



TESTING CENTER FOR OIL AND GAS LEMIGAS

SCIENTIFIC CONTRIBUTIONS OIL AND GAS (SCOG)

Vol. 49. No. 3, September: 107 – 129 - <http://journal.lemigas.esdm.go.id>

ISSN: 2089-3361, e-ISSN: 2541-0520

Published by:

LEMIGAS

Effect of Cone Design on the Structural Integrity Assessment of an Oil & Gas Conical Perforated Strainer Using CFD- Coupled Finite Element Analysis

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Received: 30 March 2026; Revised: in revised form 22 July 2026; Accepted: 03 September 2026
Available: online 30 September 2026.

ABSTRACT - Cone strainers play an important role in oil and gas process piping systems and are often subjected to high-pressure fluid flow. The early failure of a cone strainer in a Low Temperature Separation Unit (LTSU) during prolonged service motivated a detailed structural integrity investigation. A combined Computational Fluid Dynamics (CFD) and Finite Element Analysis (FEA) method was used to assess the structural behaviour of a conical perforated strainer under actual operating conditions (30.51 bar (a), 94.3 °C) and contingency conditions (36.61 bar (a), 113.16 °C), the latter representing the pressure and temperature excursions encountered intermittently during normal service. Pressure, temperature and wall shear stress obtained from steady-state CFD were applied to the structural model through one-way fluid–structure interaction, and structural adequacy was expressed as a Unity Check (UC), the ratio of the peak von Mises stress to an allowable stress of 336 MPa. The baseline 2 mm punching plate (Geometry A) was adequate under actual conditions (UC = 0.87) but failed under contingency conditions, reaching 371.8 MPa and UC = 1.11, as a result of flow-induced bending across the large, low-stiffness perforated cone surface. Thickening the punching plate to 4 mm (Geometry B) reduced the contingency peak stress by 13.4% to 322.0 MPa and the UC to 0.96, but transferred load into the ring plate, whose UC rose from 0.95 to 0.99 and which thereby became the new critical component. Repositioning the ring plate (Geometry C) reduced the punching plate UC to 0.91 and the ring plate UC to 0.30, at the cost of a 19.7% increase in pressure drop, from 7273 Pa to 10894 Pa. Geometry C is therefore proposed as the optimal configuration, and the CFD-informed FEA workflow provides a reliable framework for integrity evaluation and design optimisation of conical strainers used in the oil and gas industry.

Keywords: conical strainer, finite element analysis, CFD- FEA coupling, fluid structure interaction, pressure induced failure, structural integrity.



How to cite this article:

Mohd Azni Md Kasim, Mohd Azahari Razali, Nur Syafiqah Adha Narrudin, Norfakhira Mohd Nor, Iman Fitri Ismail, Masataro Suzuki, and Amnur Akhyan, 2026, Effect of Cone Design on the Structural Integrity Assessment of an Oil & Gas Conical Perforated Strainer Using CFD- Coupled Finite Element Analysis, *Scientific Contributions Oil and Gas*, 49 (3) pp. 107-129.

1. Introduction

Strainers are mechanical filtering devices used to prevent solid particles from entering and damaging the downstream equipment. Various types of strainers are used across numerous industries, including conical strainers, Y-strainers, and T-strainers (Patel et al., 2021). A conical strainer is a cone-shaped filtering element installed directly between two flanges in a pipeline system, designed to remove solid contaminants from fluids before they enter sensitive equipment such as heat exchangers, separators, and downstream valves. Conical strainers are widely used in oil and gas, chemical, and process industries due to their effective filtration performance, compact design, and low pressure drop (Mahmood et al., 2025; Sotoodeh 2019; Sumit et al., 2018). Although a conical strainer is an important piece of equipment in many industries, it is often exposed to extreme operating conditions, including high internal pressure, continuous fluid flow, and cyclic loading, which can lead to structural degradation and premature failure. A structural failure of a strainer can cause equipment damage, diminished flow efficiency, and compromised long-term process safety, therefore highlighting the importance of reliable structural design and assessment to ensure the security of process equipment (Sayinci 2015).

Under operating conditions, the punching plate of the conical strainer is subjected to fluid pressure and flow acceleration through its perforations. Although these perforations are essential for the filtration process, they considerably reduce the effective stiffness of the structure and lead to stress concentrations at the edges of the holes (Ghaderi et al., 2015; Lasenko et al., 2025; Luan et al., 2025; Xi et al., 2024). Additionally, the inclined design of the conical shape introduces non-uniform stresses and complicates load transfer to supporting components, such as handles and ring plates, which makes it difficult to accurately predict the structure's behavior using conventional analytical methods (Carlomagno et al., 2012; Ruiz et al., 2015; Sayinci et al., 2015).

Finite Element Analysis (FEA) is a computer-based engineering method used to predict how a structure or component will respond to real-world conditions, including forces, stress, pressure, and other loads. Various studies have used FEA to evaluate the integrity of pressure-containing components, such as pressure vessels, piping systems, and perforated plates (Balac et al., 2018; Ganhao et al., 2021; Norske Veritas 2008), particularly for different types of strainers. However, prior analyses often relied on simplified geometries and assumptions about internal pressure loading, which did not accurately reflect the actual operational strains experienced by the strainers. The pressure load on strainers is significantly affected by fluid flow characteristics, obstructions, and flow acceleration in the vicinity of the strainers (Akhyan et al., 2025; Mahmood et al., 2025; Sotoodeh 2019).

In 2018, (Umashankar et al., 2018) conducted FEA to assess stress and deformation in strainer components subjected to internal pressure, comparing the FEA results with theoretical calculations. Their findings indicated that the FEA model was validated and that incorporating a stiffener into the conical strainer significantly enhanced safety and reduced deformation. Additionally, (Haiqal et al., 2025) used FEA to investigate stress concentration around holes subjected to internal pressure and flow-induced loads in strainer applications. This study demonstrated that FEA can accurately identify the location of maximum stress and assess the influence of hole patterns, plate thickness, and pressure on stress levels in strainer components. Computational Fluid Dynamics (CFD) is regarded as an effective tool for modeling real behavior and pressure distribution in complex geometries. Several studies have investigated the combination of fluid loading based on CFD with finite element analysis (FEA), which can significantly enhance the reliability of structural integrity analyses for components exposed to flow (Bathe et al., 2004; Gu et al., 2023; Hagmeyer et al., 2022; Fadhil et al., 2021; Shakya et al., 2024; Yuan et al., 2022). However, there has been limited research on CFD-FEA coupled analysis specifically for conical perforated strainers under contingency operating conditions in real industry applications.

While previous studies have analysed strainer stresses under idealised internal pressure, few have employed one-way FSI to evaluate a conical perforated strainer under contingency flow conditions in a Low Temperature Separation Unit, and fewer still have examined how a design modification intended to relieve one component redistributes load onto another. This study addresses that gap by coupling steady-state CFD with FEA for an in-service strainer geometry, and by tracking the Unity Check of every structural component — punching plate, strainer cap, handle and ring plate — across three design configurations and two operating conditions.

In this study, prolonged operation of the conical strainer in the Low Temperature Separation Unit (LTSU) led to structural failure, damaging the upper part of the conical punching plate near the handle connection. This failure indicates the need for a thorough numerical analysis to determine the governing failure mechanism and to assess potential design improvements. Therefore, this study aims to evaluate the structural integrity of each conical strainer component using Finite Element Analysis (FEA) under fluid-induced loads. The fluid temperature, pressure and wall shear stress obtained from CFD are transferred to the FEA model using one-way Fluid-Structure Interaction (FSI) to ensure precise predictions of structural behaviour. The FEA results comprise the von Mises stress, displacement, strain and Unity Check (UC) of each component. Three configurations are compared: the as-built 2 mm punching plate, a thickened 4 mm punching plate, and a 4 mm punching plate with the ring plate repositioned, so that changes in flow behaviour can be directly correlated with the corresponding structural response and the associated hydraulic penalty.

2. Methodology

2.1 Geometry and design configurations

- **Fundamental geometry**

Figure 1 illustrates the conical strainer's geometry. This geometry was developed using the design specifications and represents the configuration used in practical. It comprises several structural and functional components, including a mesh screen serving as the primary filtering surface, a ring plate connecting the strainer to the pipeline's flange, and a conical punching plate. This punched plate distributes differential pressure throughout the cone assembly and provides mechanical stability to the mesh. A stiffener ring and strainer cap are supplementary components that enhance the strainer's strength and stability during operation. This essential component of the conical strainer enables accurate modeling for coupled CFD-FEA analysis.

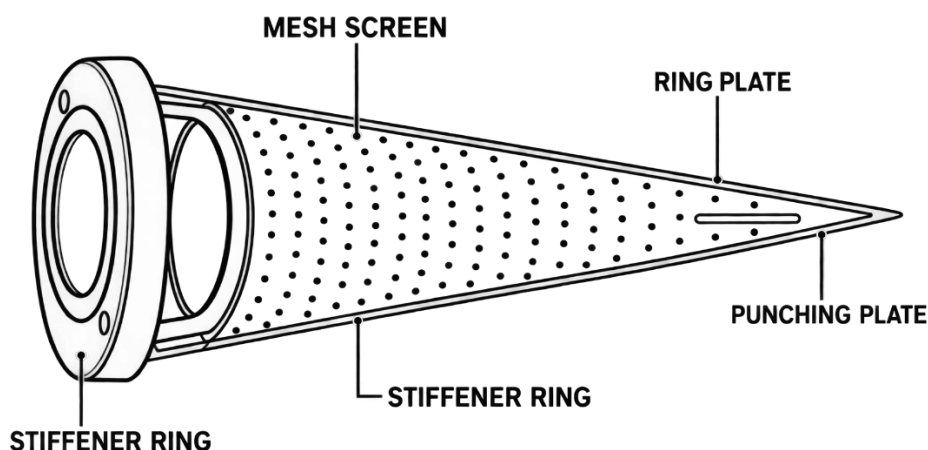


Figure 1.
LTSU Strainer Geometry

• Simulation geometry

The fundamental geometry of the conical strainer was redrawn in SolidWorks to reflect the design modifications and to support both the CFD and the FEA simulations. The parameters varied in this study are the thickness of the conical punching plate and the axial position of the ring plate. Geometry A is the baseline model, reproducing the as-built strainer with a 2 mm punching plate and the ring plate in its original position. Geometry B doubles the punching plate thickness to 4 mm while retaining the original ring plate position, isolating the effect of cone stiffness alone. Geometry C retains the 4 mm punching plate but repositions the ring plate, achieved by extending the stiffener ring towards the region of peak stress on the upper cone, so that the effect of relocating the structural support can be separated from the effect of plate thickness. The three configurations are summarised in Table 1 and illustrated in Figures 2 and 3.

Table 1.
Conical Strainer Geometry Configuration in CFD (SolidWorks)

Geometry	Modification	Punching Plate Thickness
A	As-built baseline model, original ring plate position	2 mm
B	Punching plate thickness doubled from the baseline model	4 mm
C	4 mm punching plate with the ring plate repositioned by extending the stiffener ring	4 mm

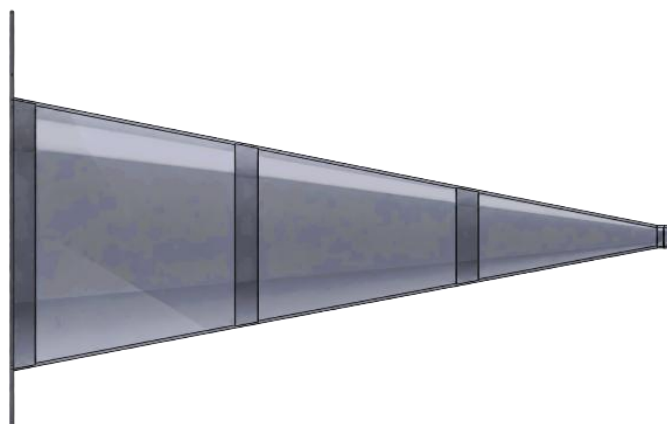


Figure 2.
Conical Strainer Model for Geometry A and B in SolidWorks (CFD), original ring plate position.

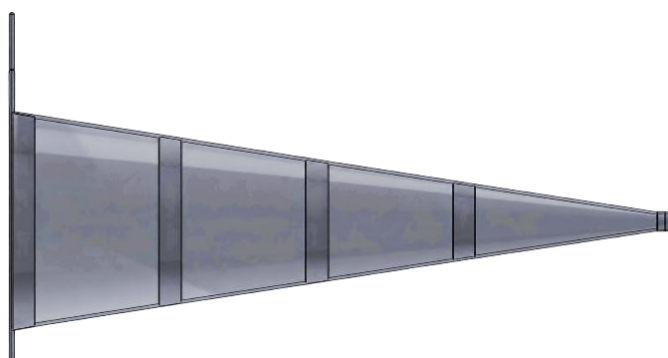


Figure 3.
Conical Strainer Model for Geometry C in SolidWorks (CFD), with the ring plate repositioned.

2.2 Fluid properties

• Thermophysical properties

For this study, thermophysical properties were calculated from the PCHE datasheet to support computational fluid dynamics (CFD) and finite element analysis (FEA) of the conical strainers. These properties are essential for determining the specific heat at constant volume, the specific heat ratio, and the molecular mass. CFD and FEA analyses depend on these thermophysical properties to understand fluid movement and heat transfer in pipelines and strainers. Tables 2 and 3 present the thermophysical properties for both the standard and contingency cases in this study.

- Actual Case: $P = 30.51$ bar (a), $T = 94.3$ °C

Table 2.
Thermophysical Properties for Standard Case

Property	Value	Unit
Specific Heat Ratio (Cp/Cv)	1.202	-
Molecular Mass	0.02897	kg/mol
Dynamic Viscosity	0.000016	Pa*s
Specific Heat (Cp)	1706	J/kg*K
Thermal Conductivity	0.0370	W/(m*K)

- Contingency Case: $P = 36.61$ bar (a), $T = 113.16$ °C

Table 3.
Thermophysical Properties for Contingency Case

Property	Value	Unit
Specific Heat Ratio (Cp/Cv)	1.236	-
Molecular Mass	0.02546	kg/mol
Dynamic Viscosity	0.000016	Pa*s
Specific Heat (Cp)	1706	J/kg*K
Thermal Conductivity	0.0370	W/(m*K)

• Boundary Conditions

The initial and boundary conditions at the strainer inlet and outlet were obtained from the PCHE datasheet in order to reproduce the operating flow conditions and the resulting structural response. All pressures are specified as absolute values. Case 1 corresponds to the actual operating condition of the conical strainer. Case 2 corresponds to the contingency condition, in which the fluid pressure and temperature are 20% above Case 1. The contingency condition is not a hypothetical excursion: pressure spikes occur intermittently during normal LTSU operation, and Case 2 represents the envelope of those spikes. Simulating this condition is the standard means of exposing uncertainties and deficiencies in the existing strainer design, by examining the effect of elevated thermal and pressure loads on the pressure distribution, the pressure drop and the resulting mechanical stress. Table 4 summarises the boundary conditions for the two cases.

Table 4.
Boundary Conditions for Case 1 and Case 2

Boundary condition	Case 1: actual	Case 2: contingency	Unit
Inlet Flow Rate	653219	783862.8	kg/hr
Outlet Flow Rate	650475	780570	kg/hr
Inlet Fluid Pressure	30.51	36.61	bar (a)
Temperature	94.3	113.16	°C

- **Strainer porous media**

The filtering surface of the strainer consists of a 50-mesh plain-weave wire screen supported by a perforated punching plate with a 60° staggered hole pattern. Resolving every perforation explicitly would require a prohibitive cell count, so the filtering surface was represented as a porous medium of equivalent resistance. This treatment reproduces the pressure drop and the pressure distribution across the strainer without modelling the individual apertures, while preserving the differential pressure that constitutes the structural load on the cone. The open area of each layer and the corresponding pressure drop were evaluated following Perry's Chemical Engineers' Handbook, Sections 6.19–6.21 (Green et al., 2008), as set out below.

For the perforated punching plate with a 60° staggered pitch, the open-area (porous) ratio is

$$dp = 0.91 \left(\frac{d}{P} \right)^2 \quad (1)$$

where d is the pore diameter (10.0 mm) and P is the hole pitch (12.0 mm), giving $dp = 0.6319$.

For the plain-weave screen, the open-area ratio follows from the aperture and the wire pitch,

$$ds = \left(\frac{O}{Pt} \right)^2 = \left(\frac{Pt - Wd}{Pt} \right)^2, \quad Pt = \frac{25.4}{n} \quad (2)$$

where O is the screen opening (0.315 mm), Wd the wire diameter (0.193 mm), and n the mesh number (50), giving $Pt = 0.508$ mm and $ds = 0.3845$.

The effective opening area of the element, A_{pe} , and the open-area ratio referred to the nozzle section area A_n are then,

$$A_{pe} = A_t dp ds, \quad R = \left(\frac{A_{pe}}{A_n} \right) \times 100\% \quad (3)$$

with $A_t = 1.3671 \text{ m}^2$ and $A_n = 0.2739 \text{ m}^2$, giving $A_{pe} = 0.3321 \text{ m}^2$ and $R = 122\%$.

The total initial pressure drop across the strainer is the sum of the losses at the container nozzle, the wire screen, and the punching plate,

$$\Delta Ph = \frac{K \rho V n^2}{2gc}, \quad Vn = \frac{Qv}{A_n} \quad (4)$$

$$\Delta Ps = \left(\frac{1}{C_D^2} \right) \cdot \left(\frac{(1 - ds^2)}{ds^2} \right) \cdot \frac{\rho V s^2}{(2gc)} \quad (5)$$

$$\Delta Pp = Qw^2 \frac{\left(1 - \left(\frac{A_f}{A_t} \right)^2 \right)}{2gc \rho C_o^2 A_f^2} \quad (6)$$

$$\Delta Pi = \Delta Ph + \Delta Ps + \Delta Pp \quad (7)$$

where k , ρ , g_c , Q_v , Q_w , $Vs = Qv/A_t$, A_f , $C_D = 0.1 NRe^{0.5}$, and C_o , are the nozzle loss coefficient (0.4), the fluid density, the gravitational conversion constant, the volumetric flow rate, the mass flow rate, the screen velocity, the open area of the punching plate (0.8638 m^2), the screen discharge coefficient and the orifice coefficient of the punching plate, obtained from $C_o(P/d)0.1 = 0.746$ as 0.7325. The resulting parameters and losses are listed in Table 5.

Table 5.
Open-area and pressure-drop parameters of the strainer element

Parameter	Symbol	Value	Unit
Pore diameter	d	10.0	mm
Hole pitch	P	12.0	mm
Plate porous ratio	dp	0.6319	–
Screen mesh number	n	50	–
Wire diameter	Wd	0.193	mm
Screen opening	O	0.315	mm
Screen porous ratio	ds	0.3845	–
Element area	At	1.3671	m ²
Punching plate open area	Af	0.8638	m ²
Nozzle section area	An	0.2739	m ²
Element opening area	Ape	0.3321	m ²
Open-area ratio	R	122	%
Nozzle loss	ΔPh	0.0519	kg/cm ²
Screen loss	ΔPs	0.000388	kg/cm ²
Punching plate loss	ΔPp	0.0146	kg/cm ²
Total initial pressure drop	ΔPi	0.0669 (0.0656 bar)	kg/cm ²

2.3 CFD and FEA analysis

• CFD analysis and fluid loading

The CFD simulation evaluates the fluid flow behaviour through the conical strainer under the two operating conditions defined above. The analysis was performed as a steady-state, compressible calculation in SolidWorks Flow Simulation, which solves the Favre-averaged Navier–Stokes equations for mass, momentum and energy on a Cartesian immersed-body mesh.

Turbulence was closed using the k – ϵ model with the damping functions of (Lam et al., 1981), which describes laminar, turbulent and transitional flow of a homogeneous fluid through transport equations for the turbulent kinetic energy k and its dissipation rate ϵ :

$$\partial \rho k / \partial t + \partial \rho k u_i / \partial x_i = \partial / \partial x_i [(\mu + \mu_t / \sigma_k) \partial k / \partial x_i] + \tau_{ij} R \partial u_i / \partial x_j - \rho \epsilon + \mu_t P B \quad (8)$$

$$\begin{aligned} \partial \rho \epsilon / \partial t + \partial \rho \epsilon u_i / \partial x_i = & \partial / \partial x_i [(\mu + \mu_t / \sigma_\epsilon) \partial \epsilon / \partial x_i] + C_\epsilon 1 (\epsilon / k) (f_1 \tau_{ij} R \frac{\partial u_i}{\partial x_j} \\ & + C_B \mu_t P B) - f_2 C_\epsilon 2 \rho \epsilon^2 / k \end{aligned} \quad (9)$$

$$\mu_t = f_\mu C_\mu \rho k^2 / \epsilon \quad (10)$$

with the standard coefficients $C_\mu = 0.09$, $C_\epsilon 1 = 1.44$, $C_\epsilon 2 = 1.92$, $\sigma_k = 1$, $\sigma_\epsilon = 1.3$ and $\sigma_B = 0.9$, and with the damping functions f_μ , f_1 , and f_2 acting to reduce the turbulent viscosity as the wall is approached (Sobachkin et al., 2014). Near-wall behaviour is captured by the two-scale wall function treatment, which switches between an integrated Prandtl boundary-layer solution where the near-wall mesh is coarse and a direct resolution of the boundary layer where it is sufficiently refined. This treatment is important here because the filtering surface is represented as a porous medium rather than as resolved perforations: the wall shear stress transferred to the structural model is obtained from the boundary-layer solution on the equivalent porous surface rather than from the flow through individual apertures.

Meshing used the Cartesian immersed-body approach with solution-adaptive refinement enabled at refinement level 3 and a cell limit of 1,563,182, so that cells were concentrated automatically in the regions of steepest gradient, principally the upstream face of the cone and the annulus around the ring plate. Convergence was monitored on goal-based criteria for inlet and outlet pressure, and the solution was

accepted once these goals had stabilised. It should be noted that solution-adaptive refinement improves the resolution of a given mesh budget but does not by itself constitute a formal grid-convergence study, since the refinement level and the cell limit still bound the final mesh. The solution is ran using solution-adaptive feature until three levels of refinement no longer changes the result of the solution by more than 3%. Because the comparison between the three geometries uses an identical meshing strategy and cell limit throughout, the relative differences between configurations, which form the basis of the conclusions drawn here, are considerably less sensitive to this uncertainty.

- **Fluid – structure interaction (FSI)**

A one-way fluid-structure interaction was adopted, whereby the pressure, temperature and wall shear stress fields obtained from the CFD solution were mapped onto the structural model as external loads. The deformation of the structure was not fed back into the CFD model, because the computed displacements are small relative to the flow passage dimensions — the largest resultant displacement obtained in any case was 0.59 mm against a cone base diameter of 570 mm — and therefore have no appreciable influence on the flow field. This is an efficient and appropriate modelling strategy for the integrity assessment of pressure-retaining structural components.

- **Finite Element Analysis (FEA)**

- **Material Properties**

The strainer is fabricated entirely from UNS S31803 duplex stainless steel, supplied to three product specifications depending on the component, as listed in Table 6. Since all components share a single alloy, a common set of mechanical properties was applied throughout the model; these are given in Table 7.

Table 6.
Component material specifications

Component	Material specification
Mesh screen	ASTM A470 UNS S31803
Punching plate	ASTM A240 UNS S31803
Handle	ASTM A240 UNS S31803
Strainer cap	ASTM A240 UNS S31803
Stiffener ring	ASTM A815 UNS S31803
Ring plate	ASTM A815 UNS S31803

Table 7.
Mechanical properties of UNS S31803 duplex stainless steel used in the FEA

Property	Value	Unit
Young's modulus, E	200	GPa
Poisson's ratio, ν	0.30	—
Density, ρ	7800	kg/m ³
Specified minimum yield strength	448	MPa
Allowable stress adopted	336	MPa

The allowable stress of 336 MPa was not derived from a pressure-vessel design code. It is the working stress adopted for this assessment, equal to 75% of the specified minimum yield strength of UNS S31803, so that the Unity Check functions as the reciprocal of the safety factor against that limit: $UC = 1.0$ corresponds to a factor of 1.33 on yield, and the material remains elastic at $UC = 1.0$. The same allowable stress was applied at both 94.3 °C and 113.16 °C without temperature derating. This is a stated assumption of the analysis rather than a code-based position, and it is mildly non-conservative at the contingency temperature, where the yield strength of duplex stainless steel is somewhat reduced relative

to its room-temperature value. The structural analysis of the conical strainer was conducted using finite element analysis. The finite element model used a blended curvature-based mesh comprising 88,340 nodes and 43,659 elements, as shown in Figures 4 and 5.

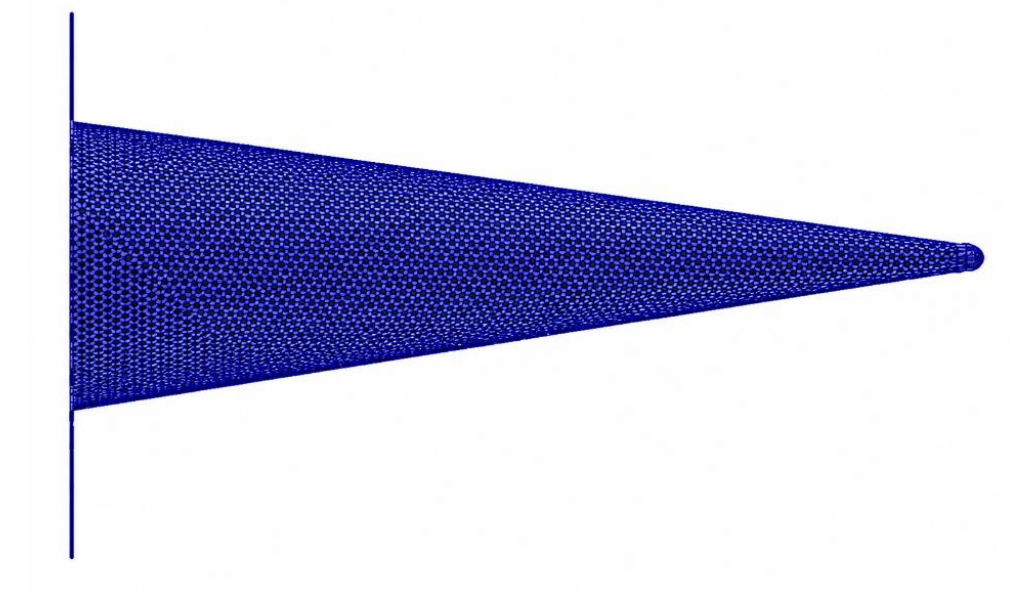


Figure 4.
Blended-Curvature Mesh

Table 8.
FEA Mesh Quality Metrics in CFD Software

Mesh quality metrics	Value	Remarks
Max element size	97.4mm	None
Min element size	4.87mm	None
Total nodes	88,340	None
Total elements	43,659	None
Max aspect ratio (AR)	75.483	None
% of Elements with AR<3 = 97.86	97.86%	Fulfilled FEA mesh criteria
% of Elements with AR > 10	1.07%	Fulfilled FEA mesh criteria

This mesh was acceptable for stress analysis since more than 97% of the elements had an aspect ratio (AR) below 3, as shown in Figure 5. The aspect ratio influences the analysis outcome appreciably, since elements with high aspect ratios degrade solution accuracy. The Unity Check (UC), defined as the ratio of the maximum von Mises stress in a component to the allowable stress of 336 MPa, was used to evaluate the structural integrity of each conical strainer component in both cases. Values below unity indicate acceptable performance; values above unity indicate that the component is not structurally acceptable.

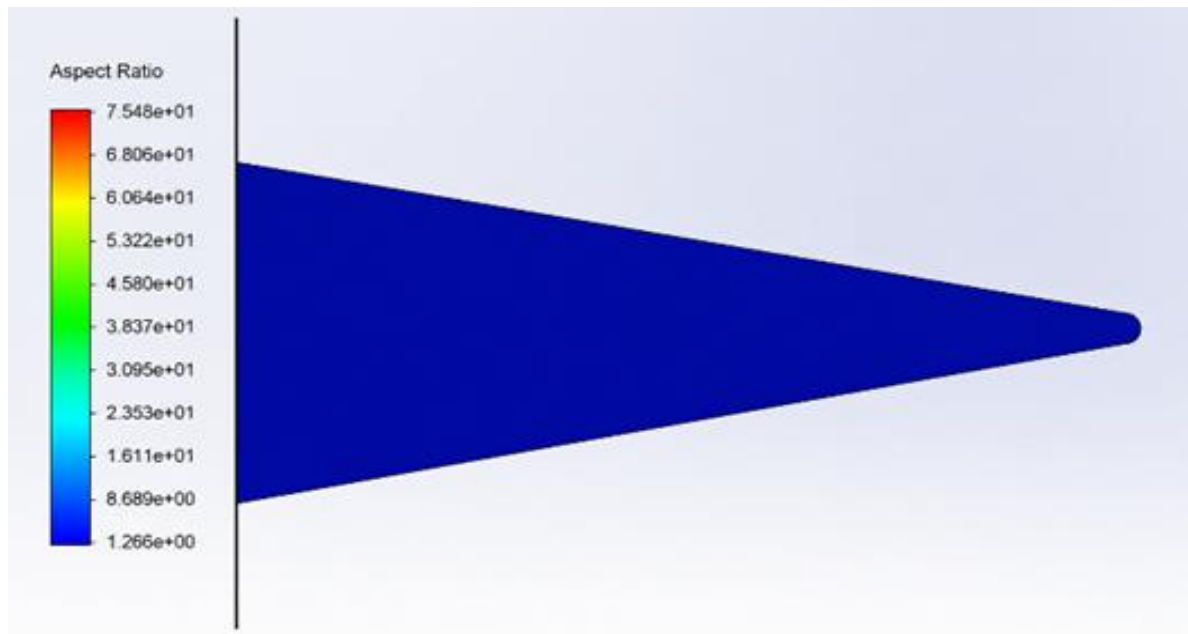


Figure 5.
Aspect Ratio (AR) of the conical strainer.

3. Result and Discussion

3.1 von Mises Stress, Displacement & Strain

- **Geometry A – 2 mm punching plate**

Figures 6 and 7 present the von Mises stress, displacement and strain distributions for Geometry A under actual (Case 1) and contingency (Case 2) conditions. In both cases the maximum stress and strain are concentrated on the upper surface of the conical punching plate near the handle connection, which is the location at which the in-service failure occurred. Under Case 1 the peak von Mises stress in the punching plate is 293.2 MPa, giving $UC = 0.87$, with a maximum resultant displacement of 0.4585 mm and a maximum equivalent strain of 1.051×10^{-3} . The baseline design is therefore acceptable under normal operating conditions, although the margin against the allowable stress is only 13%.

Under Case 2 the peak stress rises to 371.8 MPa, an increase of 26.8%, and exceeds the allowable stress to give $UC = 1.11$. The maximum displacement increases by 27.9% to 0.5863 mm and the maximum equivalent strain by 27.5% to 1.340×10^{-3} , while the high-stress region spreads over a substantially larger portion of the cone. The governing mechanism is flow-induced bending rather than membrane action. The perforations remove a large fraction of the load-bearing area — the plate porous ratio is 0.6319, so only about 37% of the nominal plate section carries load — and the resulting loss of flexural stiffness allows the unsupported span of the cone between the ring plate and the handle to bend under the differential pressure. Bending stress concentrates in the ligaments between adjacent holes at the restrained upper edge, producing the localised peak observed. These results confirm that the 2 mm punching plate of Geometry A lacks structural integrity under the pressure spikes encountered during service and is susceptible to the failure that was observed in operation.

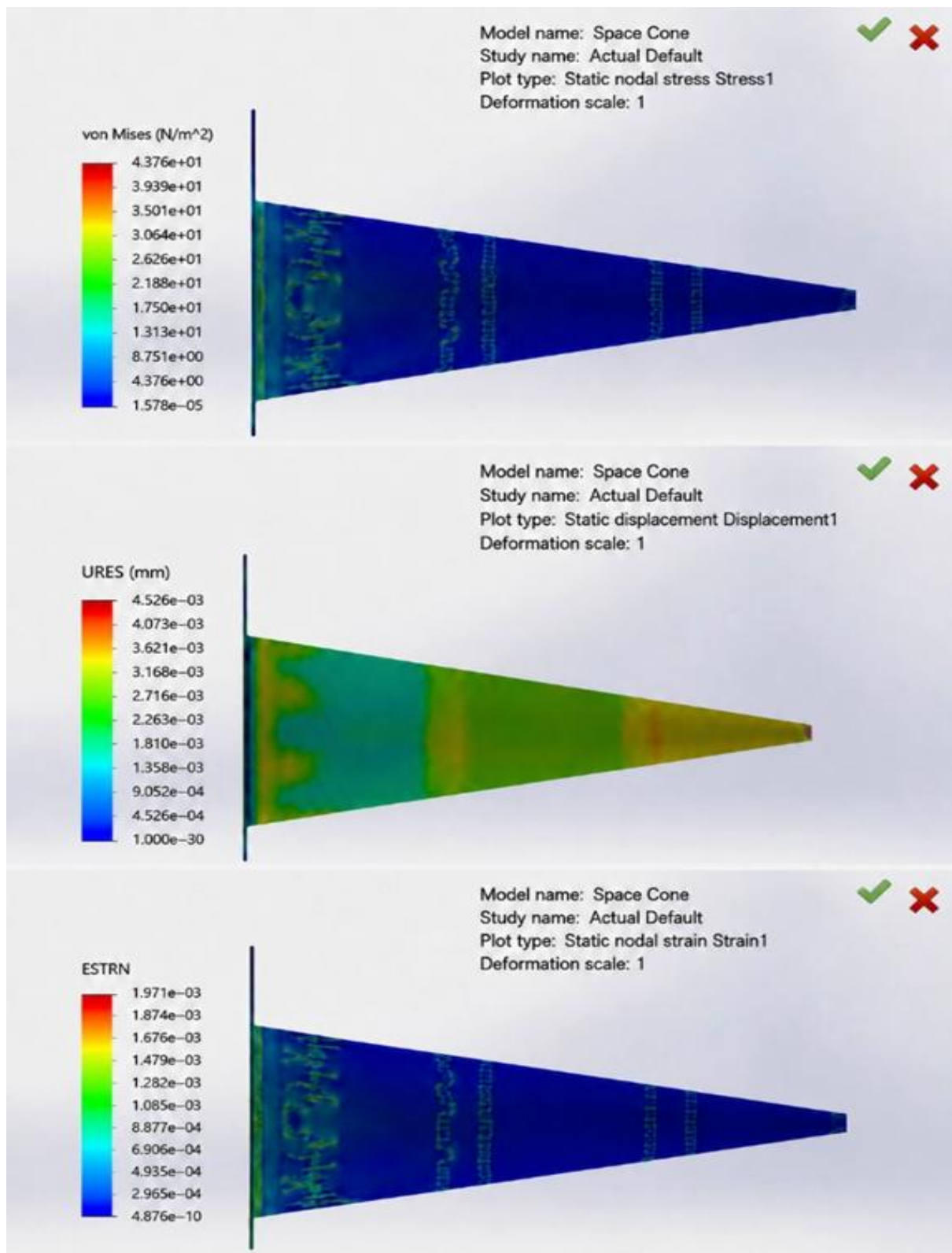


Figure 6.
von Mises Stress, Displacement and Strain for Geometry A, Case 1 (actual condition)

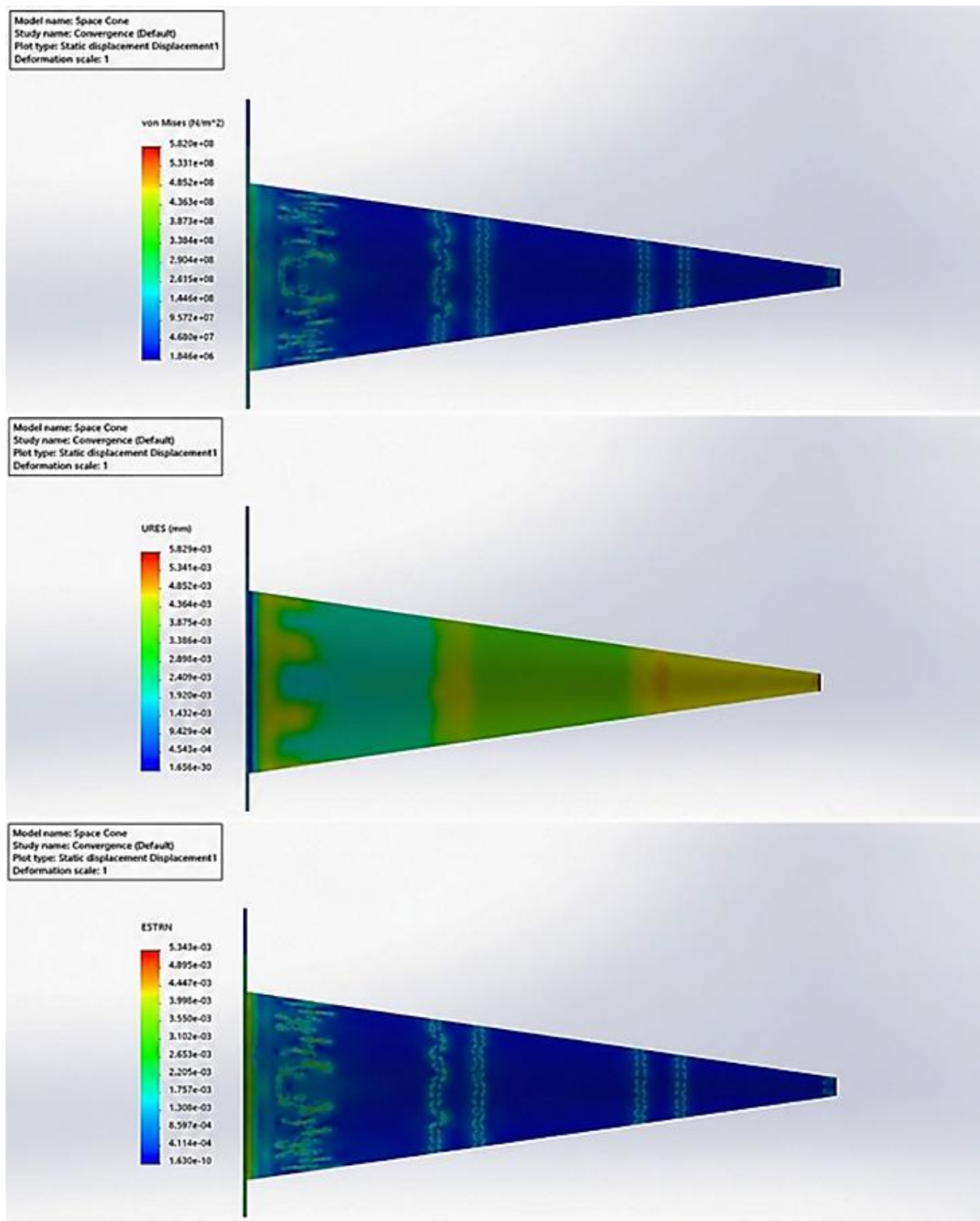


Figure 7.
von Mises Stress, Displacement and Strain for Geometry A, Case 2 (contingency condition)

- **Geometry B – 4 mm punching plate**

Doubling the punching plate thickness to 4 mm increases the flexural rigidity of the cone by approximately a factor of eight, and the effect on the punching plate is correspondingly large. Figures 8 and 10 show the results for Case 1 and Case 2 respectively. Under Case 1 the peak stress in the punching plate falls from 293.2 MPa to 253.6 MPa, a reduction of 13.5%, giving UC = 0.75, while the maximum displacement falls by 6.6% from 0.4585 mm to 0.4284 mm. Under Case 2 the peak stress falls from 371.8 MPa to 322.0 MPa, a reduction of 13.4%, bringing the Unity Check from 1.11 to 0.96 and thereby restoring compliance under contingency conditions. The maximum displacement under Case 2 falls by 7.0% from 0.5863 mm to 0.5455 mm.

The more important consequence of the modification, however, is not the reduction in punching plate stress but the redistribution of load that accompanies it. A stiffer cone transfers a greater share of the pressure load into its supports, and the peak stress in the ring plate rises from 318.8 MPa to 333.8 MPa under Case 2, taking its Unity Check from 0.95 to 0.99. The critical component therefore migrates from the punching plate to the ring plate, which retains a margin of less than 1% against the allowable stress. The same effect explains an apparent anomaly in the strain results: the maximum equivalent strain increases marginally from 1.051×10^{-3} to 1.075×10^{-3} under Case 1 and from 1.340×10^{-3} to 1.362×10^{-3} under Case 2, despite the reduction in punching plate stress, because the location of peak strain has moved off the cone and onto the more highly loaded ring plate. Geometry B is thus structurally acceptable in both cases, but it is acceptable with a very small reserve, and it is acceptable at a different location from the one that the modification was designed to protect. The hydraulic cost is modest: the pressure drop increases by 8.3% under Case 1 and 8.4% under Case 2.

- **Geometry C – 4 mm Punching Plate with Repositioned Ring Plate**

Geometry C retains the 4 mm punching plate of Geometry B but repositions the ring plate by extending the stiffener ring towards the upper cone, where the peak stress in Geometries A and B was located. Figures 10 and 11 present the corresponding stress, displacement and strain distributions. Moving the support closer to the region of maximum differential pressure shortens the unsupported span of the cone and reduces the bending moment carried by the punching plate. The peak stress in the punching plate falls further to 245.7 MPa under Case 1 (UC = 0.73) and 304.6 MPa under Case 2 (UC = 0.91), a reduction of 5.4% relative to Geometry B and 18.1% relative to the baseline under contingency conditions. The maximum displacement falls to 0.4029 mm and 0.5133 mm for the two cases, 12.1% and 12.5% below the baseline values.

The effect on the ring plate is more pronounced still. Its peak stress under Case 2 collapses from 333.8 MPa in Geometry B to 100.8 MPa, taking its Unity Check from 0.99 to 0.30, a reduction of 70%. The repositioned ring plate no longer terminates the primary bending load path of the cone but instead shares the reaction with the extended stiffener ring, so that neither component approaches its allowable stress. Geometry C is consequently the only configuration in which every component retains a substantial reserve under contingency conditions, the highest Unity Check being 0.91 in the punching plate. The strainer cap and the handle remain lightly loaded throughout, with Unity Checks not exceeding 0.15 and 0.76 respectively in any configuration or case, and neither is at any point critical.

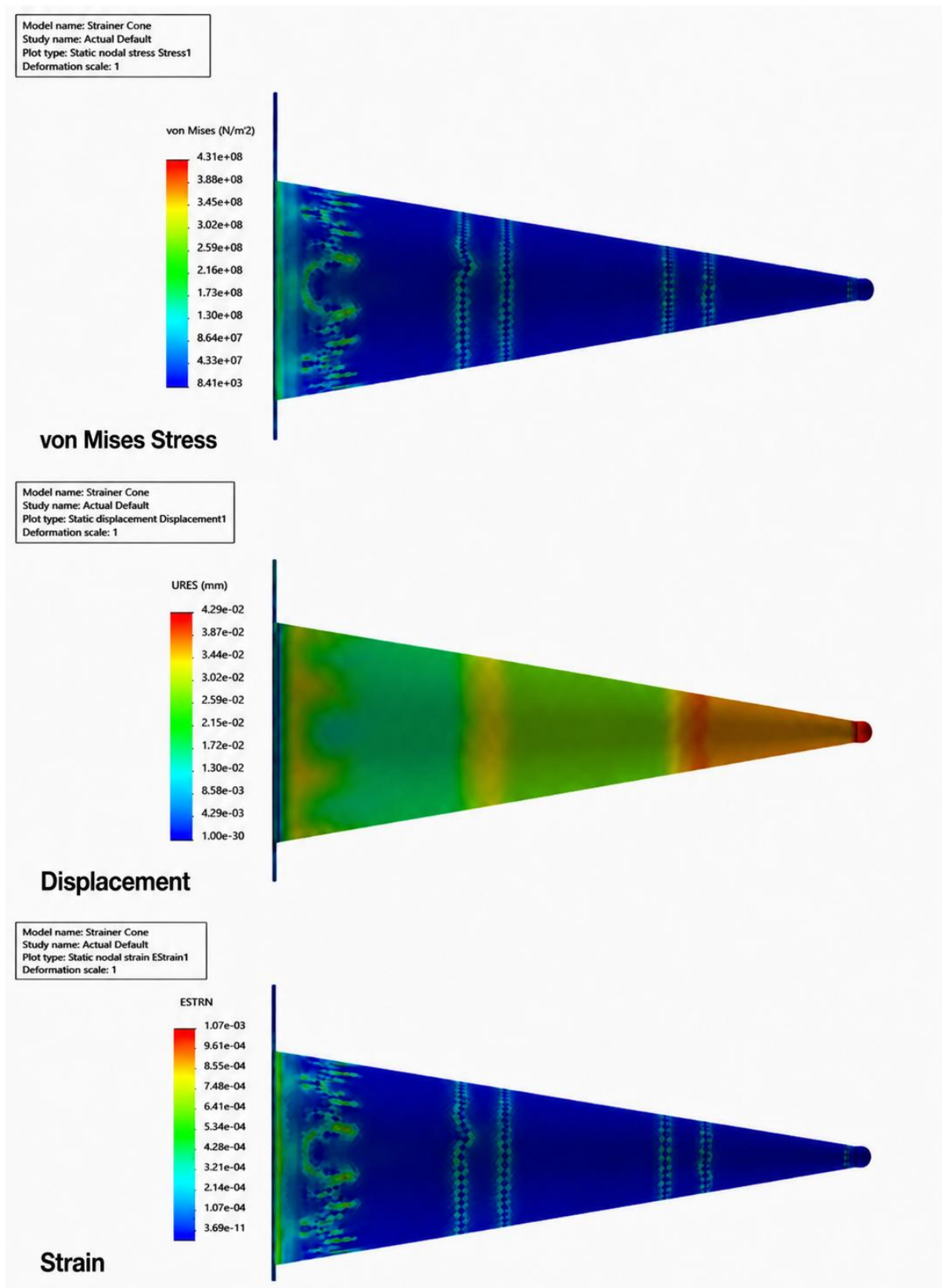
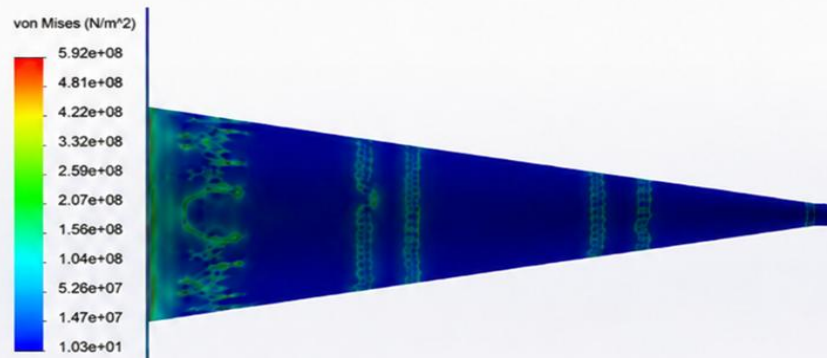


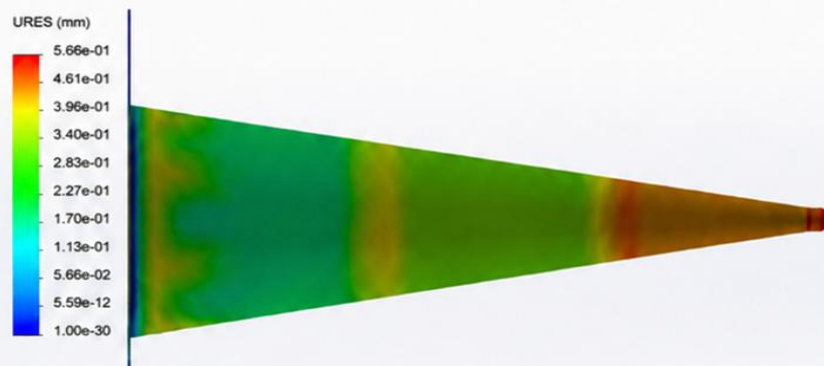
Figure 8.
 von Mises Stress, Displacement and Strain for Geometry B, Case 1 (actual condition)

Model name: Strainer Cone
 Study name: Contingency (Default)
 Plot type: Static nodal stress Stress1
 Deformation scale: 1



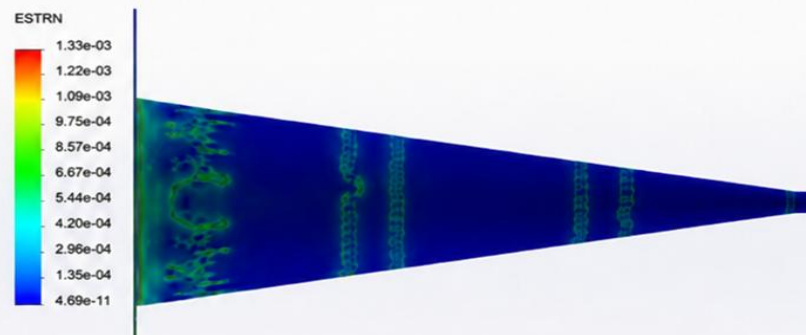
von Mises Stress

Model name: Strainer Cone
 Study name: Contingency (Default)
 Plot type: Static displacement Displacement1
 Deformation scale: 1



Displacement

Model name: Strainer Cone
 Study name: Contingency (Default)
 Plot type: Static nodal Strain1
 Deformation scale: 1



Strain

Figure 9.
 von Mises Stress, Displacement and Strain for Geometry B, Case 2 (contingency condition)

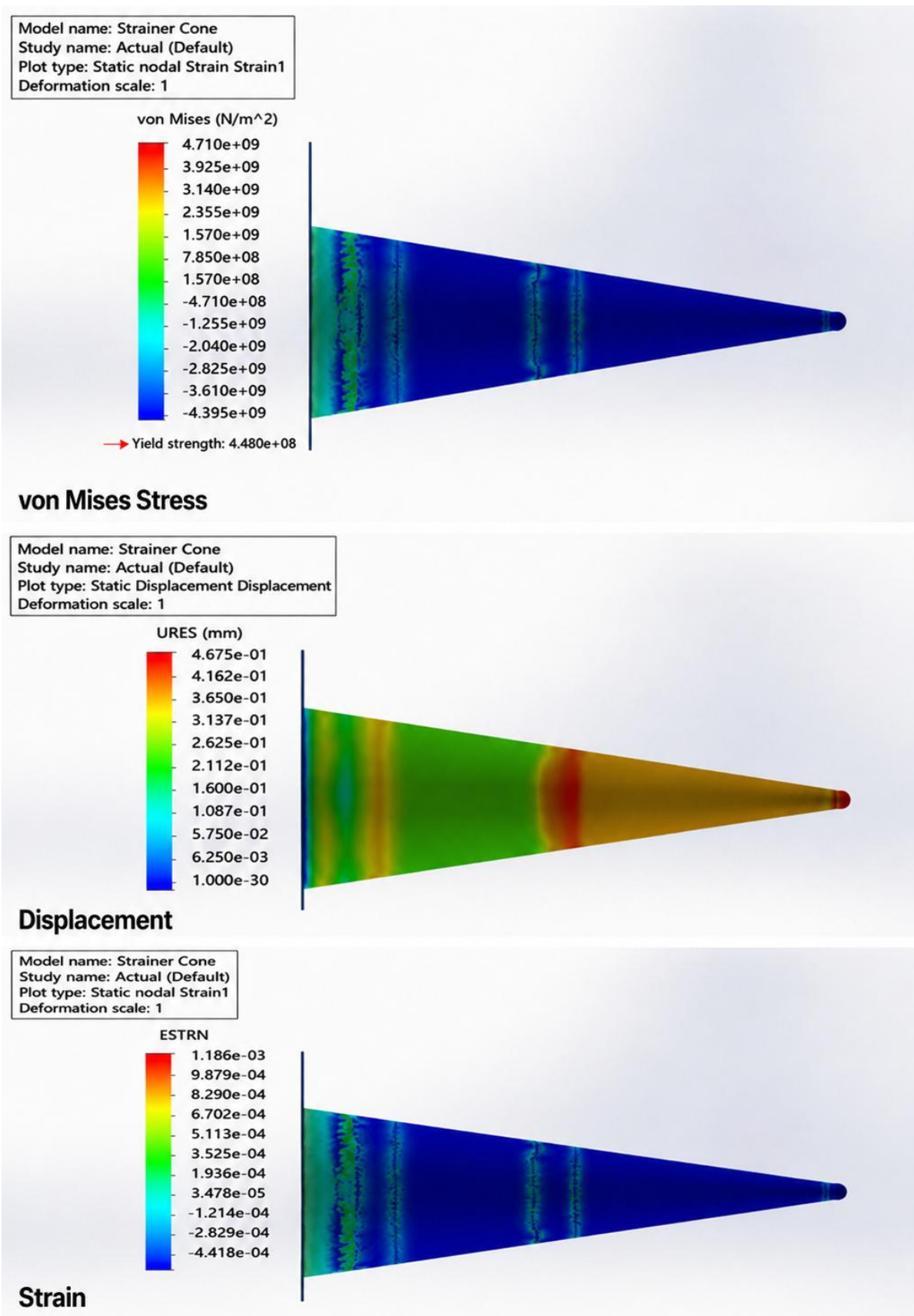


Figure 10.
von Mises Stress, Displacement and Strain for Geometry C, Case 1 (actual condition)

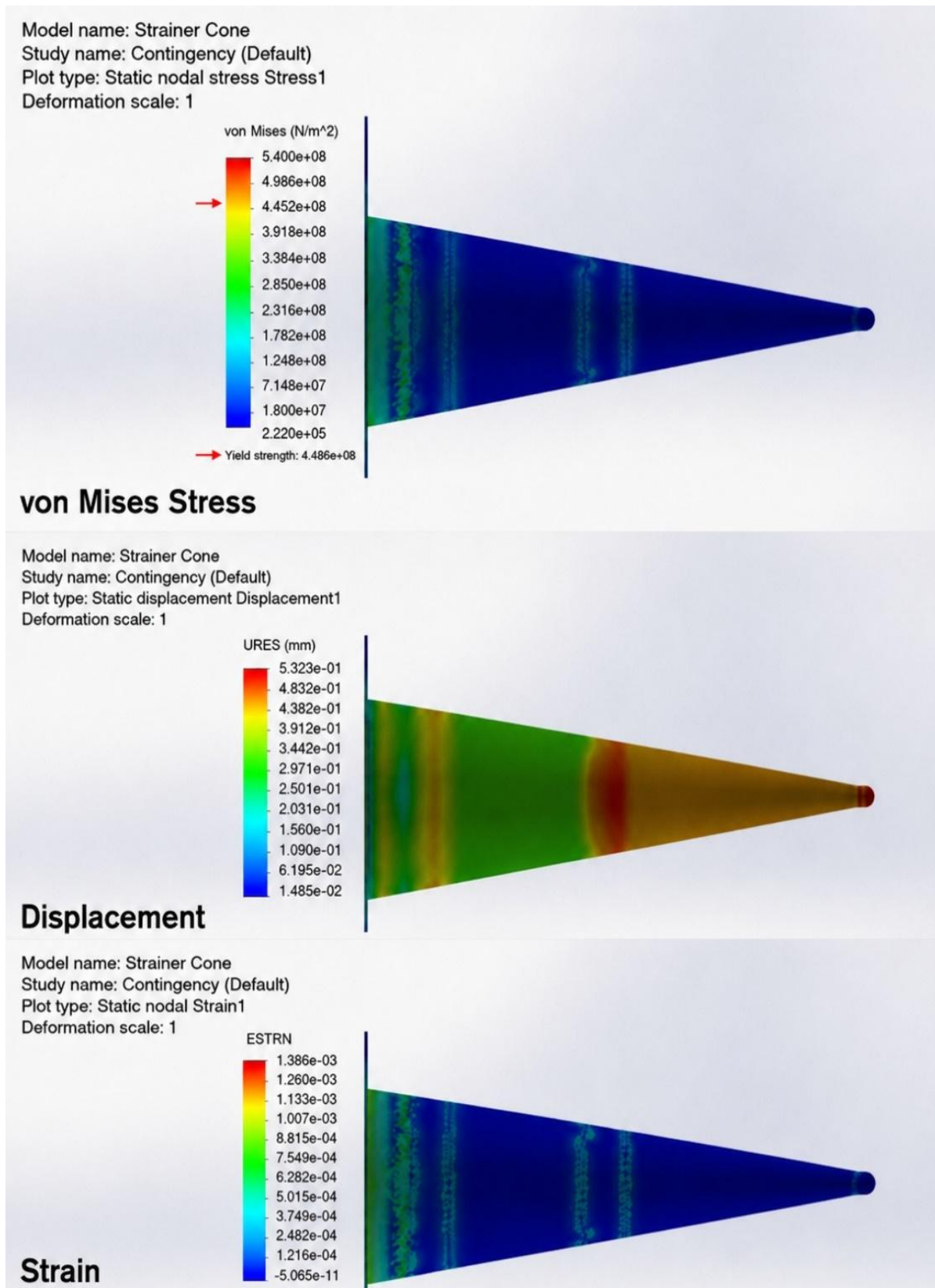


Figure 11.
von Mises Stress, Displacement and Strain for Geometry C, Case 2 (contingency condition)

3.2 Unity check values

Figure 12 and Tables 8 and 9 summarise the Unity Check values for all four components across the three geometries and both operating conditions. The punching plate governs the structural integrity of the baseline design: the Unity Check of Geometry A is 0.87 under Case 1 and 1.11 under Case 2, so that the as-built strainer is compliant in normal operation but is driven past its allowable stress by the pressure spikes represented by the contingency case. This is consistent with the observed service history, in which failure developed after prolonged operation rather than immediately on commissioning.

Thickening the punching plate to 4 mm in Geometry B lowers its Unity Check to 0.75 and 0.96 for the two cases, restoring compliance, but simultaneously raises the ring plate to 0.99 under contingency conditions. The modification therefore relocates the criticality rather than eliminating it, and the resulting design is compliant only by a margin of about 1%. Repositioning the ring plate in Geometry C addresses both components at once, reducing the punching plate to 0.73 and 0.91 and the ring plate to 0.79 and 0.30. Geometry C is thus the recommended configuration.

Table 8.
Maximum von Mises stress and Unity Check by component, Case 1 (actual condition)

Component	Geom. A	Geom. B	Geom. C
Punching plate	293.2 / 0.87	253.6 / 0.75	245.7 / 0.73
Strainer cap	39.5 / 0.12	35.1 / 0.10	35.1 / 0.10
Handle	200.5 / 0.60	200.4 / 0.60	200.5 / 0.60
Ring plate	250.4 / 0.75	262.4 / 0.78	264.0 / 0.79

Values are given as maximum von Mises stress (MPa) / Unity Check (–).

Table 9.
Maximum von Mises stress and Unity Check by component, Case 2 (contingency condition)

Component	Geom. A	Geom. B	Geom. C
Punching plate	371.8 / 1.11	322.0 / 0.96	304.6 / 0.91
Strainer cap	50.2 / 0.15	44.6 / 0.13	47.1 / 0.14
Handle	254.8 / 0.76	254.8 / 0.76	254.8 / 0.76
Ring plate	318.8 / 0.95	333.8 / 0.99	100.8 / 0.30

Values are given as maximum von Mises stress (MPa) / Unity Check (–). The Unity Check exceeding unity for Geometry A under contingency conditions is shown in bold in Figure 12.

Table 10.
Global structural response and hydraulic penalty. Δ (%) is the change in pressure drop relative to the Geometry A baseline for the same case

Geom.	Case	Displ. (mm)	Strain ($\times 10^{-3}$)	Δp (Pa)	Δ (%)
A	1	0.4585	1.051	7273	–
A	2	0.5863	1.340	9097	–
B	1	0.4284	1.075	7873	+8.3
B	2	0.5455	1.362	9862	+8.4
C	1	0.4029	1.055	8705	+19.7
C	2	0.5133	1.340	10894	+19.8

The improvement in structural performance delivered by Geometry C is not free. As Table 10 shows, repositioning the ring plate increases the pressure drop across the strainer by 19.7% under Case 1 and 19.8% under Case 2 relative to the baseline, and by approximately 10.5% relative to Geometry B. The extended stiffener ring

presents an additional obstruction to the flow on the lower side of the cone, forcing the flow to accelerate through a reduced passage and raising the local velocity on the upper side of the strainer, as shown in Figure 13. The trade-off in this design is therefore between structural reserve and hydraulic performance, and not, as might be assumed, between adding material and degrading the structure. Whether the 19.8% pressure-drop penalty is acceptable depends on the available differential pressure budget of the LTSU; the total loss remains small in absolute terms, at approximately 0.11 bar under contingency conditions.

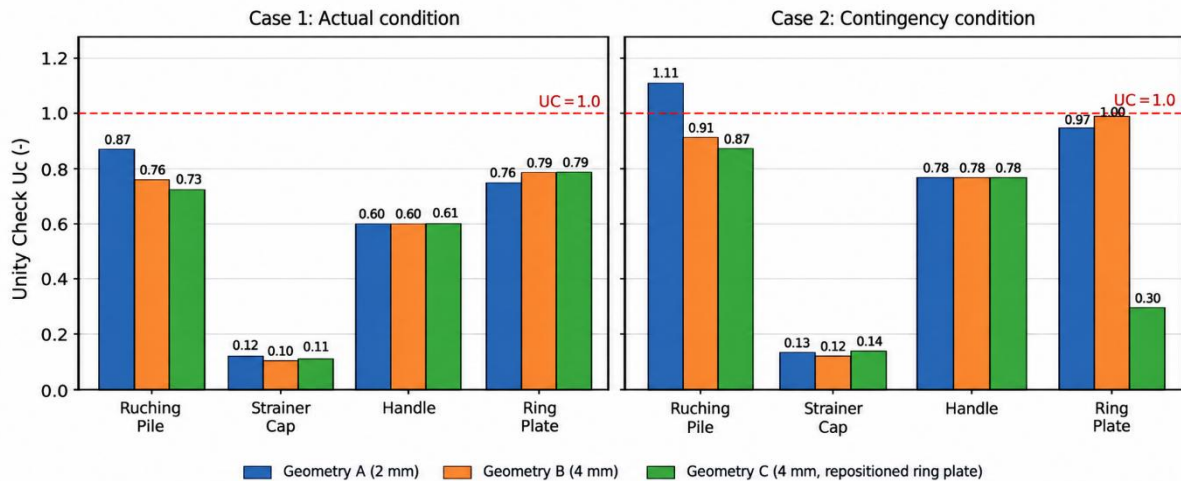


Figure 12.
Unity Check values by component for all geometries under both operating conditions

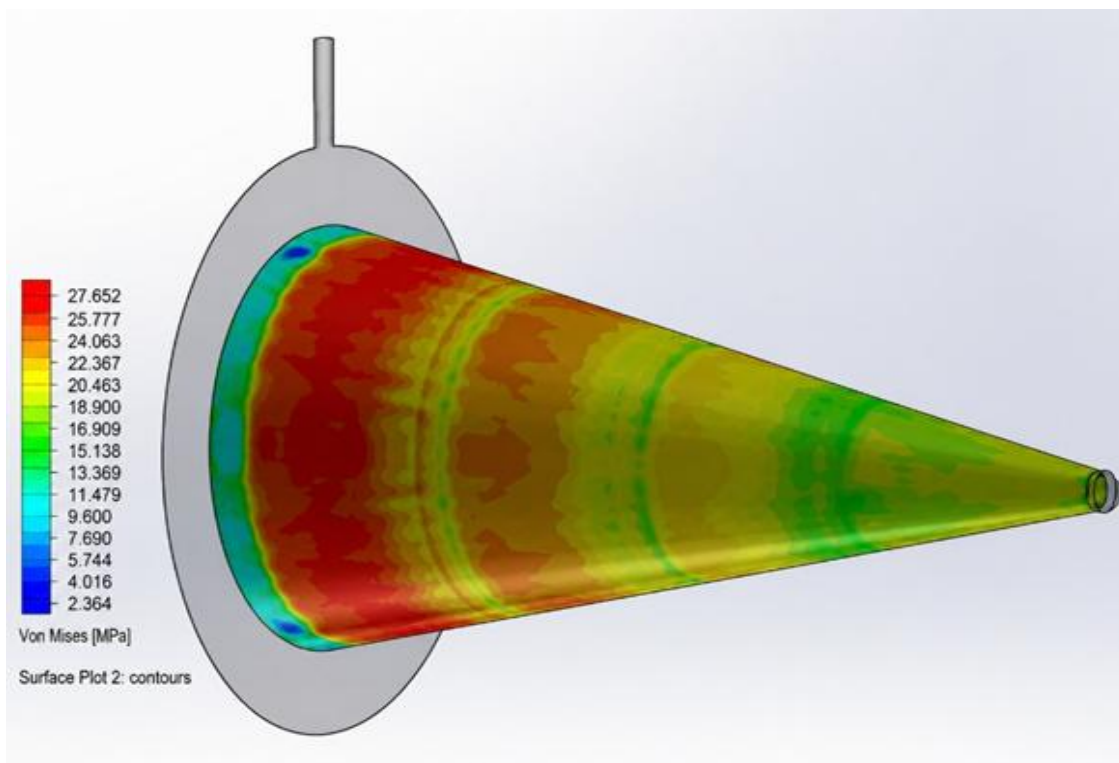


Figure 13.
Increased flow velocity on the upper side of the strainer in Geometry C (CFD plot).

4. Conclusion

This study presented a coupled CFD-FEA analysis for the structural integrity assessment of a conical perforated strainer subjected to fluid-induced loading under actual and contingency operating conditions. The punching plate was identified as the component governing the integrity of the baseline design, owing to the combination of a large pressure-acting area and the low flexural stiffness that results from its perforations. For the as-built 2 mm punching plate the Unity Check was 0.87 under actual conditions but 1.11 under contingency conditions, confirming that the design is compliant in normal operation yet driven past its allowable stress of 336 MPa by the pressure spikes that occur intermittently in service, which is consistent with the failure observed after prolonged operation. Doubling the punching plate thickness to 4 mm reduced the peak contingency stress by 13.4%, from 371.8 MPa to 322.0 MPa, and the Unity Check to 0.96. This modification did not, however, eliminate the criticality but relocated it: the stiffer cone transferred load into its supports and raised the ring plate Unity Check from 0.95 to 0.99, leaving a reserve of less than 1%. Repositioning the ring plate by extending the stiffener ring towards the region of peak stress addressed both components simultaneously, reducing the punching plate Unity Check to 0.91 and the ring plate Unity Check to 0.30, at the cost of a 19.7% increase in pressure drop, from 7273 Pa to 8705 Pa under actual conditions. This configuration is recommended as the optimal design, since it is the only one in which every component retains a substantial reserve under contingency conditions while the hydraulic penalty remains small in absolute terms.

The wider contribution of this work lies in the workflow rather than in the specific geometry. The sequence adopted here — representing the filtering surface as an equivalent porous medium, obtaining the pressure, temperature and wall shear stress fields from a steady-state CFD solution, mapping them onto the structure through one-way fluid-structure interaction, and then evaluating a Unity Check for every component rather than for the component presumed to be critical — can be standardised for the assessment of other pipeline filtration systems in the oil and gas industry, including Y-strainers, T-strainers and basket strainers. The component-wise evaluation is the essential element: had only the punching plate been assessed, Geometry B would have been reported as an adequate remedy, when in fact it leaves the ring plate within 1% of its allowable stress. Design modifications intended to relieve one component should therefore always be verified against the full set of load paths, and the associated hydraulic penalty quantified alongside the structural gain.

5. Acknowledgement

Communication of this research is made possible through monetary assistance by Universiti Tun Hussein Onn Malaysia and the UTHM Publisher's Office via Publication Fund E15216. This research was also supported by the UTHM Research Management Centre (RMC) through the GPPS Fund Q797. The authors would also like to thank the Faculty of Mechanical and Manufacturing Engineering, Universiti Tun Hussein Onn Malaysia, for its support.

6. Declarations

The authors declare that this manuscript presents original work, has not been published previously, and is not under consideration for publication elsewhere.

6.1 Author contributions

The authors confirm contribution to the paper as follows: study conception and design: Mohd Azni Md Kasim and Mohd Azahari Razali; data collection: Mohd Azni Md Kasim and Iman Fitri Ismail; analysis and interpretation of results: Mohd Azni Md Kasim, Iman Fitri Ismail, and Nur Syafiqah Adha Narrudin; draft manuscript preparation: Mohd Azni Md Kasim, Nur Syafiqah Adha Binti Narrudin, Iman Fitri Ismail, Norfakhira Mohd Nor, Amnur Akhyan, and Masataro Suzuki. All authors reviewed the results and approved the final version of the manuscript.

6.2 Funding statement

This research received monetary assistance from UTHM Publisher's Office via Publication Fund E15216 and was supported by the UTHM Research Management Center (RMC) through the GPPS Fund Q797.

6.3 Competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

6.4 The use of AI or AI-assisted technologies

During the preparation of this manuscript, the authors used AI-based writing and editing assistance tools to support language refinement, formatting, and reference organization. All scientific content, data interpretation, and conclusions are the sole responsibility of the authors, who reviewed and edited all AI-assisted output and take full responsibility for the content of this publication.

7. Glossary of terms and symbols

Terms & symbols	Definitions	Unit
LTSU	Low temperature separation unit	-
CFD	Computational fluid dynamics	-
FEA	Finite element analysis	-
UC	Unity check	-
FSI	Fluid-structure interaction	-
Cp	Specific heat	J/kg*K
AR	Aspect ratio	-
$\Delta(\%)$	The change in pressure drop relative	-

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