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Experimental and numerical studies of mono-strand anchorage

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Abstract. This paper deals with an experimental and numerical study of a mono-strand wedge anchor head mechanism. First, the experimental program is presented and monitored data such as wedge slippage, anchor deflection and strain distributions along external peripheral surfaces of the anchor head are presented and discussed. In accordance with the experimental set up, these data concern only the global behaviour of the mechanism and cannot provide valuable information such as internal stress-strains distributions, stress concentrations and percentage of yielded volume. Therefore, the second part of this paper deals with the development of an efficient numerical finite element model capable of providing mechanism of the core information. The numerical model which includes all kinematics/material/contact non-linearities is first calibrated using experimental data. Subsequently, a numerical study of the anchorage mechanism is performed and its behaviour is compared to the behaviour of a slightly geometrically modified mechanism where the external diameter has been increased by 5 mm. Finally, different topics influencing the anchorage mechanism behaviour are addressed such as lubrication and wedge shape.

Key words: anchorage; contact finite element; friction; strand; wedge; yielding.

1. Introduction

In recent years, the use of unbounded tendons for strengthening existing bridge structures, or for constructing new ones, has gained wide popularity. These unbounded tendons are either external or internal to the concrete section and their use has the advantage of eliminating the grout injection phase. In addition, the popularity of external prestressing is related to the accessibility of the prestressing and to the possibility of cable replacements or additions.

External prestressing is a construction technique recognised to increase the quality and durability of structures (SETRA 1990). However, as a consequence of the lack of a continuous bond between the tendons and the concrete, the tendons in externally prestressed bridges are only linked to the structures at specific points like the deviators and the anchorage mechanisms. Those mechanisms are therefore severely stressed since they must alone take the prestressing force. If an anchor

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mechanism fails, the concrete surrounding the tendons cannot contribute to transfer the prestressing force and this situation could result in a sudden structure failure (Bastien *et al.* 1991). Reliable behaviour of such anchorage mechanisms is therefore very important and related to the structure security.

The study of the behaviour of wedge anchor mechanisms is the purpose of a joint research program between Laval University (Québec, Canada) and the Laboratoire Central des Ponts et Chaussées (France). The research program deals with mono- and multi-strands anchor mechanisms. The present paper deals with the behaviour of a specific mono-strand anchorage. Its behaviour is examined through experimental and numerical studies. The paper briefly presents the experimental tests performed and the theory associated with the numerical analysis of the various anchorage contact interfaces, namely the wedge-anchor head and anchor head-plate interfaces. Furthermore, a comparison between experimental and numerical results shows that a good prediction of mono-strand anchor head behaviour can be achieved with the proposed numerical model.

In the first part of the numerical study, a calibration procedure, based on experimental results, leads to the determination of frictional coefficients acting at the diverse interfaces (wedge-anchor head interface and anchor head-plate interface). Using these coefficients, a mono-strand anchor mechanism and a modified one, where the external diameter of the anchor head has been increased by 5 mm, are examined. In particular, the numerical results show that yielding into the anchor heads spreads from the internal (conical) to the external surface of the anchor heads for load levels less than 80 percent of the ultimate strength of the tendon. This load level represents the maximum prestressing force allowed, at jacking, by many design codes (CAN/CSA-S6-88 1988, BPEL 1990, AASHTO 1994).

2. Experimental program

The experimental tests were performed under the supervision of the Laboratoire Central des Ponts et Chaussées. The details of the experimental procedure can be found in Bastien (1992). Briefly, they involve submitting mono-strand anchor heads to a load corresponding to 80 percent of the ultimate strength of the tendon (0.80 F_u) using an universal tension machine.

A mono-strand anchorage is an anchoring device for one prestressed strand which consists mainly of three components: an anchor head, an anchor plate and a wedge. The device under the present study is described in Fig. 1. The anchor head is a 55 mm height cylindrical metal piece with a central conical hole. The slope of the internal anchor head surface is 7 degrees from the vertical. The anchor head is set on an anchor plate at the concrete surface for anchoring purposes. The wedge is composed of three pieces linked together with a metallic ring at the upper portion. To anchor a prestressed strand, a wedge surrounding a strand is introduced into the central portion of the anchor head. A notched internal surface of the wedge grips the strand while they are both (wedge and strand) wedged into the central portion of the anchor head.

While experimental tests were performed on a 45 mm diameter anchor head, numerical analyses were performed on both 45 mm and 50 mm diameter anchor heads. The 50 mm diameter anchor head is referred to herein as the geometrically revised anchor.

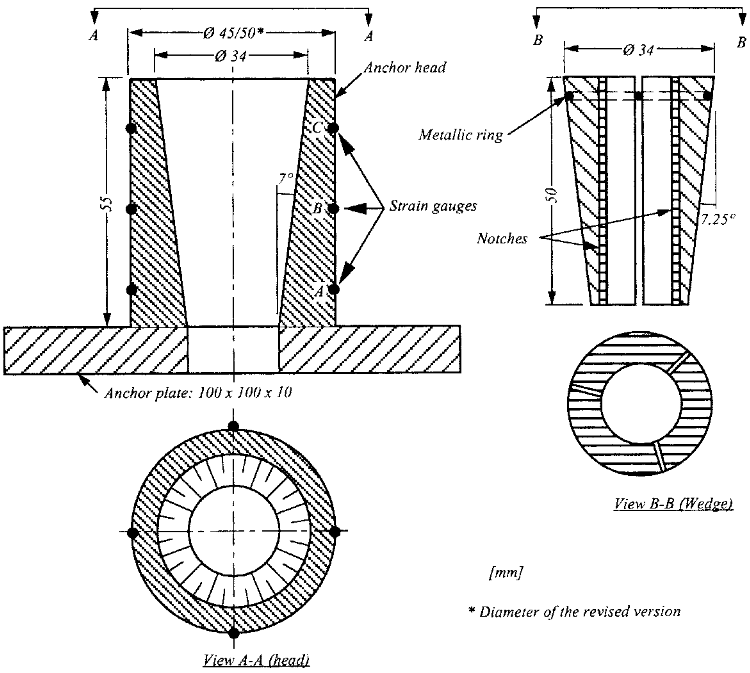


Fig. 1 Geometrical properties of the anchorage mechanism

2.1 Definition of material properties

The geometrical and mechanical characteristics of the anchorages under study have been examined. The geometry of the pieces before and after loading has been measured using a three dimensional table digitalizer especially design for this task. This investigation was done in order to assess if permanent strains (plasticity) were developed into the anchorage. It was not possible to extract samples from a mono-strand anchor head to perform tensile tests, and thus no elastic limit has been evaluated directly from such anchorage mechanisms. However, samples extracted from similar multi-strands anchor heads have been used to assess the material elastic limit. Brinell and Vickers hardness tests were also performed on the different components.

In accordance with the previous material tests, Table 1 shows the characteristics used to perform the numerical analysis. An elastoplastic behaviour is considered for the anchor plate and the anchor head while a perfectly elastic behaviour is considered for the wedge. The latter assumption is mainly governed by the very hard surface of the wedge due to heat treatment.

2.2 Instrumentation

In order to assess the global behaviour of the mono-strand anchorage under study, different measurements were monitored. In particular, strain gauges were used to evaluate the state of strain developed at the external cylindrical surface of the anchor head under loading. For each anchor tested, 24 strain gauges were distributed crosswise over three levels on the external surface. These levels are positioned at 15 mm, 30 mm and 45 mm from the bottom face of the anchor head. Two gauges are installed at each measurement point monitoring the axial and circumferential strains. To

Table 1 Mechanical properties of the anchorage mechanism

Components	F_y (MPa)	F_u (MPa)	ε_u (%)
Anchor head	400	750	16
Anchor plate	270	400	22
Wedge[1]	400	–	–
E=2.0·10^5 MPa, ν=0.30			

[1]These components have been considered elastic.
F_y: Yield strength, F_u: Ultimate strength, ε_u: Strain at F_u

evaluate the slippage of the wedge into the anchor head, mechanical gauges were used. They were set between the bench machine plates assessing their relative displacement and therefore the wedge slippage. Fig. 2(a) presents a view of a mono-strand anchor head set in place on the universal testing machine with the appropriate instrumentation.

2.3 Tests

The experimental tests were performed on an universal testing machine. The strand and wedge have been replaced by an equivalent component referred to herein as an equivalent wedge. Bastien (1992) has shown that, under loading, the strand and wedge act together and therefore their behaviour can be accurately represented by a monolithic component presenting an outer conical shape similar to a wedge (see Fig. 2b). Furthermore, the utilisation of an equivalent wedge greatly simplifies the mesh for the numerical modeling. Therefore, for the numerical analysis, an equivalent wedge has been introduced into the anchor head central hole. This equivalent wedge was considered to have perfectly elastic behaviour.

The mono-stand anchor heads were loaded by monotonous increasing steps of 0.1 F_u till 0.8 F_u

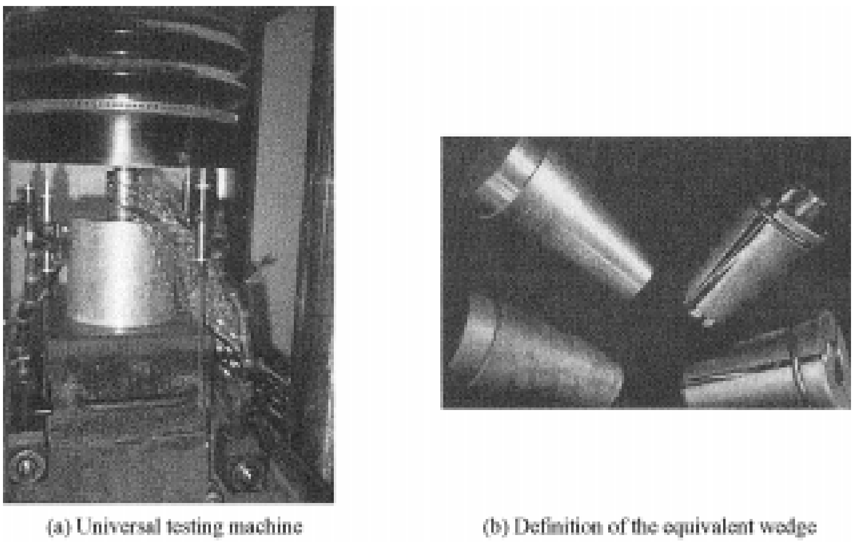


Fig. 2 Details of the experimental setup

Table 2 Experimental results

Strain gauge (see Fig. 1)	Position from bottom (mm)	ε_z (10^{-3})	ε_θ (10^{-3})
A	15	–0.70	1.10
B	30	–0.60	1.60
C	45	–0.60	1.80

Slippage of the wedge at 0.8 F_u: 0.7 mm

was reached. The maximum load was maintained from 5 to 15 minutes and then decreased by similar steps. The loading was applied through compression, i.e. the upper bench plate pressing down on the equivalent wedge top surface (see Fig. 2a).

2.4 Results

Table 2 shows typical strain results obtained at loadings of 0, 80 F_u. Axial (ε_z) and circumferential (ε_θ) strains are presented for each monitoring level position. These strains represent the average of the data of four strain gauges at each level. As expected, axial strains are negative (compression due to loading) while the circumferential strains are positive indicating a widening of the anchor head.

3. Numerical program

To this day, no theoretical approach or empirical formulation exists to adequately predict the behaviour of mono-strand anchorage mechanism. This situation is mainly due to the interaction of complex phenomena involved between component parts and their evolutive mechanical and geometrical properties under loading. It's the aim of this paper to present a numerical approach taking into consideration all of the above.

3.1 Requirements of the model

With respect to the experimental results, it has been shown that moderated slipping occurs between the wedge and anchor head as well as between the anchor head and the anchor plate. The literature dealing with contact and frictional coefficients acting between anchorage mechanism components is very sparse. Bearing in mind that these coefficients must be adequately evaluated since they play an important role on the behaviour of the anchor mechanism, an adequate numerical model must take into account the contact phenomenon with proper contact coefficients.

In addition, the experimental results show that permanent small strains occur in the anchor head but not in the wedges. This observation leads us to consider yielding with isotropic hardening in the numerical model such that elastoplastic behaviour can be evaluated (stresses and strains) during quasi-static analysis using the appropriate von-Mises yield criterion.

Generally, such mechanisms should be studied with a full three-dimensional model, especially for the cases where real wedge-tendon components are analysed. In such cases, an axisymmetric model cannot be used due to the gap between the three parts of the wedge (see Fig. 1). However, taking advantage of the equivalent wedge-tendon component, load pattern and material properties, the study of the present mono-strand anchorage mechanism could be accomplished using equilibrium equations developed in a classical axisymmetric coordinates system. If should be noted that under experimental loading, the use of the real wedge-tendon components leads to a complete yielding of the anchor head.

3.2 Proposed model

All the development referring to this model can be found elsewhere (Marceau and Fafard 1993) and is mainly based on the evaluation of a virtual work principal using a total Lagrangian formulation in an axisymmetric coordinate system, such as:

$$W(\underset{\sim}{u},\delta\underset{\sim}{u})=2\pi\int_{\Omega}\delta\underset{\sim}{\varepsilon}\cdot\underset{\sim}{S}rd\Omega+W_{\text{ext}}(\delta\underset{\sim}{u})+W_{c}(\underset{\sim}{u},\delta\underset{\sim}{u})=0, \tag{1}$$

where

$$W_{\text{ext}}(\delta\underset{\sim}{u})=-2\pi\left[\int_{\Omega}\delta\underset{\sim}{u}\cdot\underset{\sim}{F}_{v}rd\Omega-\int_{\Gamma_s}\delta\underset{\sim}{u}\cdot\underset{\sim}{F}_{s}rd\Gamma_{s}\right] \tag{2}$$

represents the external virtual work and

$$W_{c}(\underset{\sim}{u},\delta\underset{\sim}{u})=-2\pi\int_{\Gamma_c}\delta\underset{\sim}{u}\cdot\underset{\sim}{F}_{c}rd\Gamma_{c}, \tag{3}$$

the virtual work performed by external and contact forces with

$$\underset{\sim}{u}=\underset{\sim}{\bar{u}}\ \text{ on }\ \Gamma_{u};\ \ \Gamma=\Gamma_{u}\cup\Gamma_{s}\cup\Gamma_{c};\ \ \Gamma_{s}\cap\Gamma_{c}=\varnothing;\ \Gamma_{s}\cap\Gamma_{u}=\varnothing;\ \Gamma_{u}\cap\Gamma_{c}=\varnothing\,. \tag{4}$$

In Eq. (1) $\delta\underset{\sim}{\varepsilon}$ and $\underset{\sim}{S}$ are the virtual Green-Lagrange strain vector and the second Piola-Kirchhoff stress vector, respectively. In Eq. (2), vectors $\underset{\sim}{F}_{v}$ and $\underset{\sim}{F}_{s}$ are the volume and boundary loads applied on Ω and Γ_s respectively. Since the stress-strain relation must agree with the material elastoplastic behaviour, the second Piola-Kirchhoff stress vector is expressed in an incremental form as:

$$\underset{\sim}{S}=\underset{\sim}{S}_{0}^{p}+\boldsymbol{C}_{ep}\underset{\sim}{\varepsilon}_{p}^{i} \tag{5}$$

where $\underset{\sim}{S}_{0}^{p}$ corresponds to the total stress vector at the end of the step *p*, $\underset{\sim}{\varepsilon}_{p}^{i}$, the incremental Green-Lagrange strain vector between step *p* and the current iteration *i*. The elastoplastic constitutive matrix $\boldsymbol{C}_{ep}$ is given by Crisfield (1991).

Eq. (3) deals with the evaluation of the contact forces $\underset{\sim}{F}_{c}$ on Γ_c. According to the requirements of the model, the treatment of contact phenomena has been performed using a continuum-based finite element formulation (Laursen and Simo 1993, Laursen 1994, Klarbring 1995). The contact detection algorithm has been done using the slave-master approach. The frictional (Coulomb law) and contact laws have been solved using a regularized form obtained with a penalty method and integrated (frictional law) via a standard return mapping algorithm (Marceau 2001).

In order to solve the nonlinear equations, the standard Newton-Raphson iterative technique is used. The computation of the incremental form has been presented by Marceau and Fafard (1993). It should be noted that the presence of frictional contact in Eq. (1) leads to a non-symmetric system.

To preserve the quadratic rate of convergence of the Newton-Raphson technique, a consistent contact tangent matrix has been used for both sticking and sliding cases. Finally, the variational principle is used to derive the finite element equations:

$$W=\sum_{e=1}^{NE}=\delta\underset{\sim}{u}\cdot(\underset{\sim}{R}-K_T\Delta\underset{\sim}{u})=0 \tag{6}$$

where *NE* is the total number of elements.

4. Mono-strand anchorage application

In this section, a numerical study is performed on the mono-strand anchorage mechanism described previously. The aim of the study is to evaluate the internal behaviours of the mechanism such as localisation, distribution, and percentage of plastic strain in each component of the mechanism and its capability to be used under particular conditions. Finally, the results of these analyses lead us to a new version of this mechanism, the increased performance of which is verified in comparsion with the forme version.

4.1 Discretization

The discretization has been performed using an axisymmetric form of the entire anchorage mechanism. The mesh and prescribed boundary conditions are shown in Fig. 3. For the mesh, a total of 620 serendipity eight-noded elements (quadrilateral) were used while the contact zones were discretized with 80 quadratic three-noded elements. Excluding degrees of freedom affected by boundary conditions, the discretization led to a non-symmetric matrix system of 4007 unknowns.

The nonlinear problem is solved with the classical Newton-Raphson technique using 3×3 Gauss integration rule for the quadrilateral and 5 Gauss points for the integration of the contact conditions. The loading procedure has been performed with variable incremental steps applied simultaneously on the top of the wedge. To ensure convergence at extensive non-linearities, the arc-length method has been used.

4.2 Calibration

In this section, calibration of the proposed numerical model is performed in order to estimate the frictional coefficients acting at the wedge-anchor head and anchor head-anchor plate interfaces. Each of these interfaces has its proper frictional coefficient and are noted μ_{wh} (between head and wedge) and μ_{hp} (between head and plate).

According to the experimental procedure adopted by Bastien (1992), a modified anchor plate has been used in order to minimize its influence on the behaviour of the anchorage. In the experiment tests, a thick wedge grade-equivalent material has been used for the anchor plate. Consequently, for calibration purposes, frictional coefficients acting at the wedge-anchor head and head-anchor plate interfaces were considered equal and noted μ_{cal}. First, numerical analyses were performed using different frictional coefficients ranging from 0.0 to 0.12. Numerical results such as slippage of the wedge and axial/circumferential strains developed at the external cylindrical surface of the anchor head were compared to experimental results shown in Table 2. Finally, the influence of the frictional

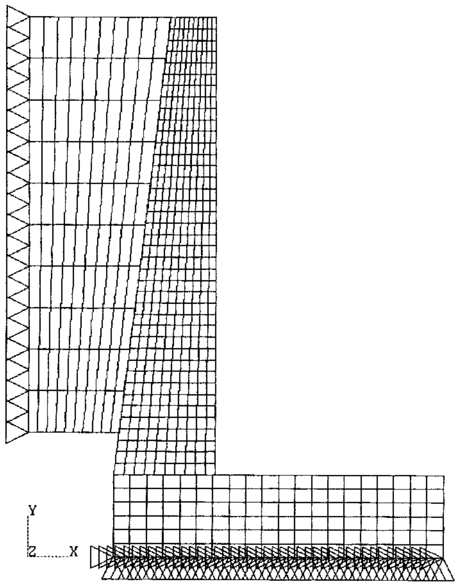


Fig. 3 Mesh and boundary condition of the anchorage mechanism

coefficient at the anchor plate and anchor head interfaces is examined by considering the standard anchor plate associated with the anchorage mechanism.

4.2.1 Frictional coefficient between wedge and anchorage head

Due to the hardened surface of wedges, it is expected that the frictional coefficient between the wedge and the anchor head will be small. According to the experiments, the wedge slippage at 0.8 F_u is in the order of 0.7 mm (see Table 2). Fig. 4 shows that the numerical results agree with the experiments for a frictional coefficient μ_{cal} ranging from 0.10 to 0.12. At design prestressing load (0.8 F_u), Fig. 5 shows that strains obtained from numerical analysis correspond to experimental values for frictional coefficients near 0.12. In particular, it can be seen that a small variation of the frictional coefficient from 0.08 to 0.10 decreases the circumferential (ε_θ) and axial (ε_z) strains by approximately 70%. Also, the circumferential strains are more influenced by such variation than the axial strains. The last three figures show the great importance of the frictional coefficients on the behaviour of such mechanisms. Especially, Fig. 4 shows that yielding is influenced by the frictional coefficient in such a way that the design prestressing load cannot be reached for particular values of these coefficients. According to the above results, the frictional coefficient μ_{cal} can be taken as 0.11. Therefore in further sections of this paper, μ_{wh} (frictional coefficient between the wedge and the anchor head) is taken as 0.11.

4.2.2 Frictional coefficient between anchorage head and plate

In the experimental procedure, a modified rigid anchor plate has been used (in practice the anchor plate associated with the mechanism should be used). Therefore, one can assume that the frictional coefficient acting between the anchor head and the standard anchor plate (μ_{hp}) can vary from 0.20 to 0.30 which represents very acceptable values for usual steel. Fig. 6 shows that μ_{hp} has no significant influence on the slippage of the wedge. In addition, Fig. 7 shows that circumferential and axial strains are not really influenced by a variation of μ_{hp}. According to previous results, a

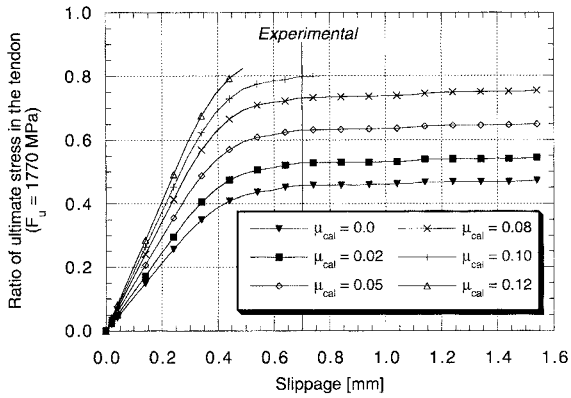
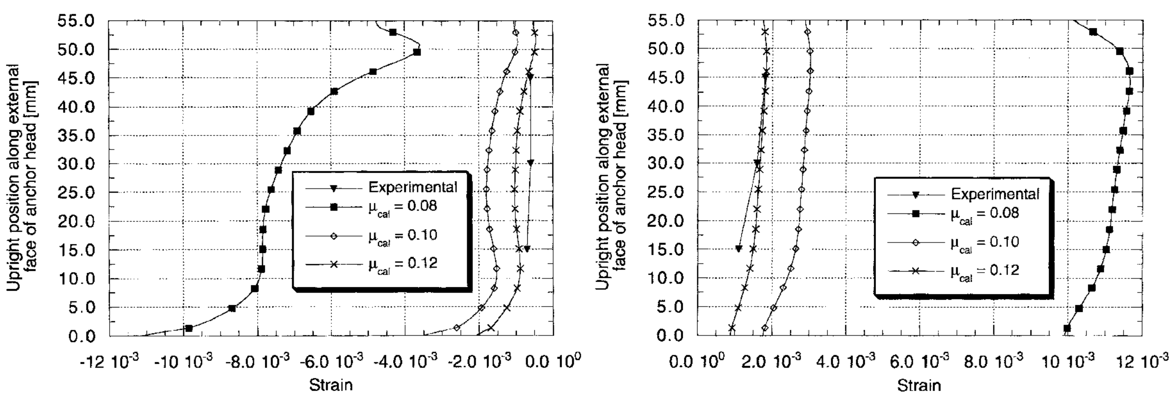


Fig. 4 Estimation of μ_{cal}: Slippage of the wedge



(a) Axial strain: ε_z (b) Circumferential strain: ε_θ

Fig. 5 Estimation of μ_{cal}: Strains along external face at 0.8 F_u

frictional coefficient between the anchor head and the standard anchor plate could be taken as 0.25.

4.3 Behaviour of the anchorage mechanism

In this section, the behaviour of the mono-strand anchorage is examined under different conditions such as the ultimate prestressing strength of standard steel strands F_u, the influence of wedge geometric tolerance and the influence of lubrication of the wedge-anchor head interface. For each condition, the slippage of the wedge and the percentage of yielding of the anchor head are evaluated.

4.3.1 Behaviour at ultimate load (F_u)

During the jacking procedure, the strands are generally stressed over the prescribed limit of $0.8F_u$. Such technique is used to make up for the loss of strength generated by the slippage of the wedge and the subsequent instantaneous effects associated with strands and concrete. However the maximum allowable prestressing force transfer from the jack to the anchorage should not exceed the prescribed value of $0.8F_u$. Therefore, it should be interesting to evaluate its behaviour and capability

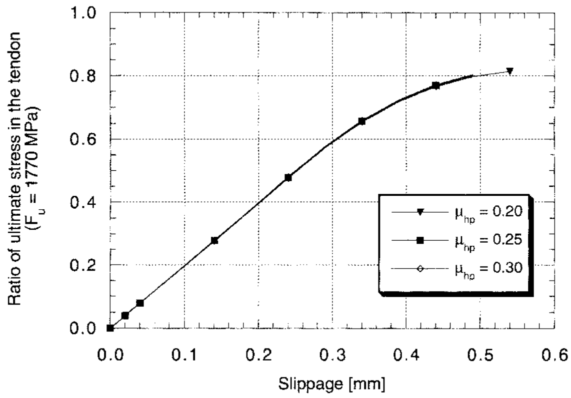
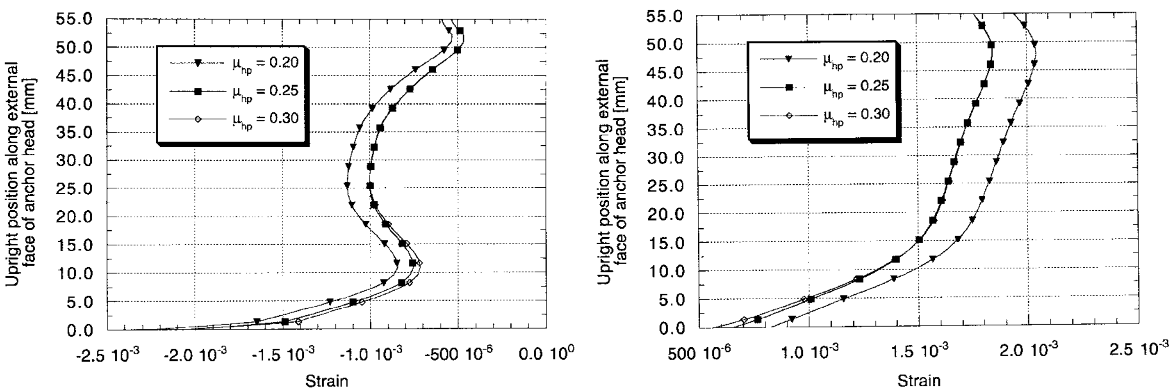


Fig. 6 Estimation of μ_{hp}: Slippage of the wedge



(a) Axial strain: ε_z (b) Circumferential strain: ε_θ

Fig. 7 Estimation of μ_{hp}: Strains along external face at 0.8 F_u

to sustain loads at prestressing higher than $0.8F_u$ caused by an erroneous prediction of loss of strength or simply by an extreme external load such as an earthquake loading.

For the frictional coefficients estimated in the previous section, Fig. 8 shows that the original anchorage mechanism has a very ductile behaviour. In particular, Fig. 8(a) shows that slippage of the wedge into the anchor head becomes very important when the load exceeds $0.8F_u$. This figure shows that yielding appears into the anchor head, but we cannot estimate the "amount" of yielding. Fig. 8(b) permits us to quantify the percentage of yielding into the anchor head and the anchor plate by estimating the total volume of the component that reaches yield (von Mises criteria f=0). In fact, this figure shows that yielding is mainly concentrated into the anchor head and starts at $0.5F_u$. At design prestressing load, 60 percent of the anchor head has reached yielding and at $0.87F_u$ the anchor head is entirely yielded. On the other hand, the plate reaches approximately one percent of yielding at $0.8F_u$ and therefore its behaviour is not really influenced by yielding. Fig. 9 shows the distribution of the equivalent plastic strain at $0.8F_u$. It starts to develop at the inner face of the anchor head, near the bottom of the wedge and, from this point, spreads to the anchor head outer face.

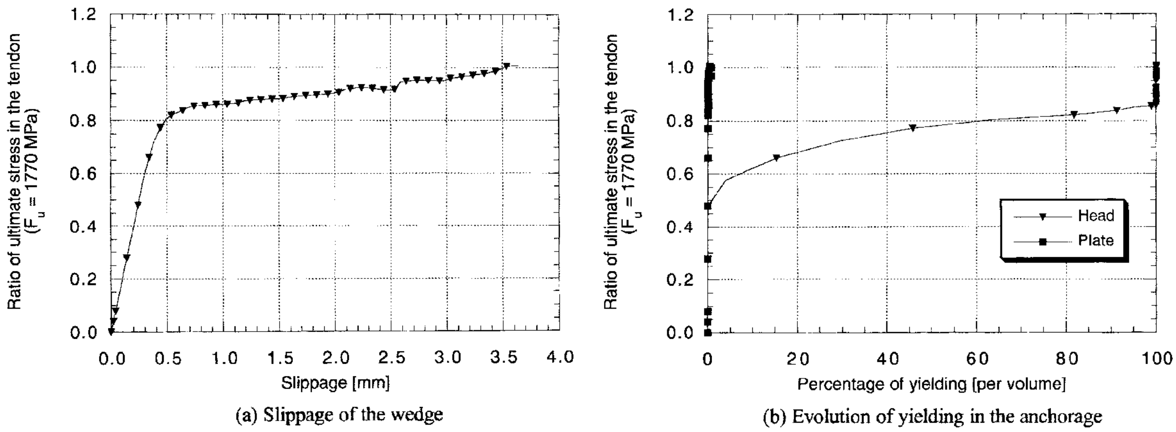


Fig. 8 Behaviour of the anchorage mechanism at F_u

4.3.2 Influence of the wedge geometric tolerance

As shown in Fig. 1, the wedge angle with respect to the anchor longitudinal axis is approximately 7.25°, compared to 7.00° for the conical shape of the anchor head. However, it should be noted that insertion of the tendon into the wedge has a widening effect and, consequently, the wedge angle tends to decrease to approximately 7.00°. Therefore, the outer surface of the wedge coincides with the inner surface of the anchor head. In practice, modification of the tolerance specifications of either the wedge or the anchor head will modify the contact interface between them affecting the expected behaviour of the entire mechanism.

The behaviour of a mono strand anchor head has been examined with two different wedge shapes. The range of angles used in this study was limited to ±0.25° due to the anchor head tolerance specifications. According to these limitations, the influence of two shapes was examined: 6.75° and 7.25°.

The influence of the wedge shape on the behaviour of the anchor head mechanism is shown in Fig. 10(a). The slippage of the wedge increased by 260 and 340 percent by changing the wedge angle from 7.00° to 6.75° and 7.00° to 7.25°, respectively. Thus, these results show that the angle of the outer face of the wedge influences significantly slippage and the stress distribution in the anchorage.

Fig. 10(b) shows that yielding starts very soon for angles different from 7.00°. For loads lower than $0.50F_u$, percentage of yielding varies according to the wedge geometry. This difference diminishes for loads higher than $0.5F_u$ due to the presence of yielding which initiates important deformation of the concerned interface and therefore rearranges its orientation. According to the initial position of the wedge, Fig. 11 shows that yielding starts at different locations depending on wedge angles. For angles lower than 7.00°, yielding starts in the bottom of the anchor head and spreads to the outer face comparatively to the other cases where yielding starts at the top of the anchor head and spreads to the bottom outer face.

4.3.3 Influence of lubrication

During installation of the anchorage mechanism, some anchor manufacturers use lubricant products to facilitate insertion of the wedge into the anchor heads. This technique allows easier

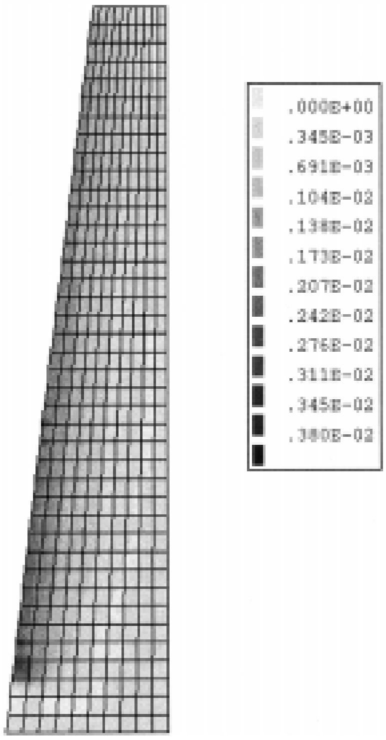
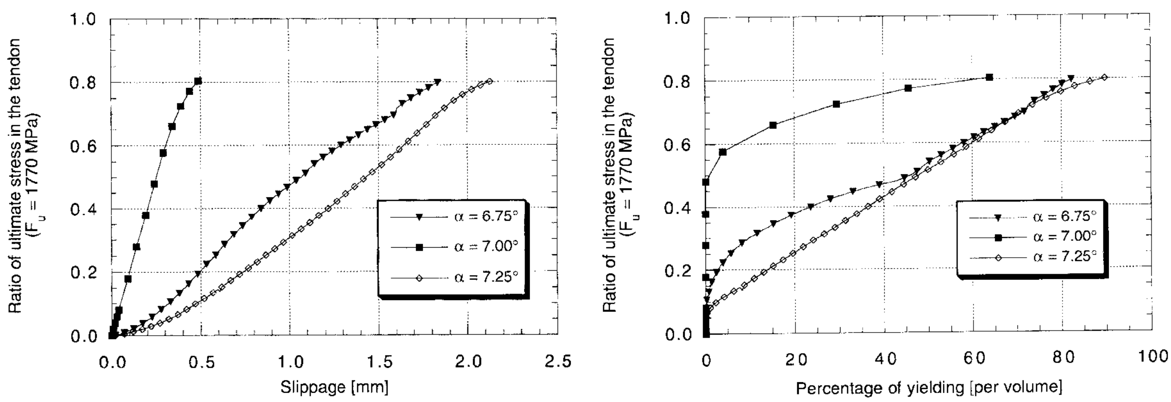


Fig. 9 Equivalent plastic strain distribution at 0.8 F_u

alignment of each component during the prestressing procedure. Lubricants used are generally grease or graphite powder, both of which are very efficient. However, this procedure decreases the frictional coefficient at the interface implying significant modification of the mechanisms behaviour.

In this paper, the lubricant has been simulated via a decrease of the frictional coefficient. Fig. 12 shows that such a lubrication modifies the mechanism performance. In particular, Fig. 12(a) shows



(a) Slippage of the wedge
(b) Evolution of yielding in the head

Fig. 10 Behaviour of the anchorage mechanism for various angles

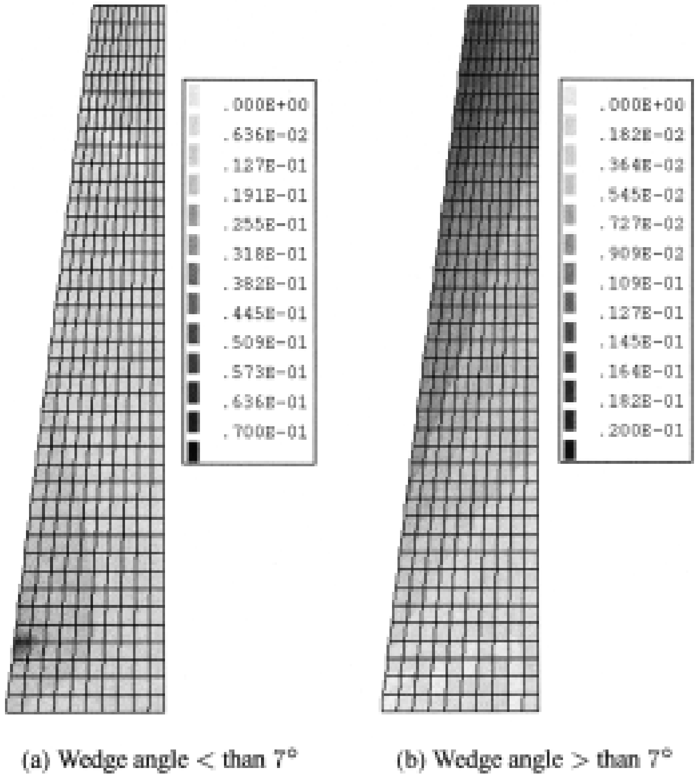


Fig. 11 Equivalent plastic strain distribution at $0.8F_u$ for various angles

that $0.8F_u$ could be reached only for frictional coefficient ranging from the non-lubricated case (0.11) to 0.04. Furthermore, Fig. 12(b) shows that these coefficients imply total yielding of the anchor head. According to these results, the actual anchorage mechanism should not be used with any lubricant to ensure that this anchor head would not experience excessive yielding under service

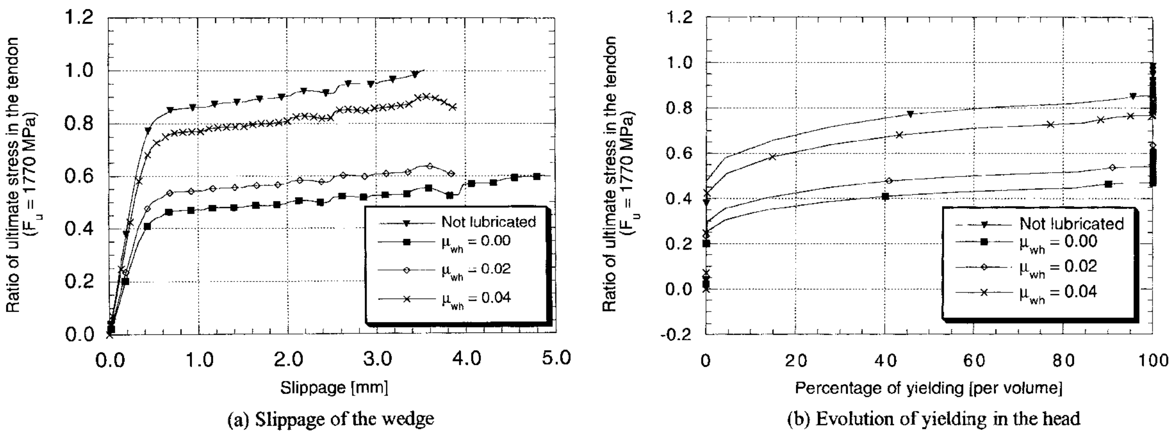
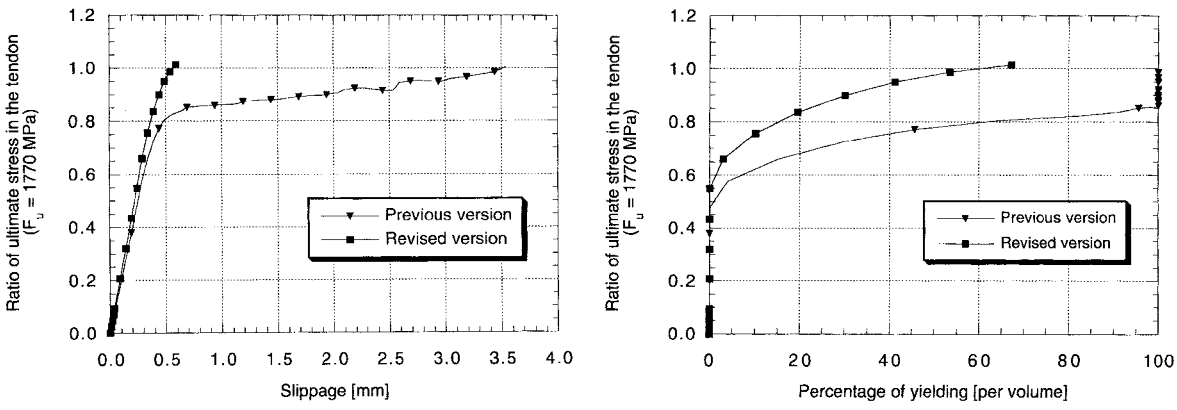


Fig. 12 Influence of lubricating the wedge-anchor interface



(a) Slippage of the wedge (b) Evolution of yielding in the head

Fig. 13 Behaviour of the revised version of the anchorage mechanism

loads.

4.4 Behaviour of the revised anchorage mechanism

It has been shown that the current mechanism provides poor performance under expected loading ($0.8F_u$). According to these results, a new geometry of this anchorage has been proposed (revised shape). Its behaviour will be compared to the former model presented in this paper. The new shape has been obtained by increasing the anchor head outer diameter by 5 mm.

Using the same mechanical properties and the same friction coefficients as the former anchor head, Fig. 13(a) shows that slippage of the wedge does not vary for prestressing loads under $0.8F_u$. Over this value, the anchor head has a very good performance since the slippage is limited to 0.6 mm at F_u comparatively to 3.6 mm for the former head. On the other hand, Fig. 13(b) shows that yielding into the anchor head decreases significantly. At $0.8F_u$, the yielded volume reaches 16 percent comparatively to 60 percent for the former anchor head. At F_u, the yielded volume of the revised anchor head reaches the same level as the former anchor head at $0.8F_u$.

5. Conclusions

This paper deals with an experimental/numerical study of an existing mono-strand wedge anchor head mechanism. The experimental program has been detailed and information such as wedge slippage and strain distributions along external peripheral surfaces of the anchor head have been presented. On the other hand, an efficient finite element model capable of providing internal information, not obtainable via experimental tests, has been developed taking into account all non-linearities that have been observed in the experimental tests.

The calibration of the numerical model led to the determination of frictional coefficients acting on the different anchorage mechanism interfaces. The numerical results showed that the frictional coefficient at the wedge-anchor head interface is the most important coefficient influencing the behaviour of the mechanism. For certain values it may lead to complete yielding of the anchor head. On the other hand, the frictional coefficient at the anchor head-plate interface does not have significant effect on the global behaviour of the mechanism.

Parametric studies have been undertaken to evaluate the behaviour of the mechanism under particular conditions. It has been shown that the anchorage tested in laboratory should not be used for prestressing loads over $0.8F_u$ of standard strength strands (1770 MPa). Also, the change of angle of the wedge outer surface (different from 7 degrees) and lubrication of the wedge-anchor head interface can lead to very poor performance of the mechanism; collapse of the anchor head could happen.

The numerical results obtained with the modified anchor mechanism, where the diameter has been increased by 5 mm, are very good. We observed that the slippage has been reduced by 600% at loads equal to F_u in comparison to results obtained with the former anchor mechanism behaviour. Furthermore, the yielded volume of the revised anchor head reaches 16% at $0.8F_u$ and 60% at F_u. In comparison, the yielded volume of the former anchor mechanism reaches 60% at $0.8F_u$ and 100% at F_u. Finally, it should be underlined that it is not recommended to use the new anchor mechanism and, *a fortiori*, the former one, with high strength strands (ultimate strength higher than 1770 MPa) or with a lubricating procedure due to the anticipated rate of yielding developed into the anchor heads.

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References

AASHTO (1994), *AASHTO LRDF Bridge Design Specification*, American Association of State Highway and Transportation Officials, Section 5, U.S.A.

Bastien, J. (1992), "Comportement des têtes d'ancrage mono et multi-torons: Étude expérimentale et étude par élements finis", Ph.D. thesis, Département de Génie Civil, Université Laval, Quebec.

Bastien, J., Chabert, A. and Picard, A. (1991), "Behaviour of external anchor heads", *Int. Symposium on Modern Application of Prestressed Concrete*, International Academic Publisher, Beijing, China, **1**, 182-189.

BPEL (1990), Règles techniques de conception et de calcul des ouvrages et constructions en béton précontraint suivant la méthode des états limites, Marché public des travaux, Fascicule no. 5, Cahier des clauses générales, France.

CAN/CSA-S6-88 (1988), Calcul des ponts-routes, Une norme Nationale du Québec, Association Canadienne de Normalisation, Ont. Canada.

Crisfield, M. A. (1991), *Non-linear Finite Element Analysis of Solids and Structures*, John Wiley and Sons Ltd, **1**.

Klarbring, A. (1995), "Large displacement frictional contact: A continuum framework for finite element discretization", *Eur. J. Mech. A/Solids*, **14**, 237-253.

Laursen, T. A. (1994), "The convected description in large deformation frictional contact problem", *Int. J. Solids Structures*, **31**, 669-681.

Laursen, T. A. and Simo, J. C. (1993), "A continuum-based finite element formulation for the implicit solution of multibody, large deformation frictional contact problem", *Int. J. Numer. Meth. Eng*., **36**, 3451-3485.

Marceau, D. (2001), Modélisation du contact tridimensionnel en grands déplacements relatifs et son application à l'étude des dispositifs d'ancrages multi-torons, Ph.D. thesis, Département de Génie Civil, Université Laval, Québec.

Marceau, D. and Fafard, M. (1993), "Modélisation du contact bidimensionnel et axisymétrique en grands déplacements relatifs et son application à l'étude des têtes d'ancrage mono-toron", Technical Report, Département de, Génie Civil, Université Laval, Québec, GCT-93-09.

SETRA (1990), Précontrainte Extérieure, Guide Technique, SETRA, France.