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Parametric Study of Propagation Buckling Mitigation for Submarine Pipeline System by Using Integral Buckle Arrestor

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Abstract. One of the failure modes that appears in the submarine pipeline system is a pipe collapse or local buckling. This failure might be occurred as the result of high external hydrostatic pressure. Local buckling will propagate to another section of pipeline which causes propagation buckling along the pipeline system. Therefore, buckle arrestors are necessary to be installed along the critical location to prevent catastrophic failure. The aim of this study was to investigate the effect of geometry and material type variation of integral buckle arrestor to buckling load factor.

Parameter variations were covered from, (i) pipe diameter, (ii) pipe thickness, (iii) buckle arrestor length, (iv) the ratio of buckle arrestor thickness to pipe thickness, and (v) type of buckle arrestor material. The load case was taken from Masela Block, Abadi Gas Field preliminary route survey that will pass through a 5,309 ft of depth. The numerical simulations using ANSYS™ was conducted to demonstrate the effect of each parameter variation. Nominal Pipe Sizes of 26", 28", 30", 32", 34" with varied API 5L X42-X80 grade of material were used in the numerical simulation. The simulation was validated using previous experimental study and refers to standard DNVGL-OS-F101.

This study indicates that pipe thickness and pipe diameter are two significant factors to influenced buckling load factor. Ratio of buckle arrestor thickness to pipe thickness, length of buckle arrestor type of material pipe shows a modest change on the buckle load factor.

Keyword: Propagation buckling; Integral buckle arrestor; Buckling load factor

Nomenclature

D	pipe outside diameter	$R1$	curvature radius 1
h	buckle arrestor thickness	$R2$	curvature radius 2
L	system length	P_{pr}	propagating pressure
L_1	buckle arrestor inner length	$P_{pr,BA}$	propagating pressure of buckle arrestor
L_2	buckle arrestor outer length	γ_m	material resistance factor
L_{BA}	total length of buckle arrestor	γ_{SC}	safety class factor
t	pipe thickness	P_e	external pressure
t_2	pipe thickness characteristic 2	P_X	crossover pressure

1 Introduction

The rapid growth of energy demand drives the exploration of oil and gas not only in the onshore but also in the offshore. Advanced technology development enables people to go deeper to look for abundant resources of gas and crude oil. Indonesia, as one of the major producers and exporters of natural gas, has an enormous resources located in the offshore. However, this condition has to deal with high risk of safety and infrastructure challenges. In fact, this has an impact not only for the exploration, refinery, and transportation sectors but also for the transmission activity. One of the common infrastructures for oil and gas transmission is by using a pipeline system.

Recently, Indonesia has found a new natural gas field located in the east part of Timor Island or known as Masela Abadi Field. The Japan Exploration company predicted gas reserves in the Masela block reached a level of 10.73 TCF. The Indonesian government announced that this block will be explored by using an onshore scheme. Therefore, the pipeline system has to be laid and installed in the middle of the sea spanning from the reservoir to the nearest possible island (Jamdena). Using this scenario, the pipeline system will pass through a maximum depth of 5,309 ft.

The major failure mode that is most likely occurs in the deepwater pipeline installation is a collapse or known as local buckling. One of the key factors causing this failure is a high external hydrostatic pressure acting on the pipeline. Adequate pipeline design is necessary so that local buckling does not propagate along the entire pipeline body or compromise the pipeline integrity. DNVGL-OS-F101 as the main code used for designing offshore pipeline provided wall thickness calculation to prevent local buckling. The calculation result may not be feasible due to excessive design values. Consequently, the mitigation plan to avoid possible catastrophic failure is to install buckle arrestors in certain dimensions and distances. An integral buckle arrestor was used to consider the easiness of installation and maintenance factor.

This study aims to investigate the effect of geometry variation and material assignment of integral buckle arrestor to the buckling load factor. The buckling load factor is the ratio of the buckling loads to the current applied loads then it indicates the factor of safety against buckling. The factors were extracted from a numerical simulation that was conducted by ANSYS^{TM}. Five nominal pipe size (NPS) of 26”, 28”, 30”, 32”, and 34”, were examined in the numerical simulation.

2 Methodology

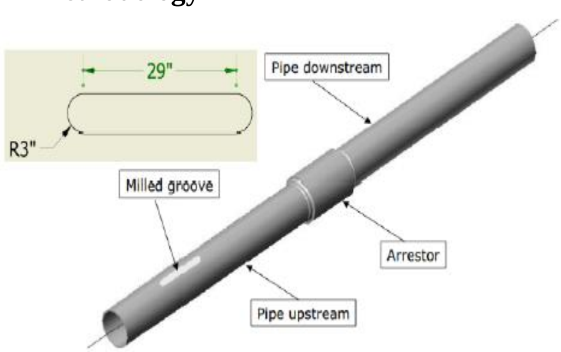

Fig. 1. Integral buckle arrestor configuration [1]

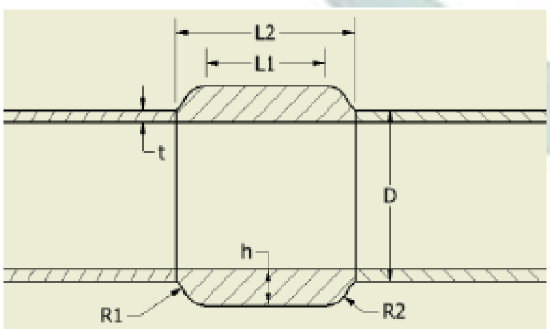

Fig. 2. Cross-section of integral buckle arrestor

Numerical simulation was conducted to examine the effect of geometry variation and material assignment on the buckling load factor. The five main parameters that varied in the geometries were pipe thickness, a ratio of buckle arrestor thickness to pipe thickness, length of buckle arrestor, and type of buckle arrestor material. Material assignment based on API 5L standard class X42 to X80 was assigned to the buckle arrestor. The generated solid element consists of an upstream pipe, downstream pipe, and buckle arrestor as shown in Fig. 1. The upstream pipe has a milled groove, notch, in order to initiate buckle similar to study conducted by Mantovano, *et al*. [3]. Fig. 2 exhibits the cross-section of integral buckle arrestor. The results of the numerical simulation will be validated by comparing the result with the previous experimental test [3] and checked with DNVGL-OS-F101 [2] compliance.

2.1 Numerical Simulations Parameters

A Solid model of integral buckle arrestor was generated as a 3D model according to pipeline system configuration. After that, material assignment, meshing parameter, initial, and boundary condition were four main parameters that necessary to be assigned in numerical simulation. The geometry notch on the upstream pipe, milled groove, was dependent to pipe thickness. The ratio of milled groove depth to pipe thickness was set to be 0.23.

2.1.1 Geometry Variations

The geometry variations are (i) pipe thickness; (ii) ratio of buckle arrestor thickness to pipe thickness; (iii) length of buckle arrestor; and (iv) pipe diameter. These parameters were varied in certain value as shown in Table 1. In the process of numerical simulation, while one parameter is varied, the other parameters are kept constant.

Table 1. Geometry Variations

Symbol	Value (in)				
NPS	26	28	30	32	34
t	1	1.25	1.5	1.75	2
L2	30	35	40	45	50
L1 = L2 - 2(in)	Calculated				
R = h - t (in)	Calculated				
L	600				
Material	API 5L X42-X80				

2.1.2 Material Assignments

Specification of material that assigned to the pipeline configuration is determined based on API 5L latest edition varied from class X42 to X80 [4]. Material properties can be seen in Table 2.

Table 2. Material Specifications

Material	Density, ρ (kg/m^3)	Young Modulus, E (GPa)	Poisson Ratio, v	SMYS (MPa)
X42	7850	207	0.3	290
X46	7850	207	0.3	320
X52	7850	207	0.3	360
X56	7850	207	0.3	390
X60	7850	207	0.3	415
X65	7850	207	0.3	450
X70	7850	207	0.3	485
X80	7850	207	0.3	555

2.1.3 Meshing parameter, Boundary and Initial Conditions

Meshing solid element was considered to be automatic and program-controlled. The meshing process used patch method with tetrahedron element type. A shape transition parameter is set to be slow. The span angle center and a smooth parameter were set to be fine. From this parameter, the sizing of meshing result was very good which was determined by the skewness criterion. The average mesh metric value was 0.31. Fig. 3 shows a solid model meshing in ANSYS™. For the boundary and initial condition, the external pressure of 2,358 psi (hydrostatic pressure at 5,309 ft) acting on the entire solid model geometry. Support type of the system was considered as fixed support at both ends of the pipe. An eigenvalue (linear) buckling feature was used to determine the system mode shape of buckling due to hydrostatic pressure.

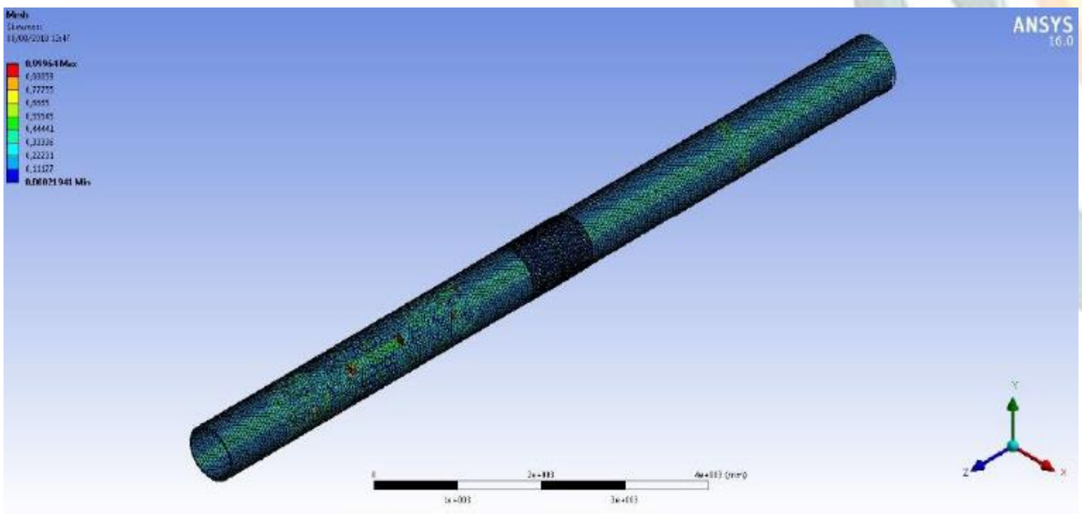

Figure 3. Solid model meshing

2.2 Validation

Two validation methods were conducted to check the result of numerical simulation, i.e comparison of the result of numerical simulation to experimental data that was conducted by Montovano, *et al* [3] and the comparison to the criterion recommended by DNVGL-OS-F101 [2].

2.2.1 Comparison to Previous Study

To ensure that the parameters used were correct, the experimental testing of ref [3] was re-modelled and simulated. The simulation setup was made as similar as possible to ref [3] experiment data. Experimental parameters that were used for comparison shown in Table 3.

Table 3. Experimental parameters [3]

Modul	Outside Diameter [mm}	Pipe Thickness [mm]	Pipe Material	Buckle arrestor to pipe thickness ratio	Modul Length [m]	Buckle arrestor length [mm]
1	141,3	6,55	X42	2,5	2.3	70.7
2	141,3	6,55	X42	3	2.3	106
3	141,4	6,55	X42	3	2.3	141.3

The collapse pressure obtained from experimental testing of ref [3] was used as the input of external pressure in the numerical simulation. The buckling load factor will be correct if the result is 1.00. From Table 4, the difference of buckling load factor between experimental data ref [3] to the numerical simulation is less than or equal to 5%. Thus, it can be concluded that the parameters used in the simulation are valid. Therefore, those parameters will be carried out to the next parametric study.

Table 4. Numerical simulation comparison with Mantovano, et al. [3]

Modul	Collapse pressure (MPa) From Ref [3]	External pressure input to numerical (MPa)	Buckling load factor ANSYS	Difference (%)
1	27,37	27,37	0,98	2%
2	26,53	26,53	1,05	5%
3	28,47	28,47	1	0%

2.2.2 DNVGL-OS-F101 Compliance Check

The second validation was conducted to check the cross-over pressure on the buckle arrestor. DNVGL-OS-F101 [2] provides manual calculation criteria for buckle arrestor thickness to prevent propagation buckling as given in Eqs. (1) and (2).

The data used for this calculation is shown in Table 5. It can be seen that the external pressure (2,358 psi) does not exceed the cross-over pressure of integral buckle arrestor (7,424 psi). Thus, Eq (1) fulfilled.

$$P_e \leq \frac{P_X}{1{,}1 \cdot \gamma_m \cdot \gamma_{SC}} \tag{1}$$

$$P_X = P_{pr} + \left(P_{pr,BA} - P_{pr}\right) \cdot \left[1 - EXP\left(-20\frac{t_2 \cdot L_{BA}}{D^2}\right)\right] \tag{2}$$

Table 5. Code compliance check calculated by Eqs. (1) and (2)

Symbol	Value	Unit
P_{pr}	3,416.74	psi
$P_{pr,BA}$	8,045.67	psi
D	32.8	in
L_{BA}	30	in
t_2	3.6	In
γ_m	1.15	ULS
γ_{SC}	1,260	High
P_e	2,358	Psi
P_X	7,424	Psi

3 Result and Analysis

The numerical simulation had been successfully conducted and the results are shown in Figs 4 to 9. There were 5 case studies of simulations that were executed separately for each configuration. The case studies covered (i) varied pipe thickness; (ii) varied buckle arrestor length; (iii) varied ratio of buckle arrestor to pipe thickness; (iv) varied pipe diameter; (v) varied of buckle arrestor material. The values of buckling load factor are extracted directly from numerical simulations. If the buckle load factor was higher than 1.0, the pipe was safe or no local buckling occurs.

The results of the first group simulation are shown in Fig. 4. In this case, the pipe thickness was varied in 5 thickness range from 1” to 2” with 0.25” increments at each step. The parameter of buckle arrestor length (L_{BA}), ratio buckle arrestor to pipe thickness (*h/t*), and buckle arrestor material were kept constant with a value of 40 in, 2, and steel grade X60, respectively. As can be seen in the Fig. 4, a slight increment of pipe thickness has a significant influence to the buckling load factor value. Various pipe diameter generates different buckling load factor but with a similar trend.

The simulation results of the second group are shown in Fig. 5. This case study aims to examine the effect of buckle arrestor length to the buckling load factor. The pipe thickness (*t*) was set at 2.0”, buckle arrestor material was set as steel grade X60 and the ratio of buckle arrestor to pipe thickness (*h/t*) was set at 0.25” while the buckle arrestor length (L_{BA}) was varied from 30” to 50” with an increment of 5” for every step. As shown in Fig. 5, there is no significant change in the buckling load factor with the increase of buckle arrestor length. A similar condition also occurs for different pipe diameters.

The simulation of the third group was varied the ratio of buckle arrestor to pipe thickness. Buckle arrestor length (L_{BA}), pipe thickness (*t*), and buckle arrestor material were determined to be at 40”, 1.25”, and steel grade X60, respectively, while the ratio of buckle arrestor to pipe thickness (*h/t*) was varied in the range of 1.5 to 3.5. The simulation result was shown in Fig. 6 which shows that buckle arrestor to pipe thickness does not significantly affect the buckling load factor at each pipe diameter.

The result of the fourth group simulation was shown in Fig. 7. In this case, the varied parameter was the pipe diameter. Meanwhile, buckle arrestor length, ratio of buckle arrestor to pipe thickness, and buckle arrestor material were maintained at 40”, 1.25”, and steel grade X60, respectively. As shown in the Fig. 7, the change of pipe diameter has a significant influence to the buckling load factor. The relationship between buckling load factor to pipe diameter is inversely proportional due to as the pipe diameter increases, the buckling load factor decreases.

The fifth group simulation was conducted by varying the material steel grade API Spec 5L from X42 to X80. The pipe thickness, pipe diameter, buckle arrestor length, and ratio buckle arrestor to pipe thickness was set at 1.25”, 30”, 2”, respectively. In Fig. 7 shows that the API Spec 5L material steel does not have a significant impact to the buckling load factor. Further study shows that the local buckling will propagate and cross over the buckle arrestor unless the ratio of buckle arrestor to pipe thickness is higher than 3 and otherwise, as shown in Figs. 9 and 10.

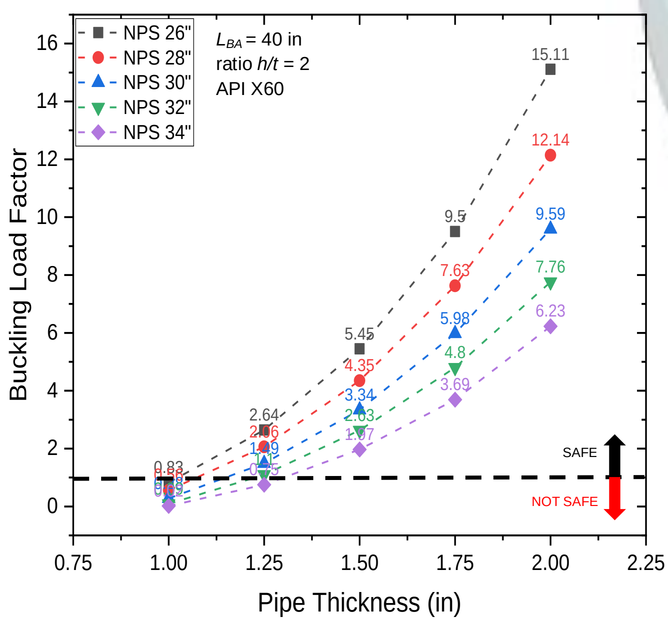

Fig. 4. Numerical simulation results: buckling load factor vs pipe thickness

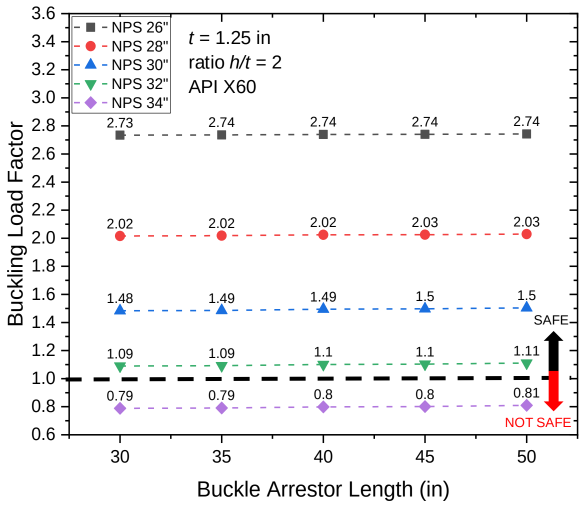

Fig. 5. Numerical simulation results: buckling load factor vs buckle arrestor length

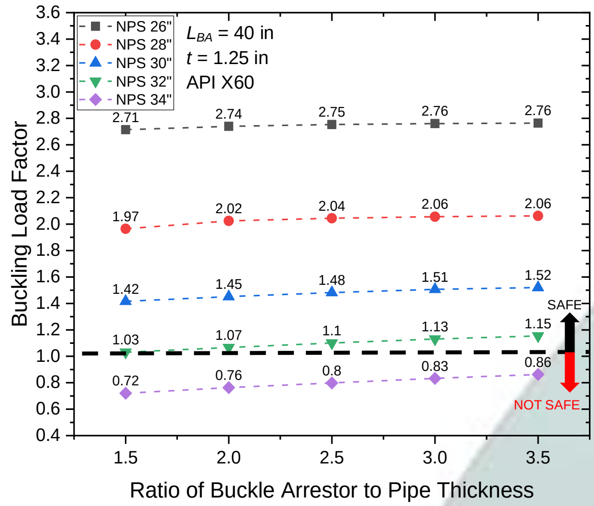

Fig. 6. Numerical simulation results: buckling load factor vs ratio of buckle arrestor to pipe thickness

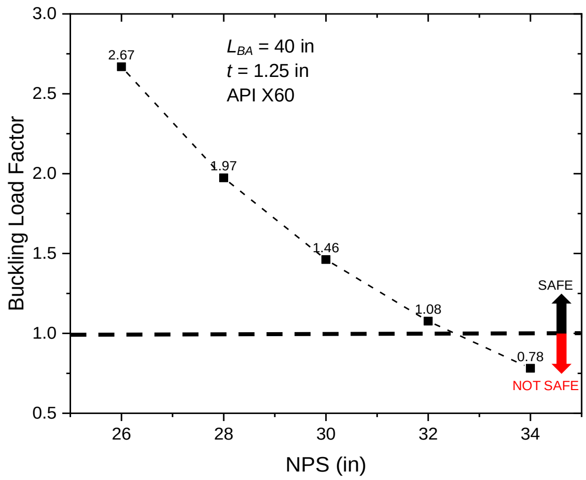

Fig. 7. Numerical simulation results: buckling load factor vs pipe diameter

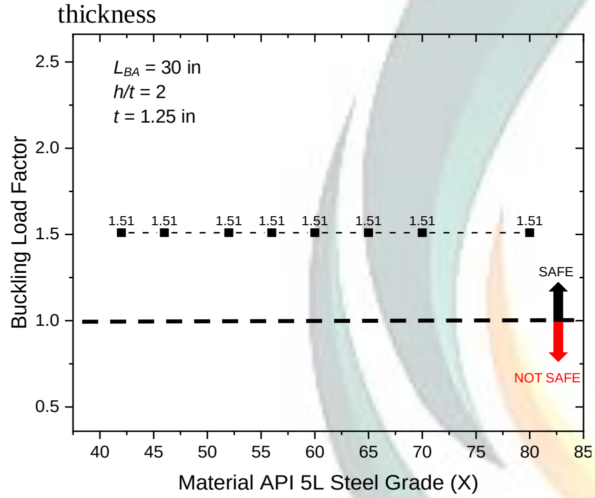

Fig. 8. Numerical simulation results: buckling load factor vs buckle arrestor material

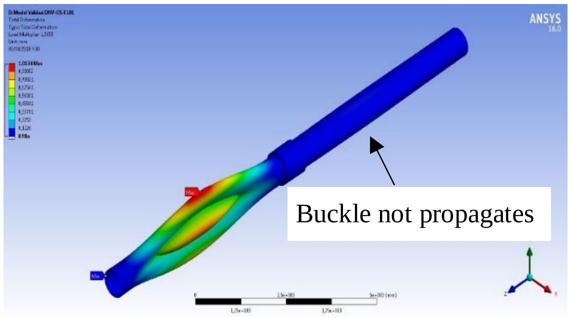

Fig. 9. Mode shape of pipe deformation NPS 28" when a ratio of buckle arrestor to pipe thickness is higher or equal to 3.0

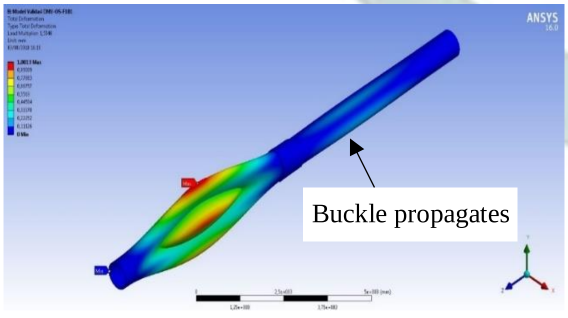

Fig. 10. Mode shape of pipe deformation NPS 28" when a ratio of buckle arrestor to pipe thickness is less than 3.0

4 Conclusion

Parametric study of the effect of geometry and material to buckling load factor has been successfully conducted by using numerical simulations. Five case study groups were carried out i.e. (i) varied pipe thickness; (ii) varied buckle arrestor length; (ii) varied ratio of buckle arrestor to pipe thickness; (iv) varied pipe diameter; (v) type of buckle arrestor material. In summary, buckling load factor was significantly influenced by pipe thickness and inversely proportional to pipe diameter. The other three parameters only provide a modest change of buckling load factor.

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