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**Sustainable Civil Engineering Structures and Construction Materials, SCESCM 2016** Development of

**the DKMQ element for buckling analysis of shear- deformable plate bending**

Foek Tjong Wonga,\* , Erwina, Alexander Richarda, Irwan Katilib aDepartment of Civil Engineering, Petra Christian University, Surabaya, Indonesia bDepartment of Civil Engineering, Universitas Indonesia, Depok, Indonesia Abstract In this paper the discrete-Kirchhoff Mindlin quadrilateral (DKMQ) element was developed for buckling analysis of plate bending including the shear deformation. In this development the potential energy corresponding to membrane stresses was incorporated in the Hu-Washizu functional. The bilinear approximations for the deflection and normal rotations were used for the membrane stress term in the functional, while the approximations for the remaining terms remain the same as in static analysis. Numerical tests showed that the element has good predictive capability both for thin and thick plates. ©

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**Keywords:**

Shear Deformable Plate Bending;DKMQ; Buckling Analysis; Kirchhoff; Reissner-Mindlin 1. Introduction Plate bending is of an utmost important structural model in engineering. To analyze practical problems of plate bending, the

finite element method (FEM) is at present the most widely used numerical method. Indeed the

plate bending problem is one of the earliest problems to which the

FEM was applied[1]. The most commonly used theories in developing finite elements for analysis of plate bending are Kirchhoff (or thin plate) and Reissner- Mindlin (or thick plate) theories. The Kirchhoff plate theory neglects the effect of shear deformation and thus it is only valid for thin plates, whereas the Reissner-Mindlin (RM) plate theory

is applicable to both thick and thin plates.

In early development of the FEM, the Kirchhoff theory was widely adopted

as the basis of the finite element

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2016.

formulation.

The difficulty with this approach is to construct the shape functions that satisfy the C1 continuity requirement. In

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the subsequent developments the RM theory is preferred since it requires only C0 continuity on the shape functions and furthermore, it is a more general theory than the Kirchhoff theory. While the use of RM theory in developing plate elements

by-passes the difficulty caused by the C1 requirement, direct application of the

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displacement-based finite element formulation, however, produces elements that overly stiff for thin plate situations. This phenomenon is known as shear locking. Early attempts

to overcome this difficulty was to employ the selective reduced integration technique

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(e.g. see [2] and the references therein).

Unfortunately this simple approach produced elements that have spurious energy modes.

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Since then there are innumerable RM plate bending elements have been proposed with different approaches to eliminate the shear locking. Some recently proposed successful plate bending elements include the refined Mindlin plate elements [3,4], a family of RM plate elements formulated using the discrete shear gap concept [5,6], and the RM plate element based on the consistent version of the Mindlin equations [7]. Among countless plate bending elements available now, the discrete-Kirchhoff Mindlin quadrilateral (DKMQ) element proposed by Katili [8] is of our interest since it has the standard nodal degrees of freedom, pass the patch test, shear locking free, and no spurious zero energy modes. Furthermore it has been proven [8] that the DKMQ has good predictive capability for thin to thick plates. This element is an extension of the DKQ (discrete Kirchhoff quadrilateral) element [9], which is a simple, efficient and reliable element for analysis of thin plates, to include the shear deformation. The DKMQ [8] results will converge to the DKQ [9] results

as the plate becomes progressively thinner. With regard to the good characteristics of

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the DKMQ element, this element has been recently further developed to the DKMQ24 shell element [10] and applied to composite plate bending structures [11,12]. However, to the authors' knowledge, there is no published report on the application of the DKMQ to plate bending buckling problems. It is thus the aim

of this paper to present the development of the DKMQ element to

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plate buckling problems. In the present development the membrane strain energy was added to the original Hu-Washizu functional for RM plates in order to account for the membrane stress effect to the plate bending stiffness. The approximate deflection and rotation fields for the membrane strain energy were taken to be the standard bilinear function, while the approximate fields for the bending and shear strain energy followed the original work [8]. The element was tested to different plate buckling problems to assess the accuracy and convergence characteristics. The results showed the DKMQ element can give accurate critical buckling loads both for thin and thick plates. 2. Formulation of the DKMQ for buckling analysis A detailed formulation

of the DKMQ for static analysis of plate bending

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have been presented in Reference [8]. In this section we present only the essential equations of the static formulation. The focus is given to formulation of the DKMQ for buckling problems. 2.1. Variational formulation We consider a plate of uniform thickness  $h$ , made from homogeneous and isotropic material

with modulus of elasticity  $E$  and Poisson's ratio  $\nu$ . Three dimensional Cartesian coordinate system is established with the  $x$ - $y$  plane lying on the plate middle surface as illustrated in Fig 1.

Based on basic assumptions of the RM plate theory, the displacement of 14

a generic point in the plate can be expressed as  $(x, y, z)$

(1) Where  $w$  is the deflection of the middle surface,  $\beta_x$  and  $\beta_y$  are the normal line rotations 5

in the  $x$ - $z$  and  $y$ - $z$  planes, respectively. The strains associated with bending deformation,  $\epsilon_x, \epsilon_y, \epsilon_{xy}$ , can be expressed in terms of the curvature,  $\kappa_x, \kappa_y, \kappa_{xy}$ , as  $\epsilon_x = -z\kappa_x, \epsilon_y = -z\kappa_y, \epsilon_{xy} = -z\kappa_{xy}$  (2a, b) Author name / Procedia Engineering 00 (2017) 000–000 3 Fig. 1. Plate with the coordinate system and positive sign convention for the displacement fields. The strains associated with transverse shear deformation are given as

(3) Here and here after symbols  $\{ \}$ ,  $\{ \}^T$ , and  $[ \ ]$  are used to denote a row matrix, a column matrix, and a square matrix, respectively. Following the static formulation [8], the Hu-Washizu functional is used as the basis to develop the DKMQ element for plate buckling analysis. In the presence of membrane (or

in-plane) pre-buckling stresses,  $\sigma_x, \sigma_y, \tau_{xy}$  and in the absence of other external forces, the Hu-Washizu functional is given 12

as [8], [13]–[16]  $\Pi = \int \Omega \{ \sigma \}^T \{ \epsilon \} - \int \Omega \{ \sigma \}^T \{ \epsilon \} + \int \Omega \{ \sigma \}^T \{ \epsilon \} - \int \Omega \{ \sigma \}^T \{ \epsilon \} - \int \Omega \{ \sigma \}^T \{ \epsilon \}$  (4)  
 (5) (6) (7) In Eqns. (4)–(7)

is the bending strain energy, is the shear strain energy, and is the 4

membrane strain energy associated with the buckling deformation. Matrix  $\{ \sigma \}$  is a matrix of assumed shear strains and matrix  $\{ \epsilon \}$  is a matrix of the shear forces. Matrices  $[ \ ]$  and  $[ \ ]$

are the bending and shear elasticity matrices, respectively, given 18

as  $[ \ ] = [ \ ]^T$  (8),  $[ \ ] = 0$  (9) Here  $D_b$  is the bending rigidity,  $D_s$  is the shear rigidity,  $k$  is the shear correction factor, which is taken to be  $5/6$ , and  $G$  is the shear modulus, ( ). The expression of  $[ \ ]$ , Eqn. (7), comes from the work of membrane stresses,  $\sigma_x, \sigma_y, \tau_{xy}$  and along the nonlinear terms of membrane Green strains [14], [16]. In this equation,  $[ \ ]$  is the matrix of membrane stresses, that is,  $[ \ ] = [ \ ]^T$  Matrices  $[ \ ]$  and  $[ \ ]$  are matrices of gradient of the deflection and rotations, that is,  $[ \ ] = [ \ ]^T$  The stationary condition of with respect to  $\{ T \}$  gives [8], [15]  $\int \Omega \{ \sigma \}^T \{ \epsilon \} = 0$  Which is a constraint equation relating the assumed shear strains  $\{ \sigma \}$  to the kinematical shear strains  $\{ \epsilon \}$ . 2.2. Approximation form (10) (11) (12) We consider a typical DKMQ element as

shown in Fig. 2. The element has four nodes and three degrees of freedom 11

per node, that is,  $w_i, \beta_{xi}, \beta_{yi}, i=1, \dots, 4$ . Mid-side nodes 5, ..., 8 are used to define a nodal parameter  $\beta_k, k=5, \dots, 8$ , which is the difference between the linearly and quadratically interpolated tangential rotation at a mid-side node. For the bending and shear strain energies and  $\beta_k$ , approximate rotations within an element are given as [8]  $\beta_k = \sum_{i=1}^4 \beta_{ki} \phi_i + \sum_{j=5}^8 \beta_{kj} \psi_j$  (13)  $\beta_k = \sum_{i=1}^4 \beta_{ki} \phi_i + \sum_{j=5}^8 \beta_{kj} \psi_j$  (14) Where  $\phi_i(x, y), i=1, \dots, 4$ , are the bilinear shape functions and  $\psi_k, k=5, \dots, 8$  are the hierarchical quadratic shape functions, all of them are expressed in terms of natural coordinates  $\xi, \eta$ .

$C_k$  and  $S_k$  are the direction cosines of side  $k$ . 4

Approximate deflection and rotations for the membrane strain energy, however, are given as  $w = \sum_{i=1}^4 \phi_i w_i + \sum_{k=5}^8 \psi_k \beta_k$  (15) The assumed shear strains  $\{ \sigma \}$  are linearly interpolated from the discrete tangential shear strains at the mid-side nodes 5, ..., 8. These mid-side tangential strains are obtained based on the moment-shear equilibrium equation along each side. The constraint equation, Eqn. (12), is implemented in discrete manner along each side using [8]  $\int_{-L_k/2}^{L_k/2} \sigma_{xy} s ds = 0$  (16) Where  $s$  is the tangential coordinate alongside  $k$  (see Fig. 2).  $j=(x_j, y_j)$   $k=5, \dots, 8$   $s = L_k \xi$   $C_k = \cos \theta_k = x_{ji}/L_k$   $S_k = \sin \theta_k = y_{ji}/L_k$   $L_k^2 = x_{ji}^2 + y_{ji}^2$   $x_{ji} = x_j - x_i$   $y_{ji} = y_j - y_i$

Author name / Procedia Engineering 00 (2017) 000–000 5 (a) (b) Fig. 2. 3

(a) DKMQ element with natural coordinates  $\xi-\eta$  and normal-tangential coordinates  $n-s$  along each side; (b) side  $k$ —nodes  $i, j$  (Taken from Reference [8]). 2.3. Geometric stiffness matrix Substituting the approximate deflection and rotations, Eqns. (15), into Eqns. (11), followed by substituting the resulting gradient matrices into Eqn. (7), the membrane

strain energy of an element can be expressed as

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, -\* + (17) in which (18)

is the nodal displacement vector and , - ∫ , - , - ∫ [ ] , - [ ] ∫ [ ] , - [ ] (19) is the geometric stiffness matrix.

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In this equation , [ where - [ ] (20a) ] [ ] [ ] (20b, c) (21) And here  $j_{11}, j_{12}, j_{21}, j_{22}$  are the components of the inverse Jacobian matrix. The reader may consult Reference [8] for a detailed formulation leading to the bending and shear stiffness matrices. 3. Numerical tests In the following tests, we perform a series of plate buckling analyses to assess the

convergence characteristics and accuracy of the developed DKMQ element in predicting a

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critical buckling load. The results

are presented in terms of a non-dimensional buckling load intensity factor

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defined as [13,17], where  $N_{cr}$  is

the critical buckling load and  $D_b$  is the bending rigidity, as defined in

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Eqn. (8). 3.1.

Simply-supported square plates subjected to an in-plane compressive load We firstly consider

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a hard

type simply-supported square plate with two different length-to-thickness ratios, that is

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$L/h=10$  and  $L/h=100$ ,

subjected to a uniaxial compressive load in x-direction. The length of

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the plate is  $L=10$  m; the material properties are  $N/m^2$  and  $\nu=0.3$ . The plate is modeled using different degrees of mesh refinement, i.e. , , , and . The resulting buckling load intensity factors are presented in Table 1 together with the analytical solutions. It is observed that the DKMQ results converge from above to the corresponding analytical solutions, both for the case of moderately-thick plate,  $L/h=10$ , as well as for the case of thin plate,  $L/h=100$ . 'Converging from above' is reasonable since the formulation basis is not a purely displacement-based formulation. Table 1. Buckling load intensity factors for hard type simply-supported square plates. Mesh  $L/h=10$   $L/h=100$   $2 \times 2$  4.694 5.016  $4 \times 4$  3.986 4.244  $8 \times 8$  3.801 4.058  $16 \times 16$  3.750 4.013 Analytical solution 3.741 4 ([18] as cited in [17]) [19] 3.2.

Rectangular plates subjected to an in-plane compressive load Secondly we consider hard type simply-supported

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rectangular plates of the length  $a$  and width  $b$

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with five length- to-width ratios,  $a/b=0.5, 1, 1.5, 2, 2.5$ , and three

thickness-to-width ratios,  $h/b=0.05, 0.1, 0.2$ ,

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as described in Reference [13]. The plate

is subjected to an in-plane compressive force  $N$  along the edges of width  $b$ .  
The

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material properties are the same as in the previous test. The plates are  
modeled using

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meshes with the number of elements along the edges of width  $b$ ,  $n=12$ , while the number of elements along the edges of length  $a$  follows the ratio of  $a/b$  (thus the shape of all elements is square of the length  $b/12$ ). The resulting buckling load intensity factors are tabulated in Table 2 together with those obtained using the meshfree method with regular particles presented by Liew et al.[13] and the

pb-2 Ritz method presented by Kitipornchai et al.

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[20]. It is observed that results from the present element are in agreement with those obtained using the meshfree and pb-2 Ritz methods.

For the case of thin plates ( $h/b=0.05$ ), the results are a

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little bit higher compared to the reference results. As the plates become thicker

( $h/b=0.1$  and  $h/b=0.2$ ),

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however, the results are between

the meshfree and pb-2 Ritz

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results. Buckling mode shapes for the

plates with length-to-width ratios  $a/b= 2$  and  $a/b= 2.5$

7

are shown in Fig. 3. These buckling modes are in agreement with those presented in Reference [13].  
Fig. 3. Buckling mode shape for hard type

simply-supported rectangular plates with length-to-width ratios: (a)  $a/b= 2$ ;  
(b)  $a/b= 2.5$ .

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Author name / Procedia Engineering 00 (2017) 000–000 7 Table 2.

Buckling load intensity factors for hard type simply-supported  
rectangular plates with different length-to-width ratios  $a/b$  and  
thickness-to-width ratios  $h/b$ .

10

$a/b$   $h/b$  Present Meshfree [13] pb-2 Ritz ([20] as cited in [13]) 0.5 0.05 0.1 0.2 6.0967 5.4085 3.7877 6.0405  
5.3116 3.7157 6.0372 5.4777 3.9963 1 0.05 0.1 0.2 3.9609 3.7637 3.1501 3.9293 3.7270 3.1471 3.9444  
3.7865 3.2637 1.5 0.05 0.1 0.2 4.2764 3.9886 3.1638 4.2116 3.8982 3.1032 4.257 4.025 3.3048 2 0.05 0.1  
0.2 3.9609 3.7637 3.1501 3.8657 3.6797 3.0783 3.9444 3.7865 3.2637 2.5 0.05 0.1 0.2 4.0817 3.8376  
3.1134 3.9600 3.7311 3.0306 4.0645 3.8683 3.2421 3.3. A square plate with a hole subjected to different  
in-plane loads Lastly we consider a hard-type simply-supported square plate with a hole of the thickness  
 $h=1$ , as shown in Fig. 4(a). The material properties are taken to be the same as in the first test (Sec. 3

.1). The plate is subjected to three different in-plane load

8

cases, namely an axial compressive load in x-direction,  $N_x$ , pure shear load, and biaxial

compressive loads in x and y directions

8

where  $N_x=N_y$ .

The plate is modelled using a mesh as shown in Fig.

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4(b). The analysis results are tabulated in Table 3 and compared to those obtained using the element-free Galerkin method (EFGM) [17].

It can be seen that the results are very close to those of the

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EFGM, that is, around 1- 3% higher than the EFGM results. Table 3. Buckling load intensity factors for

the hard type simply-supported square plate with a

17

hole. Method Axial Shear Biaxial Present 2.041 7.978 1.059 EFGM [17] 1.986 7.867 1.032 a b Fig. 4.

(a) Hard type simply-supported square plate with a

16

hole (adopted from [17], p. 474) and (b) its finite element mesh. 4. Conclusions The DKMQ element has been extended for buckling analysis of plate bending. The static part of the formulation was taken to be the same as in the original formulation, while the geometric stiffness matrix was formulated based upon the standard bilinear approximation of the deflection and rotations in the membrane strain energy expression. The convergence and accuracy of the present formulation were tested using different plate buckling problems. The results showed that the element can yield accurate solutions both

for thin and thick plates. The

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results converged from above

as the finite element mesh was refined. Therefore, the

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DKMQ element can be very useful for predicting a critical buckling load in practical applications. References [1] O. C. Zienkiewicz, R. L. Taylor, The Finite Element Method, Volume 2: Solid Mechanics, fifth ed., Butterworth-Heinemann, Oxford, 2000. [2] T. J. R. Hughes, R. L. Taylor, W. Kanoknukulchai, A simple and efficient finite element for plate bending, Int. J. Numer. Methods Eng. 11 (1977) 1529–1543. [3] C. Wanji, Y. K. Cheung, Refined quadrilateral element based on Mindlin/Reissner plate theory, Int. J. Numer. Methods Eng. 47 (2000) 605–627. [4] C. Wanji, Y. K. Cheung, Refined 9- dof triangular Mindlin plate elements, Int. J. Numer. Methods Eng. 51 (2001) 1259–1281. [5] K. U. Bletzinger, M. Bischoff, E. Ramm, A unified approach for shear-locking-free triangular and rectangular shell finite elements, Comput. Struct., 75 (2000) 321–334. [6] M. Bischoff, F. Koschnick, K. Bletzinger, Stabilized DSG elements – A new paradigm in finite element technology, Proc. 4th Eur. LS-DYNA Users Conf., Ulm, Germany, 2003. [7] G. Falsone, D. Settineri, I. Elishakoff, A new locking-free finite element method based on more consistent version of Mindlin plate equation, Arch. Appl. Mech. 84 (2014) 967–983. [8] I. Katili, A new discrete Kirchhoff-Mindlin element based on Mindlin-Reissner plate theory and assumed shear strain fields- Part II: an extended DKQ element for thick-plate bending analysis, Int. J. Numer. Methods Eng. 36 (1993) 1885–1908. [9] J. L. Batoz, M. Ben Tahar, Evaluation of a new quadrilateral thin plate bending element, Int. J. Numer. Methods Eng. 18 (1982) 1655–1677. [10] I. Katili, J. L. Batoz, I. J. Maknun, A. Hamdouni, O. Millet, The development of DKMQ plate bending element for thick to thin shell analysis based on the Naghdi/Reissner/Mindlin shell theory, Finite Elem. Anal. Des. 100 (2015) 12–27. [11] I. Katili, I. J. Maknun, A. Hamdouni, O. Millet, Application of DKMQ element for composite plate bending structures, Compos. Struct. 132 (2015) 166–174. [12] I. J. Maknun, I. Katili, H. Purnomo, Development of the DKMT element for error estimation in composite plate structures, Int. J. Technol. 5 (2015) 780–789. [13] K. M. Liew, J. Wang, T. Y. Ng, M. J. Tan, Free vibration and buckling analyses of shear-deformable plates based on FSDT meshfree method, J. Sound Vib. 276 (2004) 997–1017, 2004. [14] J. N. Reddy, An Introduction to the Finite Element Method, third ed., McGraw-Hill, Singapore, 2006. [15] J. L. Batoz, I. Katili, On a simple triangular Reissner/Mindlin plate

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