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# Engineering Structure and Strength Design of Reducer Bend under Internal Pressure

CHEN Sun-Yi[1], CHEN Jin[2], LIU Ceng-dian[2]

[1] The Challenge Petrochemical Machinery Corporation of Maoming, Guangdong maoming 525024, P. R. China

[2] Key Lab of Pressure Systems and Safety, East China University of Science and Technology, Shanghai 200237, P. R. China

**Abstract**: Standard reducer bends have structural functions of both bend and reducer pipe at the same time. However, there is a lack of strength design method in the current codes.

In order to develop the strength design method of the reducer bends subjected to internal pressure, analytical formulas are re-examined. Finite element analysis and stress measurements of the reducer bend are carried out. It is found that it is not appropriate to analyze the reducer bends by using thin membrane theory. The formula derived directly from circumferential stress formula of reducer bends under internal pressure is conservative, which is further verified by the finite element analysis results and it can thus be applied to piping design.

**Keywords**: strength design; stress analysis; reducer pipe; reducer bends; elbows

## 1 Introduction

Bends and reducers are normal pipe components used widely in pressure piping such as industrial piping, utility piping and long distance transmitting piping.

Some researchers have been focusing on reducers subjected to critical loads of instability and under external pressure, but there are very few reports about reducer bends.

Ali [1] analyzed the stability and stress of general truncated conical shells which were used as pipe-reducers, and pointed out that the critical load for a conical reducer decreases almost linearly with increasing apex angle of the conical frustum.

Rahman [2] analyzed the stability and stresses of parabolic pipe-reducers. He showed that doubly curved truncated parabolic shells can sustain a higher critical load than singly curved conical frustum. But fabrication of a parabolic shell is quite complicated. Instead of a toroidal shell, another doubly curved shell may be considered for its suitability under external loading.

Three cone cylinders subjected to external pressure and instability were tested, and test results in term of the critical pressure are compared with 2 simplified formulas, respectively[3]. Three cone cylinders with a thickness

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of 0. 5 mm are 430 mm, 570 mm and 730 mm long, respectively. Diameters of the large and small end are 260 mm and 160 mm, respectively. Cylinders with a length of 100 mm are welded to both ends. It was found that the measured critical pressures for the three cone cylinders are 41. 3%, 17. 9% and 13. 6% lower than those by one formula, and 55. 6%、30. 5% and 25. 8% lower than those by another formula, respectively.

Test results of critical external pressures are also compared with calculated results obtained from the relevant standard formula in reference[3].

Firstly, the calculation result obtained from standard formula are 56. 2%, 31% and 26. 3% larger than the test results respectively.

Secondly, the calculation by BSI rule (PD 5500) for cone cylinder elastic instability (not structural buckling) pressure is 61. 6% larger than the test result.

Finally, the calculation results of cone cylinder shell under buckling pressure are 47. 1%, 28. 7% and 18. 7% larger than the test results respectively.

These errors are probably caused by the preliminary shape tolerance of the specimens, residual stress, non-elastic action of the material, boundary non-linear sliding and friction as well as different boundary conditions.

Numerical simulation results in reference[3] shows that though the numerical outcomes were in agreement with the tests in terms of buckling load and modes of the deformations, the buckling load of numerical results rather overestimated the test values.

Reference[4] provides the theoretical basis and calculating method for the design and fabrication of reducer pipes. The finite element method was used to study the reducing bends under internal pressure and the results show that increasing the wall thickness of reducer bends is an effective way to mitigate the stress concentration.

Although reducer bend is also the standard pipe [5-6] and has structural functions of both bends and reducer bends, it is not often used in engineering and there are few reports about it. Figure 1 shows a complicated pipe component connection in some industrial plants where top elbow is connected with the bottom small elbow by an eccentric reducer bend and a nipple.

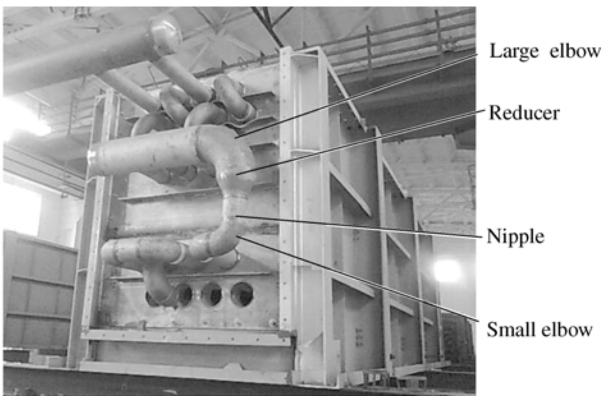



Figure 1 Complex piping components connection

In order to improve the connection quality, the above-mentioned three pipe components need to be replaced by the reducer bend shown in Figure 2.

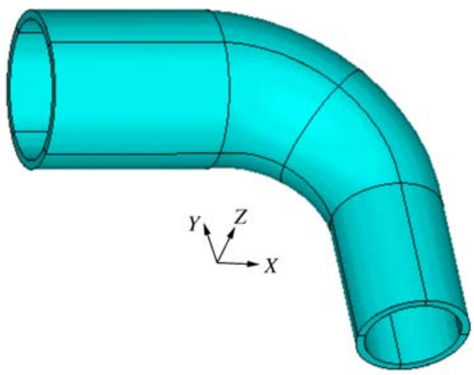



Figure 2 Model of bend reducer with tube on both ends

However, there's no corresponding standard for the strength design of reducer bends so that the strength design can only be verified by internal pressure test of the specimens. In addition, some analyses show that the pipe spoon used to the closing end of the pipe component in standard specified tests may be too short and sometimes it plays a strengthening role in the pressure resistance capability. Therefore, there are some uncertainties of this testing method. Apart from that, the minimum wall thickness of the bend specified in Part 102.4 of ASME B31.1-2011[7] or that of the reducer specified in Part 104.6 were also borrowed from minimum wall thickness of straight pipe under internal pressure in the standard.

In this paper, based on the relevant theory, strength design formulas of the reducer bends are derived. Then bearing internal pressure is analyzed to provide a basis for strength evaluation of the reducer bends used in the case shown in Figure 1.

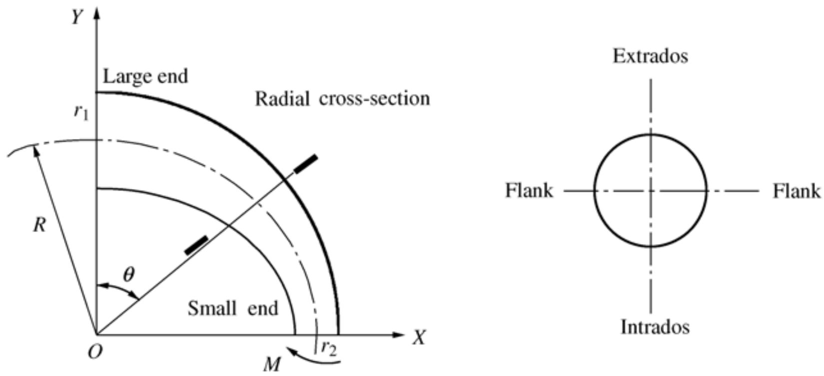
## 2 Internal pressure analysis of reducer bends

### 2.1 Parameter of the reducer bend

For convenient comparison, testing object is taken to be the same as the reducer bend under internal pressure in the stress analysis. Basic parameters of the test sample are listed in Table 1. Based on the similar test sample strength, material strength property of steel 20 is obtained by the empirical formula $\sigma_b \approx 3.5378\text{HB}$ MPa according to the China standard GB/T 1172-1999 Conversion of hardness and strength for ferrous metal.

Table 1 Main parameters of the test sample

| Size | ∅168 mm×∅89 mm×9. 5 mm |
|---|---|
| Bending radius $R$ | 226. 2 mm |
| Outside diameter of large end $D_1$ | 166. 5 mm |
| Shell thickness of large end $T_1$ | 9. 5 mm |
| Outside diameter of small end $D_2$ | 91. 7 mm |
| Shell thickness of small end $T_2$ | 12. 9 mm |
| Axis length | 355 mm |
| Large end Hardness | 109 HB |
| Large end Tensile strength | 385 MPa |
| Small end Hardness | 123 HB |
| Small end Tensile strength | 434 MPa |




a) Side section of reducing elbow b) Radial cross-section

Figure 3 Circumferential stress analysis model of reducing

## 2.2 Stress analytic formula under internal pressure

The author analyzed the structural characteristics of reducer bend: cross section radius of the reducer bends changes as the radial bending angle changes, the reducer bend is an axially dissymmetrical structure and proposed a circumferential stress formula for reducer bends without moment under internal pressure[8] that is

$$\sigma_{\varphi} = \frac{p[\pi r_{1i} - 2(r_1 - r_2)\theta]}{2t\pi\sin\alpha} \times \frac{2\pi R + [\pi r_{1i} - 2(r_1 - r_2)\theta]\sin\varphi}{\pi R + [\pi r_1 - 2(r_1 - r_2)\theta]\sin\varphi} \tag{1}$$

Where: $p$ is internal pressure, 4. 0 MPa is adopted in engineering case in Figure 1, 12 MPa and 18 MPa are selected for testing analysis respectively.

$\theta$ is radial return angle, according to testing stress analysis location, every 10 ° is determined to get the values and conversed into radian. Maximum returning angle shall be wrap angle of the reducer bends, here it is 90°

according to Figure 3.

$\varphi$ is included angle between bends radius of the cross section and neutral axial in circular segment, corresponding to the measure location of testing stress analysis. 0° is being in neutral line (Flank), 90° is being in extrados and −90° is also in intrados.

$\alpha$ is angle in bottom cone of the cross section. Here $\sin\alpha \approx 1$ is adopted.

$r_1$ is middle radius of large cross section of the reducing elbow, 78. 5 mm.

$r_2$ is middle radius of small cross section of the reducing elbow, 39. 4 mm.

$L$ is axial length of the reducer bends, 355 mm.

$R$ is bends radius at centerline of bending components, 226. 2 mm.

$r_{1i}$ is inner radius of the large cross, 73. 75 mm.

$t$ is wall thickness of the pipe components, values are obtained from measuring location with corresponding testing stress analysis.

The author proposed longitudinal stress formula of the reducing bends in reference [9] that the radial return angle $\theta$ is taken to be reducing variable, which can be obtained without moment:

$$\sigma_\theta = \frac{pr_{\theta i}^2}{2tr_\theta} \approx \frac{pr_{\theta i}}{2t} \tag{2}$$

Here, internal radius of the cross section for reducer bends with radial turning bending angle $\theta$ is:

$$r_{\theta i} = r_{1i}\left(1 - \frac{2}{\pi}\frac{r_1}{r_{1i}}\mu\theta\right) \tag{3}$$

among which reducer angle of non-dimensional coefficient reducing angle is:

$$\mu = \frac{r_1 - r_2}{r_1} = \frac{157 - 78.8}{157} \approx 0.50 \tag{4}$$

Meanwhile, it is pointed out that for normal standard reducer bend and thick wall reducer bends, the error from formula (2) is no more than 6. 0% and the calculation result is conservative. The error is within the accepted scope in engineering.

Formula for longitudinal stress in ASME B31. 1-2012 Para.102. 2

$$S_{lp} = \frac{PD_o}{4t_n} \tag{5}$$

and formula (2) have the similar form and influenced factors. Here $P$ indicates design pressure, $D_o$ indicates external radius and $t_n$ indicates bend wall thickness.

To compare formula (1) and formula (2), $\sigma_\varphi = 2\sigma_\theta$ happens at the centerline and has the same relation with that of the cylinder under internal pressure. But formula (1) and (2) haven't been verified by testing. Bring the relevant data into formula (1) and simplify it, we can obtain the circumferential stress:

$$\sigma_\varphi = \frac{(73.75 - 0.4344\theta)}{2} \times \frac{2 \times 226.2 + (73.755 - 0.4344\theta)\sin\varphi}{226.2 + (78.5 -- 0.4344\theta)\sin\varphi} \frac{p}{t} \tag{6}$$

In the spot of the intrados, we have

$$\sigma_{\varphi = -90^\circ} = (36.875 - 0.2172\theta) \times \frac{189.325 + 0.2172\theta}{73.85 + 0.2172\theta} \times \frac{p}{t} \tag{7}$$

At the neutral line, we have

$$\sigma_{\varphi = 0^\circ} = (73.75 - 0.4344\theta) \frac{p}{t} \tag{8}$$

In the spot of the extrados, we have

$$\sigma_{\varphi = 90^\circ} = (36.875 - 0.2172\theta) \times \frac{263.075 - 0.2172\theta}{152.35 - 0.2172\theta} \times \frac{p}{t} \tag{9}$$

Bring the relevant data into formula (2), longitudinal stress is simplified as

$$\sigma_\theta = (36.875 - 0.2180418750\theta) \frac{p}{t} \tag{10}$$

Draw a stress distribution curve of the testing value and the value by calculating-formula (6) to formula (9), and then make a comparison.

## 2.3 Strength calculation under internal pressure

1) Strength calculation formula. Based on the existing analysis, longitudinal stress is the maximum stress of the round cross section, and longitudinal stress in the inner intrados ($\varphi = -90°$) at the large end of reducer elbow is maximum which is the key of determining strength of the reducer elbow. It is a little risky to employ $r_1 \approx r_{1i}$, theoretically, more than 73% is thin wall in reducer elbow standard, error resulting from this replacement is usually 3%; but, it is conservative when taking $\sin\alpha \approx 1$, so these two kinds of errors can be counteracted with each other. Bring these simplified values into formula (1) and we have the following:

$$\sigma_{\varphi\max} = \frac{pr_{1i}}{2t} \frac{2R - r_{1i}}{R - r_{1i}} \tag{11}$$

Suppose the maximum circumferential stress is not larger than the allowable stress of the reducer elbow at the design temperature, the following formula can be obtained:

$$\frac{pr_{1i}}{2t} \frac{2R - r_{1i}}{R - r_{1i}} \leqslant [\sigma]^t \tag{12}$$

Taking welding efficiency $\varphi$ into account, reducer bends wall thickness strength calculation formula can be sorted out as:

$$t \geqslant \frac{pr_{1i}}{2[\sigma]^t\varphi}\frac{2R - r_{1i}}{R - r_{1i}} \tag{13}$$

If formula (13) conversion is expressed in external radius, with consideration of wall thickness additional amount, minimum wall thickness calculation formula of the elbow with internal pressure

$$t_m = \frac{PD_O}{2(SE/I + Py)} + A \tag{14}$$

has a similar form and affected factors to Para.102. 4. 5 of ASME B31. 1-2012.

In formula (14), $P$ is the design internal pressure, $D_O$ is the pipeouter diameter, $SE$ is the maximum allowable stress of material due to internal pressure and joint efficiency at the design temperature, psi (MPa), $y$ is the coefficient less than 1, in the intrados area $I>1$, in the extrados area $I<1$, in the centerline $I=1$, $t_m$ is the required minimum wall thickness, and $A$ is the required additional surplus wall thickness considering factors of the pipe components machine or medium corrosion.

2) Example of calculation. Bring parameters of the above case into formula (13), among which allowable stress is 147 MPa as per GB150. 2-2011 at the temperature of 100 ℃. The welds shall be full 100% penetration tested and thus $\varphi = 1.0$. Compared with the actual wall thickness 9. 5, there's still 7. 05 mm corrosion allowance of the required wall thickness

$$t = \frac{4.0 \times 73.75}{2 \times 147 \times 1.0}\frac{2 \times 226.2 - 73.75}{226.2 - 73.75} \approx 2.49 \text{ mm}$$

3) Engineering solution. There are six types of forming technology to manufacture reducer bends: the 1st one can be applicable to straight pipe with smaller reducing degree and hot bespoke-pulled by one step; the 2nd one can be applicable to larger reducing degree pipe bespoke-pulled step by step; the 3rd one can be applicable to assembly welding by half pressing in lower work condition; the 4th one is the hot bespoke-pull method of eccentrically reducer bends as the pipe blanket; the 5th one is combination of hot bespoke and assembly welding and the 6th one is 3D printing method. However, there's no corresponding strength design standard for any reducer bend till now. Usually, longitudinal welds are not permitted to exist in elbows or reducer bends in chemical and industrial plant, strength of the reducer bends formed by electric arc additional manufacture printing cannot meet requirements of the engineering, and the cost of hot bespoke-pulled to blank by model is much higher. Currently, one applicable method is that blank is hot pressed to be elbow by the traditional mature technology firstly, then open a narrow and long dovetail groove in the extrados at one end of elbow, remove part of the wall thickness, see Figure 4, and then hot press for second time to close the dovetail groove, and at last weld the opening. A kind of half seamless and half seaming combination structure can be manufactured by this new technology.

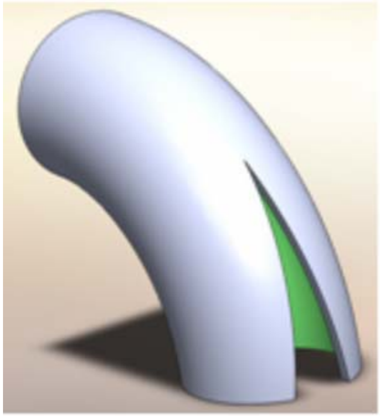


Figure 4 Dovetail opening in the wall of elbow

4) Welding location design. Formula (13) does not consider the reducer bends under combined loadings such as internal pressure, moment and torque at the same time. Under various loads, the maximum stress that determines the strength of reducer bends is in different positions, and stress constituent and its formula are different. The product forming process and the weld position are also different. For example, a dovetail groove should be opened in the neutral line of reducer bends that mainly bear moment in bending plane.

5) Weld heat treatment. In engineering, besides well-designed weld bevel, different technical methods can be employed in closing weld of different materials. PWHT can be waived for carbon steel weld; its higher strength and hardness maintained are useful for resisting abrasive corrosion for flowing fluid. Regarding stainless steel welds, solution treatment shall be done to recover its good corrosion resistance; Cr-Mo steel weld shall also be PWHT to relief its residual stress and to refine grains.

6) Equal wall thickness treatment. With regard to seamless reducer bends, hot bespoke-pulled forming is a diameter enlarging process of small diameter pipe blank. At the same time, wall thickness will decrease gradually as the diameter becomes larger. Gradually reducing the wall thickness of the blank before forming and grinding treatment after forming, can both yield reducer bends with equal wall thickness.

## 3 Testing of reducer bends under internal pressure

### 3.1 Test plan

1) Specific dimension and actual measurement hardness values of the test samples are shown in Table 2. Ultrasonic probe parting surface were placed parallel and vertically to elbow axis separately to get two readings, and then to calculate their average value. Their difference is less than 0.5% and the latter is a little larger. Wall thickness in the table is measured when probe is vertical to axis of the elbow. Cross section and orientation by equal division points are shown in Figure 5, inner diameter in AE orientation is larger than that of CG orientation, but outer

diameter is slight smaller than that of CG orientation. The result shows the ovality of the cross section.


a) Radial cross section sequence
b) Eight equalaadivision points on elbow cross

Figure 5 Cross section and equal division points

Table 2 Actual measuring parameters of reducer elbow test coupon

| Cross Sec. No. | Hardness at $A$ neutral line/ HB | Hardness at $E$ neutral line/ HB | Avg. Thk of 8 equal division points From $A$ to $H$/mm | $OD$/mm | $t/r$ | Excircle roundness of cross-section/% |
|---|---|---|---|---|---|---|
| 1 | 104. 7 | 112. 7 | 9. 5 | 166. 5 | 0. 121 | 0. 7 |
| 2 | 106. 3 | 103. 3 | 10 | 159. 5 | 0. 134 | / |
| 3 | 100. 7 | 113. 3 | 10. 38 | 149. 6 | 0. 149 | / |
| 4 | 110. 3 | 107 | 10. 53 | 144. 8 | 0. 157 | / |
| 5 | 115. 3 | 118 | 11. 28 | 134. 3 | 0. 183 | / |
| 6 | / | / | / | (130. 6) | / | 2. 2 |
| 7 | 107. 3 | 116. 3 | 11. 83 | 125. 1 | 0. 209 | / |
| 8 | 113. 7 | 120. 3 | 12. 7 | 115. 9 | 0. 246 | / |
| 9 | 120. 7 | 125. 7 | 13. 73 | 108. 2 | 0. 291 | / |
| 10 | 118. 3 | 127. 3 | 14. 03 | 99. 3 | 0. 329 | / |
| 11 | / | / | 12. 9 | 91. 7 | 0. 327 | 0. 6 |
| Avg. value | 110. 8 | 124. 2 | 11. 69 | 117. 7 | 0. 221 | / |

2) Testing system. Strain would be measured by automatically scanning via the programmable controlled YJ-33 static electric resistance strain instrument and data accumulating system. Pressure increasing instrument of the bend is the 2D1-SY pressure test pump. Generally, each measured point shall be stuck with a 90° rectangular electrical resistance strain disc along two axes in order to obtain the longitudinal strain and circumferential strain, as shown in Figure 6.

Figure 6 Test sample

Strain measured directly by bi-direction strain disc can be used to calculate principal stress via generalized Hook's law[10]:

$$\sigma_{\varphi} = \frac{E(\varepsilon_{\varphi} + \nu\varepsilon_{\theta})}{1 - \nu^2} \tag{15}$$

$$\sigma_{\theta} = \frac{E(\varepsilon_{\theta} + \nu\varepsilon_{\varphi})}{1 - \nu^2} \tag{16}$$

### 3.2 Test results

1) Obtained stress. The results of strain test are converted into stresses by formulas (15) and (16), and the stress curves could be drawn. The circumferential and longitudinal stresses at neutral line are shown in Figure 7 and Figure 8. Note that Poisson's Ratio $\nu = 0.3$ and Young's modulus $E = 196$ GPa are used in calculation according to standard.

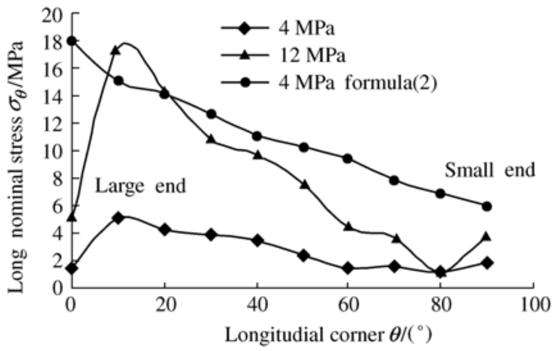



Figure 7 Longitudinal stresses curves at neutral line

2) Corresponding stress curve analysis. As shown in Figure 7, when internal pressure increases, longitudinal stress curve becomes abrupt; longitudinal stresses are all tensile stress, in terms of the level of stress at the maximum internal pressure. Sequence of the maximum longitudinal stress from high to low are in intrados, neutral line and extrados separately. Stress curves concentration degree under different internal pressure from high to low are in intrados, extrados and neutral line respectively.

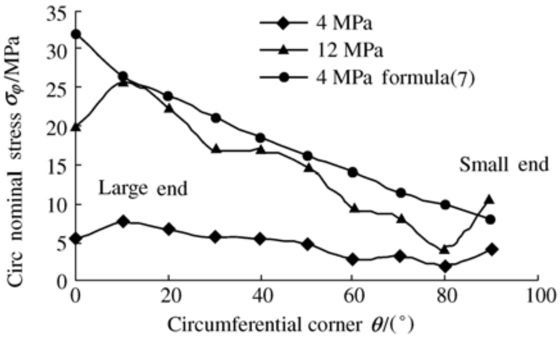



Figure 8 Circumferential stresses curves at neutral line

As shown in Figure 8, circumferential stress curve has the same tendency as that of longitudinal stress curve. Circumferential stress has not reached two times that of the longitudinal stress. By comparison of circumferential stress of different longitudinal lines, maximum, intermedium and minimum circumferential stress exist in intrados, neutral line and extrados respectively.

No matter whether it is longitudinal or circumferential stress, analytical values are apparently three to six times higher than that of measured stress which shows that analytical stress values are obviously conservative.

3) In Table 3, the FEA calculation results for the intrados of the large end on two reducer bends are compared with those of formula (1) and formula (2) respectively. The structure dimensions of the reducer bend are based on the Standard GB/T 12459-2005. The elastic modulus $E = 200$ GPa, hardening modulus $E_1 = 2$ GPa and Poisson ratio $\nu = 0.3$ of the material in FEA.

Table 3 Stress comparison

| Case | | $\sigma_\varphi$/MPa | $\sigma_\theta$/MPa |
|---|---|---|---|
| | FEM result[4] | 92.26 | 21.47 |
| 1 | Formula results | 88.27 | 33.55 |
| | Errors | −4.3% | +56.3% |
| | FEM result | 217.84 | 92.60 |
| 2 | Formula results | 254.17 | 86.26 |
| | Errors | +16.68% | −6.85% |
| | | Formula (1) | Formula (2) |

In case 1, the reducer bend is 90E(L)R-50×40 Ⅰ-Sch40 with diameter of the large end 3.91 mm and the small end 3.68 mm and under internal pressure of 10 MPa in Reference[4]. Elasticity analysis is carried out. There's no straight piping at both ends of the model and axial movement is limited at both ends.

In case 2, the reducer bend is 90E(L)R-50×40 Ⅰ-Sch40 with the thickness of 9.5mm and under internal pressure of 10 MPa. Elasticity analysis is performed in Reference[11]. The larger end of the model is fixed, and the material strength at this end is increased deliberately. Contour of longitudinal and circumferential stresses can be seen in Figure 9 and Figure 10, respectiveily.

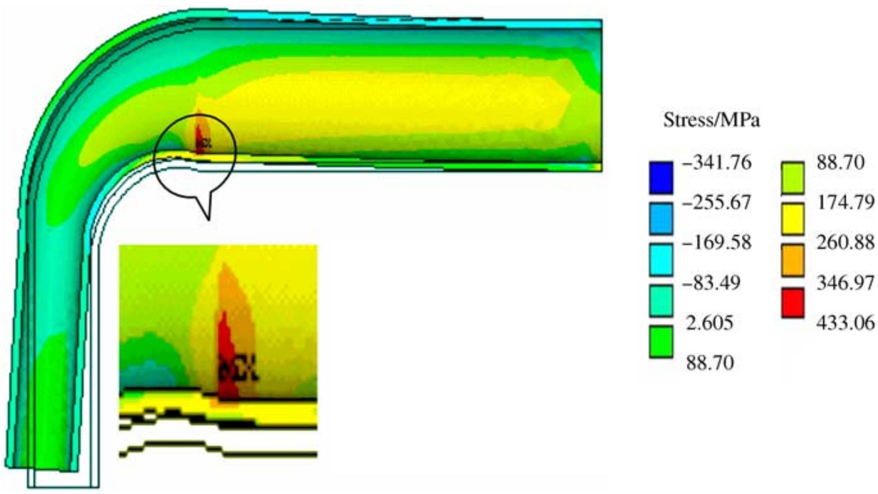



Figure 9 Circumferential stress distribution

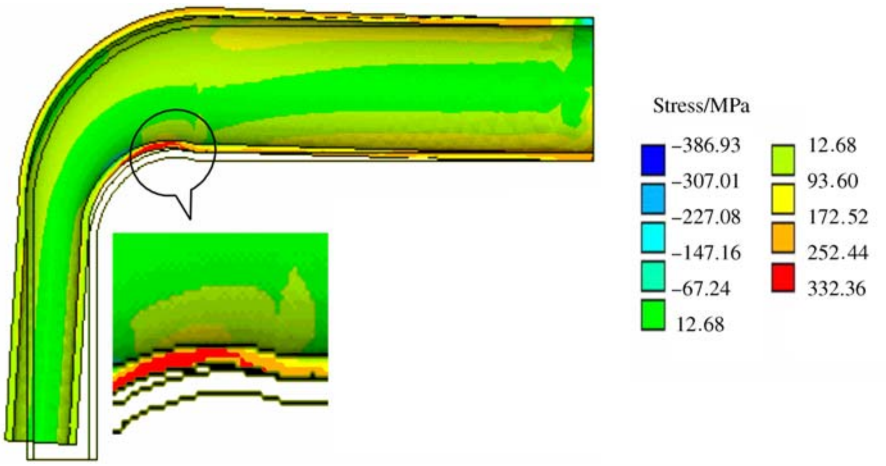



Figure 10 Longitudinal stress distribution

The data in Table 3 were analyzed considering Figure 9 and Figure 10. Both ends of the model in case 1 limit axial movement, and the effect of connecting piping is not considered. The result of formula (1) is 4.3% lower than the FE circumferential stress so that formula (1) is in danger.

In case 2, the axial tension effect due to the actual internal pressure is considered on both ends of the model both of which are connected to piping. The result of formula (1) is 16.68% higher than the FE longitudinal stress so that formula (1) is conservative and meets the requirement of the engineering safety.

## 4 Errors analysis

1) It is assumed that stress of pipe components during stress analysis is base on elastic membrane theory, but actually, reducer pipe test sample is of a thick wall structure. Some researchers found out that when $t/r$ is more than 0.05, the error caused by thin shell theory is acceptable in engineering which is verified by a lot of comparisons and trial calculation [12]. In fact, besides being subject to internal pressure, engineering pipe components are also subject to the effect of moment and gravity. As the wall thickness increases (as shown in

Table 2, $t/r$ is far larger than 0.05, as large as 342%), radial stress of the wall thickness under internal pressure cannot be ignored. The FEA results that are more realistic to show that the circumferential stress formula is conservative.

2) During the analysis of reducer bends, some deviations arise due to the idealization of the mathematic model and its assumption condition, the difficulties of FEA model building, the inaccuracy of its boundary condition, visual dimension tolerance of the specimen, etc.

3) In fact, since the reducer bend is an axially dissymmetrical structure, bending moment exists in wall thickness under internal pressure. Regarding to circumferential bending moment; on the one hand, it comes from the difference of pressurized area between extrados and intrados on the neutral line [13]; on the other hand, it comes from ovality of inner and external circles of the pipe cross section, and long and short ovals of inner and external roundcircles are not conforming and vertical to each other.

Longitudinal bending moment comes from the difference of pressurized area between extrados and intrados of the neutral line on the one hand [14]; on the other hand, it comes from the difference of pressurized area between large end and small end sections. Actually, reducer bend is belonging to neither the thin wall structure nor the rotated shell of axial symmetry, but a kind of hyperboloidal shell with two main curvatures changed gradually. Moment of the shell under internal pressure cannot be ignored, but its mathematic calculation is extremely complicated. Therefore, it is not appropriate to analyze such kind of reducer bends by the membrane theory of thin-wall and axial symmetry structures.

4) According to result in Table 2, cross section roundness of reducer bend pipe test-piece is up to 2.2%, there's only roundness tolerance for end of pipe component in relevant standard, but there's no roundness deviation requirement for the middle part of the elbow. So there should have been $\theta = 0°$ and 90° in the circumferential stress formula when $\alpha = 90°$ and $\sin\alpha = 1$. But, now $\sin\alpha \approx 1$ in all cases without difference and that is one of the errors.

## 5 Conclusions

1) Stress analytical formulas value and testing stress measurement result under internal pressure are not in compliance. The analytic formulas value is apparently larger so that it is conservative to be applied in engineering. This error mainly results from the total stress from the membrane theory and moment reaction from the shell thickness and the dissymmetrically bi-hyperbolical shell.

2) Based on the circumferential stress formula of the danger cross section of the reducer elbow under internal pressure, strength calculation formula of wall thickness is derived directly. Testing results show that wall thickness calculation formula is conservative as well as the stress formula. The FE results also show that the circumferential stress formula is conservative, so it can be applied to engineering piping design.

3) The quality of reducer bends fabricated with the new half seamless and half seaming technology can be guaranteed and is suitable for engineering applications.

## References

[1] Ali G M Z. Stability and stress analysis of conical reducers[M]. Bangladesh University of Engineering and Technology, 1991

[2] Rahman M A. Stability and stress analysis of parabolic pipe reducers[M]. Mechanical Engineering Department, Bangladesh University of Engineering & Technology, Dhaka, Bangladesh, 1994

[3] Ghazijahani T G, Showkati H. Experiments on conical shell reducers under uniform external pressure[J]. Journal of Constructional Steel Research, 2011,(67):1506-1515

[4] Tao C D, LI Q. Stress analysis of reducer bends under internal pressure[J]. Applied Mathematics and Mechanics, 2014,35(Z):50-54 (in Chinese)

[5] Standard GB/T 12459. Steel Butt-welding Seamless Pipe Fittings, 2005

[6] Standard ASME B16.9. Factory-made Wrought Butt-welding Fittings, 2012

[7] Standard ASME B31.1. Power Piping, 2012

[8] Chen S Y, Liu C D, Chen J, et al. Circumferential membrane stress of bend reducer[J]. Pressure Vessel Technology, 2007,24(2):35-39 (in Chinese)

[9] Chen S Y, Liu C D. Meridian membrane stress of reducing elbow[J]. Journal of Plasticity Engineering, 2005,12(2):48-51 (in Chinese)

[10] Xu Z L. Elasticity (the 4 edition,and the second half volume)[M]. Beijing:Highter Education Publishing Company, 2006:167-168、270 (in Chinese)

[11] Chen S Y. Analysis and research on stresses and limit load of reducer[D]. Shanghai: East China University of Science and Technology, 2006

[12] China Standardization Committee on Pressure Vessels. Explain on JB 4732-95 steel pressure vessels design by analysis[M]. Beijing: China Standard Publishing Company,1995:99-106 (in Chinese)

[13] Chen S Y, Liu C D, Chen J, et al. The force difference and its circumferential equi-moment induce by the inside surface difference between flank-extrados and flank-intrados of loop under internal pressure[C]// Proceedings of the Seventeenth National Conference on Structural Engineering, 2008 (in Chinese)

[14] Chen S Y, Liu C D, Chen J, et al. The force difference and its meridian equi-moment induce by the inside surface difference between flank-extrados and flank-intrados of loop under internal pressure[J]. Pressure Vessel Technology, 2007,24(7):21-26 (in Chinese)

## Brief Biographies

**CHEN Sun-yi** is a Ph.D in the Challenge Petrochemical Machinery Corporation of Maoming. His research interests include design and fabrication of equipment subjected to pressure. sunyi_chen@sohu.com

**CHEN Jin** is a Senior Engineer in East China University of Science and Technology. His research interests include failure analysis and safety assessment of pressure vessels and pipelines. chenjin@ecust.edu.cn

**LIU Ceng-dian** is a professor in East China University of Science and Technology. His research interests include safety assessment and life extension technology of pressure vessels and pipelines. chenjin@ecust.edu.cn