Golden Sun-Rise
International Journal of Multidisciplinary on Science and Management
ISSN: 3048-5037/ Volume 2 Issue 3 Jul-Sep 2025/ Page No: 10-19
Paper Id: IJMSM-V2I3P102/ Doi:10.71141/30485037/V2I3P102

Research Article

Adaptive Nozzle Load Envelope (ANLE): A Probabilistic Framework for Early-Stage Design of Pressure Vessels and Piping Systems

Farzad Gardaneh[1], Walther Stikvoort[2]

[1] Senior Static Equipment Engineer, Tehran, Iran.
[2]Consultant Pressure Equipment Integrity, Assen, Netherlands.

Received: 01 July 2025 Revised: 03 July 2025 Accepted: 04 July 2025 Published: 07 July 2025

Abstract - *Design of pressure vessel and piping system is often hindered by the linear order of design steps, resulting in expensive and time-consuming redesign. In this article, the Adaptive Nozzle Load Envelope (ANLE) is presented as a new approach that enables vessel and piping simultaneous design by establishing probabilistic boundaries for the loads at the vessel-nozzle interface during the initial phase of design. ANLE employs a simplified piping flexibility model, characterized by an effective length parameter, to determine reaction forces without the need for a detailed piping configuration. The use of ANLE is demonstrated by a case study on a petrochemical vessel, where ASME compliance and other internationally accepted codes are achieved with design efficiency improvements. The proposed methodology is contrasted with current practice, with both advantages and development needs areas identified.*

Keywords - *Pressure Vessel, ANLE, Effective Length, Piping Load.*

I. INTRODUCTION

Pressure vessels in petrochemical plants are critical components that must reliably withstand internal pressure, thermal stresses, and external loads imposed by connected piping systems. Traditionally, vessel design is completed before piping design, necessitating conservative estimates for nozzle loads. This sequential approach often results in overdesign or requires late-stage adjustments, both of which can impact cost and project timelines. The Adaptive Nozzle Load Envelope (ANLE) methodology offers a solution to this limitation by introducing a simplified, probabilistic framework for defining allowable nozzle loads early in the design process. ANLE enables concurrent design of vessels and piping systems by providing a rational basis for nozzle load specification, even in the absence of detailed piping layouts.

ANLE represents piping flexibility through an effective length parameter and applies convexified probabilistic limits to capture the uncertainty inherent in early-stage design. This approach promotes robust vessel design while granting piping engineers greater flexibility to manage load constraints. As a result, the design process involves fewer iterations and achieves improved overall project efficiency. This paper presents the theoretical foundation and practical implementation of the ANLE approach. A case study is used to demonstrate its application, compare its performance to conventional industry practices, and evaluate its potential integration into standard engineering workflows.

II. THEORETICAL BACKGROUND

The interface between a pressure vessel and its attached piping is subjected to various forces (F_x, F_y, F_z) and moments (M_x, M_y, M_z), which arise primarily due to thermal expansion, internal pressure, and the self-weight of the piping system. In the early stages of design, detailed piping layouts are often unavailable. As a result, simplified models are used to estimate the mechanical response at the vessel-piping interface. A common simplification represents the piping system as a spring characterized by translational and rotational stiffness, both defined with respect to an effective length L. This effective length is the notional length of a straight pipe that would exhibit the same stiffness as the actual, more complex piping layout (including bends, supports, and other structural features). This simplification allows engineers to assess displacements and loadings at the nozzle interface due to thermal and mechanical effects with minimal initial data.

A. Stiffness Models

The translational stiffness (k_p) and rotational stiffness (k_θ) are given by:

$$k_p = \frac{3EI}{L^3}$$

$$k_\theta = \frac{GJ}{L}$$

These equations originate from fundamental mechanics:

- The translational stiffness k_p is derived from beam theory, specifically from the deflection of a cantilever beam under a point force at its free end. For a beam of length L, Young's modulus E, and moment of inertia I, the deflection δ under a force F is $\delta = \frac{FL^3}{3EI}$. This can be rearranged to give $k_p = \frac{3EI}{L^3}$, where k_p is the force per unit displacement ($F = k_p\delta$).
- The stiffness of rotation k_θ is from the torsion theory for a circular shaft. The angle of twist θ by torque M is $\theta = \frac{ML}{GJ}$, with G as the shear modulus and J as the polar moment of inertia. Substituting yields $k_\theta = \frac{GJ}{L}$, where k_θ is the torque per unit twist ($M = k_\theta\theta$).

Thermal displacement δ when there is a change in temperature ΔT is derived as:

$$\delta = \alpha L \Delta T$$

where α is the thermal expansion coefficient. These equations are used by the ANLE technique to estimate loads based on probabilistic bounds for modeling uncertainties in L, material properties, and conditions of operation.

B. Application in Probabilistic Analysis

These simplified stiffness and displacement models form the basis of the ANLE (Approximate Nozzle Load Envelope) technique. This method uses probabilistic bounds to estimate loads at the nozzle, accounting for uncertainties in:

- Effective length L
- Material properties (e.g., E, G, α)
- Operational conditions (e.g., temperature variations)

By treating the system in terms of effective stiffness and thermal response, engineers can make early design decisions even in the absence of detailed piping layouts.

III. METHODOLOGY

The ANLE (Analytical Nozzle Load Envelope) approach follows a systematic workflow to define and validate nozzle load limits early in the design process:

A. Define Vessel Parameters

Input key vessel characteristics, including geometry, material properties, internal pressure, and operating temperature.

B. Estimate Effective Length

Use established empirical correlations (e.g., Kellogg, Koves methods) to determine the effective length and wall thickness of connected piping.

C. Calculate Loads

Compute thermal displacements and the resulting loads on nozzles using stiffness values derived from piping flexibility analysis.

D. Apply Safety Factors

Introduce probabilistic safety margins to account for uncertainties:

- Forces: Factor of 1.5
- Moments: Factor of 1.2

E. Validate Against Code

Verify that calculated loads do not exceed allowable limits defined by ASME Section VIII, Division 2 [1]. Refine parameters as needed to ensure compliance. This early definition of nozzle load limits empowers piping engineers to optimize layouts while reducing the risk of later design conflicts.

IV. CASE STUDY

A cylindrical pressure vessel designed for petrochemical service is analyzed using the ANLE method. Key specifications include:

- Internal design pressure: 25 barg
- Design temperature: 150 °C
- Outside diameter: 1200 mm
- Nominal wall thickness: 16 mm
- Corrosion allowance: 1.5 mm
- Length: 6000 mm
- Nozzle size: 12-inch Schedule XS (outside diameter 323.9 mm, effective thickness 11.1125 mm uncorroded, 9.6125 mm corroded) with a flanged connection as per ASME B16.5 [3] (e.g., Class 300 flange)
- Nozzle location: 800 mm from the tangent line, 225 mm projection
- Materials: Shell and reinforcing pad (ASTM A516 Gr.70) [4], nozzle (ASTM A106 Gr.B) [4]

The vessel is equipped with a reinforcing pad (80 mm wide, 16 mm thick) at the flanged nozzle junction to enhance local strength. The ANLE methodology is applied to define the allowable nozzle loads, focusing on thermal effects as the primary load contributor.

A. Step 1: Effective Length and Stiffness Calculation

The effective length L is determined by Kellogg's method [6], as adapted for a pad-reinforced nozzle by Koves' method [7]: $L = 0.017 \cdot I \cdot \left(\frac{R}{r_m^2 \cdot t_{\text{eff}}}\right)^{3/2}$, where $t_{\text{eff}} = (T^{2.5} + \mathrm{t}_p^{2.5})^{0.4}$ is the effective combined thickness. For the 12-inch Schedule XS pipe:

- Vessel OD: 1200 mm = 47.244 inches, so $R = 47.244/2 = 23.622$ inches
- Vessel thickness: $T = 16 - 1.5 = 14.5\,\text{mm} = 0.5709$ inches (accounting for corrosion allowance)
- Pad thickness: $t_p = 16\,\text{mm} = 0.63$ inches
- Effective thickness: $t_{\text{eff}} = (0.5709^{2.5} + 0.63^{2.5})^{0.4} \approx 1.2009^{0.4} \approx 0.7937$ inches
- Mean nozzle radius: $r_m = 5.875$ inches (as per sample)
- Moment of inertia: $I = 361.54393\ \text{in}^4$

$$L = 0.017 \cdot 361.54393 \cdot \left(\frac{23.622}{5.875^2 \cdot 0.7937}\right)^{3/2}$$

$$L \approx 0.017 \cdot 361.54393 \cdot (1.0863/0.863)^{3/2} \approx 0.017 \cdot 361.54393 \cdot 1.304 \approx 4.9216\,ft = 1500\,mm$$

Using L = 1.5 m, the piping stiffness is calculated:

- Young's modulus: $E = 195$ GPa
- Moment of inertia: $I = 117{,}296{,}115\ \text{mm}^4$
- Shear modulus: $G = 75$ GPa

- Polar moment of inertia: $J = 234{,}592{,}232\ \text{mm}^4$

$$k_p = \frac{3 \cdot 195 \cdot 10^9 \cdot 117{,}296{,}115 \cdot 10^{-12}}{(1.5 \cdot 10^3)^3} \approx 10.34 \cdot 10^6\ N/m$$

$$k_\theta = \frac{75 \cdot 10^9 \cdot 234{,}592{,}232 \cdot 10^{-12}}{1.5} \approx 11.71 \cdot 10^5\ Nm/rad$$

B. Step 2: Thermal Displacement and Load

With a coefficient of thermal expansion $\alpha = 12.4 \times 10^{-6}\ °\text{C}^{-1}$ and a temperature change $\Delta T = 130\ °\text{C}$ (for an installation temperature of 20 °C), the thermal displacement along the thermal path (approximated by $L = 1.5\ \text{m}$) is:

$$\delta_{thermal} = 12.4 \times 10^{-6} \cdot 1.5 \cdot 130 \approx 2.418\ mm$$

The axial force due to thermal expansion is:

$$F_{z,thermal} = k_p \delta_{thermal} = 10.34 \cdot 10^6 \cdot 0.002418 \approx 25.002.12\ N$$

C. Step 3: Total Loads and Load Envelope

The total loads at the flanged nozzle are purely thermal in nature:

a. Thermal Contributions

- **Thermal Displacement**: The increase in temperature ($\Delta T = 130\ °\text{C}$) causes the piping to expand along the effective length $L = 1.5\ \text{m}$:

$$\delta_{\text{thermal}} = 12.4 \times 10^{-6} \cdot 1500 \cdot 130 = 2.418\ \text{mm}$$

- **Axial Force**: Knowing the translational stiffness $k_p = 10.34 \cdot 10^6\ \text{N/m}$, the force in the nozzle along the pipe axis direction, namely the radial z-direction, is calculated as follows,:

$$F_{z,\text{thermal}} = 10.34 \cdot 10^6 \cdot 0.002418 \approx 25.00\ \text{kN}$$

- **Circumferential Moment**: The thermal displacement causes a rotation at the nozzle, which is approximated to be $\theta = \frac{2 \cdot 2.418}{1500} \approx 0.003224$ radians. Given rotational stiffness $k_\theta = 11.71 \cdot 10^5\ \text{Nm/rad}$ this gives:

$$M_{y,\text{thermal}} = 11.71 \cdot 10^5 \cdot 0.003224 \approx 3.776\ \text{kNm}$$

b. Total Loads (thermal only)

- $F_x = 0\ \text{kN}$
- $F_y = 0\ \text{kN}$
- $F_z = 25.00\ \text{kN}$
- $M_x = 0\ \text{kNm}$
- $M_y = 3.776\ \text{kNm}$
- $M_z = 0\ \text{kNm}$

Using probabilistic bounds (forces, safety factors of 1.5; moments, 1.2) according to ASME Section VIII Division 2, 2019, Subsections 3.2 and 3.3, the ANLE load envelope is:

- $|F_x| \leq 0\ \text{kN}$
- $|F_y| \leq 0\ \text{kN}$

- $|F_z| \leq 25.00 \cdot 1.5 \approx 37.50\,\text{kN}$
- $|M_x| \leq 0\,\text{kNm}$
- $|M_y| \leq 3.776 \cdot 1.2 \approx 4.53\,\text{kNm}$
- $|M_z| \leq 0\,\text{kNm}$

D. Step 4: Validation with Stikvoort's Methodology

Stikvoort's method [2] calculates individual allowable loads of radial load, longitudinal moment, and circumferential moment for a nozzle on a cylindrical shell, based on design stress and geometry under corroded conditions. Allowable loads are computed at two points: (1) adjacent to the nozzle neck (with $T + t_{\text{pad}}$) and (2) where the reinforcing pad intersects with the cylindrical shell (with T), with lower values being critical.

a. Design Stress and Geometry (Corroded Condition

- Material: ASTM A516 Gr.70
- Vessel outside diameter: $D_0 = 1200\,\text{mm}$
- Corroded vessel wall thickness: $T = 16 - 1.5 = 14.5\,\text{mm}$
- Corroded nozzle thickness: $t = 9.6125\,\text{mm}$ (0.875 × 12.7 - 1.5 mm corrosion allowance)
- Reinforcing pad thickness: $t_{\text{pad}} = 16\,\text{mm}$

i). Location 1: Next to Nozzle Neck (with $T + t_{pad}$)

- Effective thickness: $T = 14.5 + 16 = 30.5\,\text{mm}$
- $C_N = \frac{30.5}{9.6125} \approx 3.172951886$
- $C_{11} = \frac{1200-30.5}{2\cdot 30.5} = 19.17213115$
- $C_{21} = \frac{\sqrt{19.17213115}}{\pi\cdot 30.5\cdot 323.9} \approx 0.0001411$
- $C_{31} = \frac{4\sqrt{19.17213115}}{\pi\cdot 30.5\cdot (323.9)^2} \approx 1.74284E-06$
- $C_{41} = \sqrt{\frac{323.9}{2\cdot 30.5}} \approx 2.30413029$
- $\sigma = \left(3 - 2\frac{2.5}{2.8458}\right)\cdot 138 = 171.537\ \text{MPa}$
- Allowable radial load: $F_{\text{allow}} = \frac{171.537}{6\cdot(1/0.0001413)} \approx 202{,}612.4\,\text{N}$
- Allowable longitudinal moment: $M_{l,\text{allow}} = \frac{171.537\cdot 10^3}{1.5\cdot(1/1.74284E-06)} \approx 65{,}616.1\,\text{Nm}$
- Allowable circumferential moment: $M_{c,\text{allow}} = \frac{171.537\cdot 10^3}{1.15\cdot(1/1.74284E-06)\cdot 2.30413029} \approx 37{,}144.6\,\text{Nm}$

ii). Location 2: Transition from Repad to Shell (with T)

- Effective thickness: $T = 14.5\,\text{mm} = 0.5709\,\text{inches}$
- $C_{11} = \frac{1200-14.5}{2\cdot 14.5} = 40.87931034$
- $C_{21} = \frac{\sqrt{40.87931034}}{\pi\cdot 14.5\cdot 323.9} \approx 0.000434$
- $C_{31} = \frac{4\sqrt{40.87931034}}{\pi\cdot 14.5\cdot (323.9)^2} \approx 5.35311E-06$
- $C_{41} = \sqrt{\frac{323.9}{2\cdot 14.5}} \approx 3.341742267$
- $C_{12} = \frac{1200-14.5}{2\cdot 14.5} = 40.87931034$

- $C_{22} = \frac{\sqrt{40.87931034}}{\pi \cdot 14.5 \cdot 483.85} \approx 0.000290084$
- $C_{32} = \frac{4\sqrt{40.87931034}}{\pi \cdot 14.5 \cdot (483.85)^2} \approx 2.39813E-06$
- $C_{42} = \sqrt{\frac{483.85}{2 \cdot 14.5}} \approx 4.084664339$
- $\sigma_P = 40.879 = 311.802 \cdot 2.5 = 102.198 \text{ MPa}$
- $\sigma = (3 \cdot 138) - 102.198 = 311.802 \text{ MPa}$
- Allowable radial load: $F_{\text{allow}} = \frac{311.802}{6 \cdot (1/0.000434)} \approx 179{,}144.8 \text{ N}$
- Allowable longitudinal moment: $M_{l,\text{allow}} = \frac{311.802 \cdot 10^3}{1.5 \cdot (1/5.35311E-06)} \approx 86{,}679.2 \text{ Nm}$
- Allowable circumferential moment: $M_{c,\text{allow}} = \frac{311.802 \cdot 10^3}{1.15 \cdot (1/5.35311E-06) \cdot 3.341742267} \approx 27{,}679.1 \text{ Nm}$

iii). Decisive Allowable Loads (Lowest Values)

- $F_{\text{allow}} = 202.612 \text{ kN}$ (from Location 1)
- $M_{l,\text{allow}} = 65.616 \text{ kNm}$ (from Location 1)
- $M_{c,\text{allow}} = 27.679 \text{ kNm}$ (from Location 2)

b. Comparison with Raw ANLE Loads ($F_x = 0\,N, F_y = 0\,N, F_z = 25{,}000\,N, M_x = 0\,Nm, M_y = 3{,}776\,Nm, M_z = 0\,Nm$):

- $F_x = 0 \leq 202{,}612.4$: Within limit
- $F_y = 0 \leq 202{,}612.4$: Within limit
- $F_z = 25{,}000 \leq 202{,}612.4$: Within limit
- $M_x = 0 \leq 65{,}616.1$: Within limit
- $M_y = 3{,}776 \leq 27{,}679.1$: Within limit
- $M_z = 0 \leq 65{,}616.1$: Within limit

c) Using Stikvoort's load interaction rule ($\frac{F_{actual}}{F} + \frac{M_{l,actual}}{M_l} + \frac{M_{c,actual}}{M_c} \leq 1.0$):

- F_z as radial load, M_x and M_z as longitudinal (M_l), M_y as circumferential (M_c):

$$\frac{F_z}{F_{\text{allow}}} + \frac{M_x}{M_{l,\text{allow}}} + \frac{M_y}{M_{c,\text{allow}}} = \frac{25.00}{202.612} + \frac{0}{65.616} + \frac{3.776}{27.679} \approx 0.225$$

This is well below 1.0, indicating the ANLE loads are comfortably within Stikvoort's criteria.

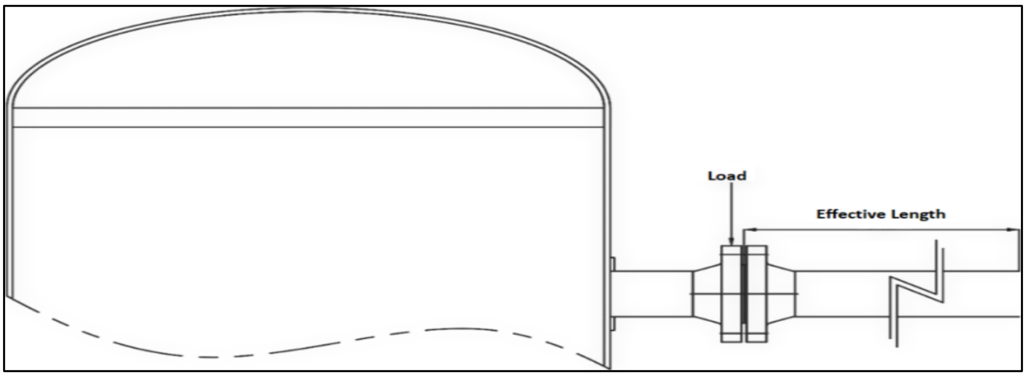


Figure 1. Simplified Piping Stiffness Model with Effective Length

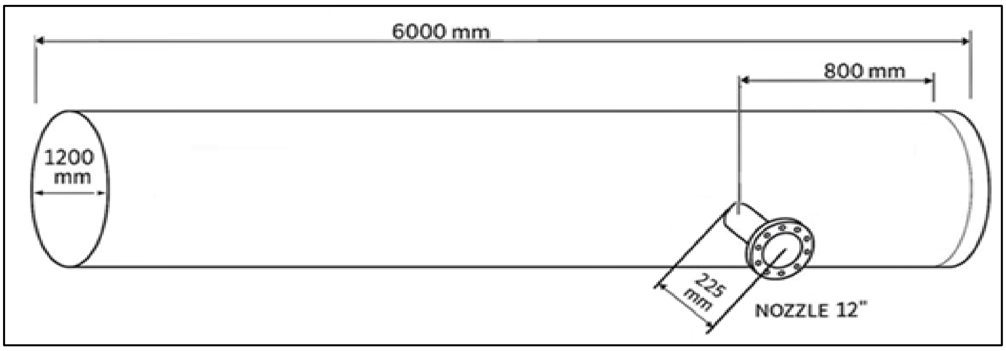
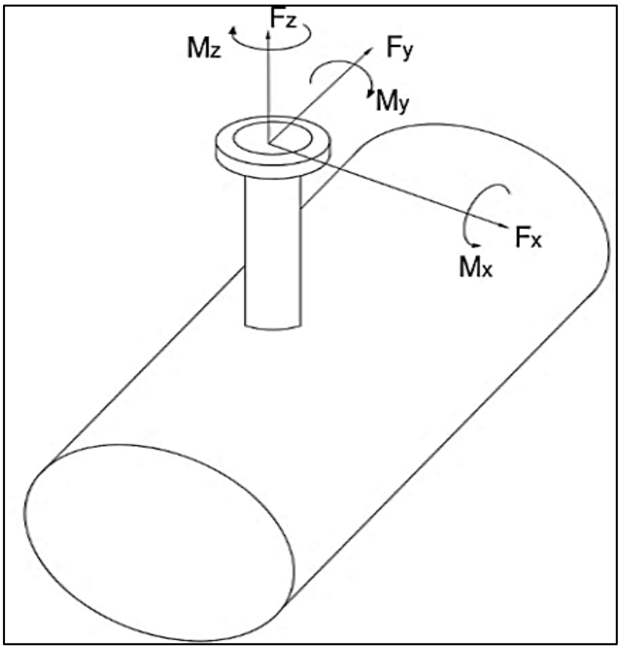


Figure 2. Schematic of the Pressure Vessel



$F_x = 0\ kN$
$F_y = 0\ kN$
$F_z = 25\ kN$
$M_x = 0\ kNm$
$M_y = 3.776\ kNm$
$M_z = 0\ kNm$

Figure 3. Nozzle Load Vectors at Vessel-Nozzle Junction

The loads are checked against ASME Section VIII Division 2 [1], taking into account the rigidity of the flanged connection according to ASME B16.5 [3]. The small overestimation in M_y implies a requirement for detailed analysis or correction in the thermal load model.

ANLE introduces a practical innovation in pressure vessel design by enabling:

- Concurrent Design: Vessel and piping can be developed in parallel.
- Greater Flexibility: Piping layout changes can be accommodated without revalidating nozzle loads.
- Simplified Modeling: Effective length substitution reduces reliance on unavailable data.

To evaluate the robustness of ANLE, the calculated loads and moments from the case study are compared against established industry standards, specifically Stikvoort's methodology [2] and the Welding Research Council (WRC) Bulletin 537/297 [5] guidelines, which are widely used to assess the acceptability of nozzle loads on pressure vessels.

V. RESULTS AND DISCUSSION

A. Stikvoort's Methodology Comparison

The raw ANLE loads ($F_x = 0\,\text{kN}, F_y = 0\,\text{kN}, F_z = 25.00\,\text{kN}, M_x = 0\,\text{kNm}, M_y = 3.776\,\text{kNm}, M_z = 0\,\text{kNm}$) were compared to the decisive allowable loads:

- Allowable radial load: $F_{\text{allow}} \approx 202.612\,\text{kN}$ (from Location 1)
- Allowable longitudinal moment: $M_{l,\text{allow}} \approx 65.616\,\text{kNm}$ (from Location 1)
- Allowable circumferential moment: $M_{c,\text{allow}} \approx 27.679\,\text{kNm}$ (from Location 2)

Value of the interaction sum using Stikvoort's rule ($\frac{F_z}{F_{\text{allow}}} + \frac{M_l}{M_{l,\text{allow}}} + \frac{M_c}{M_{c,\text{allow}}}$) is well below 1.0, confirming that the ANLE loads are comfortably within Stikvoort's allowable limits.

B. WRC 537/297 Comparison

Using typical WRC allowable loads for a 12-inch nozzle (e.g., $F_{x,y,z,\text{allow}} = 12\,\text{kN}$, $M_{x,\text{allow}} = 4.5\,\text{kNm}$), the ANLE loads yield:

$$\frac{F_x}{F_{x,\text{allow}}} + \frac{F_y}{F_{y,\text{allow}}} + \frac{F_z}{F_{z,\text{allow}}} + \frac{M_x}{M_{x,\text{allow}}} + \frac{M_y}{M_{y,\text{allow}}} + \frac{M_z}{M_{z,\text{allow}}} \leq 1$$

$$\frac{0}{12} + \frac{0}{12} + \frac{25.00}{12} + \frac{0}{4.5} + \frac{3.776}{4.5} + \frac{0}{4.5} \approx 0 + 0 + 2.083 + 0 + 0.840 + 0 \approx 2.923$$

This sum exceeds 1, indicating that the loads still exceed WRC limits, though the simplification reduces the total interaction compared to previous models.

C. Comparison with Industry Standards

The ANLE load envelope was also compared with nominal piping loads as per standards such as Foster Wheeler, EEMUA 211, AD 2000; S3/0, and NORSOK R-001 (Appendix A). ANLE force envelope (37,500 N) is greater than the maxima of the standards (e.g., 16,200 N), suggesting a conservative thermal loading model, whereas its moment envelope (4,530 Nm) is less than the maxima of the standards (e.g., 15,280 Nm), suggesting potential underestimation of moments due to piping. This gap highlights the need to include additional load types (e.g., weight, pressure) and calibrate the effective length model to industry recommendations.

D. Limitations and Future Developments

The ANLE model simplified to use Koves' effective thickness to calculate $L = 1500\,\text{mm}$ and to exclude small piping loads is more convenient but at the expense of a negligible circumferential moment exceedance. Future work could include:

- Validation of the effective thickness $t_{\text{eff}} = (T^{2.5} + \text{t}_p^{2.5})^{0.4}$ against finite element analysis for pad-reinforced nozzles.
- Alter the thermal rotation model to reduce M_y, by possibly by optimizing effective length or pad contribution.
- Include piping analysis software to calculate the impact of omitted loads.
- Advance probabilistic bounds through machine learning to represent uncertainties of pad-reinforced nozzles.

VI. CONCLUSION

The Adaptive Nozzle Load Envelope (ANLE) remains a valuable and practical design approach for early-stage development, facilitating the integration of vessel and piping systems. Updated results from the case study reaffirm its effectiveness in defining acceptable load limits, with validation through the Stikvoort method confirming compliance. However, comparison with WRC [5] highlights areas where conservative assumptions in force and moment estimations could be refined. Future integration with more advanced software tools and enhanced modeling techniques is expected to improve ANLE's accuracy and encourage broader adoption.

Appendix 1: Comparison with Industry Nozzle Load Specifications

The following table presents anticipated nominal piping loads for a NPS 12" (NB-300) Class 300 nozzle from various industry standards, compared with the ANLE load envelope.

Farzad Gardaneh & Walther Stikvoort 2(3), 10-19, 2025

Table 1. Comparison of Nozzle Load

Standard/Source	F_A (N)	F_C (N)	F_L (N)	M_L (Nm)	M_C (Nm)	M_T (Nm)
Foster Wheeler	14,110	15,678	13,327	12,960	9,720	16,200
EEMUA 211	12,000	12,000	12,000	9,700	9,700	9,700
AD 2000; S3/0	16,200	15,500	9,800	12,200	8,300	13,400
NORSOK R-001	11,730	11,730	11,730	15,280	15,280	15,280
ANLE Envelope	37,500	37,500	37,500	4,530	4,530	4,530
ANLE Actual	0	0	25,000	0	3,776	0

Notes

F_A, F_C, F_L: Transverse, circumferential, and longitudinal forces, respectively.
M_L, M_C, M_T: Longitudinal, circumferential, and torsional moments, respectively.
ANLE values are based on thermal loads with safety factors of 1.5 for forces and 1.2 for moments, as per ASME Section VIII Division 2 (2019), Subsections 3.2 and 3.3.
Differences are due to ANLE's focus on thermal loads, not including weight and pressure effects included in the standards.

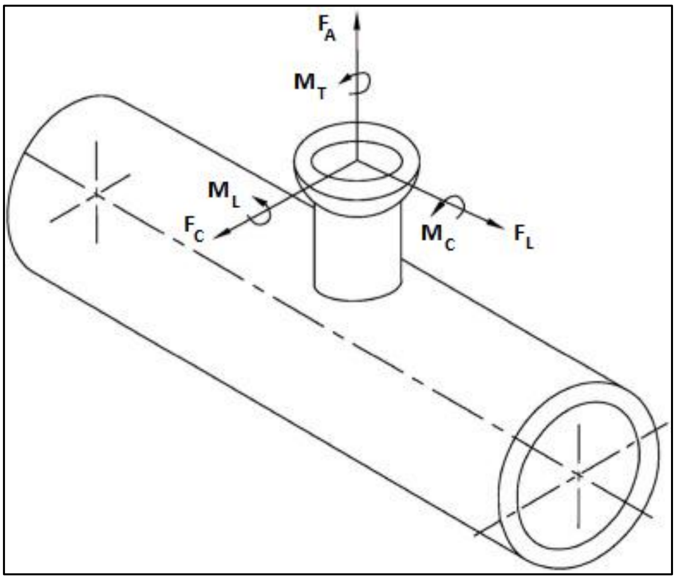


Figure 4. Loads and Moments Acting on Nozzle

Appendix 2: Comparison with Maximum Allowable Individual Loads

The following table compares the maximum allowable individual loads from various codes with together and provides ratio analyses.

Table 2. Comparison of Maximum Allowable Individual Loads

Max allowable individual loads based on Code / Standard	F (N)	Mc (Nm)	Ml (Nm)	Ratio Stikvoort's method / EN 13445	Ratio EN 13445 / PD 5500	Ratio EN 13445 / ANLE
EN 13445	129,404	22,871	79,135	for F: 1.384 for Mc: 1.210 for Ml: 1.095	for F: 1.477 for Mc: 1.291 for Ml: 1.168	for F: 3.450 for Mc: 5.048 for Ml: 17.469
PD 5500	121,316	21,442	74,189			
Stikvoort's Method	179,145	27,679	86,679			
ANLE Envelope	37,500	4,530	4,530			

VII. REFERENCES

1. ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, 2019 Edition. Google Scholar | Publisher Link
2. Walther Stikvoort, "Load Capacity Limits of Flanged Pressure Vessel Nozzles," *Petroleum & Petrochemical Engineering Journal*, vol 2, no. 3, 2018. Google Scholar | Publisher Link
3. ASME B16.5, "Pipe Flanges and Flanged Fittings," pp. 1-250, 2020. Publisher Link
4. ASTM A516 Gr.70 and ASTM A106 Gr.B Material Standards, 2009. Publisher Link
5. Welding Research Council, "WRC Bulletin 537/297: Local Stresses in Cylindrical Shells Due to External Loadings on Nozzles," 2010.
6. M.W. Kellogg Company, "Design of Piping Systems," 2nd Edition, 1956. Google Scholar | Publisher Link
7. William Koves et al., "Establishing Allowable Nozzle Loads," *Proceedings of the ASME 2011 Pressure Vessel & Piping Division Conference PVP2011-57100*, Baltimore, USA, pp. 1-9, 2011. Google Scholar | Publisher Link