STRESS ANALYSIS OF OFFSHORE PIPELINE DURING LIFTING FOR JOINTING AND REPAIR

By

JIKI, P. N.
Department of Civil Engineering
University of Agriculture, Makurdi, Benue State

ABSTRACT

The analytical stress analysis of a pipeline lifted for jointing and repair is presented. The loading considered in the analysis are the lifting forces F and P and the self weight of the pipeline q. The behaviour of the raised part of the pipeline is governed by ordinary beam theory,, in the absence of axial loads. However, the analysis is complicated because the total length of the pipeline lifted is very long compared to the lifted ' gth, thus introducing a free boundary into the problem and rendering the solution difficult. Two examples of practical significance are considered. In the first case , the line is lifted by a single lift to obtain a specific height at the end of the pipe. In the second case the line is supported at two points giving a specified height and zero end slope. Computed results which are normalized in terms of a particular known case for which a close form solution exists are presented for the two cases considered. The results for the analyses are presented in normalized form and can be used as design charts to aid repairs of damaged pipelines carrying crude oil from the seabed to the shore.

NOTATION

- *a distance from sling to separation point A.*
- *b distance sling to end of pipe.*
- *c distance of zero shear force from end of pipe*
- *d distance from second sling to A*
- *e distance from second sling to end of pipe*
- *E Youngs modulus*
- *I second moment of area*
- *M bending moment*
- *F,P sling forces*
- *q weight per unit length of pipeline(including weight of coat , less buoyancy)*
- *x, y coordinates defined in figure 1*
- *Z elastic modulus of section*
- *δ deflection*
- *σ stress*

θ rotation

o subscript which refers to the breakover special case

1. INTRODUCTION

The problem of stresses in submarine(offshore) pipelines during laying operations was first presented by Wilhoit and Merwin [7] and by Palmeret al [5] and studies on stresses caused by seabed and trench irregularities were conducted by Mousselli [4]. It is well known that high stress concentrations in offshore pipelines and tubular K ,T and TK joints give rise to catastrophic fatigue failures of such offshore structures as has been reported by Jiki [1,2,3] and by Paul et al [6]. This paper reports studies of the stresses caused when it is necessary to raise the end of a pipeline for jointing or repair operations. In particular the case of underwater repair and jointing from the seabed is reported. However, the analysis is presented herein is also applicable to the lifting of pipelines on dry land and the non dimensional presentation of results makes it easy to obtain numerical values for design purposes for any pipeline.

Two practical situations are considered in the repair. In the first case (fig.1a) ,the end of the line is raised by a sling some distance away to a specific height δ. In general the end of the line will come to rest at an angle θ. This may well be acceptable while attaching certain fittings to the line , but if a joint is to be made either to a spool piece or another pipeline it will be necessary to use the second arrangement shown in fig.1(b). In this situation two forces F and P are applied (in some circumstances one of these will need to act down wards) such that the specified height δ is achieved without any end slope. In both situations the location of point A at which the line separates from seabed is unknown. which introduces a free boundary into what is otherwise a simple bending problem. For practical pipe sizes and lift heights, the slope involved is small and since the end of the pipe is free, there is no question of any significant axial tension in the line. These two simplifying factors make it economically possible to examine the situation thoroughly, leading to a full range of numerical results.

2. STRUCTURAL ANALYSIS

2.1 Analysis: case 1- single sling lift

Just before the separation point A the pipe is straight , that is , it has zero curvature and if it is assumed that the curvature is continuous at A , then the curvature at A is also zero , that is, the bending moment is zero (this is equivalent to assuming that the seabed does not exert a concentrated external couple at A).

Thus, taking moment at A, we have:

$$Fa = \frac{q\,(a+b)^2}{2} = Q \qquad \text{---------------(1)}$$

Or

$$F = \frac{q\,(a+b)^2}{2a} \qquad \text{---------------(2)}$$

Treating the elevated part of the line as a cantilever ,it is easy to show that [8]:

$$8 = \frac{F^3\,(a1+3b/2a)}{3EI} \quad \frac{q(a+b)4}{8EI} \qquad \text{-----(3)}$$

Substitution for F from equation (2) gives

$$8 = \frac{q\,(a+b)^2\,(a^2-3b^2)}{24EI} \qquad \text{-------(4)}$$

In any particular case the only unknown in equation (4) is the length a to the lift off point. But the form of the equation is such that an iterative solution for a is necessary. Thus for the purpose of implementing the Newton iterative method to solve for the unknown length a, we write equation (4) as:

$$f(a) = \frac{q(a+b)^2(a^2-3b^2)}{24EI} - \delta \qquad (5)$$

Then we have

$$\frac{df}{da} = f\,'(a) = \frac{q(a^2-b^2)(2a+3b)}{12}$$

(6)

Using the Newton iteration and taking some initial estimate a_0 of a, an improved estimate a_1 is given as:

$$a_1 = a_0 - \frac{f(a)}{f\,'(a)} \qquad (7)$$

This procedure is repeated in a simple computer code until convergence is achieved when the sling force F is calculated, using equation (2). Two potential bending moment maxima occur (fig.2) as follows:

(a) Hogging, at the sling.

$$M = \frac{1}{2}(qb^2) \qquad (8)$$

Sagging, at the zero point of shear, that is , between the sling and A. This point is located at a istance c from the end of the line , in which :

$$c = \frac{F}{q} \qquad (9)$$

and

$$M = \frac{1}{2}qc^2 - F(c-b) \qquad (10)$$

Depending on the proportions of b to a, either of the moments in equations (8) or (10) can be the absolute maximum.

2.2 Analysis: case 2-breakover case

It is clear that there is some sling position in the single sling lift which will give a zero end rotation θ. Again treating the elevated part of the line as a cantilever, we have:

$$\theta = \frac{Fa^2}{2EI} - \frac{q(a+b)^3}{6EI} \tag{11}$$

Again substitution for F from equation (2) we have:

$$\theta = \frac{q(a+b)^2(a-2b)}{12EI} \tag{12}$$

If $\theta = 0$, then a_0=$2b_0$. This special case is being indicated by the use of a subscript $_0$.

$$\delta = \frac{3qb_0^{\;4}}{8EI} \tag{13}$$

or solving for b_0 we have :

$$b_0 = \left(\frac{8EI\delta}{3q}\right)^{1/4} \tag{14}$$

Again using a_0 =$2b_0$ in equation (2) gives :

F_0=2.25qb_0 (15)

And the largest bending moment and stress are given as:

$$M_0 = \frac{1}{2}qb_0^{\;2} \tag{16}$$

and

$$\sigma_0 = \frac{M_0}{Z} \tag{17}$$

We note here that the value for a_0 forms a satisfactory estimate of a in the Newton iteration both for single and two sling situations.

2.3 Analysis : case 3-two point supports

Referring to fig.1(b), the following equations are obtained . Taking moment at A we have:

$$Fa + Pd - \frac{1}{2}q(a+b)^2 = 0 \tag{18}$$

Deflection[8]:

$$\delta = \frac{Fa^3}{3EI}\left(1+\frac{3b}{2a}\right) + \frac{Pd^3}{3EI}\left(1+\frac{3e}{2d}\right) - \frac{q}{8EI}(a+b)^4 \tag{19}$$

Rotation [8]:

$$\theta = \frac{Fa^2}{2EI} + \frac{Pd^2}{2EI} - \frac{q}{6EI}(a+b)^3 \tag{20}$$

Using equations (18) and (20) and noting that d=a+b-e. It is possible after some careful manipulations to show that:

$$P = \frac{q(a+b)^2(2b-a)}{6(a+b-e)(b-e)} \tag{21}$$

and

$$F = \frac{q(a+b)^2(a+b-3e)}{6a(b-e)} \tag{22}$$

In principle equations (21) and (22) could be substituted into equation (19) but the outcome would not lead to any meaningful simplification of the latter equation, or to an explicit equation for the length a. Instead, the Newton iterative solution was again employed, and by rearranging equation (19) into the form similar to equation (5) ,we h

$$f(a)=\frac{Fa^3}{3EI}(1+\frac{3b}{2a})+\frac{Pd^3}{3EI}(1+\frac{3e}{2d})-\frac{q}{8EI}(a+b)^4-\delta \qquad (23)$$

and the function derivative $f'(a)$ can be calculated using the usual rule for differentiation.

To implement the Newton iterative solution, a computer code in Watford 77 Fortran was written to solve for the length a. Then sling forces P and F are computed for the current iteration using equations (21) and (22) and the value of the function $f(a)$ is evaluated. Derivatives F' and P' of F and P with respect to a are computed for the current iteration and $f'(a)$ is found using these values. Once convergence has been achieved, values for a, F, and P are output from the program. There are two potential bending moment maxima, if P is positive (see fig.7), two hogging, one sagging and one value between F and P which could in some cases be a sagging maxima. If P is negative as is the case when $b<b_0$, then the largest bending moments are located at F and between F and A.

3. RESULTS OF ANALYSIS

3.1 Results: cases 1 & 2 – single sling and break over

Results for this solution are presented in figs. 3-6, in a non-dimensional form. This form has been obtained by dividing all the results by the equivalent quantity in the break over case, except in the case of figure 6 for end rotation θ, *where* θ_0 is the rotation at the sling in the break over case. As the end rotation in the break over case is zero, this seems a natural choice. It is easy to see that:

$$\theta_0 = \frac{qb_0^3}{6EI} \qquad (24)$$

3.2 Results: case 3- two point supports

Results for this case are presented in figs.8-11 , again these results are non-dimensionalised with respect to the break over case which is also a special case of the two supports situation (when $b=b_0$) .Each figure gives a family of curves for various position of the second sling , that is , for various $\frac{e}{b}$ *ratios.*

4. DESIGN PROCEDURE

4.1 Procedure for cases 1 & 2- single sling and break over

The following design procedure which relies on the similarity laws , is suggested . For the specified E,I, δ, and q , compute the values of b_0 , a_0 , F_0 , M_0 (or σ_0) *and* θ_0 using the break over formulae. Then for a known b. enter the graphs at the calculated $\frac{b}{b_0}$ value and read off the results for $\frac{a}{a_0},\frac{F}{F_0},\frac{M}{M_0}$ *and* $\frac{\theta}{\theta_0}$ and hence values of a .M . F and θ.

Alternatively a value of $\frac{b}{b_0}$ can be chosen to minimize a particular quantity. For example if $\frac{b}{b_0} = 0.72$, the maximum stress is only 52% of that found in the break over case (seefig.5).

4.2 Procedure for case 3 : - two point supports

A similar design procedural to that employed in the single sling case is suggested. That is, compute the break over values for the line, using equations (14-17). Then enter figures 8-11 with known values of $\frac{b}{b_0}$ *and* $\frac{e}{b_0}$, and read off the results.

Alternatively values of $\frac{b}{b_0}$ *and* $\frac{e}{b_0}$ may be selected to minimize a particular quantity.
For example. if

$\frac{e}{b} = 0.70$ *and* $1.5 < \frac{b}{b_0} < 2.0$, the maximum stress is approximately 50% of that found in the break over case (see fig.11) .

5. DISCUSSION AND CONCLUSIONS

Two situations of practical interest relevant for jointing and repair work for offshore pipelines have been presented. The presentation includes details of structural analyses and design procedures that would be of interest to those involved in the design and operations of pipelines in the offshore industry.

In particular, it has been shown that by a careful selection of sling positions, the maximum stresses in a pipeline lifted for repair can be reduced by a factor close to two. Also the results presented in figures 3-6 and 8-11 have been normalized by a break over case as such they can be used as design charts for stress analysis and subsequent design of offshore pipelines, for the particular lift case considered.

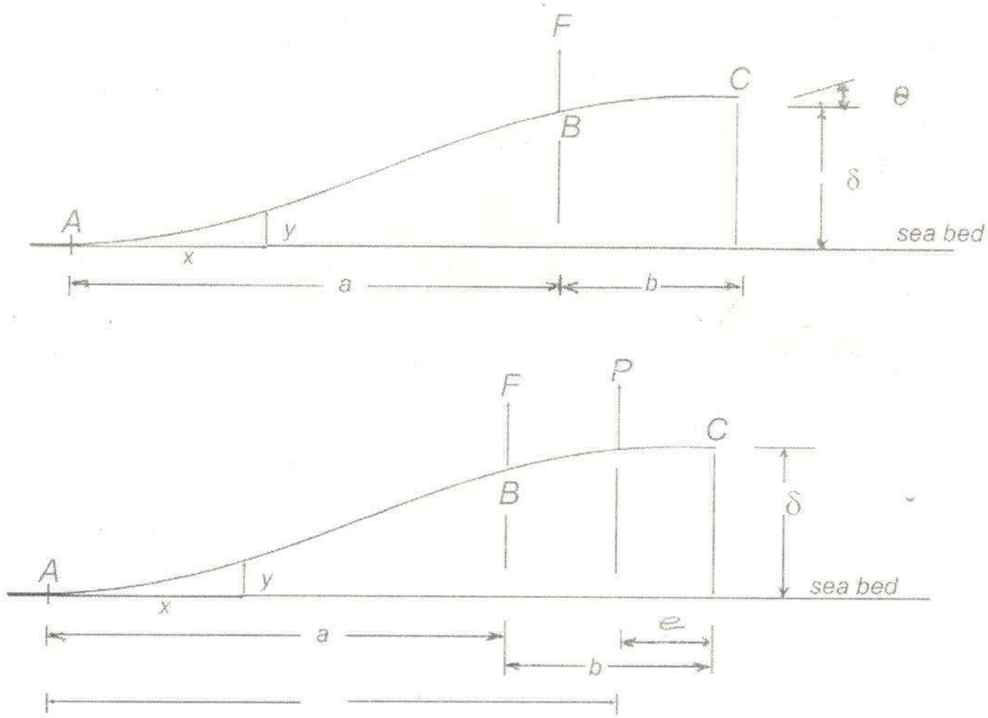

Fig. 1 Two repair and jointing situations

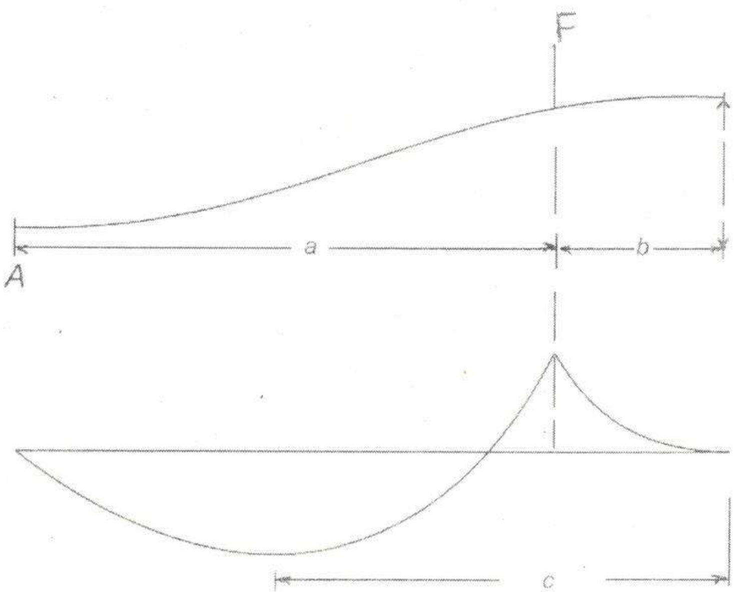

Fig. 2 Bending moment diagram - single sling case.

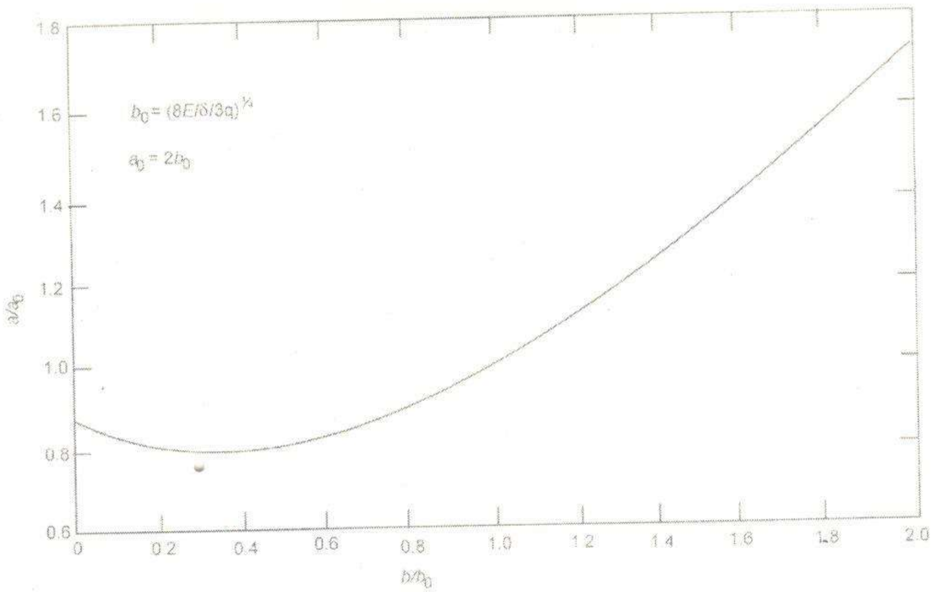

Fig. 3 Single sling case - lifted length.

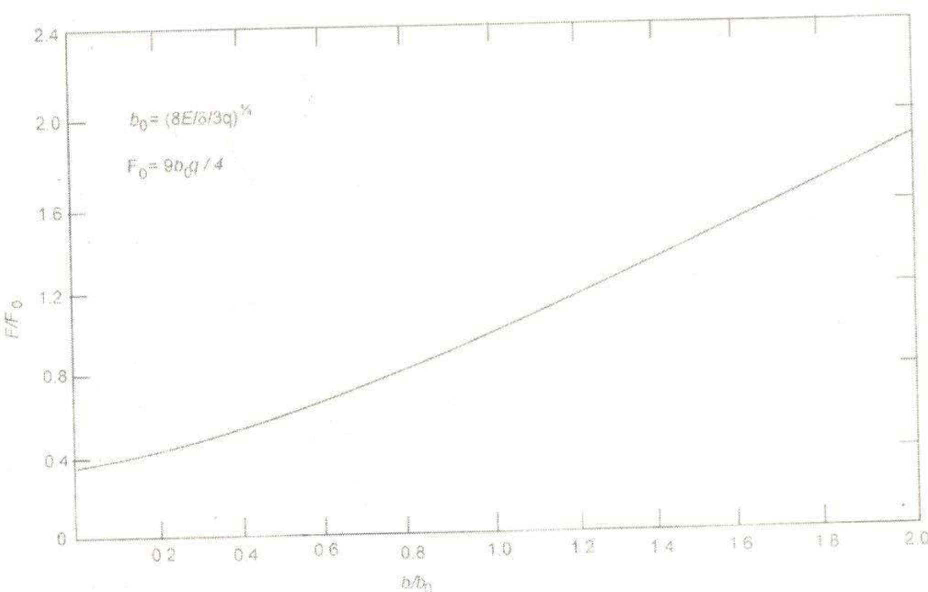

Fig. 4 Single sling case - sling force

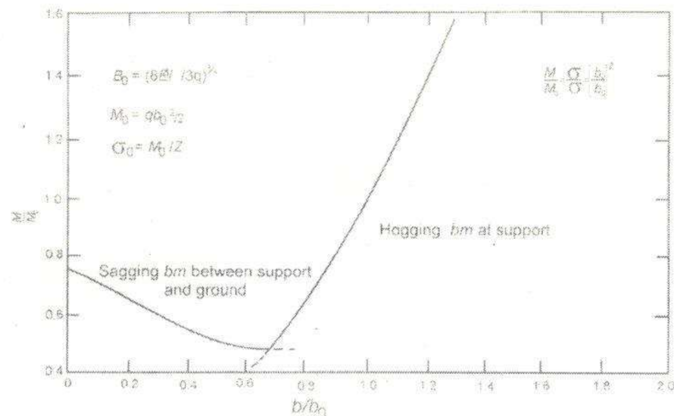

Fig. 5. Single sling case - maximum moment and stress.

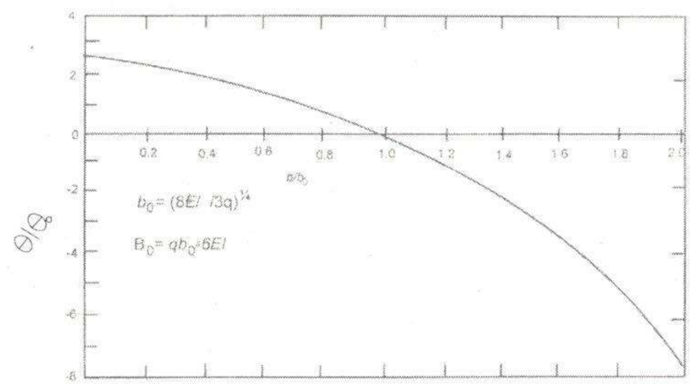

Fig. 6. Single sling case - end rotation.

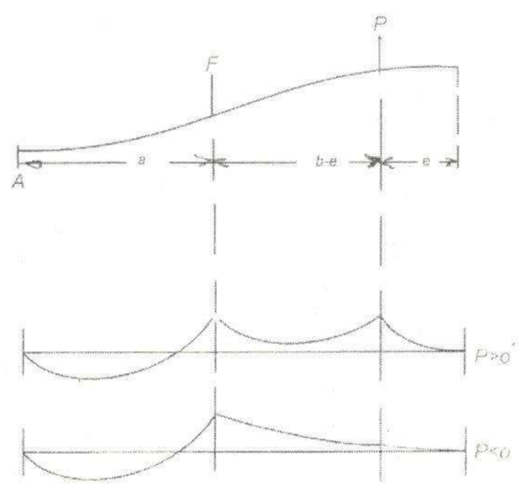

Fig. 7 Bending moment diagram two support case

A beam bending problem with a free boundary

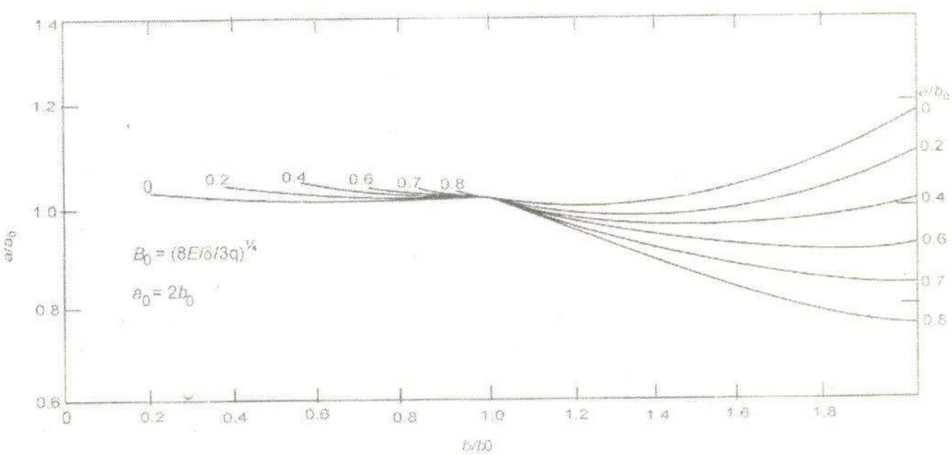

Fig. 8. Two support case - lifted length.

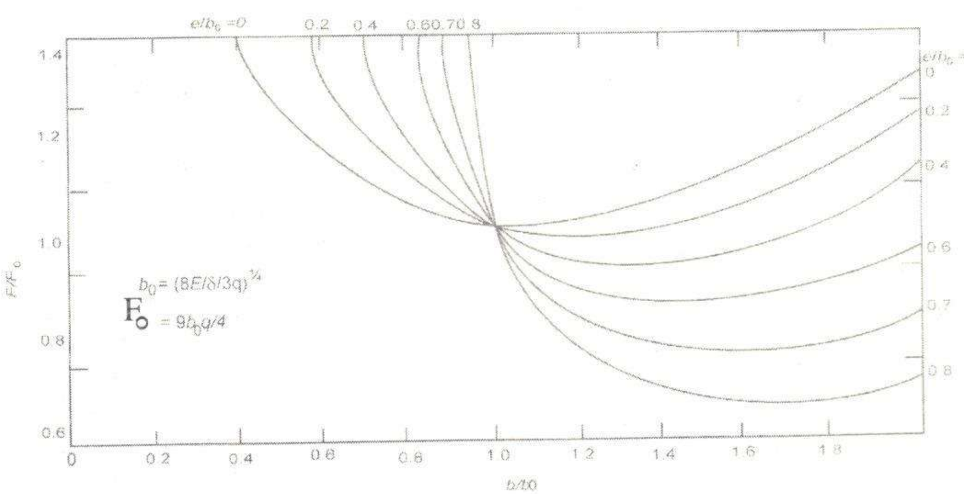

Fig 9. Two support case - sling force R.

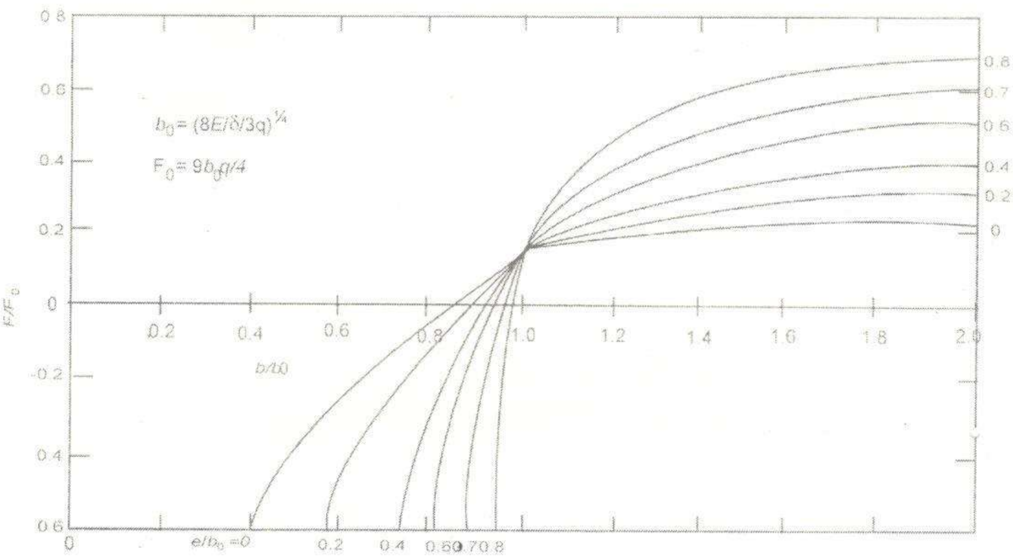

Fig. 10. Two support case - support force P.

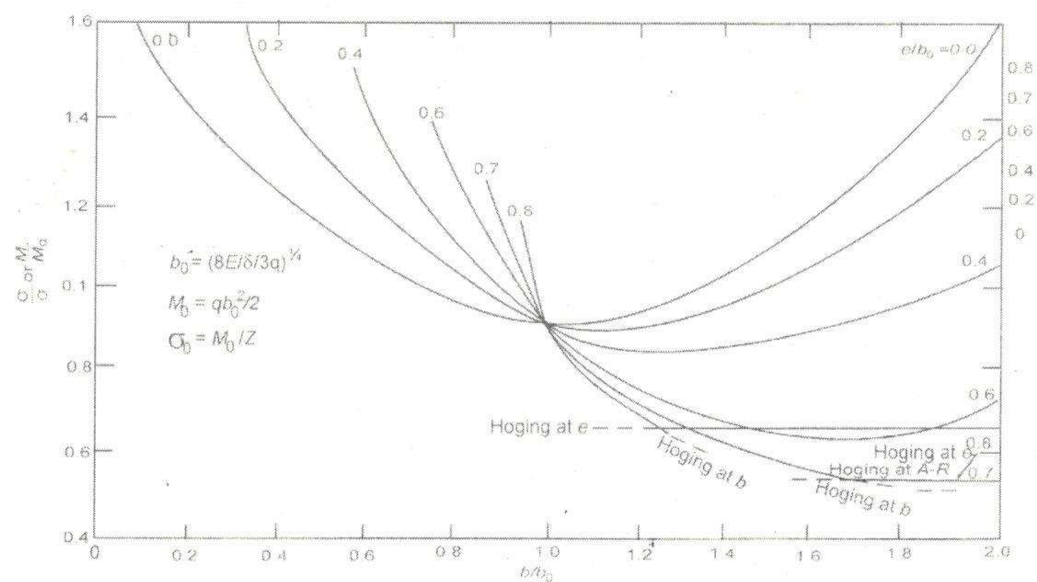

Fig 11. Two support case - maximum moment and strees.

REFERENCES

1. Jiki, P.N. (1982). An Investigation into the stress distribution using rectangular/square 'K' Joint and the finite element method .M. Sc. Thesis, Cranfield University , Cranfield England , U.K.

2 Jiki, P.N. (1994) . Effect of brace spacing on Strength of Tubular 'K' Joints Nig. J. of Tech. Education ,vol. 11, Nos. 1 & 2 , 91-106

3. Jiki , P.N. (2003) . Stress Analysis of square hollow section 'K' Joints using parametric equations. J. of Agriculture, Science and Technology (JAST) , University of Agriculture,Makurdi vol. 13 , Nos. 1 & 2 60-73 .

4 Mousselli, A. H. (1977). Pipe stresses at the seabed during installation and trenching operations .Proc. 9th Annual offshore Conference, Paper OTC 2965 .

5. Palmer, A. C., Hutchinson , g. and Ells, J. W. (1974). Configuration of submarine pipelines during laying operations . Tran A.S.M.E. J.of Engineering for Industry, vol. 96 , 1112-1118.

6. Paul, J.C., Makino, Y. and Kurobane, Y. (1994). Ultimate Resistance of Unstiffened Tubular TT-and KK-Joints. J. of Structural Engineering, ASCE.vol. 116 , No. 10 ,Oct. 2853-2870.

7. Wilhoit, Jr. J.C.and Merwin, J.E. (1967). Pipe stresses induced in lay offshore pipelines. Trans. A.S.M.E. J. Engineering for Industry, vol. 89, 37-43

8. Happold, E. (1987). Education and the routes to chartered membership. The Structural Engineer, vol. 65A, No. 5., May, 178-179.