properties which depend on the direction of twist, it was necessary to design special grips for the machine that would hold the specimen securely as it was being twisted in either direction.

Next, there was the problem of calibrating the machine. The dial gauge on the machine may be set at a high or a low setting, depending on the load being applied. The high setting runs from zero to nine thousand pounds of torque, and the low setting from zero to nine hundred pounds. Movement of the dial is caused by a Linear Variable Differential Transformer (LVDT) which measures the displacement of an arm which extends from the rotating shaft of the machine. Originally there were two LVDTs attached to the arm, one at the end of the arm for the low setting, and another at an appropriate distance on the arm for the high setting. A third LVDT was added for the purpose of our calibration.

To do the calibration, a small load cell was designed with strain gauges attached to its shaft. First, a compressive load was

applied axially to the load cell on the compression machine in the laboratory. The strain gauges were wired to a Sanborn strip chart recorder, and the loads applied were carefully recorded. The load cell was then placed on the torsion machine, on specially designed arms which were attached to the rotating shaft. One end of the load cell was placed on each arm, so that when the shaft began to rotate, one arm moved relative to the other, creating a torque in the shaft of the load cell. The strain gauges were again wired to the Sanborn, as was the LVDT. The load corresponding to the recorded strain was known from the compression test, and also the length of the arms was known, so the torque could be easily calculated. This torque was found not to be linear with the LVDT displacement, and so a plot was made of torque vs LVDT displacement. In this manner, the torque corresponding to a particular displacement could be read directly off the graph.

Also to be calibrated was a Rotational Variable Differential Transformer, which when wired to a transducer, would read the angle of twist directly. A large angle was laid out on the floor of the laboratory, and the transducer adjusted to read the angle through which the shaft on the RVDT was being moved. During the testing of the cable the RVDT was attached to the cable by an arm which rotated with the cable. A plumb line was fastened to the shaft of the RVDT, holding it in a stationary position while the casing was allowed to rotate freely.

Six samples of cable were tested; three each of the different types of cable. Each was thirty inches long, which is the

maximum length that the machine can accommodate. It is important to make the cable samples as long as possible so as to eliminate end effects. Four strain gauges were attached to the outer tapes of the cable at 90° angles to each other. These strain gauges were wired to a Sanborn along with the LVDT and the transducer for the RVDT. Each cable was twisted a few degrees in each direction initially. This cycle was repeated several times, with the angle of twist being increased slightly each time up to a maximum of about eight degrees. The cable in its in situ condition is unlikely to experience this much twist.

After the testing of the cable on the torsion machine, the outer tapes were removed with strain gauges intact, and a known axial load placed on the tape. The strain was recorded by a Sanborn, and in this manner the tapes were calibrated.

After tests on all six cables were completed, the data was gathered together, and three plots were made for each sample of cable: (1) torsion vs time; (2) angle of twist vs time; and (3) tape force vs time. These plots are included in the following section.

CALIBRATION OF TORSION APPARATUS USING THE LOAD CELL

TORSION TEST APPARATUS

ROTATIONAL VARIABLE DIFFERENTIAL TRANSFORMER

ANALYSIS

Using the work of Costello and Sinha (Ref. 6), each quadrant of the copper center is modeled as a series of concentric layers of twisted wire rope. Although the quadrant has been molded into a pie-shaped segment, it will be approximated by a cylindrical shape. The configuration of each layer of wire is that of a helix. The initial curvatures and twist for each wire are:

$$\kappa_o = 0; \quad \kappa_o' = \frac{\cos^2 \alpha}{r}; \quad \tau_o = \frac{\sin \alpha \cos \alpha}{r}$$

in which α = the original helix angle and r the initial radius of each wire. During loading, α and r may assume new values, α_1 and r_1 . The new configurations are:

$$\kappa_1 = 0; \quad \kappa_1' = \frac{\cos^2 \alpha_1}{r_1}; \quad \tau_1 = \frac{\sin \alpha_1 \cos \alpha_1}{r_1}$$

The equations of equilibrium for this configuration are:

$$\frac{dN}{ds} - N' \tau_1 + T \kappa_1' + X = 0 \quad (1)$$

$$\frac{dN'}{ds} - T \kappa_1 + N \tau_1 + Y = 0 \quad (2)$$

$$\frac{dT}{ds} - N \kappa_1' + N' \kappa_1 + Z = 0 \quad (3)$$

$$\frac{dG}{ds} - G' \tau_1 + H \kappa_1' - N' + K = 0 \quad (4)$$

$$\frac{dG'}{ds} - H \kappa_1 + G \tau_1 + N + K' = 0 \quad (5)$$

$$\frac{dH}{ds} - G \kappa_1' + G' \kappa_1 - \theta = 0 \quad (6)$$

in which N, N', T, G, G' , etc. = external and internal force and moment resultants acting along each wire. X, Y and Z are the applied forces and κ, κ' and Θ are the applied moments.

Consideration of the geometrical relation between α_1 and r_1 yields:

$$\frac{r_1}{R} = 1 + \frac{\tan^2 \left(\frac{\pi}{2} - \frac{\pi}{m} \right)^{1/2}}{\sin^2 \alpha_1} \quad (7)$$

Costello (Ref. 6) derives an equation for torsional stiffness evaluated at the initial state $F = 0$ and $\alpha_1 = \alpha$

$$C^* = r \left. \frac{dM}{d\alpha_1} \right|_{\alpha_1=\alpha} \left. \frac{1}{\frac{d\beta}{d\alpha_1}} \right|_{\alpha_1=\alpha}$$

From Eq. 7 we get

$$\left. \frac{dr_1}{d\alpha_1} \right|_{\alpha_1=\alpha} = - \frac{\tan^2 \left(\frac{\pi}{2} - \frac{\pi}{m} \right) \cos \alpha}{\frac{r}{R^2} \sin^3 \alpha}$$

and from the equations of equilibrium

$$\left. \frac{dG'}{d\alpha_1} \right|_{\alpha_1=\alpha} = \frac{\pi ER^3}{4rR} \left[2 \sin \alpha \cos \alpha - \frac{\cos^3 \alpha \tan^2 \left(\frac{\pi}{2} - \frac{\pi}{m} \right)}{\left(\frac{r}{R} \right)^2 \sin^3 \alpha} \right]$$

and

$$\left. \frac{dH}{d\alpha_1} \right|_{\alpha_1=\alpha} = \frac{\pi ER^3}{4rR} \left[\cos^2 \alpha - \sin^2 \alpha + \frac{\cos^2 \alpha \tan^2 \left(\frac{\pi}{2} - \frac{\pi}{m} \right)}{\left(\frac{r}{R} \right)^2 \sin^2 \alpha} \right]$$

We have also

$$\left. \frac{dN'}{d\alpha_1} \right|_{\alpha_1=\alpha} = - \frac{\sin \alpha \cos \alpha}{r} \left. \frac{dG'}{d\alpha_1} \right|_{\alpha_1=\alpha} + \frac{\cos^2 \alpha}{r} \left. \frac{dH}{d\alpha_1} \right|_{\alpha_1=\alpha}$$

and

$$\left. \frac{dT}{d\alpha_1} \right|_{\alpha_1=\alpha} = - \frac{\cos \alpha}{\sin \alpha} \left. \frac{dN'}{d\alpha_1} \right|_{\alpha_1=\alpha}$$

$$\begin{aligned} \left. \frac{dM}{d\alpha_1} \right|_{\alpha_1=\alpha} = m & \left(\sin \alpha \left. \frac{dH}{d\alpha_1} \right|_{\alpha_1=\alpha} + \cos \alpha \left. \frac{dG'}{d\alpha_1} \right|_{\alpha_1=\alpha} \right. \\ & \left. + r \cos \alpha \left. \frac{dT}{d\alpha} \right|_{\alpha_1=\alpha} - r \sin \alpha \left. \frac{dN'}{d\alpha} \right|_{\alpha_1=\alpha} \right) \end{aligned}$$

The axial and rotational strain of the wire helix are

$$\epsilon = (1 + \xi) \frac{\sin \alpha_1}{\sin \alpha} - 1 \text{ and } \beta = \frac{r}{r_1} (1 + \epsilon) \frac{1}{\tan \alpha_1} - \frac{1}{\tan \alpha}$$

Differentiation yields

$$\left. \frac{d\beta}{d\alpha_1} \right|_{\alpha_1=\alpha} = - \frac{1}{\sin^2 \alpha} + \frac{\cos \alpha}{\sin \alpha} \left[\left. \frac{\cos \alpha}{\sin \alpha} + \frac{d}{d\alpha_1} \right|_{\alpha_1=\alpha} + \frac{\tan^2 \left(\frac{\pi}{2} - \frac{\pi}{m} \right) \cos \alpha}{\left(\frac{r}{R} \right)^2 \sin^3 \alpha} \right]$$

in which

$$\left. \frac{d\xi}{d\alpha_1} \right|_{\alpha_1=\alpha} = \frac{1}{\pi E R^2} \left. \frac{dT}{d\alpha_1} \right|_{\alpha_1=\alpha}$$

This analysis was done for each layer of wire, with m being the number of wires in the layer. The stiffnesses of all the layers

were then added and multiplied by four to give the total stiffness of the wire core. Plots of torque vs angle have been plotted and the slopes of these graphs compared to the calculated results. The stiffness of insulation of the cable is considered to be insignificant in the analysis.

The insulation of the cable is modeled as an orthotropic, elastic thick-walled tube (Refs. 1 and 9). The constitutive equations are:

$$\epsilon_{rr} = a_{11} \sigma_{rr} + a_{12} \sigma_{\theta\theta} + a_{13} \sigma_{zz}$$

$$\epsilon_{\theta\theta} = a_{12} \sigma_{rr} + a_{22} \sigma_{\theta\theta} + a_{23} \sigma_{zz}$$

$$\epsilon_{zz} = a_{13} \sigma_{rr} + a_{23} \sigma_{\theta\theta} + a_{33} \sigma_{zz}$$

$$2 \epsilon_{\theta z} = a_{44} \sigma_{\theta z}$$

$$2 \epsilon_{rz} = a_{55} \sigma_{rz}$$

$$2 \epsilon_{r\theta} = a_{66} \sigma_{r\theta}$$

where

$\sigma_{rr}, \epsilon_{rr}$ = radial stress, strain

$\sigma_{\theta\theta}, \epsilon_{\theta\theta}$ = circumferential stress, strain

$\sigma_{zz}, \epsilon_{zz}$ = longitudinal stress, strain

$\left. \begin{array}{l} \sigma_{\theta z} \epsilon_{\theta z} \\ \sigma_{rz} \epsilon_{rz} \\ \sigma_{r\theta} \epsilon_{r\theta} \end{array} \right\} = \text{shearing stresses, strains}$

a_{ij} = elastic coefficients.

The equilibrium equation for vanishing body forces reduces to a single equation:

$$\frac{\partial \sigma_{rr}}{\partial r} + \frac{\sigma_{rr} - \sigma_{\theta\theta}}{r} = 0$$

and the compatibility equation reduces to

$$\epsilon_{rr} - r \frac{\partial \epsilon_{\theta\theta}}{\partial r} - \epsilon_{\theta\theta} = 0$$

The strain-displacement equations are:

$$\epsilon_{rr} = \frac{\partial u}{\partial r} \quad \text{and} \quad \epsilon_{\theta\theta} = \frac{u}{r}$$

Using these relations we get the following equations for stress

$$\sigma_{rr} = \frac{a_{22}}{a_{11} a_{22} - a_{12}^2} \frac{\partial u}{\partial r} + \frac{a_{11}}{a_{12}^2 - a_{11} a_{22}} \frac{u}{r}$$

and

$$\sigma_{\theta\theta} = \frac{a_{12}}{a_{12}^2 - a_{11} a_{22}} \frac{\partial u}{\partial r} + \frac{a_{11}}{a_{11} a_{22} - a_{12}^2} \frac{u}{r}$$

Plugging into the equilibrium equation, and applying the boundary conditions:

$$\sigma_{rr} = -P_0 \quad \text{at } r = a$$

$$\sigma_{rr} = - \frac{u_0 t E_{\text{shell}}}{b^2 (1 - \nu_{\text{shell}}^2)} \quad \text{at } r = b$$

where P_0 is the applied internal pressure and u_0 the radial displacement, yields the equation (Ref. 2):

$$E_r r^2 [k(k-1) A_1 r^{k-2} + k(k+1) A_2 r^{-(k+2)}] + E_r r [k A_1 r^{k-1} - k A_2 r^{-(k+1)}] - E_\theta [A_1 r^k + A_2 r^{-k}] = 0 \quad (8)$$

where the constants A_1 and A_2 are:

$$A_1 = \frac{P_o}{k E_r a^{(k-1)}} + \frac{a^{-2k} \frac{u_r t}{(1-v^2)} - \frac{P_o}{k E_r a^{(k-1)}}}{(a^{-2k} - kb^{(1-k)})}$$

$$A_2 = \frac{\frac{u_r t}{(1-v^2)} - \frac{P_o}{k E_r a^{(k-1)}}}{(a^{-2k} - kb^{(1-k)})}$$

in which

a = inside radius of insulation

b = outside radius (to metal tapes)

t = thickness of metal tapes (shell)

v = Poisson's ratio

E_r = modulus of elasticity in radial direction

E_θ = modulus of elasticity in circular direction

P_o = internal pressure

$$k = \sqrt{E_\theta / E_r}$$

Using the material constants (Ref. 2),

$$\sigma_r = E_r \epsilon_r = E_r \frac{\partial u_r}{\partial r} \quad \text{and} \quad \sigma_\theta = E_\theta \epsilon_\theta = E_\theta \frac{u_r}{r}$$

From this equation we can get the stresses at the radius $r = b$ and compare to the test results of stress in the outer shell (metal tapes).

The internal pressure, P_o is given by:

$$P_o = X/D$$

where X is the circumferential force and D is the diameter of the wire (Ref. 9). In determining the force X , we again refer to the work of Costello (Ref. 6). He assumes that the individual wires in the wire rope are not subjected to external bending moments per unit length, that is, $K = K' = 0$. The tension is assumed to be constant along the length of the wire. The equilibrium equations (Eqs. 1-6) then become:

$$-N' r_1 + T \kappa_1' + X = 0 \quad (9)$$

$$Y = 0 \quad (10)$$

$$Z = 0 \quad (11)$$

$$-G' r_1 + H \kappa_1' - N' = 0 \quad (12)$$

$$N = 0 \quad (13)$$

$$\theta = 0 \quad (14)$$

The total axial force and total twisting moments are given by:

$$F = m (T \sin \alpha_1 + N' \cos \alpha_1)$$

and

$$M = m (H \sin \alpha_1 + G' \cos \alpha_1 + T r_1 \cos \alpha_1 - N' r_1 \sin \alpha_1)$$

If we assume that $G' = 0$ and that the resultant axial force $F = 0$, we get four equations in four unknowns:

$$T \sin \alpha_1 + N' \cos \alpha_1 = 0$$

$$-N' r_1 + T \kappa_1' + X = 0$$

$$H \kappa_1' - N' = 0$$

and
$$M = m (H \sin \alpha_1 + T r_1 \cos \alpha_1 - N' r_1 \sin \alpha_1)$$

The applied twisting moment, M , is known and we can solve for the unknowns H , T , N' and the circumferential force X . The

analysis is done for each layer separately. The force exerted by the innermost layer is X_1 , that exerted by the next layer is X_2 which is the sum of the force x_1 of the first layer and the force x_2 of the second layer, and similarly for the third and fourth layer so that

$$X_4 = x_1 + x_2 + x_3 + x_4.$$

Unfortunately, the material constants for the insulation are not available at this time, so that the stresses calculated from the previous equation cannot be compared to the actual test results.

RESULTS

Using Costello's equation (Ref. 6) for torsional stiffness, we get a stiffness of the copper wire core of $20,017 \text{ lb-in}^2/\text{rad}$. The slopes on the plots of torque vs angle were difficult to interpret since the cable went through so many loading cycles, however, the range was from 10,000 to 40,000 $\text{lb-in}^2/\text{rad}$ which is very satisfactory considering that many assumptions were made in the mathematical model. These results confirm our original assumption that the insulation contributes little to the torsional stiffness of the cable.

In calculating the internal pressure exerted by the core on the insulation, we get an expression for P_o in terms of the twisting moment, M

$$P_o = 6.632 M$$

This expression can then be used in Eq. 8 to calculate the stress in the insulation at $r = b$. The data, in graphical form follows.

CONCLUSION

Although much more testing needs to be done, it appears that Costello's model for twisted wire rope is a fairly good approximation for the cable behavior. Many assumptions needed to be made to fit the cable to the mathematical model. The copper wire core, for instance, is made up of four separate segments, each of which is made up of four layers of wire. The segments are pressed into a pie-shape rather than the cylindrical shape that the model assumes. The wires of each segment are wrapped in a helical configuration, and each segment in turn is of a helical shape in the core of the cable. In the model, the segments of the core were assumed to be independent of each other; however, in reality, they do interact.

The material constants of the cable were difficult to obtain, particularly those of the paper tape insulation. An effort was made to determine the constants through laboratory tests; however, the results were not satisfactory, and more work needs to be done in this area before the test results can be analyzed.

The results of the six tests that were run were fairly consistent. The second type of cable seemed to exhibit larger values of strain for a particular angle of twist than did the first type, on the whole. The different types of cables have slightly different properties.

Since the maximum length of cable that could be accommodated was thirty inches, the end effects are likely to play a part in the

behavior of the cable. However, in this case, we have chosen to ignore them. This seems to be a reasonable assumption, as far as the test results are concerned.

Many improvements are being made in the equipment in the laboratory. The torsion equipment is being interfaced with the computer so tests can be run and the results obtained rapidly. In this manner, it is hoped that many more tests can be run in a short amount of time, and the results compared.

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