

Article

Optimal Transport Meshless Method Based Fatigue Life Calculation Method for Hydraulic Pipelines under Combined Excitation

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Abstract: The issue of fatigue in modern hydraulic pipelines is increasingly severe, and there remains a lack of effective prediction methods for pipeline fatigue life. In practical engineering, hydraulic pipelines are primarily subjected to random excitation and fluid excitation, representing a typical composite excitation. Most current research relies on solutions considering only single excitations, which leads to inaccuracies. To accurately calculate the fatigue life under composite excitation, this study incorporates resonance excitation and pulsation excitation into the optimal transport meshless method (OTM) within a strong fluid–structure coupling computational framework. Subsequently, by utilizing the hazard point method based on obtained stress data, an estimation of the service life for engine hydraulic lines can be determined. This work provides both practical guidance and theoretical insights for designing hydraulic lines in modern aircraft.

Keywords: fluid–structure interaction; fatigue life; compound incentives; meshless method; aircraft engines



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1. Introduction

The hydraulic system of an aircraft engine is a mechanism that utilizes oil as the working medium, harnesses energy from the aircraft's system, converts it into hydraulic energy, and supplies it to hydraulic components such as flight control systems and high lift systems. It constitutes a pivotal constituent within aircraft engines. With the advancement of aviation hydraulic systems towards higher pressures, heavier loads, and lighter characteristics, the safety margin of hydraulic pipelines is progressively diminishing, while the issue of fatigue caused by complex vibration environments is also becoming increasingly prominent [1]. For instance, in 2017, a Spanish aircraft experienced a severe fuel pipe failure attributed to fatigue caused by vibration [2]. In the intricate vibration environment of hydraulic pipelines, the composite excitation consists of fluid pulsation excitation and random vibration excitation, thereby imposing a significant alternating load that influences the fatigue life of pipelines [3].

Mitigating or delaying structural damage caused by fatigue has emerged as a prevalent challenge in the military and civil aviation industries, necessitating extensive scholarly research to thoroughly investigate the performance of hydraulic pipelines in aircraft. However, the predominant focus remains on analyzing their dynamic responses under singular excitation. The assessment and verification of fatigue strength in aircraft structures predominantly rely on experimental testing methodologies in most research studies [4]. Brummelen [5] investigated the impact of liquids on pipelines by considering only the added mass effect, without accounting for the mutual interaction between liquids and

hydraulic pipelines in practical scenarios. Wigger [6] predicted the liquid pressure and pipeline stress response under transient excitation using the finite difference equation derived from the feature method. Wang et al. [7] simulated the dynamic response of aircraft pipelines under the pulsating excitation of hydraulic pump sources, by combining the method of characteristic (MOC) and finite element method (FEM). Bishop N [8] pointed out that research on random excitation mostly focuses on the calculation of dynamic response under a single-point excitation source. Netto et al. [9] conducted vibration fatigue tests on flexible hydraulic pipelines and found that when the load cycle reaches 10^7 , most flexible pipelines will experience fatigue failure. Matthew [10] used Fluent software to analyze the effect of pressure pulsation on pipeline vibration characteristics. TJ George et al. [11] used basic excitation to generate high-frequency vibrations and ultimately obtained the fatigue life limit of the material. ZOU et al. [12] proposed a state variable model for analyzing fluid-induced vibration in composite pipeline systems and studied the influence of Poisson's ratio and other factors on fluid pressure. Qing [13] studied the vibration response characteristics of pipelines under different pressure pulsations from an experimental perspective. Liu G [14] studied the axial vibration, lateral vibration, and torsional vibration characteristics of pipeline systems under unsteady flow. Zhai HB et al. [15] solved the dynamic response of pipelines under random excitation using the pseudo excitation method and complex mode superposition method. Li [16,17] conducted research on the optimization of hydraulic pipeline body layout and clamp-rated layout for aircraft hydraulic pipeline systems, combined with the finite element commercial software ADINA8.4, to avoid pipeline resonance. Ouyang X [18] applied the characteristic line method to study the vibration response characteristics of aircraft hydraulic pipelines under pressure pulsation. GU J et al. [19] studied the dynamic response of flow pipelines using the method of generalized integral transformation and investigated the relationship between deflection ratio, fluid velocity, and pipeline mass. Saha [20] studied the fatigue failure problem of industrial pipeline structures and proposed a method for calculating the power spectral density within the framework of random vibration theory. Based on this method, fatigue life prediction was carried out. Zhang et al. [21] conducted a fatigue life assessment of high-pressure fuel pipelines in the frequency domain and found through fatigue tests that low-frequency resonance is the main cause of fuel pipeline failure. Jin et al. [22] conducted fatigue strength analysis of steam pipelines based on the three-interval method. Joo et al. [4] focused on the frequency domain method for estimating the fatigue life of structural vibration, calculated the fatigue life of structures under different frequency bands and rule factor stress power spectra, and proposed a stress spectrum optimization calculation method for the frequency domain method. Mehmood et al. [23] transformed the frequency domain data of hydraulic fluctuations into signals in the time domain and performed graded statistics using the rain flow counting method.

Because structures usually bear composite loads during their operation, some experts have studied the fatigue strength of structures under multi-source composite excitations. Wang et al. [24] conducted research on fatigue life prediction for metal thin-walled aircraft structures under thermal acoustic loads and verified it through fatigue tests. Zhang et al. [25,26] analyzed the influence of thermal vibration loads on the stress-strain and fatigue life of aviation hydraulic pipelines, and they established a thermal vibration coupled stress model for pipelines. Wang et al. [27] investigated a method for determining the dynamic strength safety margin and fatigue durability of pipeline structures under combined loading from pre-stressing and random excitation. Gao P et al. [28] proposed that under high-pressure fluid pulsation and strong random vibration excitation, high-vibration stress can cause cumulative damage to pipeline structures. Shuren Chen et al. [29] provided the excitation parameters and vibration response under multi-source excitation. In summary, the prediction of fatigue life in pipelines by simulating maximum stress under a single excitation is predominantly favored by most contemporary scholars. This preference stems from the inherent complexity associated with concurrently conducting frequency-domain random vibration analysis and time-domain fluid pulsation vibration

analysis. Currently, there is a scarcity of effective evaluation methods to assess the fatigue strength of pipelines subjected to composite excitation.

To address these issues, this article proposes the utilization of the optimal transport meshless algorithm (OTM) within a robust fluid–structure coupling framework. The incorporation of composite excitation into this framework serves to validate the structural integrity and fatigue durability of pipelines, while also enabling accurate estimation of their service life post-design. Resonance excitation and pulsation excitation are identified as the most critical loads in pipelines. Therefore, within the optimal transport meshless method for strong fluid–structure coupling calculations, this study evaluates stress under composite excitations such as pipeline resonance and pulsation. Subsequently, lifespan estimation is performed using the hazard point method (as depicted in Figure 1).

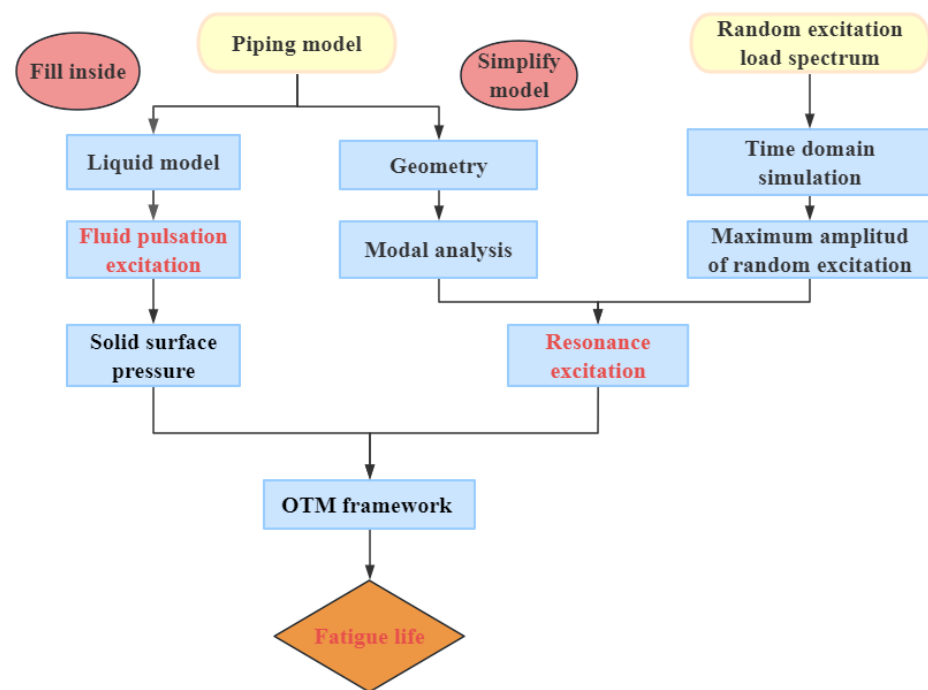


Figure 1. Research process of pipeline fatigue life under composite excitation.

2. Introduction to a Strong Fluid–Structure Coupling Framework—Optimal Transport Meshless Method

The general approach for calculating the dynamic response of hydraulic pipelines under composite excitation is based on a bidirectional fluid–structure coupling simulation, which separately calculates the stress of the pipeline under random vibration excitation and fluid pulsation. The total stress of the pipeline is then determined by superimposing these two types of stresses. It can be observed from existing literature that stress calculation for this type of pipeline essentially involves separate calculations for both fluid and solid components. Incorporating simultaneous random vibration excitation and pulsation excitation into the fluid–structure coupling framework calculation represents an effective solution surpassing that in the previous literature. The objective of this study is to achieve dynamic response calculation under composite excitation conditions in hydraulic pipelines, using an optimal transport meshless fluid–structure coupling framework.

2.1. Introduction to Optimal Grid-Free Transportation

Bo Li and Ortiz [30] proposed the optimal transport meshfree method (OTM), an explicit incremental updated Lagrangian meshless computational approach that combines material point space discretization, local maximum entropy approximation, and optimal transport theory in a cohesive manner. The fundamental principle of the OTM method lies in establishing a connection between the Benamou–Brenier differential form of the optimal

mass transportation problem and the Kantorovich Wasserstein distance. By adhering to Hamiltonian principles, discrete matter's inertial effects in both space and time are discretized within a rigorous variational framework. Consequently, this discretization yields piecewise linear motion of matter points within mass concentrations under inertial action. Building upon these foundational principles, we present a comprehensive formulation of the optimal mass-transport differential equation based on computational fluid dynamics solutions for Monge-Kantorovich mass-transport problems. This establishes a meshless computational framework for optimal transport applicable to any geometry and constitutive behavior (including general elastic solids, fluid motion, and fluid–structure coupling).

2.2. Continuum Dispersion

The OTM method employs a combination of matter points and nodes to discretize the continuous medium in meshless space, with nodes and matter points carrying all field-variable information. During the calculation process, physical information of the material, such as deformation, stress, and internal parameters, is stored at the matter points, while dynamic information of the computed neighborhood, including displacement, velocity, acceleration, and temperature, is stored at the nodes. The motion of the computed neighborhood is determined through interpolation based on dynamic information from neighboring nodes. By utilizing a neighborhood search algorithm, the OTM method eliminates any binding relationship between elements and nodes, enabling accurate prediction and simulation of large deformations in continuous media (as depicted in Figure 2).

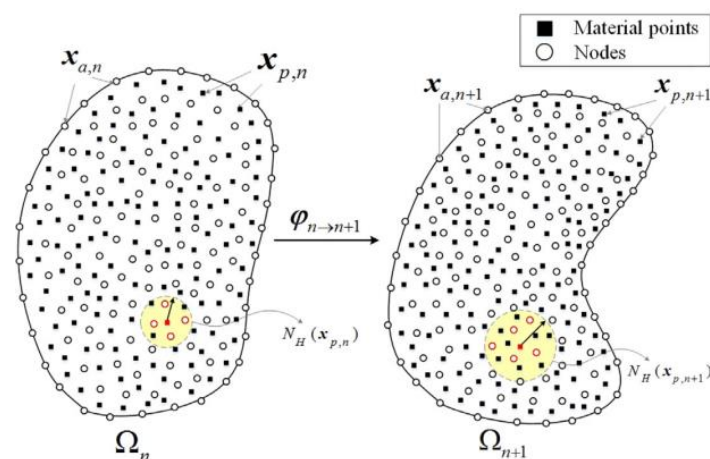


Figure 2. OTM discretizes the continuous medium into two distinct nodes [31].

2.3. Form Function

The primary distinction among different meshless methods lies in the construction of the interpolation function for the displacement field within the continuous medium domain. In the OTM method, this interpolation function takes the form of a local maximum entropy (LME) shape function. LME shape functions possess the desirable property of a weak Kronecker delta at boundaries, thereby overcoming limitations associated with interpolation functions in SPH methods. This article aims to introduce and highlight the theoretical advantages of using LME shape functions within the OTM method for simulating fluid–structure coupling problems.

Given a discrete set of nodes x_i within a continuous medium, the displacement field of that medium can be expressed as a combination of node sets:

$$u(x) = \sum_{i=1}^n p_i(x_i) u(x_i) \quad (1)$$

where $p_i(x_i)$ is the form coefficient and $u(x_i)$ is the displacement at node x_i . According to [32], the value of $p_i(x_i)$ should satisfy the following equation:

$$\min f_{\beta}(x, p) = \beta \sum_{i=0}^n p_i |x - x_i|^2 + \sum_{i=0}^n p_i \log(p_i) \quad (2)$$

where $\sum p_i \log(p_i)$ describes the information entropy of the node-set, and $\sum p_i |x - x_i|^2$ represents the effect of other nodes in the near neighborhood of the node at x . The constraint ensures the continuity of the shape function up to zero and first order, thereby satisfying the continuity requirements of the displacement field. The solution of p_i can be obtained by

$$p_i = \frac{1}{Z(x, \lambda^*(x))} \exp \left[-\beta |x - x_i|^2 + \lambda^*(x) * (x - x_i) \right] \quad (3)$$

where parameter $\lambda^*(x)$ and $Z(x, \lambda^*(x))$ satisfy the following equation:

$$\lambda^*(x) = \arg \min \log(Z(x, \lambda^*(x))) \quad (4)$$

Equation (4) can be easily solved by Newton–Raphson iteration. $\beta \in R^+$ is a Pareto optimization parameter that measures the maximum entropy and the effects of other nodes in the near neighborhood. In practice, to account for the effect of node distribution, Li [33] et al. adopted a size parameter “ γ ” to control the shape function of the LME, and γ satisfies the following equation

$$\gamma = \beta h^2 \quad (5)$$

where h is the characteristic scale of the node set and can generally be set to compute the shortest distance of the neighborhood. When “ γ ” varies from 0.8 to 6.8 (see Figure 3), the search range of the neighborhood of the LME shape functions decreases gradually from its large initial value. The derivation above reveals that the LME must encompass all nodes of the boundary, adhere to the convex approximation, and satisfy the weak Kronecker delta attribute.

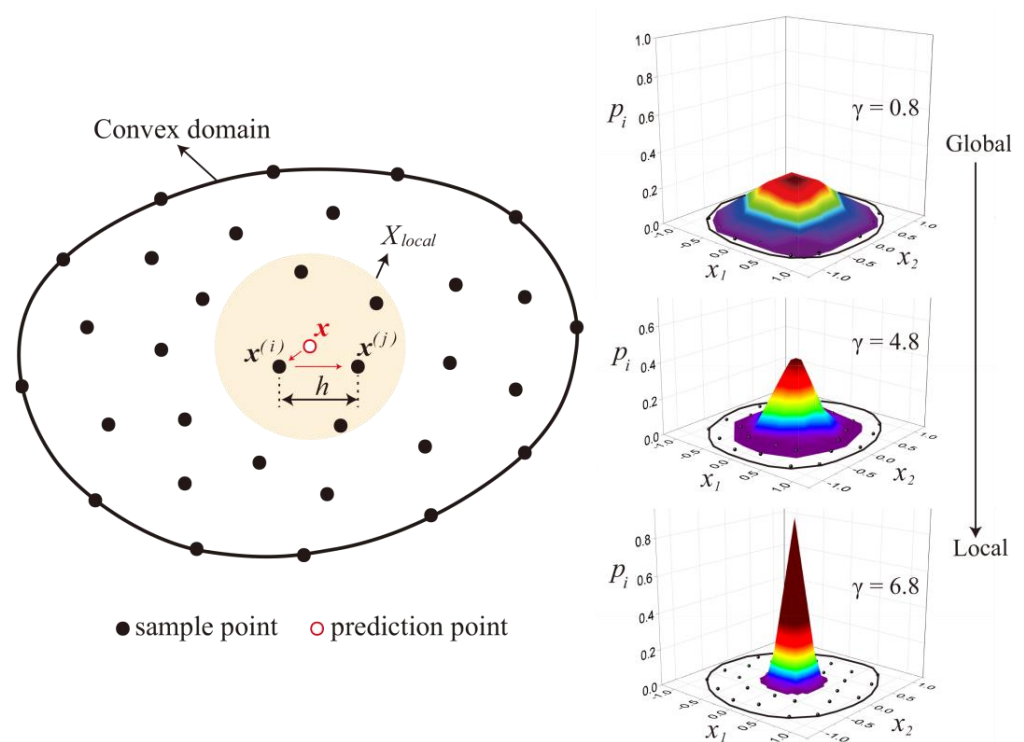


Figure 3. The neighborhood of LME follows changes in “ γ ”.

3. Framework for the Solution of Fatigue Life of Pipelines under Combined Excitation Based on the OTM Method

The pipeline system in aviation hydraulic systems plays a crucial role in connecting the pump source and actuator for the efficient transmission of hydraulic fluid. In aircraft, the aviation hydraulic pipeline system is typically routed from the engine accessory casing transmission device to the landing-gear wheel compartment and rear accessory compartment. Due to spatial limitations, numerous pipelines intersect each other, resulting in minimal gaps between adjacent accessories (as depicted in Figure 4). This configuration represents a part of the hydraulic system of Garrett GTCP85-98D gas turbine engines.

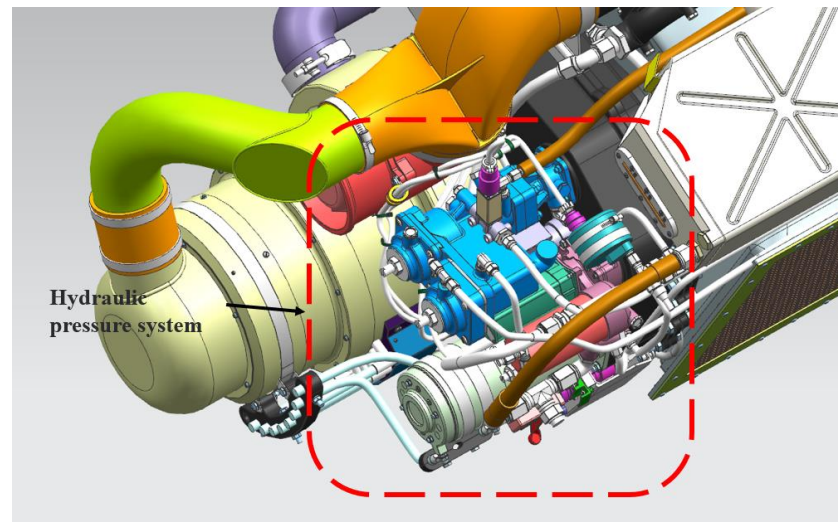


Figure 4. Hydraulic system for the GTCP85-98D gas turbine engine by Garrett.

Aero-engine pipelines operate under prolonged high-frequency vibration conditions, leading to significant fatigue in vulnerable sections of the lines and subsequently resulting in failure modes such as fractures and oil leaks, particularly at welded and bent areas. Despite designers' efforts to develop empirical formulas for pipeline design, failures remain inevitable. Testing and practice demonstrate that the failure of a liquid-filled pipeline during operation is attributed to stress fatigue caused by high-frequency variable-amplitude alternating loads. The pulsation and random vibration excitation of high-pressure pump source fluids are two types of alternating loads that exert significant influence on the fatigue strength of the pipeline. In the initial design phase of hydraulic lines in aircraft, the issue regarding fatigue strength was not prominent due to the implementation of a high safety factor and sufficient strength margins, which ensured that the working stress of liquid-filled lines remained lower than their allowable stress. The research on liquid-filled pipelines also focuses on their dynamic response under a single excitation. However, with the development of modern aviation hydraulic systems towards high pressure, high load, and lightweight designs, their strength reserves are diminishing, thereby exacerbating fatigue issues. Only by investigating the fatigue damage caused by dynamic stress values under combined excitation can the structural safety of the pipeline system be evaluated with greater accuracy.

This section will select a section of a typical aircraft hydraulic line structure model (Figure 5) as the research object. The pipeline consists of a catheter, pipe clip, bracket, and bolt. The total length of the pipeline is 0.8 m, with an outer diameter of 13.8 mm and an inner diameter of 13 mm.

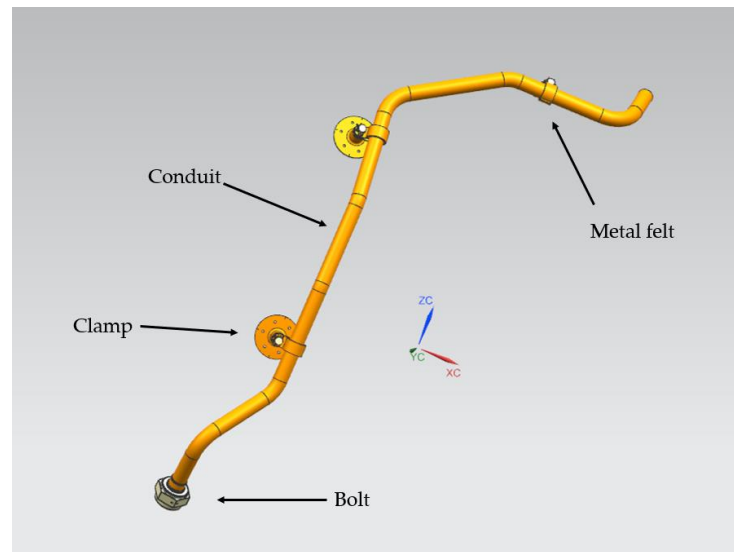


Figure 5. Typical pipeline structure geometry model.

The OTM method is employed as a simulation tool to establish a dynamic analysis approach for evaluating the structural response of pipelines subjected to resonance and fluid pulsation excitations. The critical location with maximum stress in the pipeline is identified for fatigue life assessment, thereby enabling the development of a quantitative evaluation methodology for calculating the fatigue life of composite-excited pipeline structures.

3.1. A Method of Acquiring Pipeline Resonant Excitations Based on Load Spectra

Based on the aforementioned introduction of the OTM method, it is evident that this explicit integration algorithm incorporates boundary conditions exclusively capable of imposing time-domain signals. Consequently, the OTM method is employed for simulating dynamic stress in pipelines. One approach involves extracting time-domain signals from random vibration excitations at both ends of the pipeline and applying them to the supports and endpoints. However, this approach necessitates the recording of a substantial amount of data. In the time-domain signal, components with low amplitude and frequencies significantly distant from the pipeline's natural frequency exhibit minimal energy input and contribute insignificantly to the maximum stress experienced by pipeline vibrations. Therefore, it is necessary to extract only the maximum amplitude of random vibration or the excitation signal that closely matches the fundamental frequency of the pipeline. This will enable us to obtain the vibration stress of the pipeline at its most critical moment. Consequently, this paper proposes a resonance excitation extraction method based on a load spectrum. Specific research ideas are shown in Figure 6.

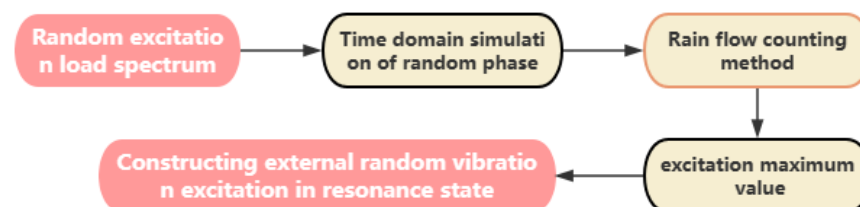


Figure 6. A resonance excitation extraction method based on a random load spectrum.

First, the acceleration power spectral density function, as depicted in Figure 7, is selected based on the guidelines specified in GJB 150.16A-2009 [34] to accurately simulate real working conditions.

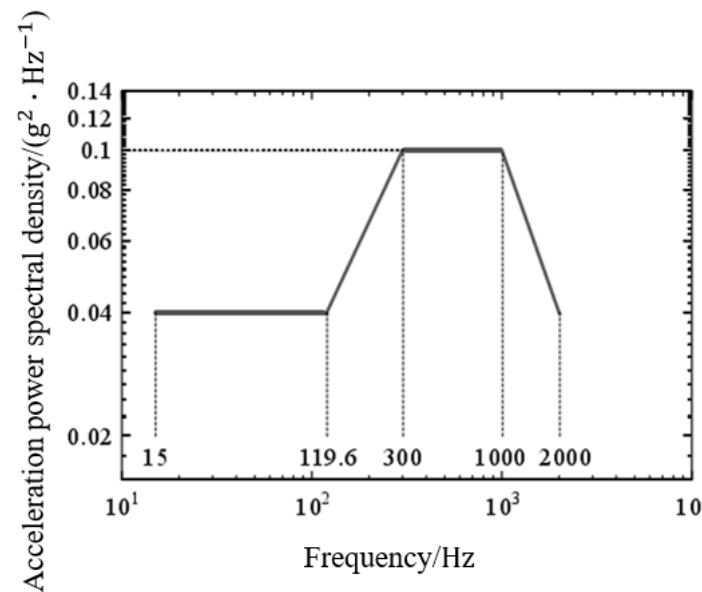


Figure 7. Load spectrum of pipeline system.

The load spectrum of the system is then simulated in the time domain using a spectrum estimation algorithm. This novel algorithm generates a set of non-deterministic linear time series data, all exhibiting a $(1/f)$ spectrum, by randomizing both the phase and amplitude of the data's Fourier transform according to their inherent randomness (refer to [35] for detailed steps).

Finally, as depicted in Figure 8, the irregular random stress time history signal obtained from the time-domain simulation is transformed into a sequence of stress cycles using the rain flow counting methodology. According to the stress cycle depicted in Figure 8, the maximum amplitude achieved through the time-domain simulation of random vibration excitation is measured at 0.0047 m/s^2 .

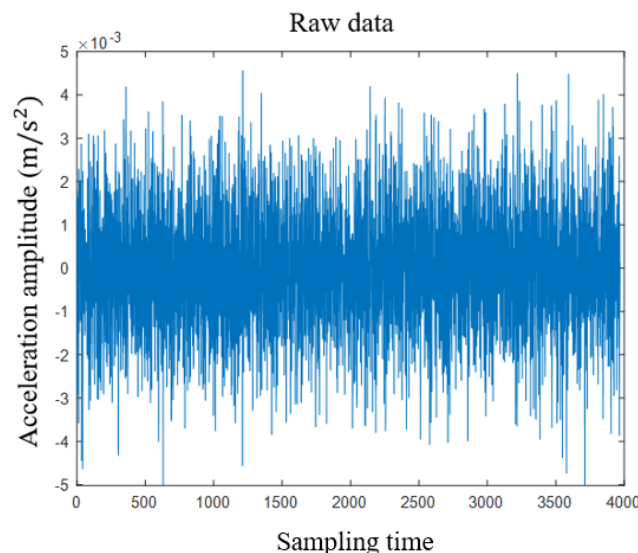


Figure 8. Time-domain simulation results of random vibration signals and rain flow counting analysis.

To acquire information such as the natural frequency of the pipeline, a modal analysis of the pipeline is essential. The material models for each component of the pipeline are presented in Table 1. The conduit was made from 0Cr18ni9, while LY12-M was utilized for the felt material. LY12-CZ served as the sleeve material, 2B06 was chosen for bracket fabrication, and bolt production employed 30CRMnSiA. By imposing a fixed constraint

on the start-stop position of the line and the contact surface of the bracket (Figure 9), the calculation yielded the mode shape of the line and stress distribution under random vibration, as depicted in Figure 10 and Table 2. Considering that the frequency band covered by the engine's random excitation power spectrum ranges from 15 to 2000 Hz, modal analysis is employed to extract the modal frequencies within this range. Within the specified frequency range of 15–2000 Hz, a total of 14 modes are identified for the hydraulic pipeline structure. The first three modes are presented in Table 2. These are the first three natural frequencies of the pipeline. Consequently, valuable information regarding the natural frequencies of the pipeline was obtained.

Table 1. Material parameters for simulating pipeline dynamic response.

Component	Assembly	Material	Density (kg/m ³)	Young's Modulus (Pa)	Poisson's Ratio
conduit	conduit	0cr18ni9	7860	199×10^9	0.295
pipe clamp	metal sheet	LY12-M	2800	66×10^9	0.33
	sleeve	LY12-CZ	2800	66×10^9	0.33
support	support	2B06	2760	68×10^9	0.33
	bolt	30CrMnSiA	7750	196×10^9	0.30

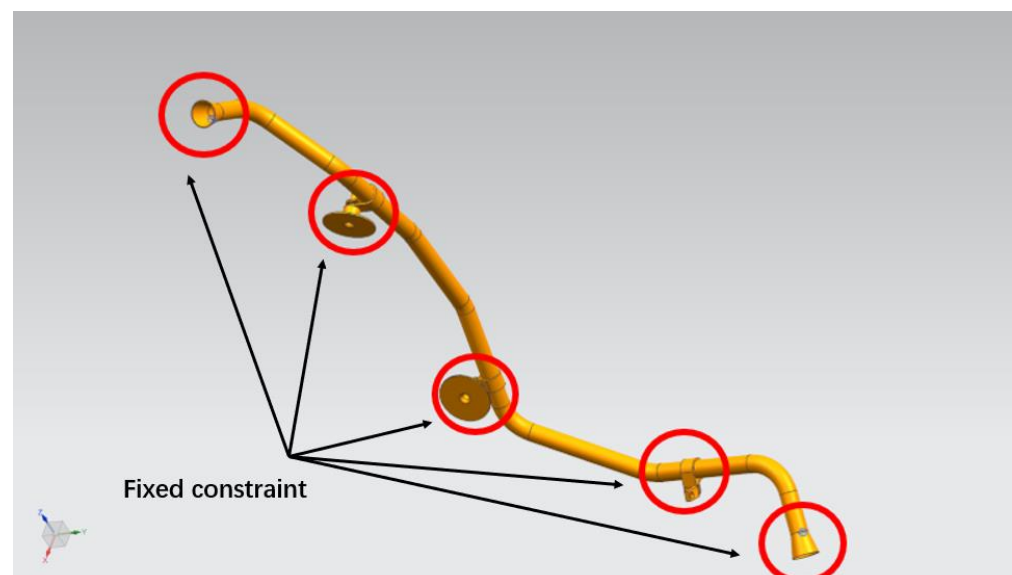


Figure 9. Imposition of a fixed constraint.

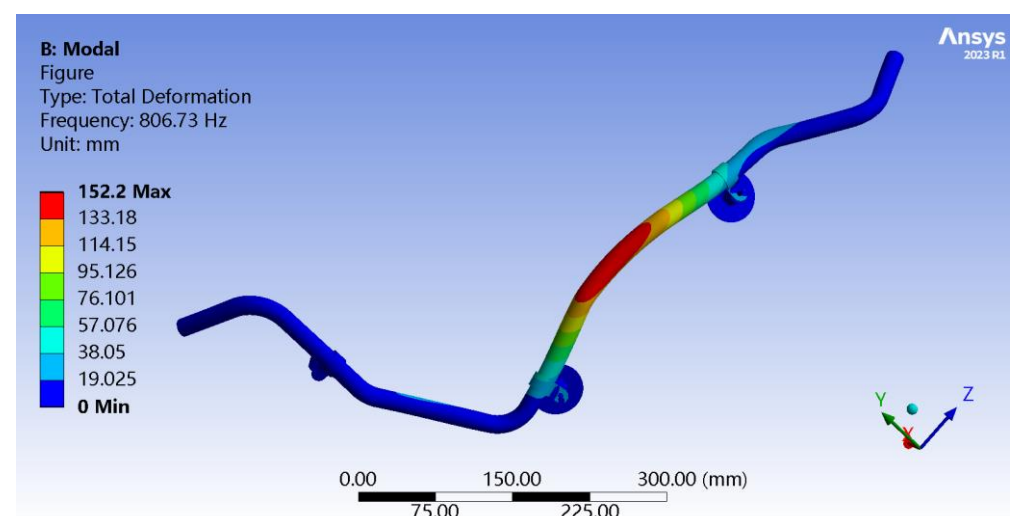


Figure 10. Pipeline modal analysis.

Table 2. The first three natural frequencies of the pipeline.

First Three Modes	Frequency (Hz)
first mode	806.73
second order mode	1080.7
Third order mode	1205.5

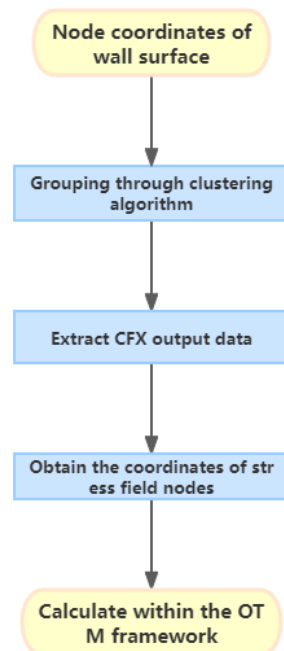
We establish the resonant excitation of the line as

$$a = A * \sin(w_0 * t) \quad (6)$$

In the formula, a represents the variation of acceleration over time between the two ends of the pipeline and the surface of the bracket, A is the maximum amplitude obtained after time-domain simulation of the random vibration excitation, and w_0 is the first-order natural frequency after the pipeline mode analysis. In the simulation of this article, $A = 0.0047 \text{ m/s}^2$ and $w_0 = 806.731/\text{s}$.

3.2. Solution of Pulsating Excitation of Fluid on the Inner Wall of a Pipeline

The most hazardous resonant excitation in the random excitation was obtained through time-domain simulation of the load spectrum. The objective of this summary is to obtain the pressure exerted on the inner wall of the pipeline under fluid pulsation excitation, and it subsequently provides the calculated pressure to the OTM for determining the dynamic stress of the pipeline under compound excitation. The specific steps are as follows: firstly, CFX calculates the pressure on the inner wall of the pipeline under pulsation excitation; then, by setting sampling points, a set of pre-stress fields is obtained through the OTM's calculations. Figure 11 illustrates the process of extracting stress.

**Figure 11.** Extracting the distribution of pulsating excitation on the inner wall pressure of pipelines.

The pipeline configuration depicted in Figure 5 employs a nine-plunger pump, featuring a maximum rotational speed of 4200 revolutions per minute and a rated flow rate of 270 milliliters per second. Given the consistent pressure pulsation cycle with the suction and exhaust cycles of the piston [36], the frequency of pressure pulsations can be expressed as follows based on literature reference [37,38]:

$$f = \frac{NZ}{60} \quad (7)$$

where N is the rotational speed and Z is the number of pistons. The frequency $f = 630$ Hz of the fluid pulsation excitation is generated by the piston pump. The period T of the fluid pulsation excitation is:

$$T = \frac{1}{f} \quad (8)$$

The period of pulsation excitation, $T = 0.00158$ s, can be determined. As the structural stress response does not reach a stable state in the initial cycles, 10 cycles are selected for simulation purposes. Consequently, the total duration of the CFX solution is set to 0.016 s. In CFX, transient analysis is employed as the solution type, with specific settings for the fluid solution time and time step:

$$u = \frac{4Q}{\pi d^2} \quad (9)$$

where u represents the oil flow velocity, measured in meters per second (m/s); Q denotes the rated flow of the piston pump, expressed in milliliters per second (mL/s); and d signifies the inner diameter of the conduit, measured in millimeters (mm). The hydraulic conduit has an inner diameter of $d = 13$ mm, while the piston pump's rated flow is $Q = 16.2$ L/min. Consequently, an oil flow velocity of $u = 2.034$ m/s is achieved. The pressure pulsation function for the oil exhibits an average value of 28 MPa and an amplitude of 2.1 MPa, which can be mathematically represented by

$$P = 28[\text{MPa}] + 2.1[\text{MPa}] * \sin(2\pi ft) \quad (10)$$

The fluid entry surface is positioned on the left, with the flow rate set perpendicular to the cross-section, while the fluid exit is located on the right and has an exit pressure of (10) (Figure 12).

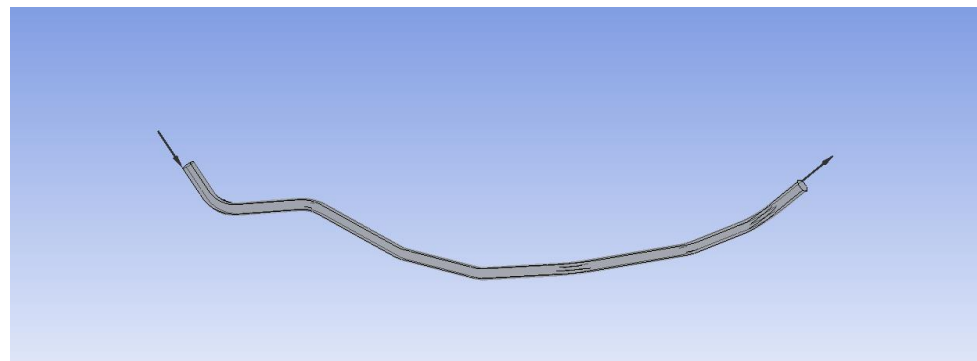


Figure 12. Fluid boundary settings.

After obtaining the pressure exerted by pulsation excitations on the inner wall of the pipeline through CFX calculations, a total of 1000 sampling points were strategically placed along the inner wall to capture and output stress variation profiles during each cycle period. The aforementioned approach was employed to extract the pressure data from within the pipeline. This enabled us to successfully acquire the preload necessary for OTM calculations. Figure 13 illustrates a schematic diagram depicting the precise locations of these randomly selected sampling points.

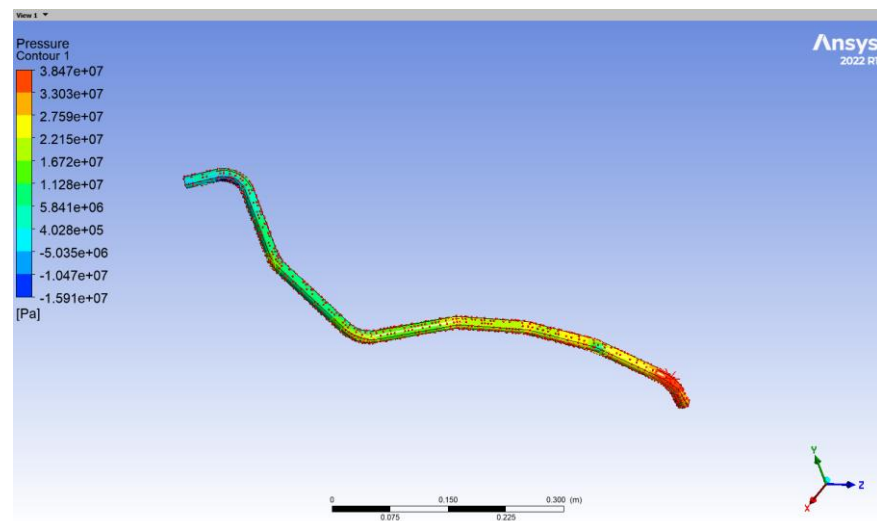


Figure 13. Positioning of sampling point along the inner wall of the pipeline.

4. Solution of Fatigue Life of Pipeline under Combined Excitation under OTM Frame

4.1. Dynamic Response Analysis of Hydraulic Pipeline Based on Optimal Transport Model (OTM)

The previous paragraph extracted the pressure and resonance excitation of fluid pulsations on the inner surface of the pipeline. This section aims to utilize the extracted pulsation excitation and resonance excitation within the OTM framework to calculate the dynamic response of the pipeline, as well as to determine its fatigue life at the maximum stress point.

The resonant loads and the pressure on the inner wall of the pipeline were previously extracted to introduce both vibration excitation and pulsating excitation of the fluid into the OTM. To account for potential misalignment between sampling points of pipeline inner wall pressure and the OTM grid file, nodes on the inner wall surface are clustered into 100 sets based on their distance from sampling points. A pressure load is then applied to each set in order to incorporate a pre-stress field excited by fluid pulsation in the OTM. Figure 14 illustrates how the inner wall surface is divided into four sets based on distance.

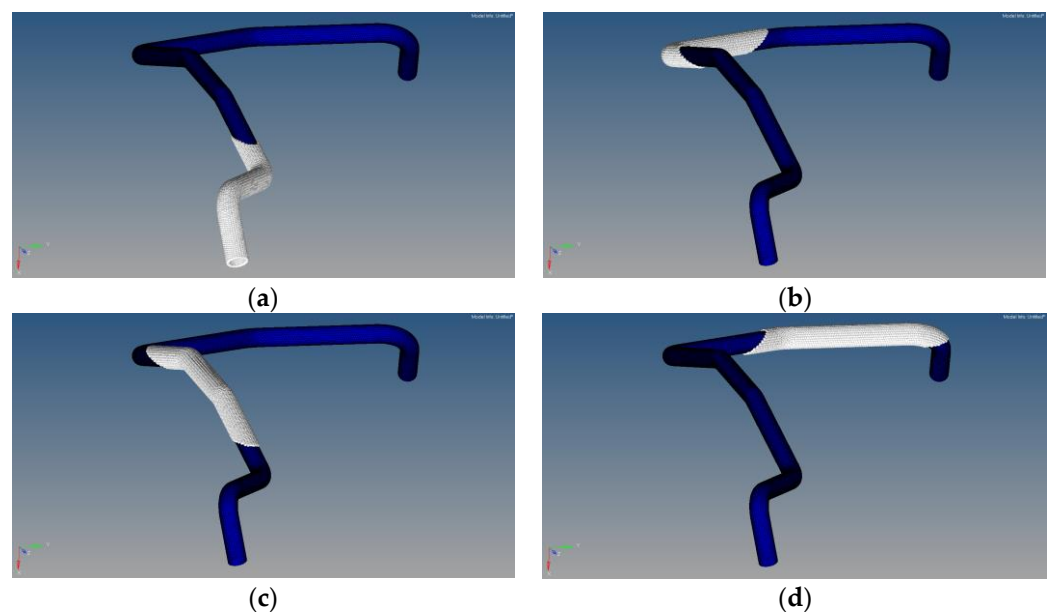


Figure 14. The nodes along the inner wall surface of the pipeline are clustered based on their proximity to the sampling point. ((a–d) are four sets of inner wall surfaces).

The discrete model used in the numerical simulation of OTM consists of 185,932 nodes and 681,965 matter points. Resonant excitation is applied to the starting and ending positions of the lines as well as the bracket contact surfaces. Table 1 provides details on the actual material parameters utilized, while Table 2 outlines key parameters relevant to the numerical simulation algorithm employed. Based on this discrete model combined with initial conditions, boundary conditions, and material parameters, we simulate dynamic stress within a pipeline under compound excitation using our model. Our total simulation time amounts to 0.0016 s (Table 3); Figure 15 displays results for the dynamic stress experienced by pipeline structures under combined stress.

Table 3. Calculation parameters related to dynamic stress in pipelines under OTM simulation of composite excitation.

Parameter	Value	Meaning of Parameters
Time_Step_Ratio	0.2	key time step long step scale
Simulation_Time	0.0016 (s)	total time simulated
Dump	20	number of visualization result files
Search_Range	1.5	neighborhood search range control parameters
Beta	1.4	shape control coefficient of shape function for local maximum entropy
Material_Type	ESCAAS_MAT_J2_POWERLAW	elastic-plastic material model
Engery_release	4×10^6	critical energy release rate

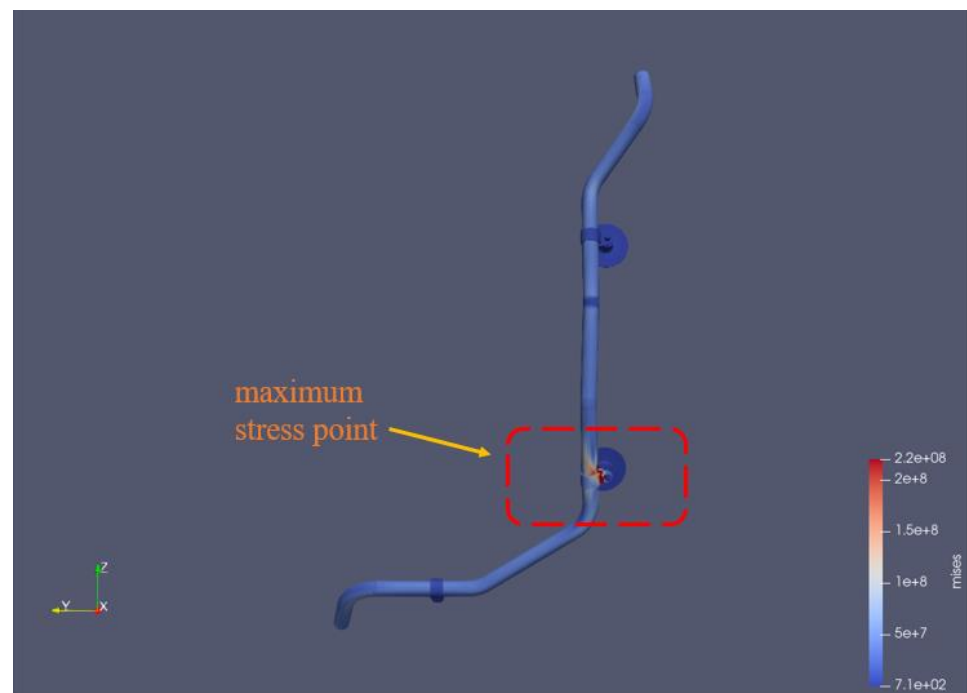


Figure 15. Dynamic stress simulation results of pipeline structure under combined excitation.

The simulation results demonstrate that the point of maximum stress in the tube body coincides precisely with the attachment location of the clamp, where a peak von Mises stress of 228 MPa is observed.

4.2. Estimation of S-N Curves Based on the Strength Limit and Yield Limit

After acquiring the stress data from the pipeline, it becomes imperative to establish the S-N curve, which depicts the relationship between stress amplitude (or maximum stress) characterizing cyclic load stress levels and the cycle times characterizing fatigue life.

Consequently, by referring to the material's S-N curve, one can determine the fatigue life of the pipeline under compound excitation.

The fatigue strength at 10^3 cycles is typically determined by employing the material's strength limit approximation, which is subsequently adjusted to incorporate factors such as reliability. The fatigue strength at 10^3 cycles, denoted as $S_{100,R}$, depends on the specific loading conditions and a correction factor accounting for reliability:

$$S_{1000,R} = S_{1000} * C_R \quad (11)$$

where S_{1000} is the fatigue strength of the standard test article at 10^3 cycles under different loading conditions; C_R is the reliability correction factor. Tests show that S_{1000} is a strength limit approximately equal to 75% at axial loading σ_b :

$$S_{1000} = 0.75\sigma_b \quad (12)$$

The fatigue limit σ_{-1} is based on the yield limit of materials [39], and it can be determined using empirical formulas, such as

$$\begin{aligned} \sigma_{-1} &= a\sigma_s + bE + c \\ a &= 0.139e^{-0.302K_t R^2} + 0.435e^{-0.395K_t R} + 0.836e^{-0.480K_t} \\ b &= (0.473e^{-0.577K_t R^2} + 1.479e^{-0.670K_t R} + 2.842e^{-0.755K_t}) * 10^{-3} \\ c &= 12.0K_t R + 28.2K_t - 82.8R - 191 \end{aligned} \quad (13)$$

where E is Young's modulus in MPa; σ_s is the yield strength of the material, in MPa; K_t is the stress concentration factor; and R is the cyclic excitation stress ratio. The strength assessment method discussed in this paper is based on the results of structural dynamic simulations, utilizing an actual structure-specific simulation model. We incorporate the stress concentration factor into the simulation; hence, there is no need for redundant consideration of its influence here. Additionally, through the implementation of an average stress correction model, asymmetric cyclic excitation can be equivalently transformed into symmetric cyclic excitation with a corresponding amplitude. Consequently, under $K_t = 1$ and $R = -1$ conditions, it becomes feasible to calculate the σ_{-1} value.

The cycles corresponding to the fatigue limit can be conservatively estimated as those corresponding to the airframe's service life, i.e.,

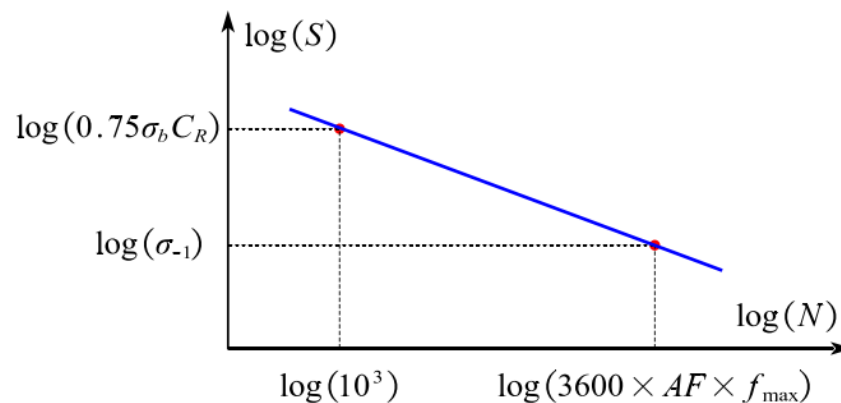
$$N_f = 3600 * AF * f_{max} \quad (14)$$

In the formula, f_{max} is the maximum value of the frequency of the fluid pulsating excitation and the random vibration excitation, in Hz. The power spectrum of the engine random excitation spans a frequency range of 15 to 2000 Hz, while the fluid pulsation excitation occurs at a specific frequency of 630 Hz. Therefore, considering these factors, the value of f_{max} is determined as 2000 Hz. AF represents the aircraft body's operational lifespan, measured in hours. The lifespan of the aircraft body encompasses two primary aspects: durability and calendar life. Durability life signifies the structural ability to withstand fatigue damage during flight, while calendar life pertains to damage caused by environmental corrosion and other factors during ground parking. Hence, this paper focuses on durability life. Based on information provided by professional institutes, it is recommended that the aircraft body should have a minimum lifespan of 6000 h. The S-N curve estimation parameters for stainless steel materials commonly used in aero-engine piping systems are provided in Table 4.

In this study, the power function expression of the S-N curve given by Equation (14) is utilized, and two points $(0.75\sigma_b C_R, 10^3)$ and $(\sigma_{-1}, 3600 \times AF \times f_{max})$ are employed for approximate fitting, as illustrated in Figure 16.

Table 4. S–N curve estimation parameters of several commonly used materials for pipelines.

Material	0Cr18Ni9	1Cr18Ni9	LY12-M	2B06	30CrMnSiA
Reliability correction factor C_R	0.8	0.8	0.8	0.8	0.8
Strength limit σ_b (MPa)	593	608	310	167.5	1080
Fatigue strength S_{1000} (MPa)	444.75	456	232.5	125.62	810
Revised fatigue strength $S_{1000,R}$ (MPa)	355.8	364.8	186	100.5	648
Stress concentration factor	1	1	1	1	1
Cyclic excitation stress ratio R	−1	−1	−1	−1	−1
Parameter a	0.3270	0.3270	0.3270	0.3270	0.3270
Parameter b	8.492×10^{-4}	8.492×10^{-4}	8.492×10^{-4}	8.492×10^{-4}	8.492×10^{-4}
Parameter c	−92	−92	−92	−92	−92
Fatigue limit σ_{-1} (MPa)	138	226	55.8	28	342.4
Young's modulus E (GPa)	199	184	72	69	190
Material yield strength σ_s (MPa)	226	235	265	46.4	835
Airframe life AF (h)	6000	6000	6000	6000	6000
Max excitation f (Hz)	2000	2000	2000	2000	2000
Predicted S–N curve	$\sigma^{20.50} \times N$ $= 6.937 \times 10^{54}$	$\sigma^{18.50} \times N$ $= 9.62 \times 10^{49}$	$\sigma^{14.6} \times N$ $= 2.58 \times 10^{35}$	$\sigma^{9.285} \times N$ $= 2.58 \times 10^{23}$	$\sigma^{27.56} \times N$ $= 2.01 \times 10^{80}$

**Figure 16.** Line S–N curve.

To further elucidate the comprehensive evaluation process, a hazardous node is taken as an example. After calculation, the S–N curve of the material corresponding to the dangerous node is fitted with two points $(355.8, 10^3)$ and $(138, 4.32 \times 10^{10})$, and the result is $\sigma^{20.50} \times N = 6.937 \times 10^{54}$. The pipe body experiences a maximum stress of 228 MPa, while its fatigue life is estimated to be 3.25×10^6 cycles.

The calculation of pipeline lifespan under compound excitation using the optimal transport meshless method has been successfully accomplished.

5. Conclusions

In this paper, the strength and fatigue life of a pipeline is checked by using the optimal transport meshless algorithm based on the strong fluid–structure coupled frame, and the pipeline life is predicted after the design of the pipeline. The most dangerous of the loads considered are the resonant excitation and the pulsating excitation of the fluid inside the line. Therefore, a method to obtain the resonance and pulsation excitation of the pipeline is proposed, and the compound excitation is added to the optimal transport meshless method of the strong fluid–structure coupling computational framework to calculate the stress of the pipeline. After stress is obtained, fatigue life is estimated based on the hazardous point method. The specific conclusions are as follows:

- (1) A method for obtaining the resonant excitation of pipelines based on the load spectrum is proposed. The OTM method is an explicit integration algorithm whose boundary conditions can impose only time-domain signals. In this chapter, the load spectrum of the pipeline system is simulated in the time domain, and the random time-domain signal relative to the load spectrum is obtained. To obtain the time-domain maximum

amplitude information of the acceleration of the random signal. The natural frequency of the pipeline is obtained via the mode analysis of the pipeline. Equation (6) incorporates the maximum amplitude of acceleration and the natural frequency of the pipeline to establish a resonance excitation in the pipeline.

- (2) The pressure on the inner wall of the pipeline caused by the fluid pulsating excitation is obtained, which is provided to the OTM to calculate the dynamic stress of the pipeline under the combined excitation. The pressure on the surface of the inner wall of the pipeline was extracted using CFX2021R2 software, and 1000 sample points were set on the surface of the inner wall of the pipeline. Due to the possibility of misalignment between the sampling points of the pipeline inner wall pressure and the OTM grid file, the nodes on the inner wall surface are clustered by the distance from the sampling points and are divided into 100 sets. A pressure sampling point load is applied to each set to achieve the addition of a fluid pulsation-excited pre-stress field in the OTM. Using the OTM software 8.4, the locations of dangerous points and the maximum stress values of the pipeline under the combined excitation of resonance and fluid pulsation were obtained. The fatigue life of the pipeline under the combined excitation was calculated by fitting the S–N curve.
- (3) For the fatigue life and strength assessment of pipelines, it is common practice to follow the methodology described in the literature [40]. By employing a fluid–structure bidirectional simulation approach, the stresses induced by random vibration excitation and fluid pulsation are separately evaluated for the pipelines. The total stress on the pipelines is then determined by superimposing these two stresses. As evident from the literature [40], this pipeline stress calculation method emphasizes the separation of fluid and solid components. Incorporating both random vibration excitations and pulsation excitations simultaneously within the framework of fluid–structure coupling analysis proves to be an effective strategy for accurately assessing stress levels and evaluating pipeline lifespans, surpassing previous approaches mentioned in the literature [40].

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